Advanced CSP Teaching Materials

Chapter 9
Thermal Energy Storage

Authors
Johannes Sattler
Bernhard Hoffschmidt
Matthias Günther
Michael Joemann

Reviewers
Salman Ajib
Anette Anthrakidis
Cristiano Teixeira Boura
Christian Faber
Joachim Göttzsche
Dörte Laing

1 Solar-Institut Jülich (SIJ), FH Aachen, Aachen University of Applied Sciences, Heinrich-Mußmann-Str. 5, 52428 Jülich, Germany
2 German Aerospace Center (DLR) - Solar Research, Linder Höhe 51147 Cologne, Germany
3 University of Kassel, Institute for Electrical Engineering, Rational Energy Conversion, Wilhelmshöher Allee 73, 34121 Kassel, Germany
4 Technische Universität Ilmenau, Postfach 100565, 98684 Ilmenau, Germany
5 German Aerospace Center (DLR) - Institute of Technical Thermodynamics, System Analysis, Pfaffenwaldring, 38-40, 70569 Stuttgart, Germany
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<td>Latin letters</td>
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<tr>
<td><strong>A</strong></td>
<td>area</td>
<td>m²</td>
</tr>
<tr>
<td><strong>A_p</strong></td>
<td>entire area of the bodies in the packed bed</td>
<td>m²</td>
</tr>
<tr>
<td><strong>B</strong></td>
<td>breadth</td>
<td>m</td>
</tr>
<tr>
<td><strong>Bi</strong></td>
<td>Biot number</td>
<td>-</td>
</tr>
<tr>
<td><strong>c_v</strong></td>
<td>volume specific heat capacity</td>
<td>kWh/m³</td>
</tr>
<tr>
<td><strong>c_p</strong></td>
<td>specific heat (capacity) at constant pressure</td>
<td>J/(kgK)</td>
</tr>
<tr>
<td><strong>c_p,air</strong></td>
<td>specific heat (capacity) of air at constant pressure</td>
<td>J/(kgK)</td>
</tr>
<tr>
<td><strong>c_p,m</strong></td>
<td>average heat capacity</td>
<td>J/(kgK)</td>
</tr>
<tr>
<td><strong>d</strong></td>
<td>diameter</td>
<td>m</td>
</tr>
<tr>
<td><strong>E_dis</strong></td>
<td>discharge energy</td>
<td>J</td>
</tr>
<tr>
<td><strong>E_cha</strong></td>
<td>charge energy</td>
<td>J</td>
</tr>
<tr>
<td><strong>f_utilisation</strong></td>
<td>storage material utilisation coefficient</td>
<td>-</td>
</tr>
<tr>
<td><strong>h</strong></td>
<td>convection heat transfer coefficient</td>
<td>W/(m²K)</td>
</tr>
<tr>
<td><strong>h_melt</strong></td>
<td>specific melting enthalpy</td>
<td>J/kg</td>
</tr>
<tr>
<td><strong>H</strong></td>
<td>enthalpy</td>
<td>J/g</td>
</tr>
<tr>
<td><strong>k</strong></td>
<td>thermal conductivity</td>
<td>W/(mK)</td>
</tr>
<tr>
<td><strong>K_St</strong></td>
<td>heat loss coefficient of a packed-bed thermal energy storage</td>
<td>W/(m²K)</td>
</tr>
<tr>
<td><strong>L</strong></td>
<td>length or height</td>
<td>m</td>
</tr>
<tr>
<td><strong>m</strong></td>
<td>mass</td>
<td>kg</td>
</tr>
<tr>
<td><strong>m</strong></td>
<td>mass flow</td>
<td>kg/s</td>
</tr>
<tr>
<td><strong>m_air</strong></td>
<td>air mass flow</td>
<td>kg/s</td>
</tr>
<tr>
<td><strong>n</strong></td>
<td>number of bodies</td>
<td>-</td>
</tr>
<tr>
<td><strong>p</strong></td>
<td>pressure</td>
<td>Pa</td>
</tr>
<tr>
<td><strong>P</strong></td>
<td>power</td>
<td>W</td>
</tr>
<tr>
<td><strong>P_th,sec</strong></td>
<td>receiver power at design point</td>
<td>W</td>
</tr>
<tr>
<td><strong>P_th,cycle</strong></td>
<td>nominal cycle inlet power</td>
<td>W</td>
</tr>
<tr>
<td><strong>Q</strong></td>
<td>heat</td>
<td>J</td>
</tr>
<tr>
<td><strong>Q_cycle</strong></td>
<td>storage capacity</td>
<td>J</td>
</tr>
<tr>
<td><strong>Q</strong></td>
<td>thermal capacity</td>
<td>W</td>
</tr>
<tr>
<td><strong>Q_loss</strong></td>
<td>heat loss to the surroundings</td>
<td>W</td>
</tr>
<tr>
<td><strong>Re</strong></td>
<td>Reynolds number</td>
<td>-</td>
</tr>
<tr>
<td><strong>t_p,e,nom,a</strong></td>
<td>hours at nominal solar electrical power per annum</td>
<td>h</td>
</tr>
<tr>
<td><strong>T</strong></td>
<td>temperature</td>
<td>°C or K</td>
</tr>
<tr>
<td><strong>T_air</strong></td>
<td>airflow temperature</td>
<td>°C or K</td>
</tr>
<tr>
<td><strong>T_b</strong></td>
<td>temperature of a body/bodies (pebbles)</td>
<td>°C or K</td>
</tr>
<tr>
<td><strong>U</strong></td>
<td>circumference of a packed-bed thermal energy storage system's cross section</td>
<td>m</td>
</tr>
<tr>
<td><strong>V</strong></td>
<td>volume m³; usable storage volume</td>
<td>m³</td>
</tr>
<tr>
<td><strong>w</strong></td>
<td>air velocity</td>
<td>m/s</td>
</tr>
<tr>
<td><strong>W_{e,a}</strong></td>
<td>total annual electrical work</td>
<td>J</td>
</tr>
<tr>
<td><strong>x</strong></td>
<td>length or height</td>
<td>m</td>
</tr>
<tr>
<td><strong>y</strong></td>
<td>hours per year</td>
<td>h</td>
</tr>
</tbody>
</table>
### Greek letters
- $\alpha$: area-related heat transfer coefficient, $W/(m^2K)$
- $\alpha_v$: volume-dependent heat transfer coefficient, $W/(m^3K)$
- $\Delta$: difference operator
- $\varepsilon$: porosity (relative to void volume)
- $\vartheta$: temperature, °C or K
- $\eta_{cha,store,dis}$: storage loss factor
- $\lambda$: effective heat transfer coefficient, $W/(mK)$
- $\nu$: kinematic viscosity, $m^2/s$
- $\rho$: mass density, $kg/m^3$
- $\sigma$: Stefan-Boltzmann constant, $W/(m^2K^4)$

### Subscripts
- $b$: body or bodies
- $e$: electric
- $liq$: liquid
- $m$: mean
- $melt$: melting
- $min$: minimum
- $max$: maximum
- $sol$: solid
- $th$: thermal

### Acronyms
- CF: electrical capacity factor
- CSP: Concentrating Solar Power
- DLR: Deutsches Zentrum für Luft- und Raumfahrt (German Aerospace Center)
- DSG: direct steam generation
- HTF: heat transfer fluid
- PCM: Phase Change Material
- PV: Photovoltaics
- SEGS: Solar Energy Generating Systems
- SM: solar multiple
Summary

This chapter discusses thermal energy storage systems for solar thermal power plants. Available storage media are presented as well as equations for doing basic and sometimes also detailed calculations. The chapter also discusses the solar field size in relation to how much energy can be made available to the energy storage and how the solar field should be sized in order to provide sufficient energy also for less favourable seasons, especially winter. Furthermore, the topic storage capacity is described, which gives, for example, details as to how the raising of the solar multiple (and enlarging of the energy storage system) can increase the plants hours at nominal power. The last section prior to the exercises section is dedicated to the topic modular storage. Modular storage is the subdivision of a single storage into several storage modules.
9 Thermal Energy Storage

Key questions

- Why is thermal energy storage so very important for solar technologies?
- What types of thermal energy storage systems are available?
- What is the solar multiple?
- What is storage capacity?
- What are the advantages of modular storage systems?

Introduction

In conventional power plants, the fuel (or the uranium) in the storage facilities of the power plant simultaneously represents the energy storage. Additional energy storage is generally not required.

Using an energy storage system in a solar thermal power plant has many advantages: higher annual solar contribution, reduction of part-load operation, power management and buffer storage [1]. An energy storage system has the function of a buffer during the day as the solar irradiation onto the earth’s surface varies with time (e.g. night and day, seasonal changes, diurnal variation and weather) [2]. With the aid of the storage the security of the energy supply is increased. In some solar thermal power plants the energy storage system delivers enough energy to operate the plant for many hours after sunset, as for example the 17 MW_solar tower Gemasolar in Spain, which has an energy storage system with a capacity of 15 hours.

A solar thermal power plant without an energy storage system can therefore only operate between sunrise and sunset, unless it is hybridised. Hence either an energy storage system and/or fossil co-firing should be integrated into the plant, as presented in Figure 1.

![Figure 1: Concept of the implementation of a heat storage system [1]](image)

When designed with an adequately sized thermal energy storage system and a fossil back-up system, CSP technologies provide firm capacity and power on demand in contrast to other renewable technologies such as PV and wind power which can only provide fluctuating power. No
other significant resources of renewable energy technologies which can deliver power on demand are available in the MENA region (e.g. no biomass, no hydro-pump-storage as, for example, found in the EU). When the share of power generated by renewable energies shall be increased to above approximately 35% of the total power generation, CSP becomes a key factor because as a dispatchable renewable power technology it can fill the gaps which are generated by PV and wind power.

When using a thermal energy storage and fossil back-up system, CSP plants can be operated as base load power plants, which is an advantage and is, for example, important for seawater desalination since desalination plants are required to operate in base load mode.

However, without thermal energy storage and fossil back-up system, CSP has no advantages against PV. This is because CSP technologies are, for example, more complicated than PV and not easily modularly extendable like PV.

This chapter discusses various types of thermal energy storage systems that have been applied in existing solar thermal power plants. Moreover, storage concepts which are still in a development stage are also introduced. Whenever available, existing solar thermal power plants with thermal energy storage are presented along with technical information.

The chapter also discusses the solar field size in relation to how much energy can be made available to the energy storage and how the solar field should be sized in order to provide sufficient energy also for less favourable seasons, especially winter. Furthermore, the topic storage capacity is described, which gives, for example, details as to how the raising of the solar multiple (and enlarging of the energy storage system) can increase the plants hours at nominal power. The last section prior to the exercises section is dedicated to the topic modular storage. Modular storage is the subdivision of a single storage into several storage modules.

### 9.1 Thermal Energy Storage Types

In the following section thermal energy storage types for the application in solar thermal power plants are described.

The current thermal energy storage types can be divided into four main groups:

1. Thermal Energy Storage Systems for Sensible Heat
   a) Indirect Storage\(^1\) Systems
   b) Direct Storage Systems

2. Latent-Heat Storage

3. Steam Accumulator

4. Thermo-Chemical Storage System

The four groups are described on the following pages.

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\(^1\) Indirect storage: When the direct storage of the HTF from the primary circuit is not possible, then the heat contained in the primary HTF can be transferred to another favourable medium, which can then be used as the storage material. Such storage is referred to as indirect storage.
9.1.1 Thermal Energy Storage Systems for Sensible Heat

The term sensible heat describes the heat which is absorbed or released by a material as a result of a change in temperature, whereupon the material does not undergo a change of aggregate state. Thermal energy storage systems for sensible heat are distinguished in: indirect storage and direct storage.

(a) Indirect Storage Systems

Four indirect storage concepts are introduced here:

i. 2-Tank Molten Salt Indirect Storage
ii. Packed-Bed Thermal Energy Storage (Regenerator)
iii. Sand Storage
iv. Concrete Storage

With indirect storage systems it must generally be taken into account that there is a pressure drop of the HTF in the charging and discharging process of the storage. The pressure drop results in an increase of the work needed to be done by the pump. This leads to reduced storage efficiency [2]. In addition there are energy losses and temperature drops in the heat exchanger when transferring the heat from the HTF to the storage medium and vice versa.
2-Tank Molten Salt Indirect Storage

The 2-tank molten salt indirect storage system is a commercially available technology, which is based on nitrate salts [1]. This storage type finds application predominantly in parabolic trough power plants and is the most widely-used technology. The recently constructed parabolic trough plants Andasol I, II, III use this technology. Figure 2 and Figure 3 show aerial views and a schematic of the Andasol I plant located in Andalusia, Spain.

