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Application of high-temperature thermal energy storage in combined heat and power plants

van Amstel, T.M.J.

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Application of high-temperature thermal energy storage in combined heat and power plants

T.M.J. van Amstel (567038)

WET 2012.

Master Thesis

Supervisor:
prof.dr. H.A. Zondag

Advisor:
dr.ir. C.C.M. Rindt

External advisors:
ir. J.B.R.M. de Jong (RWE/Essent)
MSc. E. Lonis (RWE/Essent)

Eindhoven University of Technology
Department of Mechanical Engineering
Thermo Fluids Engineering Division
Energy Technology Section

Eindhoven, April 2012
Preface

The start of this graduation project was initially difficult but then very pleasant. At an early stage, contact with energy company Essent was made. The brother of the wife of the uncle of my girlfriend, Jan de Jong, was interested in collaboration with the Eindhoven University of Technology. He told me that there were several interesting projects available and the only thing I had to do was find a supervisor at my university. This turned out to be more difficult than expected. After several disappointments, I was advised to ask Camilo Rindt. He was the first open-minded person that was really interested and willing to discuss the possibilities with Essent. After considering a project about greenhouses, the choice was made to analyze thermal energy storage in combined heat and power plants. The last piece of the puzzle that completed the team was Professor Zondag, appointed as a new professor in thermal energy storage half a year earlier.

Looking back at almost a year of hard working, there was one particular thing that I constantly experienced during every meeting at the university: The extremely pleasant cooperation with Professor Zondag and Camilo Rindt. I would like to thank both of them for their involvement, critical opinions and especially their help. Thanks to them I learned to solve problems step by step instead of everything at once and as quickly as possible. Also writing a scientific report turned out to be a slowly but instructive process.

A second word of thanks goes out to Jan de Jong for his willingness to assist me. His drive and enthusiasm for technique and energy will always remain as an example for me. Third, I would like to thank Edgar Lonis for his help and the pleasant discussions. He provided me with all the information I needed, introduced me to many useful persons and helped me, together with Jan de Jong, to understand the complex project. Fourth, I would like to thank Henk Ouwerkerk for the problem enlightening discussions and Theo the Bruijne for the plant tour and information.

Finishing this thesis also marks the end of my life as a student. I feel ready to take the new step and start my new job at Essent. The experience I gathered during this graduation project will be useful for my whole further career. A word of appreciation goes out to my parents Wout and Annet for their financial and overall support. I would also like to thank the ‘koffiegroep’ that made my days look much shorter. Their presence always kept me happy and motivated.

Last, but definitely not least, I would like to thank the person that was very important to me: My girlfriend Sabine that supported me all the time and always took care of me while I was busy graduating. Thank you very much and I am looking forward to our future!

Tim van Amstel
April 2012
Summary

The number of combined heat and power (CHP) plants with a potential efficiency of 90% is increasing. Despite the high efficiency, the flexibility of the electricity production of such a plant is limited. CHP plants generate electricity with gas turbines and use their hot exhaust gases for different heating purposes. In general these heating purposes are leading and since the electricity production is coupled to the heat production, the flexibility is limited.

This research is conducted in cooperation with RWE/Essent and focuses on one of their existing CHP plants named ELSTA. ELSTA converts its heat to superheated steam which is supplied to a nearby company. The constant steam demand of the neighboring company limits the flexible electricity production and as a result ELSTA is obligated to generate electricity in the night while the costs are higher than the revenues. During the day an excess in steam is produced when the electricity demand is high and converted to electricity in a steam turbine. Since the steam turbine has an efficiency of only 35%, the total efficiency of the plant will decrease as a function of the amount of excess steam.

Thermal energy storage (TES) can be a solution to increase the flexibility and the efficiency of the ELSTA plant. The surplus of steam can be stored during the day and used during the night to lower the electricity production and increase the flexibility. In case the efficiency of the storage system is higher than that of the steam turbine, the performance of the plant can increase even further. Another improvement can be made when the storage is able to provide enough steam to shut down one gas turbine during the night. In that case two gas turbines can run at their maximum efficiency instead of three at their minimum efficiency.

Three TES systems are selected for a detailed numerical analysis. The goal is to determine the efficiency of every storage system and to analyze the applicability. A two tank salt storage, a concrete-phase change material (PCM)-concrete storage and a salt-PCM-salt storage have been investigated. The salt-PCM-salt storage showed the best results and is very well suitable for implementation in the ELSTA plant.

The salt-PCM-salt storage system consists of liquid salt that exchanges its heat with superheated steam and hot water in a heat exchanger and a PCM inside a tank that exchanges its heat with evaporating/condensing steam. The overall efficiency of the system with the heat losses and pump power included is 74.6%. The system can cool the input steam from 500°C to 201°C which indicates that there is room for improvement.

The application of the suggested storage system decreases the steam demand for ELSTA with 18% and results in an increased flexibility of the plant. Secondly, the storage supplies a sufficient amount of steam to run two gas turbines at maximum load instead of three at minimum load during the night. As a result, the turbine efficiency increases from 27% to 33%. As a third and final improvement, the total efficiency of the plant increases since the storage efficiency of 74.6% is higher than the turbine efficiency of 35%.
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Chapter 1 Introduction

The world is running out of fossil fuels and the need for sustainable solutions rises rapidly. Energy companies are trying to increase the efficiency of their power plants and to find alternative energy sources. European guidelines force the Netherlands to increase the amount of renewable energy from 4% in 2010 to 14% in 2020 [I]. This increase will mainly be realized with extra wind and solar energy. The problem of these two renewable sources is their unpredictability and the resulting fluctuations in electricity production.

Furthermore, the demand for more efficient power plants is rising. The effective use of the remaining fossil fuels is one thing, the rising environmental concerns is another. The number of so called Combined Heat and Power (CHP) plants with a potential efficiency of 90% is increasing [2]. This trend sounds very promising but there is a major drawback. Although a CHP plant is very efficient, it is unable to react flexibly to the electricity demand. Recalling the future increase in fluctuations of electricity production by wind and sun, a problem will arise when a significant number of new CHP plants will be installed. Conventional gas power plants are well capable of compensating fluctuations in electricity production while CHP plants are not. Therefore this research will examine the possibilities of increasing the flexibility of a CHP plant.

Combined Heat and Power Plants
Conventional gas power plants have the goal to produce electricity. They contain gas-fired turbines that drive a generator. The generator converts the mechanical energy into electrical energy. Only one-third of the produced heat is used to generate electricity and the rest is wasted into the environment. The major difference of a CHP plant is that it captures its waste heat and uses it for different heating purposes.

This research will focus on a particular existing CHP plant called ELSTA. The plant is for 25% owned by energy company RWE/Essent. This research is conducted with cooperation of RWE/Essent. ELSTA uses its heat to generate steam which is distributed to a nearby company. The ELSTA plant has the obligation to supply a sufficient amount of steam constantly. Therefore the steam demand is leading instead of the electricity demand. A sufficient amount of steam can result in an excess of steam. For example when there is no wind or sun available, the electricity demand is high and the plant will increase its load. At that moment a surplus of steam could occur. The real problem arises when the opposite occurs: In case there is much wind available and the sun shines, the electricity demand will decrease. The ELSTA plant is limited in shifting down its power because the steam demand always has to be fulfilled. Therefore its flexibility is limited.

Thermal Energy Storage
Two important problems at the ELSTA plant can occur: A surplus of steam could be generated when the electricity demand is high and a surplus of electricity could be produced while the demand is low because the steam demand needs to be fulfilled. A possible solution for this problem is the
application of thermal energy storage (TES). In case an excess in steam is generated, this energy can be stored. When the electricity demand is low, steam can partly be generated by the TES. As a result the minimum load of the plant can be decreased which increases the plant flexibility. An important aspect of the application of the TES is the high operating temperature in the plant. The TES has to be able to handle high temperatures without a compromise on performance.

Research goal
The goal of this research is to increase the flexibility and efficiency of a CHP plant by applying high temperature TES. First a data analysis is required to survey the processes in the CHP plant and to quantify the amount of excess steam. The input and the output requirements of the storage need to be formulated. Secondly, other improvements for the CHP plant in combination with an implemented TES will be examined. This is followed by a literature study that examines the applicable storage techniques for the formulated demands. The three most promising techniques are selected for further detailed analysis. The applicability is tested and the efficiency will be determined. Finally the best storage technique will be recommended.

Research outline
This research will start with a detailed description of the CHP plant and performs a data analysis in Chapter 2. Next, several TES systems from literature are examined for implementation in the CHP plant in Chapter 3. Based on this study, the three best applicable storage techniques are selected for a detailed numerical analysis: a two tank salt storage, a concrete-phase change material-concrete storage and a liquid salt-phase change material-liquid salt storage. The three models will be addressed in respectively Chapter 4, Chapter 5 and Chapter 6. Subsequently the best storage technique is selected in Chapter 7 followed by a discussion. The conclusion of this thesis and recommendations for further research are summarized in Chapter 8.
Chapter 2 Combined heat and power plant ELSTA

The introduction briefly explains the working principle of a combined heat and power (CHP) plant. In this chapter a more detailed description of a CHP plant will be given. An overview of the ELSTA plant is shown and is followed by a data analysis. The goal is to determine the excess steam that can be used for a thermal energy storage and suggest other improvements. This chapter will conclude with the input and output requirements of the thermal energy storage, based on the ELSTA plant.

2.1 Plant overview

An important element of this research is the CHP plant called ELSTA. CHP plants contain gas turbines that generate electricity and their hot exhaust gasses are used to produce steam in a boiler. In case of ELSTA, the generated steam is supplied to the neighboring company Dow Chemical. The steam supply is always leading what results in a minimum load of the gas turbines to fulfill the steam demand. As a result, always an excess in steam is produced to meet the requirements. ELSTA is equipped with an extra steam turbine to convert the surplus of steam in electricity what results in extra revenues.

An important detail of the ELSTA plant is the fact that is operates at two different pressure levels. Dow Chemical has a demand for superheated 90 bar steam of 500°C but also for superheated 35 bar steam of 350°C. These requirements result in a more complex CHP plant compared to ordinary ones. The boiler that converts the energy of the hot exhaust gasses to steam, generates steam at a pressure level of 90 bar. Steam at a pressure level of 35 bar is extracted out of the steam turbine. The steam turbine is driven by the surplus of 90 bar steam. After the first stage of expansion, 35 bar steam can be extracted out of the steam turbine at a temperature of 350°C. Figure 2.1 shows a schematic overview of the plant:

ELSTA contains three 123 MW gas turbines and a 90 MW steam turbine. When the plant operates at full power a maximum of 460 MW is produced. The minimum load depends on the steam demand.
In the plant description above is stated that the electricity and the steam production are coupled. This is not completely true in the ELSTA plant. Because steam production is so critical, the boiler is equipped with so called duct burners. Exhaust gases from gas turbines contain around 15% of oxygen. It is possible to inject and ignite extra natural gas in the boiler to generate more steam. This results in a better controllable steam and electricity generation ratio but there is a threshold. Duct burning can only be applied when the gas turbines are operating and therefore the CHP still has a limited flexibility.

2.2 Efficiency

The previous paragraph shows that the CHP plant ELSTA can operate in two different modes. In the first mode the gas turbines generate electricity and the generated steam is converted into extra electricity in a steam turbine. This is called the combined cycle. In the second mode the gas turbines generate electricity and the generated steam is supplied to Dow Chemical. This is called combined heat and power. According to the plant manager of ELSTA, their maximum efficiency of the combined cycle is 45% and the maximum efficiency of combined heat and power is 80%. The difference in efficiency between the two operation modes is caused by the steam turbine that has a low efficiency. In case the plant operates in the combined heat and power mode, the steam turbine is not used and fewer losses are present. The only losses that are present are generated in the gas turbines, in the boiler and at the transport of the steam to the other production site.

It can be concluded that it is important for the ELSTA plant to operate in the combined heat and power mode as much as possible to achieve the highest efficiency. The problem is that this depends on the steam demand of Dow Chemical but also on the electricity demand. It was already concluded that there always is an excess in steam to secure sufficient steam supply. This means that the plant operates partly in the combined heat and power mode and partly in the combined cycle mode. As a result the maximum efficiency is not reached. In case thermal energy storage would be applied, the excess steam can be stored and supplied back to Dow later. In consequence the efficiency of the plant will go up. In the next paragraph the data of the ELSTA plant of 2010 will be analyzed and the amount of excess steam will be determined. Also other improvements in combination with the storage are suggested. The outcome will serve as an input for the thermal energy storage.

2.3 Data analysis over the year

This part of the report will give an overview of the dynamics of the plant and how it is controlled during the year. As stated before, there is a certain demand for 35 bar and 90 bar steam and a separate electricity demand. The complexity of the problem is that these two demands affect each other. Figure 2.2 shows that steam supply and demand are not in equilibrium. Always a surplus of steam is present, called the spare steam. The steam demand in figure 2.2 is the sum of the 35 bar and the 90 bar steam demand. The steam supply is the 90 bar steam produced by the steam boiler.
Figure 2.2: Steam supply and demand over the year 2010. The supply contains the generated 90 bar steam of the boiler and the demand is the sum of the 35 bar and 90 bar steam demand of Dow.

From October to January the overproduction of steam is relatively small while in June it gets larger. The minimum difference between supply and demand of steam is found to be 62 MT/hr and the average is 165 MT/hr. In the contract between ELSTA and DOW Chemical is stated that there always needs to be 50 MT/hr of 90 bar spare steam available. Since the fact that there is much more spare steam produced than needed, the energy plant needs to be analyzed more in detail to ascertain the cause.

The first conclusion that can be drawn from figure 2.2 is that the steam supply adjusts to the steam demand. When the steam demand increases significantly, the steam supply adapts and also increases. This is done by duct firing. Extra gas is ignited in the boiler and results in an increased steam production without changing the amount of electricity generation. Figure 2.3 confirms this assumption.

Figure 2.3: Amount of duct firing in m3 gas per hour over the year 2010

Figure 2.3 above also shows that the amount of duct firing can go up without an increasing steam demand. In that case the electricity production went down and the cofiring was used to compensate
the decreased amount of produced steam from the gasturbines. Figure 2.4 shows the electricity production over the year.

![Figure 2.4: Total electricity generation over the year 2010](image)

Comparing figure 2.4 above of the electricity production with figure 2.2 of overproduction of steam and figure 2.3 of duct firing, it can be concluded that an increased steam production is not always caused by an increased steam demand. There are exceptions. During February and March for example, there is an increased overproduction of steam while the demand is not changing. The graph of the electricity production shows that this increased overproduction of steam is also not caused by an increased electricity production. The electricity production even goes down. In this particular situation it had the following reason: one gas turbine was out of order because it was under repair. In the contract is stated that when a turbine is out of order, the amount of spare steam needs to be tripled. This results in an increased overproduction of steam.

The data analysis over the year shows an average excess in steam of 165 MT/hr. The total average steam production of ELSTA over 2010 is 575 MT/hr. The average 90 bar steam export to Dow Chemical is 336 MT/hr. A summary of the steam production in rounded percentages is depicted in figure 2.5 below.

![Figure 2.5: Schematic overview of ELSTA with the average percentages of generated steam over 2010](image)
The amount of excess steam is now known. The problem is that the previous analysis was carried out over a year. It is not clear when the surplus of steam is generated. This will be examined in the next paragraph.

2.4 Data analysis per day and per week

In the previous section the data over a whole year are analyzed. This part will focus on the data over a day and over a week. First, the power production is analyzed. The three gas turbines and the steam turbine are responsible for the electricity production. The demand changes rapidly during 24 hours. In the next two figures the average power production and the average total steam demand of 90 bar and 35 bar during the day are depicted from the month January.

![Power Production vs. Time](image)

![Steam Demand vs. Time](image)

**Figure 2.6:** The left figure shows the power production during the day and the figure on the right the total steam production during the day.

The left graph in figure 2.6 shows that early in the morning the power production is going down and at 06:00h it goes up again. Between 14:00h and 16:00h the amount of produced MW goes down a bit and starts rising between 18:00h and 19:00h. After 21:00h again a decrease can be seen which continues the next day.

The right graph in figure 2.6 clearly shows that the steam demand is nearly constant during the day. Since the average power production is changing during the day, it can be concluded that duct firing is compensating the difference between steam production by the gas turbines and steam demand. Figure 2.7 confirm this assumption, showing the average duct firing during the day.

![Duct Firing vs. Time](image)

**Figure 2.7:** Average duct firing in amount of burned gas during the day over 2010.
The previous paragraph clearly showed an overproduction of steam over the year. This paragraph shows that this excess in steam is mainly produced during the day. At night, duct firing is needed to generate enough steam. It can be concluded that the thermal storage needs to be charged during the day and discharged during the night.

Another part of the analysis is the power production during the week. It is important to know whether the trend changes during the week. The next graph gives more insight.

![Figure 2.8: Average power production during the week over 2010](image)

No real trends can be spotted in figure 2.8 so the production during the week can be assumed as constant. The same conclusion holds: The storage needs to be charged during the day and discharged during the night.

### 2.5 Possible savings

This research focuses on increasing the flexibility and the efficiency of the ELSTA plant by implementing thermal energy storage. This chapter started with an overview of plant. The plant operates in two different modes. When it operates as combined cycle, all the produced steam is converted to electricity in a steam turbine and the total efficiency is 45%. The average efficiency of the steam turbine is 35% and a lot of losses. When the plant operates in the combined heat and power mode, all the produced steam is send to DOW and the efficiency is 80%. In general, the plant operates in both modes. The steam that is demanded by DOW is supplied and the excess is used to produce extra electricity in the steam turbine.

The biggest drawback of ELSTA is the inflexibility. The plant always needs to produce the amount of demanded steam but also an extra amount of spare steam that is converted to electricity in a steam turbine. As a result a minimum load of the gas turbines is required and limits the flexibility of the plant. This can partly be compensated with duct firing but still the system is not flexible.

Thermal energy storage is suggested to increase the flexibility but also to try to increase the efficiency. There are two ways to realize this:

1. The plant needs to operate more in combined heat and power mode and less as a combined cycle. Paragraph 2.2 showed a better efficiency for the combined heat and power mode. During the day there is a surplus of steam that is send to the steam turbine (combined cycle)
while during the night extra steam is generated by duct firing. Although the duct firing is used, still too much unwanted electricity is produced. The costs of generating electricity during the night are higher than the revenues. The reason why electricity is produced is because the plant has to supply steam to DOW. The duct firing can compensate a part but this is still not enough. It would be better store the overproduction of steam during the day and use it at night. In that case the plant operates in the combined heat and power mode only, the efficiency is higher and it is more flexible.

2. The second possibility to save energy involves the load of the gas turbines. A drawback is that during the night, the gas turbines mostly operate at minimum load. Therefore the efficiency is only 27% instead of 33% at maximum load. It is not possible to run two gas turbines at maximum load instead of three on minimum load because then the steam demand cannot be fulfilled anymore. A gas turbine at minimum load is responsible for 145 MT/hr of steam and at maximum load for 195 MT/hr. The total steam production at minimum load for three gas turbines is 435 MT/hr and at maximum load for two gas turbines is 390 MT/hr. When the thermal energy storage could supply at least 40 MT/hr of steam, it would be sufficient to run two gas turbines at maximum load. This would increase the total efficiency of the plant.

2.6 Dimensioning the TES system

There are two mass flows that can be used to charge the TES. First, the hot exhaust gasses from the gas turbine and second, the superheated 90 bar steam. Unfortunately it is not possible to use the hot exhaust gasses to charge the energy storage. The ELSTA plant is designed for continuous optimal use. Converting all the hot exhaust gasses to superheated steam is part of the optimized system and all the components in the plant are specially designed for this. This is the reason why the superheated 90 bar steam of 500°C will be used to charge the TES and not the hot exhaust gasses. In order to extract as much energy out of the input steam as possible, the goal is to cool the superheated steam to water at 60°C.

The demanded output of the TES also needs to be well defined. It is clear that steam needs to be produced at the output but the plant operates at different pressure levels. Since thermal losses will be present in the TES, it is impossible to generate steam with the same temperature as the input temperature. The temperature will always drop so reheating would be needed. Reheating would cost extra energy so this option is not reasonable in order to save energy. This is why the choice is made to produce 35 bar steam with a temperature of 350°C at the output of the TES. Steam with these conditions can be supplied to DOW, retaining the combined heat and power cycle. From now on it is assumed that the average 35 bar steam demand of DOW over the year 2010 needs to be supplied by the storage. This average amount is 73 tons of superheated steam per hour with a pressure of 35 bar and a temperature of 350°C.

Discharging the TES will occur during the night. During these hours, the electricity price is so low that the costs for electricity production are higher than the revenues. Figure 2.6 of the day analysis shows that during the night the electricity production decreases at ELSTA. Duct firing is needed to generate enough steam during these moments. It is preferred to decrease the electricity production
even more but that is impossible because of the steam demand. The next figure shows the average electricity price over the day in 2010.

![Average electricity price over the day for 2010](image)

**Figure 2.9:** Average electricity price over the day for 2010

Between 23:00h and 07:00h the electricity price is the lowest so the storage will discharge during these eight hours. In order to simplify the problem, it is assumed that charging the storage will also take eight hours. This will decrease the difference between the charging and discharging mode. These eight hours of charging will occur at the moment when the electricity price is the lowest.

Summarizing, the next values for the input and the output of the TES are determined:

<table>
<thead>
<tr>
<th>Input</th>
<th>Output</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow [kg/hr]</td>
<td>73000</td>
</tr>
<tr>
<td>Temperature inlet [°C]</td>
<td>500</td>
</tr>
<tr>
<td>Temperature outlet [°C]</td>
<td>350</td>
</tr>
<tr>
<td>Discharge time [hours]</td>
<td>8</td>
</tr>
<tr>
<td>Pressure [bar]</td>
<td>90</td>
</tr>
</tbody>
</table>

**Table 2.1:** Input and output parameters for the TES

The thermal energy storage will be charged with superheated 90 bar steam of 500°C for eight hours. In order to utilize most of the energy of the input, it needs to be cooled to 60°C. The lowest possible charging mass flow will be applied to generate 73 tons of superheated 35 bar steam at 350°C per hour during eight hours of discharging. This output mass flow will be produced by heating up pressurized water of 60°C. The next chapter will examine thermal energy storage techniques that are applicable in this situation, based on the above formulated requirements.

### 2.7 Conclusions

The possible increase in flexibility can now be determined. The average total steam demand of DOW is 409 MT/hr and the 35 bar steam demand is 73 MT/hr. During the night the plant operates at
minimum load to supply a sufficient amount of steam. In case the storage system is used to generate
the demanded 35 bar steam, 18% less steam has to be produced by the ELSTA plant and increases
the flexibility.

The potential increase in efficiency can also be determined. In case the suggested TES is able to
deliver 73 MT/hr of steam, it is possible to use two gas turbines at maximum load instead of three
gas turbines at minimum load. The efficiency of the gas turbines will then go up from 27% to 33%.
The second increase in efficiency can be realized in case the efficiency of the storage system is
higher than the steam turbine efficiency.

In the next chapters of this thesis, the applicable thermal energy storage techniques for the
formulated requirements will be analyzed. If it is possible to design a suitable storage system, the
flexibility and the efficiency of the plant will go up. When the efficiency of the TES is higher than
the steam turbine efficiency, the total efficiency of the plant will increase even further.
Chapter 3 Thermal energy storage

Thermal Energy Storage (TES) can be an important element to improve the flexibility of a Combined Heat and Power (CHP) plant. In Chapter 2, the ELSTA plant is introduced which has the obligation to supply a certain amount of steam to a nearby company continuously. Therefore the plant has a minimum load which limits the flexibility. Another important aspect is that there is always an excess in steam generated. This steam is sent to a steam turbine where is in converted into electricity. The resulting problem is a lower total efficiency of the plant. The application of thermal energy storage could be the solution for both of the problems. The overproduction of steam can be stored in the TES and used to lower the minimum load of the plant when that is desired. The result is a more flexible but also a more efficient CHP plant. This part of the report will examine the different possibilities of thermal energy storage. There are several interesting options that will be discussed. The different TES techniques will be introduced first and then the advantages and disadvantages of implementing the techniques in a CHP plant will be examined. This chapter concludes with the selection of the best suitable TES techniques.

3.1 Storage techniques

The thermal energy storage materials can be divided in four different categories: liquid materials, solids, phase change materials (PCM) and thermo chemical materials (TCM). This paragraph will introduce every category separately and discuss some of the applicable materials. An important note here is that this chapter will zoom in on the used materials and not the total storage system. TES is used a lot in the concentrated solar power industry and examples will be shown from implemented techniques in such plants. The application of TES in CHP plants is new and an important part of this research.

3.1.1 Liquids

When direct liquid storage will be used for a CHP plant, the superheated 90 bar steam has to be stored directly. Since storage in pressure vessels is not useful because of the low volumetric energy density, steam accumulators are used. The steam will be stored as saturated water in a pressurized tank [3]. Figure 3.1 shows the steam accumulator in blue, implemented in a solar power plant.
In figure 3.1 steam is used as the heat transfer fluid but also as storage medium. In case the liquid storage medium is not the same as the heat transfer fluid, the system is called an indirect storage system. Then the heat transfer fluid is used to exchange its energy in a heat exchanger to a liquid storage material. The storage material is stored in a tank until the system is discharged again. During discharging the hot liquid storage material will exchange its heat back to a heat transfer fluid. The indirect storage can be implemented with a two tank storage system as explained above but also in a one tank storage system [4]. In the case of a one tank storage system, the hot and the cold fluid are both stored in the same tank. The separation of the layers is caused by the difference in stratification and the boundary is called the thermocline [5]. Figure 3.2 shows the indirect two tank storage system.
Figure 3.3 shows the indirect storage system with a single tank:

![Indirect Storage System Diagram](image)

**Figure 3.3: Indirect one tank TES [4]**

Both of the two systems from figure 3.2 and 3.3 can be implemented in a CHP plant. There are several materials available that can be used as a liquid storage medium. Table 3.1 shows the possible options.

<table>
<thead>
<tr>
<th>Storage Medium</th>
<th>Temperature (°C)</th>
<th>Average density (kg/m³)</th>
<th>Average heat conductivity (W/mK)</th>
<th>Average heat capacity (kJ/kgK)</th>
<th>Volume specific heat capacity (kWh/m³)</th>
<th>Media costs per kg ($/kg)</th>
<th>Media costs per kWh ($/kWh)</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>55</td>
<td>0.30</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>57</td>
<td>3.00</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>52</td>
<td>5.00</td>
</tr>
<tr>
<td>Nitrite salts</td>
<td>250</td>
<td>450</td>
<td>1,825</td>
<td>0.57</td>
<td>1.5</td>
<td>152</td>
<td>1.00</td>
</tr>
<tr>
<td>Nitrite salts</td>
<td>265</td>
<td>565</td>
<td>1,870</td>
<td>0.52</td>
<td>1.6</td>
<td>250</td>
<td>0.70</td>
</tr>
<tr>
<td>Carbonate salts</td>
<td>450</td>
<td>850</td>
<td>2,100</td>
<td>2.0</td>
<td>1.8</td>
<td>430</td>
<td>2.40</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>80</td>
<td>2.00</td>
</tr>
</tbody>
</table>

**Table 3.1: Liquid storage media [6]**

The table shows that liquid salts and sodium are the best options concerning the high temperatures that are reached in the ELSTA plant. Concerning the volume specific heat capacity and the costs per kWh, liquid salt is preferable over liquid sodium. The nitrate salts are very interesting because of their low costs in combination with the relatively high specific heat capacity.
3.1.2 Solids

In a solid storage medium the heat transfer fluid only passes the storage system when it is charged or discharged. The difference with a liquid material is that the storage medium does not move. There are different materials available. Table 3.2 shows some examples:

<table>
<thead>
<tr>
<th>Storage Medium</th>
<th>Temperature</th>
<th>Average density</th>
<th>Average heat conductivity (W/mK)</th>
<th>Average heat capacity (kJ/kgK)</th>
<th>Volume specific heat capacity (kWh/m³)</th>
<th>Media costs per kg ($/kg)</th>
<th>Media costs per kWh ($/kWh)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Sand-rock-mineral oil</td>
<td>200-300</td>
<td>1,700</td>
<td>1.0</td>
<td>1.30</td>
<td>60</td>
<td>0.15</td>
<td>4.2</td>
</tr>
<tr>
<td>Reinforced concrete</td>
<td>200-400</td>
<td>2,200</td>
<td>1.5</td>
<td>0.85</td>
<td>100</td>
<td>0.05</td>
<td>1.0</td>
</tr>
<tr>
<td>NaCl (solid)</td>
<td>200-500</td>
<td>2,100</td>
<td>7.0</td>
<td>0.85</td>
<td>150</td>
<td>0.15</td>
<td>1.5</td>
</tr>
<tr>
<td>Cast iron</td>
<td>200-400</td>
<td>7,200</td>
<td>37.0</td>
<td>0.56</td>
<td>180</td>
<td>1.00</td>
<td>32.0</td>
</tr>
<tr>
<td>Cast steel</td>
<td>200-700</td>
<td>7,800</td>
<td>40.0</td>
<td>0.60</td>
<td>450</td>
<td>5.00</td>
<td>60.0</td>
</tr>
<tr>
<td>Silica fire bricks</td>
<td>200-700</td>
<td>1,820</td>
<td>1.5</td>
<td>1.00</td>
<td>150</td>
<td>1.00</td>
<td>7.0</td>
</tr>
<tr>
<td>Magnesia fire bricks</td>
<td>200-1,200</td>
<td>3,000</td>
<td>5.0</td>
<td>1.15</td>
<td>600</td>
<td>2.00</td>
<td>6.0</td>
</tr>
</tbody>
</table>

Table 3.2: Solid storage media [6]

Another possible solid material is a packed bed of natural rocks. According to the literature [7], the average heat capacity is 1030 J/kgK, the average density is 2680 kg/m³, the average heat conductivity is 2.5 W/mK and the costs for the material are low.