The receivers of the collectors carry temperature-stable synthetic oil, the HTF, which is heated to 400°C [3]. In a boiler the heat from the HTF is passed to the steam cycle. The generated steam drives a steam turbine, which in turn drives a generator for electricity production (50 MW). As the storage medium is different than the HTF, a heat exchanger is used so that the heat from the HTF is passed for storage.

The indirect storage system comprises two tanks: a hot-salts and cold-salts tank, which are filled with altogether 28,500 tonnes of molten salt, allowing peak load operation for up to 7.5 h after sunset [3]. In summer, it is possible to run the turbine nearly 24 hours a day. The composition of the salt is 60% sodium nitrate (NaNO$_3$) and 40% potassium nitrate (KNO$_3$) [1].

![Figure 2: Aerial views of Andasol I](image1)

![Figure 3: Dual tank molten salt storage Andasol I](image2)
With reference to Figure 4, the storage system works as follows: During the day the hot synthetic oil from the collectors is not only passed through the boiler but also through a heat exchanger for passing the heat to the (separate) storage cycle. When charging, the salt from the cold-salts tank is pumped through the heat exchanger, upon which the salt is heated to approximately 386°C and is pumped into the hot-salts tank. During the night or in periods with low solar irradiance, the flow direction is reversed. The salt is pumped from the hot-salts tank through the heat exchanger into the cold-salts tank. In the heat exchanger the salt passes the heat to the synthetic oil. The heated synthetic oil is then passed through the boiler for producing steam.

The power plant control system ensures that the salt never falls below a temperature of 292°C, if necessary by means of electrical heating, to prevent the salt from solidifying. Solidification would take place at approximately 220°C [2].

![Figure 4: Schematic of Andasol I](image)

Figure 5 shows a typical summer day plant performance of Andasol I.
Some more technical details of the Andasol I, II and III plants are listed below ([2], [3], [5], [6]):

**General information**
- The Andasol I plant was officially inaugurated on July 1, 2009. Andasol II was completed in mid-2009 and connected to the grid. The construction works for the Andasol III plant started in spring 2009 and are scheduled to be completed in February 2011.
- Cost: about € 300 million
- Each of the plants requires 600,000 m³ of cooling water annually
- The filling of the 2000 m³ HTF prior to the commissioning of the plant took several weeks

**Solar field**
- The overall reflective mirror area amounts to 510,120 m²
- Size of solar field: 510,120 m²
- Number of parabolic mirrors: 209,664 mirrors
- Annual direct standard radiation (DNI): 2136 kWh/m²a
- Number of solar sensors: 624 sensors
- Solar field efficiency: approx. 70% peak efficiency, approx. 50% annual average
- Altogether about 90 km of absorber pipes (receivers)

**Power plant capacity**
- Each plant has (will have) a peak power output of 49.9 MWₑ
- Entire plant efficiency: Approximately 28% peak efficiency, approximately 15% annual average
- Annual operating hours: Approximately 3500 h at full-load
- Forecast gross electricity volume: about 180 GWh
- Estimated lifespan: minimum 40 years
A storage system with two tanks has also disadvantages:

- It is relatively expensive [7]. The investment cost of a storage system with two tanks is effectively doubled compared to a single tank solution because two tanks and duplicate piping are required [8].
- Using nitrate salts as storage medium has the disadvantages that it has high freezing points (120 – 220°C) [7] and that the price for nitrate salts can fluctuate because it is a commodity product that is subject to price pressures [8].

ii. Packed-Bed Thermal Energy Storage (Regenerator)

A packed-bed thermal energy storage (also known as pebble bed or rock pile storage) consists of a container filled with a bed of loosely packed particulate material with a high heat capacity e.g. pebbles, gravel or rocks (see Figure 6). This storage type was designed to utilise air as the heat transfer fluid (HTF). However, it would also be possible to use liquid media. The following descriptions are based on the HTF air.

At both the top and bottom of the storage there is a duct through which the air is forced. When charging, the hot air enters the storage through the top duct, passes through the pebble bed transferring the heat to the storage material, and leaves the storage through the bottom duct. In the discharge process, the direction of the air circulation is reversed. Cold air enters the storage through the bottom duct and is heated up as it travels upwards to the top where it exits through the top duct. The hot air can then be passed through a steam generator [4], [9].

![Figure 6: Illustration of a packed-bed thermal energy storage unit](image)

Shown in Figure 7 below is a *temperature–storage height* plot of the thermal stratification (temperature distribution) in the packed bed. The plot contains five lines, covering a range of
storage loads between 0% (completely discharged) and 100% (completely charged). For ease of explanation these conditions will be referred to as the "blue line" and the "red line", respectively.

![Storage temperature profile](image)

**Figure 7:** Temperature profile of a packed-bed thermal energy storage system for a range of load percentages [4]

**Blue line:** Up to approximately 1/3 of the storage height², ambient temperatures prevail. The temperatures then begin to rise along the height of the storage. At the top of the pebble-bed, the temperatures are close to, but still below the maximum material temperature. Therefore, this condition is referred to as having 0% load, as the maximum air temperature has not yet been reached [4].

**Red line:** In the process of charging the storage, hot air passes through the pebble-bed in the downward direction. The time needed to fully charge the storage depends on the design of the storage. When commencing the charging process, only the pebbles near the top are heated to maximum temperature and less heat is available for the lower levels. This is due to the rapid heat transfer. When the material at the top level has reached maximum temperature (i.e. the temperature of the hot air), the hot air then passes through the top layer without transferring any heat until a layer at a lower temperature is reached. Over time the lower layers will also have been heated to maximum temperature. This is a continuous process of the material layers being heated from the top to the bottom.

The storage is said to be fully loaded when the hot air passing through it transfers so little heat to the material that on exiting, the air temperature is near to the maximum allowed by system boundary conditions. Moreover, the warmer the exiting air exiting the storage at the bottom is, the greater would be the loss of energy. The condition that a storage system is defined to be fully loaded therefore depends on the power plant air system’s configuration and component limitations.

² The ratio depends on the process boundaries and storage construction
According to the red line in the graph the storage is fully charged when approximately 2/3 of the storage material\textsuperscript{3} from the top has reached maximum temperature [4].

The Solar Tower Jülisch uses air as HTF in its primary cycle (as described in the chapter Solar Tower Technology) and was built with a similar packed-bed thermal energy storage system. The major difference, however, is the storage material used as it is a porous (multi-square-channel geometry) cubic ceramic structure, similar to that applied in a regenerator or Cowper storage, as shown in Figure 8. The storage chamber is fitted with several thousand of these ceramic structures that are piled next to and on top of each other. When charging, hot air passes through the channels and transfers its heat on the thin walls. When discharging, the walls transfer the heat to cold air, heating it up on its way to the duct at the top of the chamber [4].

![Figure 8: Salt/ceramic checker bricks (left); 10-50 MW regenerator / Cowper storage; > 100 kWh (right) [10]](image)

Generally speaking, ceramic bodies can be designed in many different forms (see Figure 9). All these ceramic bodies are porous and can be easily stacked. Hexagonal ceramic checker bricks, type c), however, are favoured by industry. The ceramic bodies are used in industrial facilities, e.g. in the iron works industry in high temperature industrial regenerators (hot blast stoves), which are commercial wind heaters (the supplied air to be preheated is referred to as “wind”). In this application, several of these storage types are constructed in parallel and operated sequentially, resulting in quasi-continuous heat exchanger operation. The other ceramic bodies shown in Figure

\textsuperscript{3} The ratio depends on the process boundaries and storage construction
9 are also used in regenerator systems, but for other applications – ceramic bodies a) and b) find application in glass-kilns and ceramic body d) in coke ovens.

![Figure 9: Shapes of ceramic bodies for high temperature regenerators, edited from [2]](image)

The following presented formulae are quoted (and translated) from source [11, pp. 149-153]:

The thermal capacity $\dot{Q}$, which is transferred from the airflow to the bodies (e.g. pebbles) in the charging process is calculated by the following equation:

$$\dot{Q} = \alpha_v \cdot V \left( T_{\text{air}} - T_b \right) \text{ [W]}, \quad (1)$$

where $\alpha_v$ is the volume-dependent heat transfer coefficient between the airflow and the pebbles [W/(m$^3$K)], $V$ is the usable storage volume of the pebbles$^4$ [m$^3$], $T_{\text{air}}$ is the airflow temperature and $T_b$ is the temperature of the bodies [$^\circ$C].

The volume-dependent heat transfer coefficient $\alpha_v$ between the airflow and the bodies in the packed-bed thermal energy storage system is calculated as follows:

$$\alpha_v = 650 \left( \frac{\rho \cdot w}{d} \right)^{0.7} \text{ [W/(m$^3$K)]}, \quad (2)$$

where $\rho$ is the density of the air [kg/m$^3$], $w$ is the airflow velocity [m/s] and $d$ is the equivalent diameter of the bodies [m].

The equivalent diameter of the bodies is calculated by:

---

$^4$ As mentioned before, not the total volume of the packed bed reaches operating temperatures. Hence, the usable volume is less than the total volume of the pebbles.
\[ d = \left[ \frac{6 \ V \ (1 - \varepsilon)}{\pi \ n} \right]^{1/3} \text{[m]}, \quad (3) \]

where \( \varepsilon \) is the porosity (relative void volume) of the packed bed and \( n \) is the number of bodies in the packed bed.

The Biot number \( Bi \) is calculated from the convection heat transfer coefficient \( h \) [W/m\(^2\)K], the radius \( R \) of the body [m] and the thermal conductivity \( k \) [W/(m K)]:

\[ Bi = \frac{h \ R}{k} \quad [-] \quad (4) \]

The Biot number is the ratio of the inner resistance of a body \( (R/k) \) to the resistance at the surface \( (1/h) \) of a body. At values of \( Bi \leq 0.1 \) the inner thermal resistance of the body is small in comparison of the surface resistance and therefore the internal temperature gradients are negligible.

The pressure loss in a packed-bed thermal energy storage for a range \( 100 < Re < 13,000 \) can be calculated from the following equation (from Cole).

\[ \Delta p = \rho \ w^2 \left( \frac{L}{d} \right) \left[ \frac{(1 - \varepsilon)^2}{Re} + \frac{368}{(1 - \varepsilon)} \right] \quad [\text{Pa}], \quad (5) \]

where \( L \) is the length (height) of the packed bed in flow direction [m], \( d \) is the equivalent diameter of the bodies [m], \( w \) is the air velocity [m/s], \( \varepsilon \) is the porosity of the backed bed (from 0.35 to 0.5), \( Re \) is the Reynolds number \( Re = \frac{w \ d}{\nu} \), and \( \nu \) is the kinematic viscosity of the air [m\(^2\)/s]. Please note that in this chapter, the Reynolds number is described with the symbols according to the source and may vary from other text books.

For bodies with an uneven form a form factor \( f \) is used in order to take the deviation of the spherical shape into account. The form factor \( f \) is the ratio of the area of the body to the area of a sphere of the same size. There are no sure descriptions of \( f \) values but specific approximate values can be used. Example for assumed values: \( f = 1.5 \) for round gravel; \( f = 2.5 \) for small granite bodies that are broken into pieces and \( f = 1.5 \) if these granite bodies have a diameter > 50 mm. With the values of \( \varepsilon \) and \( f \) the pressure loss can be calculated:

\[ \Delta p = \rho \ w^2 \left( \frac{L}{d} \right) \left(1 - \varepsilon\right) \frac{f}{\varepsilon^{3/2}} \left[ 4.74 + 166 \left(1 - \varepsilon\right) \frac{f}{\varepsilon^{3/2}} \frac{Re}{50} \right] \quad [\text{Pa}], \quad (6) \]

If the values of \( \varepsilon \) and \( f \) are not known then the following simplified equation is used:

\[ \Delta p = \rho \ w^2 \left( \frac{L}{d} \right) \left(1 - \varepsilon\right) \frac{1750}{50} \quad [\text{Pa}], \quad (7) \]

Figure 10 shows the pressure drop per height of the packed bed, \( \Delta p/H \), the volume-dependent heat transfer coefficient \( \alpha_v \) for the packed-bed thermal energy storage system in dependence of the air velocity \( w \) and also the size of a body between 20 and 150 mm. Note that the symbols \( H \) and \( L \) are the same.

The entire area of the bodies in the packed bed \( A_p \) is calculated by:
The equation for calculating the area-related heat transfer coefficient $\alpha$ is:

$$\alpha = \frac{\alpha_v \cdot d}{6 \, (1 - \varepsilon) \, f} \quad [\text{W/(m}^2\text{K)}],$$  \hspace{1cm} (9)

The heat transfer in a packed-bed thermal energy storage with a total packed bed height $H$, which is divided into layers of equal length $\Delta x = H/n$, shall now be considered. The discretisation scheme of the packed-bed thermal energy storage in a mathematical model is shown in Figure 11. Due to the low thermal conductivity of the packed bed the radial temperature gradients can be neglected so that every layer $i$ at a time $t$ can be marked with a uniform temperature $T_{b,i}$ of the bodies.