Depending on the filler material, different set-ups are needed to make the medium useful as an energy storage system. For example, the German Aerospace Center (DLR) successfully made and tested a concrete storage for the solar power plant Almeria [8]. A tube register was created first and then covered with concrete. A large number of different cements were tested first to find the most useful composition. The storage will be charged when the HTF is transported through the tube register and is transferring the heat to the concrete. The concrete stores the thermal energy. Figure 3.4 below shows a concrete storage module, consisting of many parallel tubes covered with concrete.

Figure 3.4: TES of solid concrete [7]
A different storage concept is a packed bed of natural rocks or pebbles. By packing a lot of natural stones together, a porous structure arises. This structure can be implemented in an insulated tank to be able to function as a TES. When the storage needs to be charged, hot air is normally sent into the tank where it exchanges its heat to the stones. The tank will be built in vertical direction and charged from above [8]. In the tank stratification in temperature will arise during charging. The upper part, which is passed first by the steam, will have the highest temperature. The bottom has the lowest temperature. Normally the packed bed storage is used in combination with air as the heat transfer fluid. This makes discharging easy. The process can be reversed by injecting cold air at the bottom of the tank. While flowing up, the air is heated. In case the system is implemented in the CHP plant, the output of the storage needs to be steam. This can be realized by creating tubes between the rocks in the storage tank to be able to produce steam with the right requirements during discharging. Charging can be done in direct contact with the rocks.

The above explained set-up can also be realized with ceramics instead of natural stones. The set-up is the same. Hot steam is sent through the porous ceramics and will heat the material up. Steam is produced during discharging of the system by sending water through the tubes that are present between the ceramics. The geometry of the ceramics can be in the shape of checker bricks [8], spheres, saddles or cowper stones [9].

3.1.3 Phase change materials

The ELSTA plant operates with superheated steam and therefore the charging and discharging of the storage involves superheated steam. In case superheated steam needs to be generated, more than 70% of the required energy is needed for evaporation. An efficient alternative to the liquid and solid thermal energy storage systems is the use of phase change materials (PCM). A lot of energy can be stored in the latent heat of PCMs. They operate within a small temperature range because during evaporation/condensation small temperature differences are present. Figure 3.5 shows an example of the working principle.

![Figure 3.5: Working principle PCM](image)

The heat transfer fluid flows through the finned tube and exchanges its heat to the tube. The tube exchanges the heat again to the PCM. The fins are used to increase the heat transfer area. The most effective phase change materials use the solid-liquid phase change [11].
3.1.4 Thermo chemical materials

The above explained options for TES have one problem in common. Storing the heat will result in energy losses. The longer the thermal energy needs to be stored, the more heat will be lost. The next technique has the advantage that no energy will be lost during storage. This is the so called thermal sorption storage performed with thermo chemical materials (TCM). Charging the storage is based on a reversible endothermic reaction. Heat is needed to drive the reaction. When the supply of heat stops, the system cools down and drives the reaction in the other direction. This reversed reaction is exothermic so energy will be released. In order to use these properties for storage, the reversed reaction needs to be prevented. If the products of the endothermic reaction are separated and stored, the energy is stored until they are combined again.

Unfortunately this storage technique is not developed well enough for use on big scale. More investigation is needed first to make it feasible with a CHP plant. Current problems on small scale are the difficulty of the control of the process conditions during charging and discharging, the reactor design that leads to an inhomogeneous hydration and dehydration with a big pressure drop and the lack of insight in what is exactly happening in the processes [12].

3.1.5 Combinations of techniques

The four applicable storage materials are based on solids, liquids, phase change materials and thermo chemical materials. Other research also used combinations to be able to meet some specific demands for the TES. The combined techniques will be discussed briefly in the next part.

The first combination that will be discussed uses solid and liquid materials together. Earlier was stated that using one storage tank (the so called thermocline) can reduce the total costs for a TES. There can be gained even more by using low-cost filler material as the primary storage medium and molten nitrate as the HTF. The best solid filler materials for this application are quartzite rock and silica sand [13]. This system is useful for CSP plants that use molten salt as a HTF. During charging the hot molten-salt flows through the tank at the top and heats up the rocks. It exits the storage tank at the bottom at a lower temperature. Discharging the system is the exact opposite. Implementing this system into a CHP plant would change the way of charging. The heat exchanger that is normally used to generate steam out of the hot molten-salt is now also used for charging the system. Figure 3.6 shows the set-up.
The next storage technique uses both concrete and a steam accumulator. Steam accumulators can only provide saturated steam while ELSTA operates with superheated steam. Therefore an additional system is needed. A concrete storage system, as depicted below, is connected to the steam accumulator [15]. When the system is charged, superheated steam heats up the concrete storage until saturated steam remains and is then stored in the steam accumulator. Discharging the system works the other way around. The set-up is depicted in figure 3.7 below.

The next TES technique is a combination of concrete, PCM and concrete. The ELSTA plant needs to generate steam during discharging of the storage system. In order to produce steam, water is preheated, then vaporized and finally superheated. These three processes can be divided in three separated storage systems. When the system is charged, first a concrete storage is passed where superheated steam exchanges heat until saturated steam is formed. The saturated steam exchanges its
heat in the PCM when it condenses to saturated water and is cooled down further in the second concrete storage. Discharging the system works the other way around. Water is preheated in the first concrete part until saturated water is formed, then the PCM vaporizes the water and the last concrete storage is responsible for superheating the steam. Figure 3.8 shows the set-up of the system [16].

![Diagram of the TES system](image)

**Figure 3.8:** TES made of concrete-PCM-concrete [16]

The last discussed system also contains three storage sections for the three different phases of steam. It uses a combination of the two tank salt storage for the superheated phase and the liquid phase and a PCM for the vapor phase [17]. The set-up is depicted in figure 3.9 below.

![Diagram of the two tank liquid storage and steam accumulator](image)

**Figure 3.9:** Combination of a two tank liquid storage and a steam accumulator [17]

A buffer tank is applied in the set-up to be able to vary the mass flows of the salt through the heat exchanger for the superheated and the liquid phase. Since the specific heat of water and steam is different, the mass flow of salt needs to be varied to reach the wanted temperature.
3.2 Implementation of TES techniques in ELSTA plant

In this chapter the possibilities for the application of TES in the ELSTA plant are investigated. The previous paragraph summed up the existing storage technologies and this paragraph will focus on the application of the existing storage systems in the ELSTA plant. The following values for the input and output of the storage were formulated in Chapter 2:

<table>
<thead>
<tr>
<th></th>
<th>Charging</th>
<th>Discharging</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow [kg/hr]</td>
<td>?</td>
<td>73000</td>
</tr>
<tr>
<td>Temperature inlet [°C]</td>
<td>500</td>
<td>60</td>
</tr>
<tr>
<td>Temperature outlet [°C]</td>
<td>?</td>
<td>350</td>
</tr>
<tr>
<td>Discharge time [hours]</td>
<td>8</td>
<td>8</td>
</tr>
<tr>
<td>Pressure [bar]</td>
<td>90</td>
<td>35</td>
</tr>
</tbody>
</table>

Table 3.3: Charging and discharging variables of the TES

The energy that needs to be stored to generate the above given output can be determined with the next formula:

\[ E = \Delta t_{out} \cdot \dot{m}_{out} \cdot (H_{out, out} - H_{out, in}) \]  

(3.1)

where \( \Delta t_{out} \) is the discharge time, \( \dot{m}_{out} \) the mass flow of the output steam and \( H_{out, out} \) and \( H_{out, in} \) respectively the enthalpy of the steam at the prescribed outlet and the inlet during discharging. The total amount of needed energy in the storage is 1.66 TJ or 462 MWh. The storage has to be able to deliver 57.7 MW of power during 8 hours.

The TES techniques will be analyzed with several criteria and a multi criteria analysis is performed to find the three best suitable storage systems.

3.2.1 Criteria

The next requirements are formulated for a multi criteria analysis:

**Energy density**

The energy density is determined by the multiplication of the density with the specific heat capacity and the temperature range. When the density and the specific heat capacity of a material are high, little mass of the storage medium is needed so also a smaller volume. The applicable temperature range is also an important factor that can lower the needed volume. The wider the temperature range, the more energy can be stored. A positive result of these three factors results in less needed volume for the storage set-up. This can be an important criterion because the TES has to be implemented in an existing CHP plant. The amount of space available is limited.

**Heat transfer capacity of storage material**

A good heat transfer capacity will increase the dynamics of the system. The thermal energy of the steam needs to be exchanged with all the material during charging. A better conductivity allows easier charging of the TES. An important note here is that a difference will be made between liquids and solids. The conductivity of solids is in general high but this does not mean that it is easy to
charge the material. The heat transfer capacity can still be low. Heat exchanging tubes within the material are needed to spread out the heat through the whole storage. In case of liquids, a shell in tube heat exchanger can be used to exchange heat. Not all the material has to be charged at once but can be pumped through the heat exchanger to have a bigger heat exchanging area. This results in a high heat transfer capacity. This criterion will be evaluated with a qualitative value.

**Applicability of the storage technique in combination with steam**

An important part of the problem is that the thermal energy of steam needs to be transferred to a storage material but it also has to be possible to extract steam out of the material. Some TES options are easily applicable for this problem while others need to be adjusted because they normally are charged or discharged in a different way. This criterion is also evaluated with a qualitative value.

**Proven technology**

Some storage technologies are currently already used in certain applications while others are examined in new research projects. This criterion will give a qualitative value that indicates the degree of proven technology.

**Low investment costs**

The investment costs are based on existing projects or estimated based on literature. In case the estimation is too difficult, a qualitative value will be given.

**Stability**

The mechanical and chemical stability are important factors that determine the degree of possible long term use. The TES system needs to be able to run a very large number of cycles, while charging and discharging are continuously alternating. The severe restrictions of the output also play an important role here. A decreasing power output during discharging is a disadvantage.

**Chemical compatibility**

Good chemical properties of the storage medium are important to make it compatible with the system. Important factors that will be examined are the corrosiveness, toxicity and flammability. This criterion will receive a qualitative mark.

**Control**

The TES system will be adapted in an existing CHP plant. The control of the CHP plant is already very advanced and makes the implementation of TES more difficult. The need for an easy controllable system is therefore high.

The criterion efficiency is deliberately not included in the list. First of all, the efficiency is partly dependent on the conduction, the applicability of the technique in combination with steam and the stability of the system. Second of all, the efficiency strongly depends on the investment costs. In a situation when costs are no issue, a lot can be invested in the insulation of the storage. The last and most important reason is the unpredictability of the efficiency. The three best suitable storage techniques will be tested with a detailed numerical analysis. This numerical analysis will result in a approximation of the efficiency of the storage system.
3.2.2 Multi criteria analysis

The choice for the most suitable storage system will be made based on a multi criteria analysis (MCA). A MCA is a tool that can be used for decision making in a problem with conflicting goals. Multiple alternatives are available to reach a certain goal. In a MCA, criteria are formulated to evaluate the different options. The criteria will be standardized, what results in a score between 0 and 1. Then the criteria are weighted and the standardized scores are multiplied with the weighting factors. The summation of the weighting factors has to be one. The summed numbers per alternative will lead to a total score between zero and one. Based on the total score a ranking can be made. The criteria for this case are already mentioned and briefly explained above.

The weighting factors that have to be applied can be approached from two different ways. In the view of the science, criteria such as costs and proven technology are not important at all. The focus is on the most promising technique. The energy density, conductivity, applicability with steam and stability are the most important factors. The rest is less important. In case the weighting factors are set by the industry, proven technology and costs are the most important criteria. Second are the applicability with steam and the stability of the system. Third are the energy density and the conductivity. These criteria are still important but when a technique is proven, stable and the investment costs are low, the energy density and conductivity have to be good enough already. The chemical applicability and control are the least important because these criteria overlap other criteria too. The next weighting factors are given to the criteria.

<table>
<thead>
<tr>
<th>Weighting factors</th>
<th>Industry</th>
<th>Science</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy density [J/m3]</td>
<td>0.10</td>
<td>0.20</td>
</tr>
<tr>
<td>Heat transfer capacity [+/-]</td>
<td>0.10</td>
<td>0.20</td>
</tr>
<tr>
<td>Applicability with steam [+/-]</td>
<td>0.15</td>
<td>0.20</td>
</tr>
<tr>
<td>Total investment costs [Mio €]</td>
<td>0.20</td>
<td>0</td>
</tr>
<tr>
<td>Proven technology [+/-]</td>
<td>0.20</td>
<td>0</td>
</tr>
<tr>
<td>Stability [+/-]</td>
<td>0.15</td>
<td>0.20</td>
</tr>
<tr>
<td>Chemical compatibility [+/-]</td>
<td>0.05</td>
<td>0.10</td>
</tr>
<tr>
<td>Control [+/-]</td>
<td>0.05</td>
<td>0.10</td>
</tr>
</tbody>
</table>

Table 3.4: Weighting factors for the criteria per stakeholder

All the storage techniques that are introduced in paragraph 3.1 will be discussed in the multi criteria analysis. Some criteria are evaluated with numbers and others are evaluated with plus and minus signs. The results are normalized in a number between zero and one. Table 3.5 shows the results. For every storage technique, the value for the a criteria can be found in the left column. The column on the right shows the normalized number. In Appendix A the detailed argumentation and the explanation of the normalization can be found. This appendix also includes the numbers that are found for the density, heat capacity and the temperature range that are used to determine the energy density.
<table>
<thead>
<tr>
<th></th>
<th>Two tank salt</th>
<th>One tank salt</th>
<th>Steam vessel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy density [J/m³, +/−]</td>
<td>4.725E+08 0.67</td>
<td>4.725E+08 0.67</td>
<td>8.436E+07 0.12</td>
</tr>
<tr>
<td>Heat transfer capacity [+/-]</td>
<td>++ 1.00</td>
<td>++ 1.00</td>
<td>++ 1.00</td>
</tr>
<tr>
<td>Applicability with steam [+/-]</td>
<td>++ 1.00</td>
<td>++ 1.00</td>
<td>+ 0.80</td>
</tr>
<tr>
<td>Investment costs [Mio €, +/-]</td>
<td>25 0.93</td>
<td>20 1.00</td>
<td>-- 0.20</td>
</tr>
<tr>
<td>Proven technology [+/-]</td>
<td>++ 1.00</td>
<td>+ - 0.60</td>
<td>+ - 0.60</td>
</tr>
<tr>
<td>Stability [+/-]</td>
<td>+ - 0.60</td>
<td>- 0.40</td>
<td>+ - 0.60</td>
</tr>
<tr>
<td>Chemical compatibility [+/-]</td>
<td>+ - 0.60</td>
<td>+ - 0.60</td>
<td>++ 1.00</td>
</tr>
<tr>
<td>Control [+/-]</td>
<td>++ 1.00</td>
<td>- 0.40</td>
<td>- 0.40</td>
</tr>
<tr>
<td>Total Industry score</td>
<td>0.87</td>
<td>0.75</td>
<td>0.55</td>
</tr>
<tr>
<td>Total Science score</td>
<td>0.81</td>
<td>0.71</td>
<td>0.64</td>
</tr>
</tbody>
</table>

**Table 3.5a:** Results of the MCA of the first three storage techniques

<table>
<thead>
<tr>
<th></th>
<th>Concrete</th>
<th>Ceramics</th>
<th>Natural stones</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy density [J/m³, +/−]</td>
<td>9.900E+07 0.14</td>
<td>1.212E+08 0.17</td>
<td>4.173E+08 0.59</td>
</tr>
<tr>
<td>Heat transfer capacity [+/-]</td>
<td>+ 0.80</td>
<td>+ 0.80</td>
<td>+ - 0.60</td>
</tr>
<tr>
<td>Applicability with steam [+/-]</td>
<td>+ 0.80</td>
<td>- 0.40</td>
<td>- 0.40</td>
</tr>
<tr>
<td>Investment costs [Mio €, +/-]</td>
<td>20 1.00</td>
<td>-- 0.20</td>
<td>-- 0.20</td>
</tr>
<tr>
<td>Proven technology [+/-]</td>
<td>+ - 0.60</td>
<td>- 0.40</td>
<td>-- 0.20</td>
</tr>
<tr>
<td>Stability [+/-]</td>
<td>+ - 0.60</td>
<td>+ - 0.60</td>
<td>- 0.40</td>
</tr>
<tr>
<td>Chemical compatibility [+/-]</td>
<td>++ 1.00</td>
<td>++ 1.00</td>
<td>++ 1.00</td>
</tr>
<tr>
<td>Control [+/-]</td>
<td>+ - 0.60</td>
<td>+ - 0.60</td>
<td>+ - 0.60</td>
</tr>
<tr>
<td>Total Industry score</td>
<td>0.70</td>
<td>0.45</td>
<td>0.40</td>
</tr>
<tr>
<td>Total Science score</td>
<td>0.63</td>
<td>0.55</td>
<td>0.56</td>
</tr>
</tbody>
</table>

**Table 3.5b:** Results of the MCA of the second three storage techniques

<table>
<thead>
<tr>
<th></th>
<th>Phase Change Material</th>
<th>Thermal Chemical Material</th>
<th>Salt-rock thermocline</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy density [J/m³, +/−]</td>
<td>3.954E+08 0.56</td>
<td>7.092E+08 1.00</td>
<td>+ 0.80</td>
</tr>
<tr>
<td>Heat transfer capacity [+/-]</td>
<td>+ 0.80</td>
<td>+ 0.80</td>
<td>+ - 0.60</td>
</tr>
<tr>
<td>Applicability with steam [+/-]</td>
<td>+ 0.80</td>
<td>- 0.20</td>
<td>+ 0.80</td>
</tr>
<tr>
<td>Investment costs [Mio €, +/-]</td>
<td>+ - 0.60</td>
<td>-- 0.20</td>
<td>-- 0.20</td>
</tr>
<tr>
<td>Proven technology [+/-]</td>
<td>+ - 0.60</td>
<td>+ - 0.60</td>
<td>+ - 0.60</td>
</tr>
<tr>
<td>Stability [+/-]</td>
<td>+ - 0.60</td>
<td>+ - 0.60</td>
<td>+ - 0.60</td>
</tr>
<tr>
<td>Chemical compatibility [+/-]</td>
<td>+ - 0.60</td>
<td>+ - 0.60</td>
<td>+ - 0.60</td>
</tr>
<tr>
<td>Control [+/-]</td>
<td>+ - 0.60</td>
<td>+ - 0.60</td>
<td>+ - 0.60</td>
</tr>
<tr>
<td>Total Industry score</td>
<td>0.65</td>
<td>0.50</td>
<td>0.48</td>
</tr>
<tr>
<td>Total Science score</td>
<td>0.67</td>
<td>0.72</td>
<td>0.50</td>
</tr>
</tbody>
</table>

**Table 3.5c:** Results of the MCA of the third three storage techniques
3.3 Thermal energy storage selection

The MCA results in score, which can be ranked per stakeholder. The next table shows the results.

<table>
<thead>
<tr>
<th>Rank</th>
<th>Science</th>
<th>Industry</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Two tank salt</td>
<td>Two tank salt</td>
</tr>
<tr>
<td>2</td>
<td>Salt-PCM-salt</td>
<td>Salt-PCM-salt</td>
</tr>
<tr>
<td>3</td>
<td>Concrete-PCM-concrete</td>
<td>One tank salt</td>
</tr>
<tr>
<td>4</td>
<td>Thermal Chemical Material</td>
<td>Concrete-PCM-concrete</td>
</tr>
<tr>
<td>5</td>
<td>One tank salt</td>
<td>Concrete</td>
</tr>
<tr>
<td>6</td>
<td>Concrete-steam vessel</td>
<td>Phase Change Material</td>
</tr>
<tr>
<td>7</td>
<td>Phase Change Material</td>
<td>Concrete-steam vessel</td>
</tr>
<tr>
<td>8</td>
<td>Steam vessel</td>
<td>Steam vessel</td>
</tr>
<tr>
<td>9</td>
<td>Concrete</td>
<td>Thermal Chemical Material</td>
</tr>
<tr>
<td>10</td>
<td>Natural stones</td>
<td>Salt-rock thermocline</td>
</tr>
<tr>
<td>11</td>
<td>Ceramics</td>
<td>Ceramics</td>
</tr>
<tr>
<td>12</td>
<td>Salt-rock thermocline</td>
<td>Natural stones</td>
</tr>
</tbody>
</table>

The most interesting TES technique for science would be the two tank salt storage. The most suitable technique for the industry is also the two tank salt storage. The biggest advantage for science is the good applicability with steam, the very high energy density and the good conductivity. For the industry the fact that this technique is commercially available and proven is very important. It is a relatively simple technique that serves its purpose.
The second best storage system is the salt-PCM-salt technique. For both science and industry this is the second best option. The biggest advantage is the fact that this technique is designed for steam storage during charging and steam generation during discharging. It is almost the same technique as the two tank salt storage except that for the storage of vapor phase of steam PCM is used.

The third most interesting technique is the concrete-PCM-concrete storage. For science it is the third best option and for the industry the fourth best option. This technique is comparable with the salt-PCM-salt technique but instead of salt the much cheaper concrete is used.

In the next three chapters these three storage techniques will be analyzed more in detail. In Chapter 4 the two tank salt storage will be discussed, in Chapter 5 the concrete-PCM-concrete technique and finally in Chapter 5 the salt-PCM-salt storage system. This chapter performed an analysis on the general properties of the storage system that were found in literature; the next three chapters will examine the performance more into detail with numerical models.
Chapter 4 Two tank salt storage

The two tank salt storage is introduced in Chapter 3. This part of the report will go more into detail about the storage technique. First the working principle of this technique in the combined heat and power plant will be explained. An overview of the components with the known parameters will be given and the properties of the fluids will be discussed. The second part will focus on the design of the storage system and gives a final design of the system. In the third part the numerical system model will be developed. The last part of this chapter will show the results of the application of the two tank salt storage in the combined heat and power plant. These results will be compared with analytical results. The goal is to find the total efficiency of this storage system and a proper view of the applicability.

4.1 System configuration

The two tank salt storage system consists of two storage tanks. Before the storage will be charged, one tank is completely filled with salt at its minimum temperature, denoted as the cold tank. For this situation commercial available HITEC heat transfer salt will be used. The properties of the salt can be found in Appendix B. The freezing point is 142°C but for security reasons the minimum allowed temperature is assumed to be 165°C. The cold salt will be heated up with the 90 bar superheated input steam by being pumped through three separate heat exchangers. Every single heat exchanger will be designed for optimal use for a particular phase of the steam: the desuperheating, condensation or the cooling down section. The charging cycle will take 8 hours. At the end of this cycle, all cold salt is pumped from the cold tank, through the three heat exchangers in series, to the so called hot tank. The heat exchangers will operate with the counter-flow principle. The figure below shows an overview of the total system:

![Figure 4.1: Charging the two tank salt storage](image-url)
Figure 4.1 shows that the temperature of the steam during charging of the system is declining. The condensation temperature is at 90 bar is 304°C. The input steam can be cooled down to the minimum temperature of the salt or higher. The temperature of the cold salt is 16°C before charging and is heated up. The cold salt enters the heat exchanger from the opposite side compared to the steam. The temperatures X1, X2 and X3 from figure 4.1 are unknown and need to be determined.

When the system switches to discharging, the flow directions of the salt and the steam are reversed. The left tank, completely filled with hot salt, is now being emptied and used to produce steam again. The biggest difference is the magnitude of pressure of the steam that needs to be produced. The demanded output is steam at 35 bar with a temperature of 350°C. The evaporation temperature of water under these conditions is 244°C. The overview is shown below in figure 4.2:

![Figure 4.2: Discharging the two tank salt storage](image)

### 4.2 Dimensioning the system

The total storage system consists of two tanks and three heat exchangers. The first step of designing the system is determining the needed mass flows of the steam for charging and the salt. The mass flow of steam needed for discharging is already known. Second, the design of the storage tanks will be discussed and last the dimensions of the heat exchangers will be determined.

#### 4.2.1 Mass flows

The two tank salt storage will be charged in eight hours but also discharged in eight hours. During discharging 20.28 kg/s of 35 bar superheated steam at 350°C needs to be generated. The goal is to determine the minimum input mass flow to realize this output. First the ratio between the mass flow salt and a steam input of 1 kg/s needs to be determined. This is done with the next formula:

$$\dot{m}_{st}(H_{st,500°C} - H_{st,170°C}) = \dot{m}_{sa}C_p(T_{hot} - T_{cold})$$  \hspace{1cm} (4.1)

where $\dot{m}$ is the mass flow, $H$ the specific enthalpy, $C_p$ the specific heat, $T_{hot}$ the temperature of the salt in the hot tank and $T_{cold}$ the temperature of the salt in the cold tank. Earlier in this chapter was
stated that the cold temperature of the salt is 165°C. It was also concluded that it is desirable to heat up the salt to a temperature of 350°C. The problem of this analysis is that the temperature difference between salt and steam is not taken into account. The energy of the steam will be transferred to the salt through a heat exchanger. In a heat exchanger it is only possible to exchange heat when there is a temperature difference between the two fluids. Assuming only the above equation, the ideal mass flow of salt would be 9.22 kg/s. This amount of salt is able to store all the energy of the superheated steam. The temperature-enthalpy graph would look like the following:

![Temperature-enthalpy graph during charging of the storage](image)

Figure 4.3: Temperature-enthalpy graph during charging of the storage

The problem in the graph above is the conflicting salt and the steam temperature. At a certain point where the steam temperature is 304°C, the temperature of the salt becomes higher. This is physically impossible. A so called pinch point (PP) is the minimum approach between the temperatures of the two fluids. The advantage of a low pinch point is a higher efficiency but the drawback is the need for a costly heat exchanger. The pinch point for this situation is set on 5°C, based on literature [30]. Knowing the pinch point, the actual mass flow of salt can be determined to store 1 kg/s of superheated steam.

\[
\dot{m}_{st,in} (H_{\text{sat, steam,304°C}} - H_{\text{water,170°C}}) = \dot{m}_{sa} C_{p,sa} ((304 - PP) - T_{\text{cold}})
\]

This results in a salt mass flow of 9.64 kg/s. The resulting hot temperature of the salt that can be reached is 342°C. Knowing the mass of salt needed to store one kilogram of superheated steam per second, the output can be determined. This is done in the same way as above, except for the pinch point that is now at saturated water level:

\[
\dot{m}_{sa} C_{p,sa} (T_{\text{hot}} - (244 + PP)) = \dot{m}_{st, out} (H_{\text{superheated,342}} - H_{\text{sat, water,244°C}})
\]

This results in a possible output of 0.69 kg/s of superheated 35 bar steam at 342°C. The analytical approach can now be summarized. In order to store the energy of one kg of superheated 90 bar steam of 500°C per second, 9.64 kg of salt per second is needed. The salt will be heated to a temperature of...
342°C and is able to generate 0.69 kg of superheated 35 bar per second with a temperature of 342°C. Using the ratio between the charging, salt and discharging mass flow, the actual required values can be determined. The demanded steam output for discharging is 73000 kg/hr so 20.28 kg/s. The next table shows the rest of the needed values.

<table>
<thead>
<tr>
<th>Charge flow [kg/s]</th>
<th>Storage material[kg/s]</th>
<th>Discharge flow [kg/s]</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.00</td>
<td>9.64</td>
<td>0.69</td>
</tr>
<tr>
<td>Actual</td>
<td>29.39</td>
<td>283.30</td>
</tr>
</tbody>
</table>

Table 4.1: Mass flows of input steam for charging, salt and output steam for discharging

### 4.2.2 Storage tank

During charging and discharging, salt is flowing in and out of the storage tanks. There is continuously salt present inside the tanks. Therefore the heat losses from the storage tanks need to be taken into account. The heat losses from the storage tank consist of internal losses from the salt to the vessel wall, losses through the insulation and external losses by convection and radiative heat transfer to the ambient. For this thermal energy storage situation, the salt tanks will be well insulated to prevent heat losses as much as possible. The thermal resistance of the insulation is much larger and the internal and external resistance can therefore be ignored [31]. The two tank salt storage will be modeled as a closed-loop system so there are no heat losses from the top salt layer in the tank. The heat losses through the insulation depend on the contact area between the tank and the salt but also on the amount and quality of the insulation material.

In order to determine the contact area between the tank and the salt, the dimensions of the storage tank need to be determined. The mass flow of salt is 286.3 kg/s during 8 hours. Therefore 8160 tons of salt is needed. To minimize the heat losses, the shape of the tank has to be optimized. The storage tank needs a volume of 4400m³ to store the salt mass, a height of 17.8m and a diameter of 17.8m. The detailed calculations can be found in Appendix C.