The instantaneous energy balance for the $i$-th layer of the packed-bed thermal energy storage in the charging process can be described with the following differential equation:

$$\left(1 - \varepsilon\right) \rho_b \, c_b \, V_i \, \frac{dT_{b,i}}{dt} = \alpha_v \, V_i \left(T_{\text{air},i-1} - T_{b,i}\right) - \dot{Q}_{\text{loss},i} \quad [\text{W}], \hspace{1cm} (10)$$

where $\rho_b$ is the density of the body [kg/m$^3$], $c_b$ is the specific heat capacity of the body [J/(kg K)], $V_i$ is the volume of the $i$-th layer [m$^3$], $T_{b,i}$ is the temperature of the body [°C], $t$ is the time [s],
\( T_{\text{air,}i-1} \) is the temperature of the airflow from the \((i-1)\)-th layer \(^\circ\text{C}\) and \( \dot{Q}_{\text{loss,}i} \) is the heat loss to the surroundings [W].

The heat loss of the \(i\)-th layer to the surroundings is calculated from the equation:

\[
\dot{Q}_{\text{loss,}i} = K_{St} \left( T_{\text{b,}i} - T_{\text{amb}} \right) \Delta x \; U \quad [\text{W}],
\]

where \( K_{St} \) is the heat loss coefficient of the thermal energy storage [W/(m\(^2\)K)], \( T_{\text{amb}} \) is the temperature surrounding the thermal energy storage \(^\circ\text{C}\), \( \Delta x \) is the height of the \(i\)-th layer [m] and \( U \) is the circumference of the thermal energy storage’s cross-section [m].

Because the specific heat capacity of the air can be neglected when compared to that of the bodies, the following energy balance is valid for the \(i\)-th layer of the thermal energy storage in the charging process:

\[
\dot{Q} = (m_{\text{air}} \cdot c_{p,\text{air}}) \left( T_{\text{air,}i-1} - T_{\text{air,}i} \right) = \alpha_v \; V_i \left( T_{\text{air,i-1}} - T_{\text{b,}i} \right) \quad [\text{W}]
\]

This follows that the air temperature in the \(i\)-th layer of the thermal energy storage is calculated with the equation:

\[
T_{\text{air,}i} = T_{\text{air,}i-1} - \frac{\alpha_v \; V_i \left( T_{\text{air,i-1}} - T_{\text{b,}i} \right)}{(m_{\text{air}} \cdot c_{p,\text{air}})} \quad [\text{\(^\circ\text{C}\)}],
\]

where \( T_{\text{air,}i} \) is the air temperature in the \(i\)-th layer \(^\circ\text{C}\), \( T_{\text{air,i-1}} \) is the air temperature in the \((i-1)\)-th layer \(^\circ\text{C}\), \( m_{\text{air}} \) is the air mass flow [kg/s] and \( c_{p,\text{air}} \) is the specific heat capacity of air [J/(kg K)].

Similar equations can be derived for the discharging of the storage.

Figure 11: Model of the packed-bed thermal energy storage system [11, p. 151]
iii. *Sand Storage*

The sand storage may become particularly important in the future when solar tower power plants are being constructed near to or in sand-desert regions.

The advantages of sand are [4]:

- low cost
- high storage capacity
- suitable for high temperatures
- environmentally friendly

Similar to the packed-bed thermal energy storage, the sand storage also utilises the HTF air from the primary cycle of a solar tower power plant with air receiver. As well as using regenerators as thermal energy storage using ceramics or concrete as storage material, it is also possible to utilise sand as a cheaper alternative [4].

When charging, the hot air is passed directly through a packed bed of vertically sliding sand in a cross-flow arrangement. The grains of sand are thereby heated up to approximately 800°C and fall into a hot storage. In the discharging process the sand is transported from the hot storage to a so-called fluid bed cooler, which works in a similar way to a fluidised-bed combustion system. The hot sand is fluidised from below (air streams are blown into the sand from below at high speed, causing the grains of sand to begin to float). The fluidised sand comes into contact with water pipes and transfers its heat to the water. The water will have evaporated by the time it has left the fluid bed cooler. The generated steam can then be fed to the steam turbine [4]. The cooled sand exits the fluid bed cooler at a temperature of approximately 150°C and returns to either the air-sand heat exchanger or it is stored in the cold storage tank [4].

![Figure 12: Sand storage concept in a solar thermal power plant [4]](image_url)

For the implementation of this concept, a suitable air-sand heat exchanger is required [4].

The air and sand inlet temperatures are 800°C and 200°C, respectively (see Figure 13). In the test rig, sand is filled into a storage bin at the top of the heat exchanger. The sand slides vertically...
through a rectangular narrow-width passage of the heat exchanger. In a cross-flow direction, the hot air enters the heat exchanger through a porous wall, passes through the moving bed of sand, transfers the heat, and exits at a much lower temperature through another porous wall. The hot sand exits at the bottom. The sand grain size is approximately 1 mm [4].

The simulated air and sand temperature distribution and the air flow vector field are shown in Figure 14. As shown in the model, the hot air shall transfer as much of its heat to the sand as possible and exit the heat exchanger at a low temperature of approximately 200°C (blue colour). The sand on the other hand shall exit the heat exchanger with an optimal temperature close to the hot inlet air temperature (red colour). The temperature distribution in the sand passage shows the typical cross-flow characteristic with a near-diagonal isotherm. For the application of this heat exchanger in solar thermal power plants upscaling will be necessary [4].
If, on the other hand, sand grain sizes larger than 1 mm are preferred in the future, then the sand storage concept of Figure 12 is no longer optimal. The larger the grain size the greater the erosion and therefore the greater the damage of the heating surfaces of the fluid bed cooler. Hence the storage concept would have to be modified as shown in Figure 15 [4].

This storage concept is similar to that of the Solar Tower Jülich (packed-bed storage, air as HTF in the primary cycle) with the difference that an air-sand heat exchanger, a hot and cold storage tank and a sand transportation system are used. When charging, the cold sand is transported from the cold storage to the air-sand heat exchanger where it is heated by the hot air. The heated sand is then stored in the hot storage. When discharging, the hot sand from the hot storage is transported to the
air-sand heat exchanger where it then heats the cold air. The heated air is then passed through the steam generator. After the sand has transferred its heat it is returned to the cold storage [4].

iv. Concrete Storage

Storing energy in concrete is a very promising option that has been investigated in the recent years and it is expected to be an attractive option regarding investment and maintenance costs [12]. Based on the typical and state-of-the-art boundary conditions for a parabolic trough power plant (HTF temperature: 400°C; HTF pressure: 25 bar; charging/discharging time: 6h/6h), two concrete storage test modules have been developed [13], [12]. After successfully testing the first module, the second module (Figure 16 and Figure 17) was designed by Ed. Züblin AG and DLR in a joint effort and a major focus was given to cost reduction. Basic components of the concrete storage comprise a tube register and a storage concrete. The tube register, which consists of 132 seamless tubes (arranged in a triangular pitch) and each having a length of 9 m and outer diameter of 18 mm, is embedded within the concrete and it is required for the transportation and distribution of the heat transfer fluid while the fluid pressure is still sustained. The thermal energy is stored by the concrete [12].

The dimensions of the test module are: total length: 10 m; length of storage concrete: 8.60 m; height/width: 1.70 m × 1.30 m [13]. As the tube register is embedded within the concrete, possible leaks must be avoided. Hence, as a measure against leaks, the storage was designed such that no joints are found within the storage concrete [12].

![Figure 16: Storage module with visible tube register](image)
The concrete storage can be referred to as a regenerative storage as in the charging and discharging process, which takes place in cycles, a hot and a cold HTF passes through the storage module (i.e. through the tube register). By means of heat transfer by conduction the HTF passes the heat onto the concrete as it passes through tubes that are distributed throughout the structure of the concrete. The charging and discharging process works as follows: To charge the storage, the HTF heated by the solar field is passed through the tubes and heats up the concrete. When discharging the storage, cold HTF flows in the opposite flow direction as in the charging process and is heated up by the concrete. The storage has reached a fully charged state when the HTF exiting the storage has a higher temperature than the maximum inlet temperature for the collector. In the discharge process, the concrete passes the heat to the HTF and the HTF passes the heat to the steam cycle. The state when the storage is said to be fully discharged depends on the minimum inlet steam temperature with which the steam turbine can be operated [12].

For the application in a parabolic trough power plant like the Andasol I, II or III plant types, which use thermal oil as HTF and produce 50 MWₑₑₑ, it has been determined that the volume of the concrete required for storing such a plant’s overall storage capacity of 1,100 MWhₑₑₑ must be approximately 50,000 m³. As the volume is too large to have a single storage block built in this size, it must be designed modularly. Such a basic storage module is shown in Figure 18, which has the preferable dimensions of 18 m length, 4 m height and 2.6 m width and weighs 400 tonnes. The basic module cannot be sized bigger than this due to a number of limitations which include: the
desired tube length must be available on the market; the components must be producible and not be too large for transportation [13].

Altogether 252 of these basic storage modules are required to make up the entire storage system’s volume. Grouped in 4 units of 63 basic storage modules, the modules are interconnected in series and in parallel, as presented in Figure 19. The land requirement is approximately 300 m × 100 m. An estimated €38 million in investment costs are required of the described storage system for a parabolic trough power plant of the Andasol design [13].
b) Direct Storage Systems

Three direct storage concepts are introduced here:

i. 2-Tank Molten Salt Direct Storage
ii. Single-Tank Thermocline Storage
iii. 2-Tank Oil Storage

i. 2-Tank Molten Salt Direct Storage

The direct storage system has been successfully applied in the solar tower technology. The usage of molten salt as HTF in parabolic trough power plants is still under investigation. For both CSP technologies, the 2-tank molten salt direct storage is discussed in more detail.

2-tank molten salt direct storage in solar tower power plants

In direct storage systems, the HFT, which is heated by a receiver, is used directly as a storage medium. The HTF/storage medium can be stored either in a single or dual tank. The nowadays very promising direct storage system consists of two tanks and the HTF/storage medium is molten salt. By the example of the solar tower power plant Solar Two, this storage type is explained in more detail. The plant had a 2-tank direct storage system consisting of a hot-salts and a cold-salts storage tank (see Figure 20 and Figure 21).

Figure 20: Direct storage system of the “Solar Two” solar tower power plant, Barstow, USA [15]
The storage medium was a salt mixture of sodium nitrate (NaNO$_3$) and potassium nitrate (KNO$_3$) in the proportion 60% NaNO$_3$ + 40% KNO$_3$. The total weight of the nitrate salts in the system was 1,500 tonnes. The 2-tank system consisted of a hot-salts and a cold-salts tank. In the cold-salts tank, the salt temperature is maintained at 290°C at all times (even if the plant was not in operation) because the salt mixture would have solidified at approximately 221°C. This was achieved with an electrical heating system in the tank at times when the plant was not in operation. However, the salt in the piping between the hot-salt tank and the receiver was allowed to solidify though because when starting up the plant, an electric trace heating system preheated the pipe until the salt was molten again.

When the plant was in operation, the salt was pumped from the cold-salts tank through the receiver, where the salt was heated to process temperature (565°C). From the receiver, the salt was directed into the hot-salts tank. The salt in this tank had a temperature of 565°C. From the hot-salts tank the salt was passed through the steam generator. On leaving the steam generator, the salt was passed back into the cold-salts tank. The capacity of the storage system was approximately 115 MWh$_{th}$, allowing for 3 hours of nominal power generation. The efficiency of charging, storing and discharging of this storage system was approximately 97%. Therefore, a 2-tank storage system comes closest to the ideal storage [2].

The criteria for an ideal storage are given below [2]:

- No storage losses, i.e. the insulation is ideal
- No exergy storage losses, i.e. the exergy of the stored heat is equal to the exergy that is withdrawn (no mixing of hot and cold storage medium, no temperature balance during storage, no temperature loss through heat transfer during the charge and discharge processes.
- The discharge capacity is equal to the charge capacity

The 2-tank direct storage system technology is also applied in a new power plant. The Gemasolar solar tower power plant is, at the date of writing, the currently newest solar tower power plant that
has been completed in construction. The Gemasolar plant is also the first commercial plant that uses molten salts (sodium and potassium nitrates) as HTF/storage medium in an external receiver and the 2-tank direct storage system [16]. It is planned that operation starts in 2011 [17]. Figure 23 shows a schematic of the plant and Figure 24 the construction site.