Knowing the dimensions, the decreasing temperature in time can be expressed as:

\[
\frac{dT}{dt} = -\frac{U \cdot A(t) \cdot (T_{hot} - T_a)}{M(t) \cdot C_p} \tag{4.4}
\]

The average heat losses \(U\) in a tank for salt are known from experiments at Solar Two and therefore assumed to be 0.22 W/m²K [32]. During charging and discharging of the storage tank, the salt mass \(M(t)\) in the tank changes in time so also the contact area \(A(t)\) between the salt and the tank changes in time. This results in heat losses that are not constant. The hot salt temperature \(T_{hot}\) is assumed to be 342°C, the ambient temperature \(T_a\) 20°C and the area is a summation of the wall and the bottom surface area. The maximum heat losses are 82 kJ/s and result in 1.56 GJ lost energy after eight hours. This would result in a temperature decrease of the salt of 0.12°C after eight hours, based on the salt properties of Appendix B. Since the decrease is so small, the heat losses from the storage tank will be neglected.
4.2.3 Heat exchanger

The most important part of this storage technique is the use of three heat exchangers that operate in counter flow. Compared to co-current, exchanging heat in counter-current flows is more effective because of the slowly declining temperature difference. In a co-current flow, the initial temperature gradient is higher but goes down very quickly and results in a less effective process. The total heat transfer in the heat exchanger can be represented by:

\[ q = U A \Delta T_{lm} \]  

(4.5)

In this equation \( U \) is the overall heat transfer coefficient. The overall heat transfer coefficient \( U \) is a function of the individual heat transfer coefficients of the two mass flows, the wall thickness, the diameter and the thermal conductivity of the tube and the fouling resistances. Equation (4.11) in paragraph 4.3.1 shows the total heat transfer coefficient. Since \( U \) is not a constant, the average temperature per heat exchanger is chosen to give the fluid properties a constant value. It is assumed that the heat exchanger is a double pipe heat exchanger. It consists of one pipe inside another larger pipe. The diameter of the small pipe is 0.2m and the large pipe has a diameter of 0.46m. Knowing the dimensions, the six different mean values for \( U \) can be determined. In equation (4.5) of the total heat transfer, \( \Delta T_{lm} \) is the logarithmic mean temperature difference and can be expressed as:

\[ \Delta T_{lm} = \frac{\Delta T_{tube} - \Delta T_{shell}}{\ln \frac{\Delta T_{tube}}{\Delta T_{shell}}} \]  

(4.6)

The logarithmic mean temperature difference can be determined with the temperature differences between the two flows per heat exchanger. The next important equation links the mass flows to the total heat transfer:

\[ q = m_{tube} c_{p,tube} \Delta T_{tube} = m_{shell} c_{p,shell} \Delta T_{shell} = U A \Delta T_m \]  

(4.7)

Now all variables are known except for the effective surface area per heat exchanger. This is the first important design parameter that needs to be set. A numerical iterative process is used to fulfill the prescribed temperature demands. It is important that the area is large enough to be able to exchange enough heat. There are three heat exchangers connected in series. The above explained method for determining the optimal area per heat exchanger results in six different values for the surface area but the largest surface area per section is chosen. This allows a sufficient surface area for exchanging heat continuously, during charging and discharging. In case the surface area of the heat exchanger is much larger than the idealized one, the heat transfer will occur faster.

<table>
<thead>
<tr>
<th>Surface Area [m²]</th>
<th>Liquid section</th>
<th>Two-phase section</th>
<th>Superheated section</th>
</tr>
</thead>
<tbody>
<tr>
<td>543</td>
<td>1676</td>
<td>623</td>
<td></td>
</tr>
</tbody>
</table>

Table 4.2: Surface area of heat exchangers for the three sections

During charging the condensing heat exchanger is over dimensioned and during discharging the ones for preheating and superheating. This results in a not fully optimized process. For example evaporation could occur earlier in the wrong heat exchanger and the same holds for condensation.
4.3 Numerical model full storage system

This part of the chapter will start with the mathematical model for the two tank salt storage. Secondly, the flow chart of the numerical model is shown. Since the three heat exchangers are connected in series and an analytical solution exists, the validation is performed for the total storage model. This is shown in the third part.

4.3.1 Mathematical model

The three heat exchangers in series will be modeled separately. The performance of every single heat exchanger is based on the energy equation. The next assumptions are made:

1. The inlet velocity of the steam/water is fully developed;
2. The tube wall is made of thin steel and the thermal resistance can be ignored;
3. The pressure drop of the steam through the pipe is neglected and therefore a constant boiling temperature is assumed;
4. The outer pipe is perfectly insulated and heat losses to the surroundings are therefore negligible;
5. The steam/water and salt velocity and temperature are radially uniform;
6. Conductive heat transfer is neglected;

The next figure represents the scheme that is used:

In order to take the temperature along the heat exchanger into account, the heat exchanger is segmented along the flow direction. The larger the number of segments, the more accurate the solution will be. The next equations are used to solve the model. The energy content of the fluid per segment can be expressed as:

\[ Q_{st}(j) = C_{p, st}(j) \dot{m}_{st} T_{st}(j) \]  \hspace{1cm} (4.8)

\[ Q_{sa}(j) = C_{p, sa} \dot{m}_{sa} T_{sa}(j) \]  \hspace{1cm} (4.9)

Subscript \( st \) represents the steam and subscript \( sa \) the salt. The specific heat of steam is temperature dependent and therefore also a function of the position. The amount of transferred heat per segment can be expressed as:
where \( n \) is the number of segments, \( F_1 \) the correction factor and \( U_1 \) the overall heat transfer coefficient. The correction factor can be applied in case more tubes or bent tubes are used to correct the changing flow patterns. The overall heat transfer coefficient is a function of the individual heat transfer coefficients of salt and steam \((h_{st}, h_{sa})\), the inner and outer diameter \((D_{in}, D_{out})\), the thermal conductivity of the tube \(k_{tube}\) and the fouling resistances \((R_{st}, R_{sa})\). It can be expressed as:

\[
\frac{1}{U_1(j)} = \frac{D_{out}}{h_{st}(j)D_{in}} + \frac{R_{st} D_{out}}{D_{in}} + \frac{D_{out} ln \left( \frac{D_{out}}{D_{in}} \right)}{2k_{tube}} + R_{sa} + \frac{1}{h_{sat}(j)}
\]

The heat transfer coefficient of the evaporation/condensation phase of the steam is different from the liquid or superheated phase of steam. For the liquid phase of steam, the superheated phase of steam and the liquid salt, the specific heat transfer coefficient can be determined by:

\[
h(j) = \frac{N u(j)k}{D}
\]

where \( k \) is the conductivity and \( D \) the diameter. The Nusselt number \( Nu(j) \) can be determined with the Dittus-Boelter relation:

\[
Nu(j) = 0.023 \left( \frac{Re(j)^{4/5}}{Pr(j)^{1/3}} \right)^N
\]

Here \( N \) is 0.3 for cooling and 0.4 for heating. The Reynolds number \( Re(j) \) and the Prandtl number \( Pr(j) \) change on every position since the specific heat and dynamic viscosity are a function of the changing temperature. As a result the Nusselt number changes in axial direction.

For the condensation/evaporation phase of the steam, a constant heat transfer coefficient is assumed. It is based on the equation from Shah (1979) and can be expressed as:

\[
Nu_{st}(j) = 0.023 \left( \frac{Re(j)^{4/5}}{Pr(j)^{1/3}} \right)^N \left[ 1 - x \right]^{0.8} \frac{3.8x^{0.76}(1 - x)^{0.04}}{P_r^{0.35}}
\]

It can be seen that a constant is multiplied with the Dittus-Boelter equation for the liquid phase. Variable \( x \) is the vapor fraction and \( P_r \) the reduced pressure. The average value of the multiplication factor depends on the reduced pressure and differs for charging and discharging. For charging, the average value of this constant is 3.5 and for discharging it is 4.7. This constant is applied on the Dittus-Boelter equation for determining the heat transfer coefficient. Since the heat transfer coefficient of the salt is much lower, it has little influence on the overall heat transfer coefficient of the heat exchanger in the two-phase section.

Combining equations 4.8, 4.9, 4.10 and 4.11, results in expressions for the new temperature for the next axial position. These new temperatures can be expressed as:

\[
dq(j) = \left( T_{st}(j) - T_{sa}(j) \right) \frac{U_1(j) \cdot A}{n} \cdot F_1
\]
\[ T_{st}(j + 1) = \frac{Q_{st}(j) + dq(j)}{c_{p,sa}(j) m_{st}} \]  
(4.15)

\[ T_{sa}(j + 1) = \frac{Q_{sa}(j) + dq(j)}{c_{p,sa}(j) m_{sa}} \]  
(4.16)

4.3.2 Flow chart

The numerical model of the full storage system consists of six mathematical models of every heat exchanging section. It starts with solving the eight hour charging part. First the cooling down, then the condensation and finally the desuperheating section is solved. The charging section is iterated until the prescribed temperature of the superheated steam input is reached. Then the eight hour discharging is solved in the same way. It uses the hot salt temperature that is reached during charging as an input for the discharging section. First the superheated section is solved, then the evaporation section and finally the preheat section. The discharging cycle iterates until the prescribed inlet temperature of the water is reached. Finally the eight hours of standby time are solved. The temperature of the cooled salt will be used in the next cycle as the input temperature of the cold salt. Figure 4.5 shows the solution procedure.
4.3.3 Numerical system model validation

The number of segments of the heat exchangers determines the accurateness of the solution. An infinite number of segments would come close to a continuous solution. Figure 4.6 will show the difference in number of segments. When the number of segments is increased for the charging section, the salt temperature has to approach the analytically determined value 342°C from paragraph 4.2.1.
The model for the heat exchanger is now validated. At least 500 segments are needed for a reliable solution.

4.4 Results

The numerical model neglects the heat losses of the storage tank and determines the performance of the heat exchangers. The next two figures show the results of the first storage cycle of 24 hours. The left figure shows the eight hour charging cycle and the right figure shows the eight hour discharging cycle.

It can be seen that the input steam is cooled from 500°C to 171°C and the salt heated from 165°C to 342°C during charging. During discharging, 35 bar steam of 342°C is generated with water of 60°C. The salt is cooled to 211°C during that cycle.
Now the energy that is extracted out of the input steam compared to the absorbed energy by the generated steam can be determined. The percentage of energy in compared to energy out is defined as:

\[ E1 = \frac{m_2 \left( H_{2,342°C} - H_{2,60°C} \right)}{m_1 \left( H_{1,500°C} - H_{1,171°C} \right)} \]  

(4.17)

where \( m \) is the mass flow of the steam and \( H \) the enthalpy of the corresponding temperature. This results in a percentage of 62%. The lost energy remains in the not fully cooled salt that has a temperature of 211°C at the end of eight hours discharging. It can be concluded that the solution is not stable yet. The second cycle uses the cooled salt temperature as an input for the charging cycle. The following results are obtained:

![Figure 4.8: Results of charging on the left and discharging on the right of the temperatures over the length of the heat exchanger during the second cycle](image)

It can be seen that the increased cold salt temperature has a serious effect on the heat capacity of the salt for charging. The input steam can only be cooled down to the condensation point of 304°C and has a vapor percentage of 95.5%. The salt heated to a temperature of 342°C again. The temperatures of discharging are the same as in the first cycle. Calculating the percentage of energy in compared to the energy out \( E1 \) results in 100%.

Knowing that the system now approaches a stable solution, the efficiency of the system can be determined. The practical efficiency \( E2 \) will be determined. It assumes that the cooled steam will be dumped and that the energy content is lost. This efficiency can be determined with the next formula:

\[ E2 = \frac{m_2 \left( H_{2,342°C} - H_{2,60°C} \right)}{m_1 \left( H_{1,500°C} - H_{1,60°C} \right)} \]  

(4.18)

The formula assumes that the 90 bar input steam can be cooled down to 60°C so all the available energy is utilized. The main problem is the fact that the input can only be cooled down to 304°C instead of 60°C. The resulting practical efficiency of the two tank salt storage is 62.4%. Not all the energy from the input steam is used and results in a low efficiency.
The advantages and disadvantages of the two tank salt storage can now be summarized. It is positive that the exit temperature of the salt is constant. The drawbacks of the system are that the needed temperature of the superheated 35 bar steam at the outlet cannot be reached and that the efficiency is low. An auxiliary heater would be needed to reach the needed temperature which decreases the efficiency. These conclusions will be used in Chapter 7 where the three different storage systems will be compared and the best technique is chosen.
Chapter 5 Concrete-PCM-Concrete storage

The second thermal energy storage option that will be discussed is the combination of concrete with phase change material (PCM). This chapter has the same structure as the former chapter except for the fact that now two materials are analyzed. It will start with general information about the storage technique and gives an overview of the set-up. The second part is divided in a concrete and a PCM part. For both materials, it will analyze the design of all components. In the third part the numerical system model is considered and finally the results will be discussed. The goal is to find the efficiency of the system and to conclude whether this technique is useful for thermal energy storage.

5.1 System configuration

The main problem of storing energy of superheated steam is the evaporation/condensation phase. During this process the temperature is constant and causes problems for exchanging heat since a temperature difference is needed. This is clearly shown in the former chapter where salt was used as a storage material. The next storage technique will store the energy of every phase of the water/steam cycle in a different material: the energy of the hot water inside the first block of concrete, the evaporating/condensing steam in a PCM and the superheated part in the second block of concrete. The set-up is depicted in figure 5.1 below.

![Figure 5.1: Overview of concrete-PCM-concrete storage during charging](image)

The figure above shows that during charging the steam enters the system from the right and flows through the first block of concrete. Then it enters the PCM from above where the steam condenses and the liquid water will fall to the bottom. The saturated water is further cooled in the second block of concrete on the left side. During discharging, water enters at the left and is preheated close to the
boiling temperature in the concrete. The preheated water will then be pushed in the PCM evaporator, where the water evaporates. The steam is then superheated in the concrete on the right side.

An important note here is the time dependency of the system. During charging the temperature of the concrete increases in time and during discharging it decreases. This will make the system design more complicated and the need for a numerical model higher.

## 5.2 Dimensioning the System

The storage system is a combination of two different storage materials. Firstly the design of the concrete and secondly the design of the PCM will be analyzed.

### 5.2.1 Concrete

The biggest drawback of concrete is that the material is not easy to charge because the conductivity is moderate. A big amount of heat exchanging tubes inside the material are needed to transfer the heat in a fast and effective way. The amount of concrete around a tube is an essential part of the design of the concrete storage.