A selection of technical details is given below [18], [16]:

- The storage system consists of a hot-salts and a cold-salts tank, which store the salt at 565°C and 290°C, respectively
- Capacity of storage system: 15 hours
- Receiver inlet temperature: 290°C
- Receiver outlet temperature: 565°C
- Height of tower: 150 m
- Electricity generation: 17 MW_e
- Heliostat field area: 318,000 m²
- Number of heliostats: 2,650
- Net electrical production expected: 110 GWh/year

2-tank molten salt direct storage in parabolic trough power plants
The usage of molten salt as HTF in parabolic trough power plants is still under investigation. The pioneer in this field is the Italian Archimede Solar Energy that developed and produces receivers for molten salt as HTF. These receivers can be operated at high temperatures (operation range from 290 to 550°C) [19]. The salt is a eutectic mixture of 60% NaNO₃ and 40% KNO₃. As the solar field uses molten salt as HTF, the molten salt can be stored directly, as shown in the example configuration in Figure 22.

The costs are low in comparison to thermo oil and the required salts are available in large quantities. The mixture has high heat capacity and high heat conductivity. The freezing temperature is roughly between 220 to 240°C. The high freezing temperature requires the implementation of a freeze protection system.

![Figure 22: Example of a parabolic trough power plant using molten salt as HTF and a direct molten salt storage][19]

As mentioned above, the operation temperature of the solar field can be higher than with thermo oil. The higher temperatures, which can be reached with molten salts, make possible the use of
steam turbines that operate at conditions that are closer to the temperature conditions in fossil-fuel power plants. This means that standard power plant components can be integrated more easily into parabolic trough plants. A big problem, however, is the freeze protection of the solar field if it is operated with molten salts that have a high freezing temperature.

**Figure 23:** Gemasolar Solar Tower, Fuentes de Andalucía, Spain [20]
The storage capacity $Q$ is calculated via the product of the salt mass, the average specific heat capacity and the temperature difference between the charging and discharging temperatures [2]:

$$Q = m \cdot c_p, m \cdot \Delta \vartheta_{\text{min} - \text{max}} \ [\text{J}] \quad (14)$$

The disadvantages of using a storage system with two tanks as well as the use of nitrate salts as storage medium are the same as those given in the previous section “i. 2-Tank Molten Salt Indirect Storage”.

**ii. Single-tank thermocline storage**

The single-tank thermocline storage technology was investigated and provides one possibility for further reducing the cost of a 2-tank molten salt direct storage. In this design, both the hot and the cold storage fluid are stored in a single tank, where the hot fluid is at the top and the cold fluid at the bottom of the vessel. The thermocline is the zone between the hot and cold fluids.

The major advantage of this type storage is that it is possible to replace most of the storage fluid with a low-cost filler material [7] (the requirement is as low as 30% of entire storage medium). The filler material provides the bulk of the thermal capacitance of the system (i.e. it stores heat well) and also prevents convective mixing [8].

Research was conducted by Sandia National Laboratories and a 2.5 MWh, backed-bed thermocline storage system was designed (Figure 25). The storage fluid was a binary molten-salt and the filler material quartzite rock and sand [7].

**Figure 24**: Construction site of the Gemasolar solar tower [17]
A system diagram of the Solar One experimental plant is shown in Figure 26. The water/steam receiver is located at the top left corner of the diagram. Water enters the receiver and the generated steam can be directed to a steam turbine or storage. Instead of using a steam accumulator for storage, a thermocline rock/oil storage tank was used. To charge the storage, steam from the receiver is passed through an oil heater, through which cold oil was pumped. The heated oil is then stored in the thermocline storage tank. For discharging the storage, hot oil was pumped from the thermocline storage tank to an oil/steam generator. The generated steam was then passed to the steam turbine, where it entered at a lower pressure stage.

Figure 25: Thermocline test at Sandia National Laboratories [7]
iii. 2-tank oil storage

The commercially operating parabolic trough power plant SEGS I, which is located in the Mojave Desert in USA, began operation in 1985 and delivers a peak electrical output of 13.8 MW. The plant is still in operation, except for its thermal energy storage [22]. The plant was originally designed with a 2-tank oil storage system consisting of a hot oil tank and cold oil tank [2]. Figure 27 shows the schematic of SEGS I.

![Figure 27: Schematic of SEGS I [2]](image)

The cold storage tank had a diameter of 21 m and a height of 12.2 m. The volume without the roof curvature amounted to 4223 m³. With 4640 m³, the hot storage tank is designed slightly larger than the cold tank to compensate the expansion of the HTF after it was heated by the solar field. Approximately 3.4 million litres of HTF are used for the storage [2].

With reference to Figure 27 the storage system of SEGS I is described by the following:

The petroleum-based HTF is operated with a maximum and minimum temperature of 307°C and 240°C, respectively.

The primary cycle is situated within the boundaries of the red dashed line. The “cold” HTF (240°C) is pumped from the cold tank through the parabolic trough collector field where it is heated up (307°C), and passed into the hot oil tank. From there the HTF is pumped through the steam generator. Having passed through the steam generator, the HTF, now at a much lower temperature (240°C), is pumped back into the cold oil tank.

The secondary cycle, shown within the boundaries of the blue dashed line, is the steam cycle with steam turbine and generator for electricity production.

The storage system had a storage capacity of 115 MWth, equivalent to 2 hours of operation by storage, for example, after sunset [2].
On February 26, 1999 the storage burned down (see Figure 28). In addition to the failure of the safety valve, two other reasons for the incident were given: The basis of the HTF was petroleum which was easily inflammable and the tanks were charged with natural gas instead of an inert gas (nitrogen), due to financial reasons. This incident showed that mistakes were being made in the design and planning of the thermal energy storage [2].

In total, 9 SEGS plants (SEGS I – IX) were built which started operation between 1985 and 1991 and all are still in operation [22].
In this section storage media for sensible storage systems are presented in two tables, one table for solid storage media and another for liquid storage media. Following each of the tables, basic formulae are given for the calculation of the storage capacity \( Q \).

Solid storage media: Table 1 shows the available solid storage media for indirect storage of thermal energy, i.e. the thermal energy from a HTF (e.g. thermal oil, air) is transferred to a solid storage medium.

**Table 1: Solid storage media, edited from [21, p. 203]**

<table>
<thead>
<tr>
<th>Solid storage media</th>
<th>Temperature Cold °C</th>
<th>Temperature Hot °C</th>
<th>Average density (kg/m³)</th>
<th>Average heat conductivity (W/(mK))</th>
<th>Average heat capacity (kJ/(kgK))</th>
<th>Thermal diffusivity (m²/s)</th>
<th>Volume specific heat capacity (kWh/m³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sand-rock-mineral oil</td>
<td>200</td>
<td>300</td>
<td>1700</td>
<td>1.0</td>
<td>1.30</td>
<td>4.5×10^-7</td>
<td>60</td>
</tr>
<tr>
<td>Reinforced concrete</td>
<td>200</td>
<td>400</td>
<td>2200</td>
<td>1.5</td>
<td>0.85</td>
<td>8.0×10^-7</td>
<td>100</td>
</tr>
<tr>
<td>NaCl (solid)</td>
<td>200</td>
<td>500</td>
<td>2160</td>
<td>7.0</td>
<td>0.85</td>
<td>3.8×10^-6</td>
<td>150</td>
</tr>
<tr>
<td>Cast iron</td>
<td>200</td>
<td>400</td>
<td>7200</td>
<td>37.0</td>
<td>0.56</td>
<td>9.2×10^-6</td>
<td>160</td>
</tr>
<tr>
<td>Cast steel</td>
<td>200</td>
<td>700</td>
<td>7800</td>
<td>40.0</td>
<td>0.60</td>
<td>8.6×10^-6</td>
<td>450</td>
</tr>
<tr>
<td>Silica fire bricks</td>
<td>200</td>
<td>700</td>
<td>1820</td>
<td>1.5</td>
<td>1.00</td>
<td>8.2×10^-7</td>
<td>150</td>
</tr>
<tr>
<td>Magnesia fire bricks</td>
<td>200</td>
<td>1200</td>
<td>3000</td>
<td>5.0</td>
<td>1.15</td>
<td>1.4×10^-6</td>
<td>600</td>
</tr>
</tbody>
</table>

According to the properties of regenerators it would be necessary to include the actual storage material utilisation coefficient \( f_{\text{utilisation}} \) for the calculation of the true storage capacity. The factor depends on the heat conductivity of the storage medium and the operational mode of the storage [2].

The true storage capacity \( Q \) is hence calculated [2] with the equation

\[
Q = m \cdot f_{\text{utilisation}} \cdot c_{p,m} \left( \Delta \vartheta_{\text{min}/\text{max}} \right), \quad [\text{J}]
\]  

where \( m \) is the mass [kg], \( c_{p,m} \) is the mean heat capacity [J/(kgK)], and \( \Delta \vartheta_{\text{min}/\text{max}} \) is the temperature difference of the working fluid \( (\vartheta_{\text{max}} - \vartheta_{\text{min}}) \) [K].
For basic calculations of regenerators there are simplified analytical solutions which are described in literature [2].

- Important thermo-physical properties for storage materials:
  - *Specific heat capacity*: The higher the heat capacity, the less material is required and thus the lower the costs.
  - *Heat conductivity*: The higher the heat conductivity, the faster the material is heated, thus the better the material utilisation and hence the lower the material usage. Also, for fluid storage materials, high heat conductivity is important in enabling small heat exchangers to be sufficient.
  - *Temperature and cycle stability*: must be sufficiently high.
  - *Compatibility of materials/corrosivity*: “Aggressive” storage materials require high quality construction materials and thus negate a possible cost advantage.
  - *Inflammability/combustibility*: Should be as low as possible. The same applies to the toxicology of the storage material.

Liquid storage media:

Table 2 gives an overview of the current technically available liquid storage media [21]. The liquid storage media can also be used as HTF, i.e. the HTF is the same as the storage medium. For the storage the crucial factors are the possible operating temperatures and the achievable temperature ranges, the storage capacity in kWhₘₘₜ/m³ (influence of tank size and construction costs) and finally the storage media costs [2].

**Table 2:** Liquid storage media, edited from [21, p. 203]

<table>
<thead>
<tr>
<th>Liquid storage media</th>
<th>Temperature Cold (°C)</th>
<th>Temperature Hot (°C)</th>
<th>Average density (kg/m³)</th>
<th>Average heat conductivity (W/(mK))</th>
<th>Average heat capacity (kJ/(kgK))</th>
<th>Thermal diffusivity (m²/s)</th>
<th>Volume specific heat capacity (kWhₘₜ/m³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mineral oil</td>
<td>200</td>
<td>300</td>
<td>770</td>
<td>0.12</td>
<td>2.6</td>
<td>6.0×10⁻⁸</td>
<td>55</td>
</tr>
<tr>
<td>Synthetic oil</td>
<td>250</td>
<td>350</td>
<td>900</td>
<td>0.11</td>
<td>2.3</td>
<td>5.3×10⁻⁸</td>
<td>57</td>
</tr>
<tr>
<td>Silicone oil</td>
<td>300</td>
<td>400</td>
<td>900</td>
<td>0.10</td>
<td>2.1</td>
<td>5.3×10⁻⁸</td>
<td>52</td>
</tr>
<tr>
<td>Nitrate salts</td>
<td>250</td>
<td>450</td>
<td>1825</td>
<td>0.57</td>
<td>1.5</td>
<td>2.1×10⁻⁷</td>
<td>152</td>
</tr>
<tr>
<td>Nitrate salts</td>
<td>265</td>
<td>565</td>
<td>1870</td>
<td>0.52</td>
<td>1.6</td>
<td>1.7×10⁻⁷</td>
<td>250</td>
</tr>
<tr>
<td>Carbonate salts</td>
<td>450</td>
<td>850</td>
<td>2100</td>
<td>2.0</td>
<td>1.8</td>
<td>5.3×10⁻⁷</td>
<td>430</td>
</tr>
<tr>
<td>Liquid sodium</td>
<td>270</td>
<td>530</td>
<td>850</td>
<td>71.0</td>
<td>1.3</td>
<td>6.4×10⁻⁵</td>
<td>80</td>
</tr>
</tbody>
</table>

---

5 Definition of inflammability: Easily ignited and capable of burning rapidly; flammable.
Further information: The liquid sodium listed in the last row of Table 2 was also used as a HTF in the solar thermal power plant technology. Its high value of heat conductivity at still sufficiently good heat capacity leads to excellent heat transfer properties, which are very advantageous for the receiver design. Its fire hazard and the high costs, however, disqualify this medium as a viable alternative and therefore liquid sodium is no longer used in solar thermal power plants.

The storage capacity $Q$ is calculated with the following equation:

$$Q = m \cdot c_{p,m} (\Delta \theta_{\text{min/max}}), \quad [\text{J}]$$

(16)

where $m$ is the mass (kg), $c_{p,m}$ is the mean heat capacity [J/(kgK)], and $\Delta \theta_{\text{min/max}}$ is the temperature difference of the working fluid ($\theta_{\text{max}} - \theta_{\text{min}}$) [K].
9.1.2 Latent-Heat Storage System

When storing latent heat, the property of materials to absorb or release heat energy during a phase change is used. Latent-heat storage systems therefore use a phase change material (PCM) as storage medium. In principle there are three possible phase changes [2]:

- solid – solid
- solid – liquid
- liquid – vapour

The phase change solid – solid is seldom used. Here the phase change takes place by the change of the crystalline form [11, p. 155] of the solid PCM. Also the phase change liquid – vapour is seldom used. In today’s development of latent-heat storage systems, the phase change solid – liquid plays an important role, which will now be discussed further.

When charging the storage, the solid PCM is melted and when discharging, the fluid PCM becomes colder and solidifies [11, p. 155]. The melting and solidification process takes place either at a constant melting temperature or in a temperature range, depending on the substance/composite of the PCM. Achieving constant process temperatures is generally desired for the stable operation of a power plant.

For latent-heat storage systems used in medium and high temperature applications, materials are chosen whose melting temperatures lie within the temperature range of the process. Based on the mass or the volume of the storage medium, it is possible to store considerably more energy with such a phase change material than, for example, with concrete.

For applications in medium or high temperature ranges, nitrate and nitrite salt phase change materials in particular are considered. Several salts, e.g. hydroxide, partly have very high melting enthalpies, but are not considered because of high corrosivity properties or too high costs [2].

The heat $Q$ (in J) stored (i.e. storage capacity) in a latent-heat storage system is composed of three terms [11, p. 155]:

- $Q_{sol}$ (sensible heat): required to heat up the PCM from a solid state at temperature $\vartheta_{min}$ to the melting temperature $\vartheta_{melt}$
- $Q_{melt}$ (latent heat): required for melting the PCM
- $Q_{liq}$ (sensible heat): required for heating the liquid PCM to the storage temperature $\vartheta_{max}$
The aforementioned storage capacity $Q$ of a latent-heat storage system is calculated via the temperature increase, which the storage experiences during operation [2]:

$$Q = m \cdot f_{utilisation} \cdot \left( c_{p, sol, m} \cdot \Delta \vartheta_{min/melt} + \Delta h_{melt} + c_{p, liq, m} \cdot \Delta \vartheta_{melt/max} \right), \quad [J] \quad (17)$$

where

- $f_{utilisation}$ is the actual storage material utilisation coefficient,
- $m$ is the mass of the storage material [kg],
- $c_{p, m}$ is the mean heat capacity (note: subscript $m$ stands for mean, $sol$ for solid and $liq$ for liquid) [J/(kgK)],
- $\Delta \vartheta_{min/melt}$ is the temperature difference of the melting temperature and the minimum temperature of the phase change material in solid phase [K],
- $\Delta \vartheta_{melt/max}$ is the temperature difference of the maximum temperature and the melting temperature of the phase change material in liquid phase [K], and
- $\Delta h_{melt}$ is the specific melting enthalpy [J/kg]

Since the physical properties in the liquid phase are different than those in the solid phase, the equation must include two different specific heat capacities $c_p$ – the average heat capacity of the solid phase of the storage medium and the average heat capacity of the liquid phase of the storage medium. This allows the portions of the storable heat to be determined whilst the medium is in the two different phases. Each portion is calculated using the applicable temperature change which occurs. For the solid phase, the minimum temperature and the melting temperature are the boundaries, whilst for the liquid phase the melting temperature and the maximum storage temperature are the boundaries [2].

One problem with PCM is encountered in the discharging process. As the liquid PCM transfers the heat to the heat transfer fluid, which flows through a pipe, it crystallises (becomes solid). This, however, occurs at first on the pipe which carries the heat transfer fluid because there the
Temperature is at the lowest in the entire storage (especially at the heat transfer fluid inlet). The solidification, which progresses concentrically away from the pipe, has the negative effect that is lowers the heat transfer coefficient $\lambda$ significantly [11, p. 156]. One approach to overcome this problem, the finned-tube design, was developed by the DLR as portrayed in Figure 30 [1]. The greater the heat transfer coefficient of a PCM in solid state is, the better the heat transfer will be [11, p. 156]. With the finned-tube design, effective heat transfer coefficient values of $\lambda > 10$ W/(mK) can be achieved [1].

Figure 30: Discharging of a latent-heat storage - heat passes from PCM to heat transfer fluid (left), Finned tube design (right) [1]

Latent-heat storage systems can be useful for both solar power generation and industrial process heat generation. The melting enthalpy [J/g] and melting temperature [°C] of several phase change materials is shown in Figure 31. For today’s application the common temperature range of PCM is 200 – 350°C.
Phase change storage media: Table 3 and Table 4 contain several phase change storage media and their properties for use in latent-heat storage systems.

**Table 3:** Possible heat transfer media for latent-heat storage systems, edited from [21, p. 203]

<table>
<thead>
<tr>
<th>Phase change storage media</th>
<th>Melting Temperature T_m (°C)</th>
<th>Cold Hot (°C)</th>
<th>Average density (kg/m³)</th>
<th>Average heat conductivity (W/(mK))</th>
<th>Average heat capacity (kJ/(kgK))</th>
<th>Thermal diffusivity (m²/s)</th>
<th>Volume specific heat capacity (kWh/m³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>NaNO₃</td>
<td>306</td>
<td>2257</td>
<td>0.5</td>
<td>200</td>
<td>1.1×10⁻⁹</td>
<td>125</td>
<td></td>
</tr>
<tr>
<td>KNO₃</td>
<td>334</td>
<td>2110</td>
<td>0.5</td>
<td>267</td>
<td>8.9×10⁻¹⁰</td>
<td>156</td>
<td></td>
</tr>
<tr>
<td>KOH</td>
<td>380</td>
<td>2044</td>
<td>0.5</td>
<td>150</td>
<td>1.6×10⁻⁹</td>
<td>85</td>
<td></td>
</tr>
<tr>
<td>Salt-ceramics (Na₂CO₃-BaCO₃-MgO)</td>
<td>500-850</td>
<td>2600</td>
<td>5.0</td>
<td>420</td>
<td>4.6×10⁻⁹</td>
<td>300</td>
<td></td>
</tr>
<tr>
<td>NaCl</td>
<td>802</td>
<td>2160</td>
<td>5.0</td>
<td>520</td>
<td>4.5×10⁻⁹</td>
<td>280</td>
<td></td>
</tr>
<tr>
<td>Na₂CO₃</td>
<td>854</td>
<td>2533</td>
<td>2.0</td>
<td>276</td>
<td>2.9×10⁻⁹</td>
<td>194</td>
<td></td>
</tr>
<tr>
<td>K₂CO₃</td>
<td>897</td>
<td>2290</td>
<td>2.0</td>
<td>236</td>
<td>3.7×10⁻⁹</td>
<td>150</td>
<td></td>
</tr>
</tbody>
</table>

*Figure 31: Applicability of PCM for industrial process heat and for solar power generation [1]*
Table 4: PCMs in the temperature range 200 – 350°C with work reported on material examinations, edited from [23]

<table>
<thead>
<tr>
<th>Phase change storage media</th>
<th>Melting temperature $T_m$ (°C)</th>
<th>Enthalpy $H$ (J/g)</th>
</tr>
</thead>
<tbody>
<tr>
<td>LiOH(20wt%)-NaOH(80wt%)</td>
<td>215</td>
<td>280</td>
</tr>
<tr>
<td>LiNO$_3$</td>
<td>252</td>
<td>380-530</td>
</tr>
<tr>
<td>KNO$_3$(54wt%)-NaNO$_3$(46wt%)</td>
<td>222</td>
<td>100</td>
</tr>
<tr>
<td>Ca(NO$_3$)$_2$(45wt%)-NaNO$_3$(55wt%)</td>
<td>230</td>
<td>~110</td>
</tr>
<tr>
<td>KCl-ZnCl$_2$ #</td>
<td>230-430</td>
<td>200-220</td>
</tr>
<tr>
<td>NaNO$_2$-NaOH #</td>
<td>232-265</td>
<td>250-300</td>
</tr>
<tr>
<td>LiCl-LiOH #</td>
<td>262</td>
<td>485</td>
</tr>
<tr>
<td>NaOH-NaNO$_3$ #</td>
<td>247-292</td>
<td>240-300</td>
</tr>
<tr>
<td>NaNO$_3$</td>
<td>306</td>
<td>178</td>
</tr>
<tr>
<td>KNO$_3$(94wt%)-KCl(6wt%)</td>
<td>~320</td>
<td>~80</td>
</tr>
<tr>
<td>KNO$_3$</td>
<td>334</td>
<td>98</td>
</tr>
</tbody>
</table>

Note: * Complex phase diagram, PCMs with different compositions feasible.

The most recent developments in the latent-heat storage technology are now described in more detail:

Developments in direct steam generating solar thermal power plants have shown that the cost of power generation may be reduced if steam as HTF is used directly. CSP Technologies that can employ direct steam generation (DSG) are solar towers, parabolic trough power plants and Fresnel collectors. Extensive work is done in this field, including the further development of the storage technology. The demand for 2-phase water/steam storage systems is increasing rapidly, but apart from steam accumulators no such storage technology has reached a commercial status. Development by the DLR and Ed. Züblin AG has lead to a proposal of a three-part storage system, designed to meet the demand of storage of the CSP technologies using water/steam as HTF. This storage system consists of a concrete storage (developed by DLR and Ed. Züblin AG), which is required for the transfer of sensible heat for one-phase preheating of water as well as for superheating of steam, and a latent-heat storage (PCM-storage) which is required for the two-phase evaporation [24], as shown in Figure 34.
The difference in a single thermal energy storage system and a combined sensible and PCM storage system is shown by their characteristics in the temperature-enthalpy graphs of Figure 33. The combined sensible and PCM storage is very efficient as during the storage the temperature differences between the fluid and the storage material is small. In the future, direct steam generating parabolic trough power plants, for example, shall be designed to handle steam pressures of 100 bar. At around this pressure level nearly 70% of the energy is needed for the evaporation/condensation of the heat transfer fluid, hence the major capacity is stored in the PCM storage [25].

As the original design of the concrete storage is based on the use of thermal oil as HTF, efforts are being made to adapt it to the requirements of the HTF water/steam and increased operating temperatures of 500°C. The design and construction of a superheating storage test module was the next step. Still, this test module is limited to operating temperature of 400°C, but high temperature concrete, which can handle temperatures of 500°C, has been developed along the way. The PCM-
storage employs sodium nitrate as phase change material, which has a melting temperature of 305°C. In the charging process the steam pressure is 112 bar and during the discharging process 78 bar. After a successful demonstration of the concept at a laboratory of DLR, the system was then scaled up to about 14 tonnes of salt, having a latent heat capacity of about 700 kWh. It is worth mentioning that the PCM-storage is not filled with salt in a molten state. Instead, fine to medium grained solid salt is simply filled in directly. The melting is done solely by the PCM-storage. Hence there is no need for an external melting unit [24].

Demonstration: In 2009, a demo-storage system (Figure 34) with concrete superheater and PCM-storage in the water/steam test loop was built at the power plant Litoral of Endesa in Carboneras, Spain. Having a total storage capacity of around 1000 kWh, this high temperature storage system is the first storage of its kind that was built to demonstration scale. The volume of the concrete storage is 22 m$^3$ and the PCM-storage has a volume of 8.5 m$^3$. The maximum internal pressure of the fluid water/steam is 128 bar [24].

![Figure 34: Demo-storage system with concrete superheater (left) and PCM-storage (right) in the water/steam test loop at the power plant Litoral of Endesa in Carboneras, Spain [24]](image_url)
9.1.3 Steam Accumulator

Steam accumulators are pressure vessels in which a charging system feeds steam in the hot water (well distributed). Through the input of heat of condensation, the temperature of the hot water in the storage vessel rises. Steam accumulators utilise the ecologically most worthwhile and cheapest HTF available. The necessary pressure vessels, however, dominate the investment costs.

Although listed as a separate type of thermal energy storage system, the steam accumulator (see Figure 35) can also be categorised as a special form of direct storage system. The following must be distinguished here: If the HTF of the primary circuit is steam, and a steam accumulator is being used, then the steam would be directly stored. In this case, the steam accumulator could be regarded as a direct storage system. If, on the other hand, the HTF of the primary circuit is not steam, then the steam accumulator could – by option – be placed into the steam cycle. This, however, no longer fulfils the criteria of a direct storage system, nor, strictly speaking, those of an indirect storage system.