Since the energy of the superheated steam and hot water are stored in separate concrete blocks, the dimensions need to be determined separately too. They can be optimized for both systems. The amount of concrete that is needed depends on the design of the storage module. The storage module is considered as parallel iron tubes where steam/water is pumped through. The tubes are surrounded by a concrete cylinder. The size, the arrangement and the distance between the heat exchanging tubes determine the amount of needed concrete. The next figure shows the storage module:

![Concrete Storage Module](image)

**Figure 5.2:** The left figure shows the intersection of the storage cylinder and the right figure the concrete storage module

In order to reduce computational time, only one cylinder will be analyzed in the numerical model. The next figure shows the storage unit:
Figurc 5.3: Concrete storage cylinder

The output steam demand is known and therefore also the amount of energy that needs to be stored. With the next formula, the volume of the needed concrete can be determined:

\[
\Delta t \cdot m_{\text{st, out}} (H_{\text{st, 350C}} - H_{\text{st, 244C}}) = V_c \cdot \rho_c \cdot C_p \cdot \Delta T_c
\]  

(5.1)

Here \(\Delta t\) represents time, \(m\) the mass flow, \(H\) the enthalpy, \(\rho\) the density and \(C_p\) the specific heat. The volume \(V_c\) of the concrete can be determined in case the temperature range \(\Delta T_c\) would be known. The problem is that this temperature differs throughout the concrete but also through time and can therefore not be predicted. In the multi criteria analysis in Chapter 3 a temperature difference of 40°C was assumed but now the numerical model will determine the exact temperatures.

The second important issue is the requirement that all the energy can be transferred to the concrete. The amount of needed power is:

\[
m_{\text{st, out}} (H_{\text{st, 350C}} - H_{\text{water, 244C}}) = n_{\text{total}} \cdot A_{\text{tube}} \cdot (T_{\text{steam}} - T_c)_{\text{min}}
\]

(5.2)

where \(A_{\text{tube}}\) is the contact area between the tube and the concrete, \(n_{\text{total}}\) the heat transfer coefficient from steam to concrete and \(n_{\text{tubes}}\) the number of tubes. Again the same problem occurs. The concrete temperature \(T_c\) and the steam temperature \(T_{\text{steam}}\) change through time and space and make it difficult to solve this problem analytically. In Appendix D an analytical approach is suggested with some assumptions. This could serve as a first guess of the dimensions in the numerical model. In the following part a numerical simulation model will be presented to be able to design the storage and analyze the performance.

5.2.2 Phase change material

The condensation energy of steam will be stored in a phase change material (PCM). Both the steam and the PCM will have a constant temperature during melting/condensing or solidification/evaporation and therefore a constant heat transfer.

The design of the storage module is comparable with the concrete model. One tube surrounded by PCM will be modeled. The composition of the storage material is an important design issue. During charging the condensation temperature of the 90 bar steam is 304°C and during discharging the evaporation temperature of the 35 bar steam is 244°C. For an optimal exchange of heat, the solidification/fusion temperature of the phase change material has to be in the middle of these temperatures. Therefore the inorganic salt ZnCl₂ is chosen with a melting temperature of 280°C as the PCM. This is also the initial temperature of the PCM.

Dimensioning the storage system is again very difficult because this problem is time dependent. During charging the PCM will absorb heat from the steam and undergoes a solid-liquid phase
change. This can be seen as a moving liquid front through the material in time. After the melting
enthalpy is reached, the temperature of the PCM will go up when more energy is absorbed.

5.3 Numerical model full storage system

In paragraph 5.2 the design of the concrete and PCM is discussed and some examples of charging
and discharging with random mass flows are shown. This paragraph will analyze the structure of the
mathematical model for both the materials. Since the model of the complete storage system is time
dependent, it is not possible to predict the results analytically. The mathematical models will be
linked to one complete system model in this part. Finally, the most decisive parameter will be varied
to optimize the storage and the dimensions will be given.

5.3.1 Concrete

This paragraph will analyze the concrete that will be used in the numerical model of the full storage
system. First a mathematical model is suggested and this model will be validated in the second part.
Some examples of the temperature distribution in the concrete will be given.

5.3.1.1 Mathematical model concrete

The numerical model considers one cylinder of concrete. The extra concrete between the cylinders is
not taken into account. Other assumptions to simplify the model are:

1. The inlet velocity of the steam/water is fully developed
2. In the first second the steam flows through the whole length of the cylinder
3. The tube wall is made of very thin iron so the thermal resistance can be neglected
4. The pressure drop of the steam through the tube is neglected
5. The concrete cylinder is perfectly insulated and thus the outer surface of the cylinder is
   adiabatic
6. The temperature distribution in the concrete cylinder is symmetrical around the axis and
   therefore the problem is two dimensional
7. The steam/water flow velocity is radially uniform

Based on these assumptions, the energy equation for concrete can be expressed as:

$$\rho_c c_{p.c} \frac{\partial T_c}{\partial t} = -k_c \left( \frac{\partial^2 T_c}{\partial r^2} + \frac{1}{r} \frac{\partial T_c}{\partial r} + \frac{\partial^2 T_c}{\partial z^2} \right)$$

(5.3)

The result is a two dimensional, non-linear, second order and time dependent problem in cylindrical
coordinates. It can be solved with a finite difference solution method. The discretization of the
equation is done with the following scheme:
The approximations of the second derivatives are:

\[
\frac{\partial^2 T}{\partial r^2}_{i,j} = \frac{T_{i+1,j} - 2T_{i,j} + T_{i-1,j}}{(\Delta r)^2} \quad (5.4)
\]

\[
\frac{\partial^2 T}{\partial z^2}_{i,j} = \frac{T_{i,j+1} - 2T_{i,j} + T_{i,j-1}}{(\Delta z)^2} \quad (5.5)
\]

The approximation of the first derivative:

\[
\frac{\partial T}{\partial r}_{i,j} = \frac{T_{i+1,j} - T_{i-1,j}}{2\Delta r} \quad (5.6)
\]

The differential equation (5.3) is solved with an explicit method so the new temperature can be determined by:

\[
T_{i,j}^{t+1} = \frac{-k_c}{\rho_c C_p,c} \left( \frac{T_{i+1,j}^t - 2T_{i,j}^t + T_{i-1,j}^t}{(\Delta r)^2} + \frac{1}{r} \frac{T_{i+1,j}^t - T_{i-1,j}^t}{2\Delta r} + \frac{T_{i,j+1}^t - 2T_{i,j}^t + T_{i,j-1}^t}{(\Delta z)^2} \right) dt + T_{i,j}^t \quad (5.7)
\]

It is assumed that the concrete is fully insulated so no heat losses occur on the outside boundaries except for the contact area between steam and concrete. The applied boundary conditions are:

\[
k_c \frac{\partial T_c}{\partial z} = 0, \quad R_{in} \leq r \leq R_{out} \quad \text{and} \quad z = 0; \quad R_{in} \leq r \leq R_{out} \quad \text{and} \quad z = L \quad (5.8)
\]

\[
k_c \frac{\partial T_c}{\partial r} = 0, \quad r = R_{out} \quad \text{and} \quad 0 \leq z \leq L \quad (5.9)
\]
The thermal conditions of the concrete that is in contact with the steam flow at \( r = R_{in} \) is determined with an energy balance equation. Figure 5.5 below is a close-up of the lower left corner of the nodal scheme and shows the convective steam at the concrete surface. The concrete at the bottom side is assigned a thickness that is half the radial distance between interior nodes.

Figure 5.5: Surface nodes with convection and conduction

The steam temperature distribution for the liquid and superheated phase along the axial direction \( T_{st} \) is determined with the energy conservation equation. It shows that the amount of energy transferred from steam to concrete, determines the energy content of the steam on the next axial position:

\[
\rho_{st} C_{p, st} \frac{\pi}{4} D^2 u \frac{\partial T_{st}}{\partial z} = h \pi D (T_{st} - T_c) \tag{5.10}
\]

Rearranging the equation results in an expression for the new steam temperature on the next axial position:

\[
T_{st}(z + dz) = \frac{h(z) A (T_{st}(z) - T_c(z))}{C_{p, st}(z) \bar{m}} + T_{st}(z) \tag{5.11}
\]

where \( A = \pi \cdot D \cdot dz \). The heat transfer coefficient \( h(z) \) changes per axial position but also in time. The Nusselt relations from equation (4.13) and (4.14) from Chapter 4 are again applied for this model.

For points at the interface between the steam and the concrete, a similar energy conservation equation can be expressed. The energy content of the concrete depends on the amount of transferred heat by convection of steam and the conduction further into the concrete. It can be expressed as:

\[
\rho_c C_{p, c} \frac{\partial T_c}{\partial t} = h A (T_{st} - T_c) + k_c A \frac{\partial T_c}{\partial r} \tag{5.12}
\]
where $V_j$ is the volume of concrete at the axial position. The steam temperature $T_{st}$ follows from equation (5.11).

The initial temperature of the concrete is assumed to be uniform through all the material:

$$T_c = T_{cr}, \quad R_{in} \leq r \leq R_{out} \quad \text{and} \quad 0 \leq z \leq L \quad (5.13)$$

The goal is to design a system model of the storage and therefore the relatively simple but computationally fast explicit method is chosen. Since the problem is time dependent and parameters change continuously, it is important that the steam percentages are monitored for every time step. The procedure for solving the concrete model can be found in paragraph 5.3.3 where the flow chart for the complete storage system is depicted.

**5.3.1.2 Validation and results concrete**

The cylindrical numerical model, suggested above, is difficult to validate because no analytical solution exists. The temperature distribution of the steam in axial direction and the conduction in the concrete in radial direction need to be validated.

First, the temperature distribution of steam over the axial direction needs to be validated. The temperature increase or decrease of steam also depends on the number of nodal points. The next figure shows the percentage of condensed steam in the first concrete storage section during the first seconds of charging with an increasing number of segments.

![Figure 5.6: Percentage of condensed steam over the axial position during the second of charging in the concrete block. The number of segments is increased and shows that the solution converges.](image)

The figure shows that the exiting steam percentage at 26m converges to a constant value when the number of nodes are increased. For the concrete-PCM-concrete storage 51 nodal points in axial direction will be used to determine the solution. With 51 nodal points an accurate solution will be obtained while keeping the computation time low.
In order to validate the heat transfer through the concrete, the model is adjusted to a one dimensional problem in Cartesian coordinates. The heat transfer in the originally radial direction is analyzed. The problem is reduced to a transient temperature distribution in a semi-infinite solid with surface convection. Steam of 500°C is constantly transferring heat to concrete while the concrete temperature increases. The next figure shows the situation:

![Figure 5.7: Semi-infinite 1D solid that is charged by surface convection](image)

Energy equation (5.3) reduces to a one dimensional situation and can be expressed as:

\[
\rho_c C_{p,c} \frac{\partial T_c}{\partial t} = -k_c \frac{\partial^2 T_c}{\partial x^2}
\]  

(5.14)

For points at the interface between the steam and the concrete, equation (5.12) changes to:

\[
\rho_c C_{p,c} \frac{\partial T_c}{\partial t} = hA(T_{st} - T_c) + k_c \frac{\partial T_c}{\partial x}, \quad x = 0
\]  

(5.15)

Since the validation model will be analyzed as a semi-infinite solid, the boundary condition of insulation is not applied. The analytical solution for this transient temperature distribution in a semi-infinite solid exists and can be expressed as [33]:

\[
\frac{T(x,t) - T_i}{T_{st} - T_i} = \text{erfc}\left(\frac{x}{2\sqrt{at}}\right) - \exp\left(\frac{hx}{k} + \frac{h^2at}{k^2}\right) \cdot \text{erf}\left(\frac{x}{2\sqrt{at}} + \frac{h\sqrt{at}}{k}\right)
\]  

(5.16)

Where \(T_i\) is the initial temperature of the concrete, \(\alpha\) is the thermal diffusivity and \(t\) the time. In figure 5.8 the temperature over position \(x\) is plotted after 3600 seconds of charging. The initial temperature of the concrete is assumed to be 300°C. The numerical result with an increasing number of nodal points is compared to the analytical result.
Figure 5.8: Numerical solution with increasing segments and analytical solution after 3600s for 1D convection-conduction problem.

This graph validates the numerical model and shows that at least 21 nodes in the radial direction of the concrete are needed to gain an accurate result.

Based on experimental research from the literature [15] and the validation, the dimensions and parameters of the concrete are assumed as follows:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Sign</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner tube radius</td>
<td>R_{in}</td>
<td>0.010</td>
<td>m</td>
</tr>
<tr>
<td>Radial thickness</td>
<td>t_t</td>
<td>0.05</td>
<td>m</td>
</tr>
<tr>
<td>Length tube</td>
<td>L</td>
<td>26</td>
<td>m</td>
</tr>
<tr>
<td>Number of tubes</td>
<td>n_{tubes}</td>
<td>11000</td>
<td>-</td>
</tr>
<tr>
<td>Specific heat capacity</td>
<td>C_p</td>
<td>1100</td>
<td>J/kgK</td>
</tr>
<tr>
<td>Density</td>
<td>\rho</td>
<td>2250</td>
<td>kg/m^3</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>k</td>
<td>1</td>
<td>W/mK</td>
</tr>
<tr>
<td>Initial temperature</td>
<td>T_i</td>
<td>300</td>
<td>°C</td>
</tr>
<tr>
<td>Nodes in radial direction</td>
<td>m</td>
<td>21</td>
<td>-</td>
</tr>
<tr>
<td>Nodes in axial direction</td>
<td>n</td>
<td>51</td>
<td>-</td>
</tr>
</tbody>
</table>

Table 5.1: Dimensions and parameters for concrete

To illustrate the thermal storage process of concrete, a charging mass flow of steam of 23.5 kg/s is assumed with the prescribed discharging mass flow as 20.28 kg/s. The next results show the temperature distribution of the concrete for different moments in time. The axial and radial axes are depicted with the element number. The actual radial height is 0.05m and the actual axial length is 26m. In the color bar on the right side the temperature can be read.
During charging:

![Temperature distribution concrete over time during charging with 500°C steam. Upper figure shows the result after 1 hour charging and the lower figure after 8 hours charging.](image)

The concrete temperature increases very fast in the bottom left corner. This is caused by the superheated steam that enters the storage at that position. In the beginning, the temperature difference between the steam and the concrete is very big and therefore the heat transfer is high and fast. The increased temperature is also responsible for the heat transfer in radial and axial direction by conduction. For example an hour later, the temperature difference between the lower left corner and the steam is much smaller. The steam will transfer its heat further along the axial direction.

The results of the temperature distribution through the concrete during discharging can be found in Appendix E. This numerical model of concrete can be used to analyze the total system. For the cooling down/preheating part, the same model with different parameters will be used. The phase change material model needs to be implemented too to determine the optimized storage dimensions. This will be done in the next part.

### 5.3.2 Phase change material

This paragraph will focus on modeling the PCM. The mathematical model is derived first and validated in the second part. Examples of charging the PCM will be given in the third part.

#### 5.3.2.1 Mathematical model PCM

The physical model and discretization scheme for concrete can also be used for the PCM storage. The same assumptions are applied on this part of the model. An extra assumption is that the heat transfer in the PCM is controlled purely by conduction. The convection plays a negligible role taking into account the strong viscosity of the PCM in the liquid phase.
The alternative form of the energy equation for two-dimensional heat transfer in the PCM can be written in terms of the enthalpy:

\[ \rho \frac{\partial H}{\partial t} = -k \left( \frac{\partial^2 T}{\partial r^2} + \frac{1}{r} \frac{\partial T}{\partial r} + \frac{\partial^2 T}{\partial z^2} \right) \]  (5.17)

The discretized derivatives can be found in equations (5.4), (5.5) and (5.6) from the concrete model in paragraph 5.3.1.1. The differential equation will also be solved explicitly. The solution procedure will follow the same steps as the concrete part but also needs one extra step to solve the problem. Since the enthalpy of a new time step is based on temperature derivatives, the enthalpy of the old time step also needs to be expressed in temperature. As stated before, the temperature of the PCM will only rise if it is in fully liquid state. Therefore the new enthalpy is lowered with the total heat of fusion and if this number is positive, a new temperature will be defined. When the enthalpy is negative, a temperature decrease will be defined. These conditions are stated in the next equations:

\[ H_{t+1}^{i+1} = H_{t+1}^{i+1} - \Delta H_{\text{melt}} \]  (5.18)
\[ H_{t+1}^{i} = H_{t+1}^{i} + C_{\text{PCM}}Q \]  (5.19)
\[ T_{t+1}^{i+1} = T_{t+1}^{i} - Q \]  (5.20)

Again, the steam percentage is monitored to assure proper condensation or evaporation. The procedure for solving the PCM model can be found in paragraph 5.3.1 where the flow chart for the complete storage system is depicted.

5.3.2.2 Validation and results PCM
For charging PCM, no analytical results exist for the two-dimensional situation in cylindrical coordinates. Therefore, the numerical model will be adjusted to a one-dimensional situation in Cartesian coordinates of which an analytical solution does exist. The new formulas can be derived in the same way as shown for the concrete in equations (5.14) and (5.15). The analytic solution for the position of the melting front as a function of time is:

\[ s(t) = \frac{2k_{\text{pcm}}(T_{\text{st}} - T_{\text{PCM,melt}})}{\rho_{\text{PCM}}\Delta H_{\text{melt}}} \cdot t \]  (5.21)

The position of the melting front \( s(t) \) indicates the point where the enthalpy of the PCM has reached the melting enthalpy of the material at time \( t \). At that point the phase of the material is fully liquid. The numerical model is also able to show the position of the melting front. The melting front moves through the material. After the material is melted, the temperature will go up. Plotting the results of the temperature over de radial position in the PCM from the numerical model and the position of the melting front of the analytical solution for different values of time results in the next graphs.
Figure 5.10: Comparison of the results of the numerical and the analytical model with a moving melting front in PCM. The left figure shows the result after 500 seconds and the right figure after 2000 seconds.

The numerical model shows a temperature gradient. In case the temperature is higher than 280°C, the material is melded. At the most left axial position where the temperature is 280°C, the melting front is located. The analytical result of the position of the melting front is shown with the red dot. It can be seen that there is a small difference between the analytical model and the numerical model. This can be explained by the fact that the analytical solution assumes that no more heat can be stored when the material is melted. The numerical shows the opposite. The melted material increases in temperature and stores more heat. The following PCM dimensions and parameters are used for the numerical model:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Sign</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner tube radius</td>
<td>( R_{in} )</td>
<td>0.010</td>
<td>m</td>
</tr>
<tr>
<td>Radial thickness</td>
<td>( t_i )</td>
<td>0.025</td>
<td>m</td>
</tr>
<tr>
<td>Length tube</td>
<td>( L )</td>
<td>75</td>
<td>m</td>
</tr>
<tr>
<td>Number of tubes</td>
<td>( n_{tubes} )</td>
<td>10000</td>
<td>-</td>
</tr>
<tr>
<td>Specific heat capacity</td>
<td>( C_p )</td>
<td>1470</td>
<td>J/kgK</td>
</tr>
<tr>
<td>Density</td>
<td>( \rho )</td>
<td>2907</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>( k )</td>
<td>0.5</td>
<td>W/mK</td>
</tr>
<tr>
<td>Initial temperature</td>
<td>( T_i )</td>
<td>280</td>
<td>°C</td>
</tr>
<tr>
<td>Nodes in radial direction</td>
<td>( m )</td>
<td>21</td>
<td>-</td>
</tr>
<tr>
<td>Nodes in axial direction</td>
<td>( n )</td>
<td>81</td>
<td>-</td>
</tr>
<tr>
<td>Melting temperature PCM</td>
<td>( T_{melt} )</td>
<td>280</td>
<td>°C</td>
</tr>
<tr>
<td>Melting enthalpy PCM</td>
<td>( H_{melt} )</td>
<td>75000</td>
<td>J/kg</td>
</tr>
</tbody>
</table>

Table 5.2: Dimensions and parameters for PCM

To illustrate the thermal storage process of PCM, a charging mass flow of steam of 26.39 kg/s is assumed with the prescribed discharging mass flow as 20.28 kg/s. The next results show the temperature distribution of the PCM after eight hours charging and after eight hours discharging with these mass flows.
Figure 5.11: Temperature distribution in PCM. Upper figure shows the result after eight hours charging and the lower figure after eight hours discharging.

The input steam enters the storage at 304°C on the left and exits at 280°C at the right during charging. The output steam enters the storage on the right at 244°C and exits at 280°C at the left after discharging. The temperature plots of the PCM above, clearly show a melting front in both figures. The material with a temperature of 280°C (dark blue in the upper figure, dark red in the lower figure) is in phase transition while a higher temperature indicates a liquid phase and a lower temperature a solid phase.

5.3.3 Flow chart

Knowing the mathematical models of concrete and PCM, the numerical system model of the full storage system can be designed. It first solves three time dependent problems: Concrete for desuperheating the input steam, PCM for condensation and concrete again for cooling down the water. First the steam temperature is determined, second the new heat transfer coefficient and then the conduction in the storage material is solved. This is done for all positions on the z axes per section. When all three sections are solved, the same is done for the next time steps. When eight hours of charging are solved, eight hours of discharging are solved in a similar way. The day cycle ends with eight hours of standby time. After solving these 24 hours, the next cycle can be performed. Figure 5.12 shows the flow chart of the numerical system model that is used.
Figure 5.12: Flow chart of numerical system model
5.3.4 Thermal conductivity

The results of the charging and discharging process are strongly dependent of the thermal conductivity. The next graph shows the difference in generated steam output after discharging for a thermal conductivity of concrete of 0.5 W/mK and 1.0 W/mK.

![Graph showing the difference in steam output for different conductivities](image)

**Figure 5.13:** Exit temperature of generated 35 bar steam for different conductivities

A higher thermal conductivity will lead to a more uniform temperature distribution along the radial direction of the concrete. Heat flows faster into the concrete and the steam is desuperheated in a shorter distance. During discharging the opposite occurs. Heat is extracted faster out of the concrete so the temperature of the steam increases faster. This will lead to a higher temperature rise of the steam along the flow direction and therefore a more efficient system. For the PCM the effect of a higher conductivity is even larger. The low conductivity of 0.5 W/mK of the material is a real restriction. In Appendix F the contour plots of the concrete and the PCM with a different conductivity are shown. The melting front moves much faster through the PCM with a higher conductivity. As discussed in Chapter 3, a maximum conductivity of 1.0 W/mK for concrete and 3.0 W/mK for PCM could be reached.

Increasing the conductivity has two major benefits on the storage system. First, the heat exchange is faster and therefore a higher power can be realized. Second, the thickness of the concrete or the PCM layer could be increased and the length of the tubes decreased, retaining the same volume of storage material. The result is a less expensive system because the heat exchanging tubes are costly compared to the storage material.

Finding the optimized dimensions is done by trial-and-error. The goal is to minimize the amount of surface area for the tubes. Table 5.3 gives the final design dimension, based on a conductivity of 3.0 W/mK. The inner tube radius, specific heat, density, melting enthalpy, initial temperature and number of nodal points are the same as shown in table 5.1 and 5.2 for concrete and PCM.
This part of the chapter will show the results of combining the concrete, PCM and concrete to one storage system. During the charging the mass flow through all the three components is equal and the same holds during discharging. The final design is based on the minimum needed input mass flow to realize the prescribed output of 73 tons of superheated steam per hour. The second important issue is the difference in number of cycles but this will be explained later. The storage has to operate for many years and needs to be able to run a big amount of cycles. The performance has to be stable.

For the concrete-PCM-concrete storage a numerical model is developed. The model assumes an initial temperature or enthalpy of every storage component at the beginning of the first cycle. After the first cycle of charging and discharging, these initial values have changed for the second cycle. These start-up effects need to be examined and the minimal needed input steam to generate 20.28 kg/s of 35 bar output steam has to be determined for a stable system.

The next graphs show the exit temperatures of the input steam after being cooled and the output steam after being superheated:

![Graphs showing exit temperatures](image)

**Figure 5.14:** Exit temperature of cooled 90 bar steam over time on the left and exit temperature of generated 35 bar steam on the right. Multiple cycles are plotted to show the differences.

The figure on the left shows an increasing steam exit temperature in time. While the concrete and the PCM are charged, the input steam is cooled less in time. Increasing the number of cycles results in an even higher exit temperature. The figure on the right shows that during discharging, the temperature of the generated steam is decreasing in time. In the next cycle the same occurs but at a higher temperature level. After running more cycles, the output steam temperature approaches the input steam temperature of 500°C. The needed temperature of the output steam is 350°C and
therefore the higher temperature steam will not be efficiently used. The input mass flow of 90 bar superheated steam that is needed to derive a stable solution is 26.9 kg/s. This value is found by trial and error. Decreasing the input mass flow results in unreached output requirements. An example is shown in Appendix G. The dimensions that are mentioned in paragraph 5.3.4 are used.

The efficiency can be determined by using the results of the last cycle of figure 5.13. They show the exit temperature of the 90 bar water after charging over time and the exit temperature of the generated 35 bar steam after discharging over time. The first important percentage that will be determined with the results is the amount of absorbed energy by the generated steam over the released energy by the cooled input. The mean steam temperature of the water exiting the charging cycle is 213°C. The mean temperature of the generated steam after discharging is 490°C. Knowing these mean temperatures, the amount of released energy through steam cooling during charging compared to the absorbed energy for steam generation during discharging can be determined with the next formula:

\[
E_1 = \frac{\dot{m}_2(H_{2.490} - H_{2.600})}{\dot{m}_1(H_{1.500} - H_{1.213})}
\] (5.22)

Here \(\dot{m}_1\) is the mass flow of the 90 bar steam input, \(H_1\) the specific enthalpy of the corresponding temperature at 90 bar, \(\dot{m}_2\) is the mass flow of the 35 bar steam output and \(H_2\) the specific enthalpy of the corresponding temperature at 35 bar. The percentage of used energy compared to the absorbed energy is 97%. This validates that the amount of energy that goes in also comes out. The reason that this efficiency is not 100% is caused by approaching the efficiency with the mean temperature, small deviations in steam parameters and the large amount of time steps and cycles that increase these deviations.

The second percentage is a more important value and indicates the practical efficiency of the storage system. It assumes that the cooled water after eight hours charging will be dumped and that the containing energy is lost. The second assumption is that the generated steam with a higher temperature than prescribed will be cooled to 350°C. The energy that is released will also be lost. This efficiency can be determined with the next formula:

\[
E_2 = \frac{\dot{m}_2(H_{2.350} - H_{2.600})}{\dot{m}_1(H_{1.500} - H_{1.600})}
\] (5.23)

The resulting practical efficiency of the salt-PCM-salt storage is 68.7%.

The numerical model is solved and conclusions can be drawn. The advantage of the concrete-PCM-concrete storage technique is that the required temperature of 350°C can be reached. The realized efficiency of 69% is low. Energy is lost because the input steam can only be cooled to 213°C instead of 60°C. Also the mean temperature of the generated 35 bar output steam is too high. The steam needs to be cooled to the prescribed temperature and the energy released is lost. The drawbacks of the system are the declining power output during discharging. The temperature of the generated steam depends on the time the storage has been discharging. This also results in a system that is not dynamic. It is not easy to suddenly increase the mass flows while maintaining the same output temperature.
Chapter 6 Salt-PCM-Salt storage

The third and last storage technique that will be discussed is a combination of the first and the second storage technique from Chapter 4 and Chapter 5. It contains liquid salt and a phase change material (PCM). Since the three separate storage technique are already extensively discussed in the previous chapters, this chapter will focus on the utilization of the different systems in one optimal storage system. First the general set-up will be discussed, second the design parameters and a further optimization, third the numerical model and finally the results will be given and discussed.

6.1 System configuration

The two other storage techniques that are discussed in the previous chapters have some drawbacks. The two tank salt storage experiences problems because of the pinch point. The combination of concrete and PCM is not able to deliver a constant temperature output which results in energy losses. The combination of these two techniques could be a good solution. Applying a PCM for the condensation/evaporation phase in combination with molten salt for the superheated steam and hot liquid water phase, would exclude the pinch point problem and could deliver a more constant temperature output. Another drawback of the fully liquid salt system is that it is not possible to generate the demanded steam temperature after discharging. The liquid salt mass flow through all three heat exchangers needs to be the same. This mass flow is too high to produce superheated 35 bar steam with a temperature of 350°C. In this chapter a solution for this problem will be adapted in the storage system. In order to reach the wanted salt temperatures in every heat exchanger, different mass flows for the superheated and the hot liquid section are necessary. Therefore a buffer tank will be added to the system which enables varying mass flows. In figure 6.1 an overview of the system during charging is given.

Figure 6.1: Overview of the Salt-PCM-Salt storage

During charging of the storage, superheated steam transfers its heat to the molten salt that is pumped through the desuperheating heat exchanger to the hot tank. The saturated steam is condensed in the PCM part where it melts. The saturated water is cooled by liquid salt that was stored in the cold tank and flows through the cooler. As depicted in figure 6.1, a part of the salt is pumped to the buffer storage. The decreased salt mass flow is able to increase fast in temperature and generate the
prescribed output temperature during discharging. This is caused by the difference in specific heat between steam and water. Discharging is realized by reversing the flow directions.