Depending on the operating method / discharge characteristics, one distinguishes between two accumulator types that find use in power plants:

1. In sliding pressure systems (Ruth's storage), saturated steam is withdrawn during discharging. As a result, the pressure in the vessel drops. The drop in pressure also results in the lowering of the vaporisation point. Hence, when the pressure is reduced, there is a transition of hot water into the steam phase and a reduction of the hot water temperature.
Over the duration of discharge, the pressure and temperature of the extracted steam is continuously reduced.

When steam is supplied to the steam turbine it must be dry, i.e. superheated, (except in nuclear power plants where special turbines are used) in order to ensure a longer turbine operating life. Sliding pressure systems only deliver saturated steam, which means that further steps must be undertaken to ensure the steam is dry. The two common methods are:

- to further expand the saturated steam, as lowering the pressure leads to a lowering of the vaporisation point and hence the steam will become dryer,
- to pass the steam through a superheater, which is supplied with heat from an extra thermal storage for superheating.

Since the process parameters of the extracted steam steadily sink during discharging of the accumulator, the steam is first supplied to the medium-pressure turbine and later to the low-pressure turbine. During charging, the temperature and pressure in the storage rise. Accordingly, the charging process should also be carried out sequentially, at first with steam from the low-pressure turbine and then with live steam [2].

The PS10 solar tower in Seville, Spain, uses such a steam storage system. The general concept of the plant is illustrated in Figure 36. The steam storage system installed on the PS10 site is shown in Figure 37 [1]:

![Figure 36: Schematic of the PS10 solar tower, edited from [1]](image-url)
2. In the displacement accumulator (developed by Marguerre) hot water is withdrawn from the top of the vessel for the duration of discharging and supplied to the evaporator. Simultaneously, cold feedwater flows in at the bottom of the vessel. Due to the regulation of the outflow and inflow, the pressure in the vessel remains constant. At the same time, the steam extraction from the turbine which is used for preheating the feedwater is halted, so that all the live steam can be reduced in the turbine. A power reduction of the evaporator can thus be counterbalanced and the turbine continues to run constantly. A failure of the evaporator, however, cannot be compensated for with this storage type. Therefore this type of storage is unsuitable as an energy storage system for a solar thermal power plant.

Steam accumulators are nowadays used in industrial facilities, but became obsolete for electrical power generation because of the power grid. In Germany, one of the last applications was in the capital city Berlin [2].

9.1.4 Thermo-Chemical Energy Storage

In thermo-chemical energy storage systems, reversible reactions are used to store energy, which are endothermic when charging and exothermic when discharging. The useful specific reaction enthalpy of such reactions is merely one order of magnitude smaller than that in the combustion of fossil fuels and hence considerably larger than that in the storage of sensible or latent heat [2].

However, this technology is currently at a very early stage. In the long term, though, this storage system could offer relatively low costs and offer the prospect of systems with inherent energy storage for continuous (24 h, round-the-clock) generation of electricity [26].

Ideally, the reaction products can be easily separated and no further reactions take place [9, p. 404].

**Figure 37: PS10’s steam storing system [1]**
Certain reaction steps of catalytic reactions, as for example the back reaction, can only be reached by the addition of a catalyst and not through the change of the chemical balance. This is why after the reaction the products can be stored side by side without them undergoing a reverse reaction. If the reacting substances and the catalyst form a homogeneous mixture, i.e. both are either gaseous or dissolved in a fluid then this is referred to a homogeneous catalysis. In a heterogeneous catalysis (surface catalysis) the reacting substances are gaseous or dissolved in a fluid while the catalyst exists in a solid form [2].

In the thermal dissociation the starting substance AB exists solid or liquid. Through the supply of the molar reaction enthalpy at temperature $T_1$ and a corresponding pressure $p_1$, the chemical bond of the substance pair AB is removed. The product A then also exists solid or liquid while B is gaseous. This reaction is also referred to as heterogeneous evaporation. The reversal of the process – this is the absorption – occurs when the balance is disrupted by a temperature reduction or a pressure increase. Here, B is the absorptive, A is the absorbent and AB is the absorbate. In the process the employed reaction enthalpy of the dissociation is released. For the storage of energy the products A and B must be stored separately [2].

The basic principle of the storage technology with adsorption processes is the ability of highly porous bodies like charcoal, pumice stone, silica gel or zeolith to retain gases or vapours in great quantities over their active surfaces. This leads to a reaction with corresponding reaction enthalpy. Moreover, a so-called capillary condensation occurs. In moistened capillaries, under the influence of the surface forces, the vapour pressure compared to the value at the free surfaces is reduced. If vapours come into contact with porous substances, then a lowering of vapour pressure occurs such that the vapours condense at a pressure which lies below the saturated vapour pressure of the vapour. The operating principle of adsorptive storage types begins with the drying of the adsorbent, e.g. by using air with a low relative moisture. This is how the storage unit is charged. The storage unit is discharged by means of introducing cold and moist air. The moist air passes the moistures to the hygroscopic adsorbent, whereby bond energy is released, which contributes to the heating of the air. Hence, the dry storage condition corresponds to stored energy, which, when hermetically sealing the storage material, is practically storable for an unlimited time [2].

Advantages and disadvantages of this storage system [2], [26], [11, p. 171]:

**Advantages**
- after separating the reaction products the energy can practically be stored loss-free over a longer period of time
- high storage energy densities
- heat-pumping capability

**Disadvantages**
- thermo-chemical energy storage systems are more complex than conventional sensible energy or latent-heat storage systems
- the storage material costs are very high

Table 5 shows different media used in sensible energy storage, latent-heat storage and thermo-chemical energy storage systems. The mass $m$ and volume $V$ is given for each storage material and represent the mass and volume required for a storage capacity $Q_s = 1$ MWh = 3.6 GJ at a temperature difference of 20K [11, p. 172]. On the example of NH$_4$Br, the advantage of high storage energy densities which thermo-chemical energy storage systems have over other
technologies can be emphasised. Moreover, thermo-chemical energy storage materials have very low masses and volumes compared to sensible energy storage materials.

Table 5: Energy density $q$ and the required mass $m$ and volume $V$ of the storage material for sensible energy storage, latent-heat storage and thermo-chemical energy storage systems. Storage capacity $Q_s = 1 \text{ MWh} = 3.6 \text{ GJ}$. Temperature difference $\Delta T = 20$ [11, p. 172]

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Sensible Energy Storage</th>
<th>Latent-Heat Storage</th>
<th>Thermo-Chemical Energy Storage</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Water</td>
<td>Rock</td>
<td>Concrete</td>
</tr>
<tr>
<td>Heat capacity $c$ (kJ/(kgK))</td>
<td>4.19</td>
<td>0.84</td>
<td>1.1</td>
</tr>
<tr>
<td>Density $\rho$ (kg/m$^3$)</td>
<td>1000</td>
<td>1600</td>
<td>2240</td>
</tr>
<tr>
<td>Enthalpy $h$ (kJ/kg)</td>
<td>333.5</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Energy density:</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>$q$ (kJ/kg)</td>
<td>84</td>
<td>17</td>
<td>22</td>
</tr>
<tr>
<td>$q_r$ (MJ/m$^3$)</td>
<td>84</td>
<td>27</td>
<td>49</td>
</tr>
<tr>
<td>Mass $m$ (10$^3$ kg)</td>
<td>42.8</td>
<td>211.7</td>
<td>163.4</td>
</tr>
<tr>
<td>Relative mass</td>
<td>1</td>
<td>4.9</td>
<td>3.8</td>
</tr>
<tr>
<td>Volume in m$^3$</td>
<td>42.8</td>
<td>203.6*</td>
<td>73.4</td>
</tr>
<tr>
<td>Relative volume</td>
<td>1</td>
<td>4.76</td>
<td>1.7</td>
</tr>
</tbody>
</table>

* rock bulk with an ullage of 35%

How important the thermo-chemical energy storage can become for CSP plants in the future can be explained by the example of a heat pipe, as shown in Figure 38. The heat pipe is a system which could one day be used for the storage and also for the long-distance transfer of solar energy. Hence, the heat pipe constitutes a very interesting option for transporting energy from e.g. North Africa or the Mediterranean area to Central Europe. The heat pipe can be used, for example, in the solar tower technology. The solar tower technology has a central receiver which absorbs solar radiation and heats up to very high temperatures. The heat from the receiver can be used for a catalytic reforming process (this is the so-called Eva process) in the reformer of the heat pipe. In the reformer, methane reacts with water in an endothermic reaction at a temperature of 960°C (heat is provided by solar tower). The reaction is as follows:

$$\text{CH}_4 + \text{H}_2\text{O} = \text{CO} + 3\text{H}_2 - 6020 \text{ kJ per kg CH}_4$$  \(18\)

The produced gas mixture (CO + 3H$_2$) is a synthetic or energy carrier gas. The gas can be transported in a pipeline. In a further process, methane can be produced from the gas mixture in a so-called methanator (the methanator uses the so-called Adam process). In the methanator, heat is released, which, having a temperature of 500 – 700°C, is sufficient for the generation of steam. As a catalyst for both reactors Rhodium or Nickel can be used [11, pp. 172-173].
The thermo-chemical energy transport can be realised in subterranean pipelines. The advantages of the pipeline transportation are practically non-existing energy losses and the high energy density of the storage material [11. p. 173].
9.2 Solar Multiple

The solar multiple $SM$ is defined as “the ratio of the receiver power at design point $P_{th,rec}$, to the nominal cycle inlet power $P_{th,cycle}$” [21, p. 200].

$$SM = \frac{P_{th,rec}}{P_{th,cycle}}$$

(19)

This term shall be clarified with an example: The solar power tower plant Solar Two near Barstow, California, USA, was designed with a solar multiple of 1.2 [27], meaning that the heliostat field delivers a 20% greater thermal power capacity than is required to generate the nominal power for the design day. The 20% excess thermal energy can be delivered to the thermal energy storage.

Solar technologies operating without thermal storages are limited to annual capacity factors near 25%, meaning the plant can operate a maximum of 2200 hours per year in solar-only\(^6\) operating mode. If on the other hand a molten-salt solar power tower located in the California Mojave desert is designed with a very large solar field that has a solar multiple of 2.7 and a thermal storage, then the annual capacity factor could be raised to approximately 65%, according to simulations at the Sandia National Laboratories (USA) [28].

Figure 39 illustrates the process of storage charging, storing and discharging over the course of a day, with respect to the powers $P_{th,rec}$ and $P_{th,cycle}$.

\(^6\) Solar-only means that only solar irradiation as energy source is used.
There are several good reasons why it is important to design a solar field which has a solar multiple SM > 1.0 and a thermal energy storage system to store the excess energy [2]:

- An increase of the annual solar hours at nominal power at correspondingly enlarged solar multiple has the following effects:
  - better financial utilisation of the power plant block-unit
  - compliance with the maximum allowable fossil co-firing
  - useful in places where energy from fossil fuels is expensive (e.g. islands)
- Increase of the annual hours of nominal solar load operation
- Shifting of the solar electricity generation from day into night hours, if higher revenues can be obtained at night
- Buffering during times when clouds block the direct solar irradiation
- Homogenising of plant operation when in solar-only mode
- Preheating of the power plant when daily start-up necessary
- Anti-freeze measure for primary (water-steam) cycle on cold nights (synthetic heat transfer oils already freeze at +12°C)

The left graph of Figure 40 illustrates the process of storage charging, storing and discharging over the course of a day, with respect to the solar multiple SM for Summer solstice. The right graph shows corresponding curves for winter solstice.

![Figure 40: Process of storage charging, storing and discharging over the course of a day, edited from [29, p. 641], [2]](image)

Description of Figure 40 [2]:
The blue line (SM = 1.0) in the left diagram represents the progression of the thermal power of a solar field over the course of the day, which was dimensioned to exactly the size required, so that at
noon (12 o’clock) local time on June 21, the nominal power of the plant is reached. In the right-
hand diagram, it is shown that the solar field can only supply approximately 60% of the nominal
thermal power on the day of the winter solstice (December 21) at noon (12 o’clock) local time.
In order to increase the solar fraction, one can enlarge the solar field so that the progression of the
power output of the solar field is changed according to the black line (labelled SM > 0) for June 21.
The progression of the black line was adapted such that on winter solstice (December 21) at noon
(12 o’clock) local time the thermal nominal thermal power of the plant was reached.
With reference to the time axis for June 21 the power plant can be operated with solar-only input at
nominal power from ~7 am until ~5 pm. The excess energy of the solar field (yellow area in left-
hand diagram) can be controlled by defocusing a portion of the mirrors. If the solar thermal power
plant is equipped with a thermal energy storage system, then the excess energy can be diverted into
the thermal energy storage. The thermal energy in the storage can then be utilised and discharged,
for example, in the hours after sunset (red area).
The thermal energy discharged (red area) is less than the energy stored (yellow area), due to the
energy losses in the charging, storing (for some hours) and discharging process of the storage.

The discharge energy $E_{dis}$ is calculated via the charge energy $E_{cha}$ and the storage loss factor
$\eta_{cha,store,dis}$, which contains the energy losses in the charging, storing and discharging process:

$$E_{dis} = E_{cha} \cdot \eta_{cha,store,dis} \quad [J] \quad (20)$$
9.3 Storage Capacity

The graph below (Figure 41) shows the number of attainable hours at nominal solar power per year in dependence of the solar field size (the solar multiple) and the storage size. The solar multiple is based on the equinox (March 21). This example is for the parabolic trough power plant SEGS VIII, which has a nominal electrical output of \( P_e = 80 \text{ MW}_e \). The annual direct solar irradiation is given as 2500 kWh/m² [2].

![Figure 41: Hours at nominal power per year in dependence of the solar multiple and storage size, edited from [29, p. 642]](image)

*Description of Figure 41 [2]:*

The diagram shows the attainable number of hours at nominal power plotted against the solar multiple SM (i.e. field size). Without the use of a thermal energy storage system the doubling of the solar field size from SM = 1 to SM = 2 leads to a gain of additional 300 solar annual hours at nominal power – hence between 2100 h to 2400 h can be reached. For a SM = 1.0, the use of a thermal energy storage system with a storage capacity of 1 h yields a gain of annual hours at nominal power from approximately 2100 h to 2200 h. The deployment of a thermal energy storage system with a capacity of 6 h does not result in a further gain. Hence, it becomes clear that the capacity of the thermal energy storage system and the solar multiple must be matched with one another.

The attainable annual hours at nominal power are required for determining the capacity factor of a power plant. The capacity factor \( CF \) is the ratio of the hours of solar rated output power operation to the total hours per year (8760 h).

*Note: A solar thermal power plant does not always reach rated power due to the daily and annual irradiation cycles, but is very often operated in part load. As a result, “virtual” hours at rated output power were introduced, by relating the plant's total annual electrical work to the annual rated output power. These equations are given below [2]:*
Hours at nominal solar electrical power per annum, $t_{p,e,nom,a} = \frac{W_{e,a}}{P_{e,out}}$ \hspace{1cm} (21)

with $W_{e,a}$: total annual electrical work; $P_{e,out}$: rated output power.

Capacity factor: $CF = \frac{t_{p,e,nom,a}}{y}$ \hspace{1cm} (22)

with $CF$: electrical capacity factor; $y$: hours per year (8760h).

The capacity factor correlates directly with the SM, the capacity of the thermal energy storage system and the annual irradiation at the location of the plant.

In the discussion about the capacity factor and hours at rated output power it is important to always clearly specify whether one is referring to hours at nominal solar power or the total hours at nominal power (i.e. including fossil co-firing).

The following graph (Figure 42) shows the electricity generation costs (cost per kWh\textsubscript{e}) in dependence of the solar multiple and the storage size, which are to be compared to the targeted revenues for the parabolic trough power plant SEGS VIII, which has a rated power output of $P_e = 80$ MW\textsubscript{e}. The annual direct solar irradiation is given as 2500 kWh/m\textsuperscript{2}. The solar multiple is based on the equinox (March 21) [2].

![Figure 42](image_url)

**Figure 42**: Electricity generation costs (cost per kWh\textsubscript{e}) as a function of the solar multiple and the storage size, edited from [2]

**Questions to be considered:**

- How can the dimension of the thermal energy storage system be matched to the solar multiple?
- What is the target quantity?
Description of Figure 42 [2]:
A target quantity could be the energetic plant efficiency, averaged over the year. This is the amount of annually generated solar electricity energy from the plant in proportion to the annual solar irradiation energy that is incident on the entire solar field.

To find answers to the questions stated above, simulation programmes are used which perform time-discrete computations of the power flows in solar tower power plants and thus allow a consideration of the sum of these power flows for the entire year.

For solar thermal power plants, especially with regard to thermal energy storage systems, the ultimate target is to eventually keep the cost of solar generated electricity as low as possible.

For a solar thermal power plant with a nominal power of 80 MW\textsubscript{e}, the investment and operating costs for the power station block-unit are fixed. A very small solar field leads to few hours at nominal solar power per year and therefore to higher allocated electricity generation costs, which are given in the diagram as €/kWh\textsubscript{e}. An increase in the solar multiple leads to more hours at nominal power, consequently to a significantly better utilisation of the entire plant and to lower electricity generation costs. A further increase of the solar multiple above 1.0 without the integration of a thermal energy storage system, however, barely increases the hours at nominal power and causes the electricity generation costs to increase again. Because of this, the minimum in the curve progression occurs.

The use of a thermal energy storage system, however, leads to a better utilisation of a larger solar field, hence to more hours at nominal solar power and to even lower electricity generation costs as the power plant block-unit is better utilised.

The conditions for such calculations are that specific costs for the solar field and the thermal energy storage can be adhered to.

In reality the electricity generation costs must be compared to the respective possible revenue when designing a power plant.
9.4 Modular Storage

Until now, thermal storage systems in CSP plants are used that consist of one large storage body. The most important examples are the large molten salt storages that were used first in the Andasol power plants. We can call this type of storage “monolithic storage”. The following figure shows such a system.

An alternative concept is the modular storage. The characteristic difference to monolithic storage systems is that the storage capacity in a modular storage system is subdivided into several storage modules. The subdivision allows an independent charging of the different modules as well as an independent discharging. Modular systems have the following important advantages: They allow a better storage utilization (higher temperature spreads) (a), and they allow a reduction of exergy losses in the discharging process and can be adapted to different temperature levels if different temperature levels are needed simultaneously (b).

a) The following figure shows how a modular storage can make a better usage of the installed storage. In the left system (the monolithic system), the temperature at the outlet of the storage and the inlet of the solar field is limited to 315°C. If the temperature was higher, the thermo oil would acquire higher temperatures than 400°C in the solar field, which would affect its chemical structure and finally destroy it. This limits the average temperature of the storage and, hence, its capacity.

If the maximum temperature at the outlet of the storage and the inlet of the solar field got higher, then the average temperature of the storage would increase and more useable thermal energy could be stored. This is achieved in the modular system (right), where the storage is subdivided into two modules, which are charged separately from each other. Now, and with
shorter collector lengths, the maximum temperature at the outlet of the storage can be increased and the average storage temperature raises, which enhances the storage capacity\(^7\).

\[\text{Figure 44: Maximum temperature levels in a monolithic storage (left) and a modular storage (right) [Laing et al. (2006)], edited from [31]}\]

b) Real heat transfer processes imply exergy losses. This is the case because of temperature difference between the warmer body, which releases the heat, and the colder body, which absorbs the heat. Even if no thermal energy gets lost in the process, the transferred energy is “devaluated” because the change from a higher temperature to a lower one makes it lose a part of its ability to be converted into mechanical work (which is needed to drive the electric generator). The exergy loss depends on the temperature difference between the warm body and cold body. In order to minimize exergy losses, the temperature difference must be maintained low.

A modular storage allows a reduction of exergy losses. This is possible in quite a practical way if each of the processes preheating, evaporation and superheating gets its own storage module that is adapted to the specific characteristics of these processes. The following figure shows the scheme of such a system. In this case, each storage module can be discharged separately.

\(^7\) See Laing et al. (2006) [31]
By means of an appropriate adaptation of the storage modules to the preheating, evaporation and superheating, the temperature differences in the heat transfer can be maintained lower than with a monolithic storage. The following figure illustrates this.

The usage of a latent heat storage module for the evaporation stage could reduce the exergy loss even more, because the temperature differences in the evaporation section could be reduced even more.
Indeed, modular storage systems are very appropriate for CSP plants, because the steam generation itself is, so to say, a modular process. It consists of the three stages preheating, evaporation and superheating, which are realized at their respective temperature levels or within their respective temperature ranges. In typical currently realized systems the following values hold: Evaporation takes place at 311°C (at 100 bar), preheating between, say, 80°C and 311°C and superheating between 311°C and the maximum HTF temperature, which is normally 393°C. A modular storage is especially appropriate for the steam generation process because for each process stage a storage module can be designed that corresponds to its thermal characteristics.

The following diagram shows the heat consumption share of the three steam generation stages and their respective temperature level.

Modular storage systems could be used for direct steam generation (DSG) systems. Until now, there are no large thermal storage systems for DSG systems. Currently, research and development efforts are done to provide an appropriate storage system. The most promising system is a three

---

stage modular storage with high temperature concrete storage modules (sensible heat) for preheating and superheating and a phase change material (PCM) storage module (latent heat) in between for isothermal evaporation. An isothermal storage module for evaporation allows charging (PCM freezing and water condensation) and discharging (PCM melting and steam condensation) at nearly constant pressure.

At discharging, feed water is heated up close to the evaporation point in the preheater storage module (sensible heat storage). Evaporation takes place in the PCM module with a connected steam separator and a recirculation pump. Superheating takes place in a sensible heat storage module again. At storage charging, the steam/water flows in the opposite direction: The superheated steam is cooled down in the superheater storage, condensed in the PCM module and the resulting water is cooled down in the preheater.
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a GREA Innovacio´ Concurrent, Universitat de Lleida, Pere de Cabrera s/n, 25001-Lleida, Spain
b Instituto de Investigación en Ingeniería de Aragón, I3A, Grupo de Ingeniería Térmica y Sistemas Energéticos (GITSE), Dpto. Ingeniería Mecánica, Área de Máquinas y Motores Térmicos, Universidad de Zaragoza, Campus Politécnico Río Ebro, Edificio “Agustín de Betancourt”, María de Luna s/n, 50018 Zaragoza, Spain
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**Exercises**

**Problem 1:**
Calculate the required storage medium volume for the following thermal energy storage systems in order to store 1 MWh of thermal energy for the given temperature range.

Hint: The mean heat capacity of the thermal energy storage systems can be calculated using the equations given below:

\begin{align*}
(1) \quad Q &= m \cdot c_{p,m} \cdot \Delta \theta_{\text{min/max}} \quad \text{(direct storage for sensible heat)} \\
(2) \quad Q &= m \cdot \dot{f}_{\text{utilisation}} \cdot c_{p,m} \cdot \Delta \theta_{\text{min/max}} \quad \text{(indirect storage for sensible heat)} \\
(3) \quad Q &= m \cdot \dot{f}_{\text{utilisation}} \left( c_{p,\text{sol,m}} \cdot \Delta \theta_{\text{min/melt}} + \Delta h_{\text{melt}} + c_{p,\text{liq,m}} \cdot \Delta \theta_{\text{melt/max}} \right) \quad \text{(latent-heat storage)}
\end{align*}

Note: Subscript *m* stands for *mean*, *sol* for *solid* and *liq* for *liquid.*
Direct storage for sensible heat

1. with salt nitrate (plant: Solar Two)

Given:
- Temperature range: 450 – 550°C
- \( Q = 1000 \text{ kWh} \)
- \( c_{p,m} = 1.6 \text{ kJ/(kgK)} \)
- \( \rho = 1870 \text{ kg/m}^3 \)
- \( \Delta \vartheta_{\text{min/max}} = 100K \)

Solution:
\[
m = \frac{Q}{c_{p,m} \cdot \Delta \vartheta} = 22500 \text{ kg}
\]
\[
V = \frac{m}{\rho} = 12.03 \text{ m}^3
\]

2. with synthetic oil (plant: SEGS)

Given:
- Temperature range: 250 – 350°C
- \( Q = 1000 \text{ kWh} \)
- \( c_{p,m} = 2.3 \text{ kJ/(kgK)} \)
- \( \rho = 900 \text{ kg/m}^3 \)
- \( \Delta \vartheta_{\text{min/max}} = 100K \)

Solution:
\[
m = \frac{Q}{c_{p,m} \cdot \Delta \vartheta} = 15652 \text{ kg}
\]
\[
V = \frac{m}{\rho} = 17.39 \text{ m}^3
\]

Indirect storage for sensible heat

3. with concrete

Given:
- Temperature range: 300 – 400°C
- \( Q = 1000 \text{ kWh} \)
- \( c_{p,m} = 0.85 \text{ kJ/(kgK)} \)
- \( \rho = 2200 \text{ kg/m}^3 \)
- \( \Delta \vartheta_{\text{min/max}} = 100K \)
- \( f_{\text{utilisation}} = 0.85 \)

Solution:
\[
m = \frac{Q}{f_{\text{utilisation}} \cdot c_{p,m} \cdot \Delta \vartheta} = 49827 \text{ kg}
\]
\[
V = \frac{m}{\rho} = 22.65 \text{ m}^3
\]

4. with sand

Given:
- Temperature range: 600 – 700°C
- \( Q = 1000 \text{ kWh} \)
- \( c_{p,m} = 0.90 \text{ kJ/(kgK)} \)
- \( \rho = 1500 \text{ kg/m}^3 \)
- \( \Delta \vartheta_{\text{min/max}} = 100K \)
- \( f_{\text{utilisation}} = 0.85 \)

Solution:
\[
m = \frac{Q}{f_{\text{utilisation}} \cdot c_{p,m} \cdot \Delta \vartheta} = 47059 \text{ kg}
\]
\[
V = \frac{m}{\rho} = 31.37 \text{ m}^3
\]
Latent-heat storage

5. with water

Given:
- Temperature range: -10 – 90°C (phase change at 0°C)
- \( Q = 1000 \text{ kWh} \)
- \( c_{p,\text{sol},m} = 2.40 \text{ kJ/(kgK)} \)
- \( c_{p,\text{liq},m} = 4.18 \text{ kJ/(kgK)} \)
- \( \rho = 1000 \text{ kg/m}^3 \)
- \( \Delta \theta_{\text{min/melt}} = 10\text{K} \)
- \( \Delta \theta_{\text{melt/max}} = 90\text{K} \)
- \( f_{\text{utilisation}} = 0.95 \)
- \( \Delta h_{\text{melt}} = 333 \text{ kJ/kg} \)

Solution:

\[
m = \frac{Q}{f_{\text{utilisation}} \cdot \left( c_{p,\text{sol},m} \cdot \Delta \theta_{\text{min/melt}} + c_{p,\text{liq},m} \cdot \Delta \theta_{\text{melt/max}} \right)} = 5168 \text{ kg}
\]

\[
V = \frac{m}{\rho} = 5.17 \text{ m}^3
\]

6. with salt mix

Given:
- Temperature range: 210 – 310°C (phase change at 220°C)
- \( Q = 1000 \text{ kWh} \)
- \( c_{p,\text{sol},m} = 0.95 \text{ kJ/(kgK)} \)
- \( c_{p,\text{liq},m} = 1.36 \text{ kJ/(kgK)} \)
- \( \rho = 1790 \text{ kg/m}^3 \)
- \( \Delta \theta_{\text{min/melt}} = 10\text{K} \)
- \( \Delta \theta_{\text{melt/max}} = 90\text{K} \)
- \( f_{\text{utilisation}} = 0.95 \)
- \( \Delta h_{\text{melt}} = 100 \text{ kJ/kg} \)

Salt mix: 60% KNO\(_3\) + 40% NaNO\(_3\)
(KNO\(_3\): potassium nitrate, NaNO\(_3\): sodium nitrate)

Solution:

\[
m = \frac{Q}{f_{\text{utilisation}} \cdot \left( c_{p,\text{sol},m} \cdot \Delta \theta_{\text{min/melt}} + c_{p,\text{liq},m} \cdot \Delta \theta_{\text{melt/max}} \right)} = 16341 \text{ kg}
\]

\[
V = \frac{m}{\rho} = 9.13 \text{ m}^3
\]
**Summary of results [2]:**

<table>
<thead>
<tr>
<th>Storage type; medium</th>
<th>Volume</th>
<th>Temp. range</th>
<th>Poisonous</th>
</tr>
</thead>
<tbody>
<tr>
<td>direct storage; salt nitrate</td>
<td>12.03 m³</td>
<td>450 – 550°C</td>
<td>Yes</td>
</tr>
<tr>
<td>direct storage; synthetic oil</td>
<td>17.39 m³</td>
<td>250 – 350°C</td>
<td>Yes</td>
</tr>
<tr>
<td>Indirect storage; concrete</td>
<td>22.65 m³</td>
<td>300 – 400°C</td>
<td>No</td>
</tr>
<tr>
<td>Indirect storage; sand</td>
<td>31.37 m³</td>
<td>600 – 700°C</td>
<td>No</td>
</tr>
<tr>
<td>Storage for latent heat; water</td>
<td>5.17 m³</td>
<td>-10 – 90°C</td>
<td>No</td>
</tr>
<tr>
<td>Storage for latent heat; salt mix</td>
<td>9.13 m³</td>
<td>210 – 310°C</td>
<td>Yes</td>
</tr>
</tbody>
</table>

**Problem 2:**

(i) Write the equation for calculating the storage capacity of direct storage systems for sensible heat.

(ii) Write the equation for calculating the storage capacity of indirect storage systems for sensible heat.

(iii) Write the equation for calculating the storage capacity of latent-heat storage systems.

**Problem 3:**

What effects/advantages does thermal energy storage have on a solar thermal power plant?
Problem 4:
Using the below diagram, explain the solar multiple “SM”

![Diagram showing solar multiple with SM = 1.0 and SM > 1.0]

Problem 5:
(i) Describe the 2-tank direct storage system of the solar tower power plant Solar Two.
(ii) Write down the chemical formulae of the heat transfer fluid used. Write out the names of the salts.

Problem 6:
(i) What are the two phase change materials used in latent-heat storage systems for the application in the medium and high temperature range?
(ii) Which three phase changes are possible in a latent-heat storage system?

Problem 7:
Describe the terms
(i) indirect storage and direct storage
(ii) sensible heat
(iii) latent heat

Problem 8:
If no thermal energy storage was used in a solar thermal power plant, what effect would the doubling of the solar multiple (SM) from SM = 1 to 2 have on the annual hours at nominal solar power?

Problem 9:
What is the capacity factor $CF$? Describe it in detail and give all relevant equations.
Problem 10:
Define the solar multiple SM.

Problem 11:
Name the different types of thermal energy storage systems, give an example for each and also state a possible HTF (do not write down chemical formulae).

Problem 12:
Name three common indirect storage designs.

Problem 13:
(i) Describe the packed-bed thermal energy storage system (regenerator) and how the basic charging and discharging processes work. Sketch the temperature profile of the storage system for 100% and 0% load and describe how the profile changes when the storage is being charged.

(ii) Sketch the sand storage concept for a solar thermal power plant for sand grain sizes up to 1 mm and describe it with the help of a sketch. In which component is the steam produced? What is meant by fluidisation?

Problem 14:
Sketch the heat exchanger of the sand storage. Include also the type of heat transfer fluid (HTF), the HTF temperature and flow direction as well as the sand flow direction (do not give any dimensions).

Problem 15:
A 2-tank direct storage system comes closest to the ideal storage. What are the conditions for an ideal storage?

Problem 16:
Name 1 advantage of latent-heat storage systems.

Problem 17:
Sketch and explain the temperature profile of the packed-bed thermal energy storage system for a range of load percentages.

Problem 18:
Write the equation for the storage discharge energy \( E_{dis} \) and give a brief explanation.

Problem 19:
What is the ultimate target for solar thermal power plants, especially with regard to thermal energy storage systems?
Problem 20:
Sketch a solar multiple SM vs. time of day diagram for the boundary condition solar multiple SM = 1.4 on June 21, with utilisation of the solar power between 6 am and 5 pm. Mark and label the areas representing the energy stored and the energy discharged.

Problem 21:
Name 4 advantages of sand as a storage medium.

Problem 22:
For a packed-bed thermal energy storage, calculate the following:

- energy storage dimensions: \(H\) (height), \(L\) (length) and \(B\) (breadth) \((L = 2B)\)
- pressure loss \(\Delta p\)
- volume-dependent heat transfer coefficient \(\alpha_v\) between airflow and bodies
- area-related heat transfer coefficient \(\alpha\) between airflow and bodies

The storage cpacity \(Q_s\), shall be 1600 MJ. The minimal and maximal temperature of the storage are \(T_{\text{min}} = 100^\circ\text{C}\) and \(T_{\text{max}} = 500^\circ\text{C}\), respectively. The airflow \(\dot{V}\) is given with \(\dot{V} = 650 \text{ m}^3/\text{h}\) and the air velocity \(w\) is given with \(w = 0.09 \text{ m/s}\). The storage material is gravel which has a uniform body diameter of \(d = 60 \text{ mm}\) and a form factor \(f = 1.5\). The porosity \(\varepsilon\) of the packed bed is \(\varepsilon = 0.4\).

The material properties of the gravel are: specific heat capacity \(c = 0.84 \text{ kJ/(kg K)}\), \(\rho_b = 1600 \text{ kg/m}^3\).

Hint: Begin with the calculation of the required air properties at the given temperatures.

Solution:

1) The density and kinematic viscosity of air is listed in literature in a table. For this calculation problem the values were obtained from German literature (VDI-Wärmeatlas, Springer-Verlag Berlin Heidelberg 2006). The used table includes several air properties for various air temperatures in a range of -200°C to 750°C. Note that dry air at a pressure of 1 bar is assumed in this calculation problem.

Values are obtained for the mean air temperature: \(T_m = \frac{(100 + 500)^\circ\text{C}}{2} = 300^\circ\text{C}\)

- air density \(\rho_{\text{air}} = 0.6075 \text{ kg/m}^3\)
- kinematic viscosity \(\nu_{\text{air}} = 490.7 \times 10^{-7} \text{ m}^2/\text{s}\)

2) Calculation of the usable storage volume and dimensions of the storage

\[
V = \frac{Q_s}{\rho_b \cdot c \cdot (1 - \varepsilon) \cdot (T_{\text{max}} - T_{\text{min}})}
\]

\[
V = \frac{1600000 \text{ kJ}}{1600 \text{ kg/m}^3 \cdot 0.84 \text{ kJ/(kgK)} \cdot (1 - 0.4) \cdot (773.15 - 373.15) \text{K}} = 4.96 \text{ m}^3
\]
The cross-section area of the storage is \( A = \frac{\dot{V}}{3600w} = \frac{650 \text{ m}^3 / \text{h}}{3600 \text{ s} / \text{h} \cdot 0.07 \text{ m/s}} = 2.58 \text{ m}^2 \)

The height of the storage is \( H = \frac{V}{A} = \frac{4.96 \text{ m}^3}{2.58 \text{ m}^2} = 1.92 \text{ m} \).

With \( A = L \cdot B = 2.58 \text{ m}^2 \) and \( L = 2 \cdot B \), the breadth \( B \) is calculated as follows:

\[
B = \sqrt{(0.5A)} = 1.135 \text{ m} \\
L = 2B = 2 \cdot 1.135 \text{ m} = 2.27 \text{ m}.
\]

3) Calculation of the pressure loss \( \Delta p \)

\[
\text{Re} = \frac{w \cdot d}{\nu_{\text{air}}} = \frac{0.09 \text{ kg/s} \cdot 0.06 \text{ m}}{490.7 \times 10^{-7} \text{ m}^2 / \text{s}} = 110.04
\]

For Reynolds numbers in the range \( 100 < \text{Re} < 13000 \), the following equation can be used:

\[
\Delta p = \rho \cdot w^2 \left( \frac{L}{d} \right) \left( 1 - \varepsilon \right)^2 \left[ \frac{1.24 \cdot \text{Re} + 368}{\text{Re}^3} \right] \left[ \frac{(1 - 0.4)^2}{110.04 \cdot 0.4^3} \right] = 4.76 \text{ Pa}
\]

As \( f \) and \( \varepsilon \) are known also the following equation can be used:

\[
\Delta p = \rho \cdot w^2 \left( \frac{L}{d} \right) \left( 1 - \varepsilon \right) \left[ \frac{1.24 + 166 \cdot (1 - \varepsilon) \cdot f}{\varepsilon^{\frac{3}{2}} \cdot \text{Re}} \right] \left[ \frac{(1 - 0.4)}{0.4^{\frac{3}{2}}} \right] = 5.76 \text{ Pa}
\]

Using the simplified equation, the pressure loss results in:

\[
\Delta p = \rho \cdot w^2 \left( \frac{L}{d} \right) \left( 21 + \frac{1750}{\text{Re}} \right) \left[ \frac{1.24 + 166 \cdot (1 - 0.4)}{0.4^{\frac{3}{2}} \cdot 110.04} \right] = 5.90 \text{ Pa}
\]
4) Volume-dependent heat transfer coefficient $\alpha_v$ between airflow and bodies:

$$\alpha_v = 650 \left( \frac{\rho w}{d} \right)^{0.7} = 650 \left( \frac{0.6075 \cdot 0.09}{0.06} \right)^{0.7} = 609.06 \, W/(m^3K)$$

5) Area-related heat transfer coefficient $\alpha$ between airflow and bodies

$$\alpha = \frac{\alpha_v d}{6 \left( 1 - \epsilon \right) f} = \frac{609.06 \, W/(m^3K) \cdot 0.06 \, m}{6 \left( 1 - 0.4 \right) \cdot 1.5} = 6.77 \, W/(m^2K)$$