6.2 Dimensioning the system

The design of the storage is based on the previously obtained results from Chapter 4 and 5. Since this storage technique is a combination of earlier discussed materials, this chapter will only discuss the new design issues. More detailed information can be found in the previous chapters. The advantage of the liquid salt heat exchanger is the good controllability of the process and especially the achievable constant outlet temperature for every heat exchanger during discharging. A difficult design aspect for the new system is the replacement of liquid salt by PCM for the condensation/evaporation phase. It is important that this change has little impact on the performance of the storage. Therefore the outlet conditions of the steam exiting the PCM storage have to be as close as possible to the design temperature. The second important design issue is that this temperature is as constant as possible. The time dependency of the PCM is the limiting factor. During charging, the PCM needs to condensate saturated steam to saturated water of 304°C and during discharging it needs to evaporate saturated water at 244°C to fully saturated steam. In the first stage of the design, the PCM will be optimized to fulfill these demands. In the second part, the mass flows of the liquid salt will be determined.

6.2.1 PCM optimization

In order to create a proper model, the amount of charging/discharging cycles is very important. A constant output can only be realized when the solution becomes constant in time. Initially, the parameters of the PCM from table 5.3 in Chapter 5 will be used. For charging it is assumed that the inlet conditions of the PCM are saturated steam of 304°C and for discharging saturated water of 244°C. The next graph shows the solution of the steam temperature over time for the PCM per cycle.

![Figure 6.2: Steam fraction at the outlet of the PCM during discharging. The solution stabilizes after increasing the number of cycles to 15](image)
Figure 6.2 shows a constant solution after 15 cycles. For the next optimization steps, always 15 cycles are executed and the given solution is the result of the 15th cycle. Figure 6.2 also shows that the percentage of generated steam at the outlet of the PCM is declining in steps. This has the following explanation. After the PCM is charged, the temperature has increased and most of the PCM is in liquid phase. Generating steam in the first 100 seconds during discharging goes very well because the large temperature difference between the PCM and steam results in a good heat transfer. The temperature of the PCM will decrease in time and as a result less steam is produced. When the enthalpy of the first radial layer of segments of PCM that is the closest to the tube with steam will go below the melting enthalpy, the temperature remains constant. Then the heat transfer to the steam is constant and results in a constant percentage of generated steam. This can be seen in figure 6.2. This first layer also receives energy by the conductive heat transfer of the second PCM layer with a higher temperature. When the enthalpy of the second PCM layer approaches the melting enthalpy, the conductive heat transfer will go down and as a result the heat transfer from layer one to the steam also decreases. When the melting enthalpy is reached, the result is a constant heat transfer to steam again. This will continue for each layer in the PCM. These effects can be seen in figure 6.2 and look like sudden jumps in the generated steam percentage. They are caused by the discrete solution method.

In paragraph 5.3.4 from the previous chapter, the conductivity has been discussed. It was concluded that a higher conductivity results in a better performance. Therefore the highest achieved thermal conductivity of 3 W/mK from literature is used for this model. This conductivity is based on an applied graphite sandwich structure between the PCM. The next figure shows the temperature of the generated steam at the outlet during discharging using a different number of tubes.

![Figure 6.3: Steam fraction at the outlet of the PCM storage during discharging for a different number of applied tubes](image)

It can be seen that an increased amount of tubes results in a more constant output of the steam percentage. This is caused by the fact that the mass flow is lower and therefore it takes longer before the first layer of PCM reaches the melting enthalpy. For the next optimization step, 8000 tubes are used. In chapter five was explained that ZnCl₂ is used as a PCM. One of the problems of this material is the relatively low latent heat. The next figure shows the difference in performance when a higher latent heat of the storage material would be available.
Figure 6.4: Steam fraction at the outlet of the PCM storage during discharging for different values of the latent heat of the PCM.

Figure 6.4 shows that a higher latent heat results in a more constant steam outlet temperature. This part of the research will not focus on an existing PCM but on the best performance. The graph above shows that for a better performance of this storage technique, a PCM composition needs to be found with the highest possible latent heat. A maximum of 300 kJ/kg is assumed according to existing PCMs from literature [31]. The same is done by changing the radial thickness of the layer PCM around the tube with a latent heat of 300 kJ/kg. The next results are found:

Figure 6.5: Steam fraction at the outlet of the PCM storage during discharging for a different thickness of PCM material around a tube.

The graph above shows that a thin layer of PCM has a positive effect on the performance. This is caused by the fully insulated PCM and results in more concentrated heat close to the tube. The power output is therefore higher during discharging.
The above performed optimization can be further improved. A maximum of 12000 tubes is assumed and the minimal radial thickness of PCM is assumed to be 0.1m. The next values are found and will be used in the rest of the analysis:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Sign</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of tubes</td>
<td>$n_{tubes}$</td>
<td>12000</td>
<td>-</td>
</tr>
<tr>
<td>Heat of fusion</td>
<td>$H_{\text{melt}}$</td>
<td>300000</td>
<td>J/kg</td>
</tr>
<tr>
<td>Radial thickness</td>
<td>$t_t$</td>
<td>0.10</td>
<td>m</td>
</tr>
<tr>
<td>Tube Length</td>
<td>$L$</td>
<td>15</td>
<td>m</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>$k$</td>
<td>3</td>
<td>W/mK</td>
</tr>
</tbody>
</table>

Table 6.1: Optimized dimensions of PCM storage for numerical analysis

6.2.2 Liquid salt parameters

The efficiency of this storage needs to be as high as possible. The amount of energy that is needed for the generation of superheated 35 bar steam also has to be absorbed during charging. Cooling superheated 90 bar steam of 500°C to liquid water of 165°C provides 2.66 MJ/kg of energy and generating superheated 35 bar steam of 350°C with water of 60°C, needs 2.85 MJ/kg. Since 20.28 kg/s 35 bar steam needs to be produced, the input mass flow of steam should be at least 21.69 kg/s. The same assumptions and set-up of the liquid salt heat exchangers are used as in Chapter 4. The heat exchanger is assumed to be a double pipe heat exchanger.

After the numerical model is realized, the mass flows of the salt need to be optimized. Since the charging and discharging of PCM is time dependent, it is not possible to calculate the optimized mass flows analytically. The determined minimum mass flows from this paragraph will be used as input parameters in the model and optimized later. The minimum input parameters of the salt mass flows also need to be determined. This is done in a similar way as in Chapter 4 with equation (4.1). During discharging, salt will be cooled from 244°C to 165°C in the liquid water part. This results in a needed mass flow of 132.13 kg/s of liquid salt. For the superheated section, the salt will be cooled from 500°C to 244°C which equals a mass flow of 15.17 kg/s during discharging. The same mass flows are applied for the charging period.

6.3 Numerical model full storage system

The PCM and the liquid salt are both discussed earlier in this report. The applied mathematical model per section will be the same. However, the combination of the three sections results in a different system model. The liquid salt model is independent of time while the PCM model changes every time step. The result is a time dependent model of the total storage system. Another adjustment is the implementation of the buffer storage. Salt with a certain temperature is stored in the tank during charging and added to the salt flow during discharging. This results in a sudden temperature jump of the salt before entering the heat exchanger when these temperatures differ. Figure 6.6 shows the solution procedure:
Figure 6.6: Flow chart of numerical system model of Salt-PCM-Salt storage
The explicit method is used to determine the solution. The major difference with the two tank liquid salt model is the change in conditions during time for the liquid salt heat exchanger. Every time step a new iteration is needed to reach the right solution.

6.4 Results

The characteristics of the total storage system will be shown and discussed in this section. The optimized PCM is implemented in combination with the minimum mass flows from paragraph 6.2.2. Running the model and optimizing the mass flows of salt and steam will result in the required output conditions. The next values are used:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flow salt superheated part</td>
<td>52.45</td>
<td>Kg/s</td>
</tr>
<tr>
<td>Mass flow salt liquid part</td>
<td>106.13</td>
<td>Kg/s</td>
</tr>
<tr>
<td>Mass flow steam input</td>
<td>24.39</td>
<td>Kg/s</td>
</tr>
<tr>
<td>Mass flow steam output</td>
<td>20.28</td>
<td>Kg/s</td>
</tr>
<tr>
<td>Area Heat Exchanger superheated part</td>
<td>623</td>
<td>m²</td>
</tr>
<tr>
<td>Area Heat Exchanger liquid part</td>
<td>400</td>
<td>m²</td>
</tr>
</tbody>
</table>

Table 6.2: Mass flows and dimensions for Salt-PCM-Salt storage

The model is time dependent and the initial conditions change until a stable solution is reached. The next graph shows the temperature of the generated 35 bar steam over time for different cycles:

Figure 6.7: Temperature of the generated 35 bar steam at the outlet of the TES in time. In total 15 cycles are shown that reach a stable solution.

Figure 6.7 above shows 15 discharging cycles. After nine cycles the solution becomes constant. This result is used for the further analysis and always nine cycles are performed.
The first second of charging in the heat exchangers is depicted in the graphs of figure 6.8:

![Graph 6.8](image)

Figure 6.8: First second of charging the superheated section of the first heat exchanger in the left graph and the first second of charging the liquid section of the third heat exchanger in the right graph

And during discharging:

![Graph 6.9](image)

Figure 6.9: First second of discharging the superheated section of the first heat exchanger in the left graph and the first second of discharging the liquid section of the third heat exchanger in the right graph

In figure 6.9 a difference can be seen between the temperature of the salt that exits the first heat exchanger (left graph, blue line on the right side) and the temperature of the salt that enters the third heat exchanger (right graph, blue line on the left). This is caused by the buffer storage that adds a colder mass flow to the salt flow that comes from the first heat exchanger. Figure 6.1 shows the buffer storage in the system overview. It can also be seen that during discharging, steam is not fully superheated in the heat exchanger and evaporation occurs in the cooling heat exchanger. This is shown in the left and the right graph of figure 6.9. In Appendix H the graphs of the temperature of the hot salt and the cold salt during time are shown.

To verify the stability of the solution, the ratio is calculated of the amount of absorbed energy by the generated steam and the released energy by the cooled input. The mean steam temperature of the water exiting the charging cycle is 201°C. The mean temperature of the generated steam after discharging is 445°C. Knowing these mean temperatures, the amount of released energy through
steam cooling during charging compared to the absorbed energy for steam generation during discharging can be determined with the next formula:

\[
E_1 = \frac{m_2(H_{2,445C} - H_{2,60C})}{m_1(H_{1,500C} - H_{1,251C})}
\]  \hspace{1cm} (6.1)

Here \(m_1\) is the mass flow of the 90 bar steam input, \(H_1\) the specific enthalpy of the corresponding temperature at 90 bar, \(m_2\) is the mass flow of the 35 bar steam output and \(H_2\) the specific enthalpy of the corresponding temperature at 35 bar. The percentage of used energy compared to the absorbed energy is 101%. This validates that the solution is stable. The small difference of 1% is caused by approaching the efficiency with the mean temperature, small deviations in steam parameters and the large amount of time steps and cycles that increase these deviations.

The second percentage is a more important value and indicates the practical efficiency of the storage system. It assumes that the cooled water after eight hours charging will be dumped and that the containing energy is lost. The second assumption is that the generated steam with a higher temperature than prescribed will be cooled to 350°C. The energy that is released will also be lost. This efficiency can be determined with the next formula:

\[
E_2 = \frac{m_2(H_{2,350C} - H_{2,60C})}{m_1(H_{1,500C} - H_{1,60C})}
\]  \hspace{1cm} (6.2)

The resulting practical efficiency of the salt-PCM-salt storage is 75.7%.

The advantages and disadvantages of the salt-PCM-salt storage can now be summarized. The salt-PCM-salt storage is able to generate a relatively constant steam output during discharging. After seven hours discharging the reached temperature at the outlet decreases but the rest of the time it is constant. The system is easy to control because the mass flows through the steam/salt heat exchangers could be varied to meet specific output demands. The storage system is able to meet the prescribed demands while reaching a good efficiency.
Chapter 7 Thermal energy storage selection

In Chapter 4, 5 and 6 the performance of different storage techniques is analyzed with numerical system models. In this chapter a comparison between the three systems will be made. First the results per system will be summarized. Then the determined dimensions and performances will be compared and the best applicable storage technique will be chosen. The best storage system will be briefly further analyzed and a more realistic efficiency will be determined. In order to qualify the results and select the best TES system, the requirements are important. In table 7.1 below, the storage requirements from Chapter 2 are shown again:

<table>
<thead>
<tr>
<th>Steam inlet temperature for charging</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Desired outlet temperature cooled steam for discharging</td>
<td>500</td>
<td>°C</td>
</tr>
<tr>
<td>Steam inlet temperature for discharging</td>
<td>60</td>
<td>°C</td>
</tr>
<tr>
<td>Desired outlet temperature generated steam for discharging</td>
<td>60</td>
<td>°C</td>
</tr>
<tr>
<td>Prescribed mass flow steam for discharging</td>
<td>350</td>
<td>°C</td>
</tr>
</tbody>
</table>

Table 7.1: Requirements thermal energy storage system

7.1 Selection of best storage technique

The results of the stable solution of the three different storage techniques are summarized in table 7.2 below.

<table>
<thead>
<tr>
<th>Two Tank Salt Storage</th>
<th>Concrete-PCM-Concrete</th>
<th>Salt-PCM-Salt</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input mass flow steam [kg/s]</td>
<td>29.39</td>
<td>26.90</td>
</tr>
<tr>
<td>Output mass flow steam [kg/s]</td>
<td>20.28</td>
<td>20.28</td>
</tr>
<tr>
<td>Mean T cooled input steam [°C]</td>
<td>304</td>
<td>213</td>
</tr>
<tr>
<td>Mean T generated output steam [°C]</td>
<td>342</td>
<td>490</td>
</tr>
<tr>
<td>Practical efficiency [%]</td>
<td>62.4</td>
<td>68.7</td>
</tr>
<tr>
<td>Surface Area Superheated Section [m^2]</td>
<td>623</td>
<td>17970</td>
</tr>
<tr>
<td>Surface Area Evaporation Section [m^2]</td>
<td>1676</td>
<td>12566</td>
</tr>
<tr>
<td>Surface Area Preheat Section [m^2]</td>
<td>543</td>
<td>6283</td>
</tr>
</tbody>
</table>

Table 7.2: Summary of storage parameters for the three different thermal energy storage techniques

The practical efficiency assumes that the cooled input steam is dumped and the energy that the fluid contains is lost. The surface area is the total surface area of all the heat exchanging tubes in the storage system. In case of the two tank salt storage, every section contains one tube. For the concrete-PCM-concrete, the superheated section of the concrete contains 11000 tubes, the evaporation section of the PCM 5000 tubes and the preheat section of the concrete 10000 tubes. The salt-PCM-salt storage assumes one tube for the superheated and preheat section of the liquid salt heat exchanger and 12000 tubes for the evaporation section of the PCM.
The numerical models of the different thermal energy storage techniques showed some advantages but also some disadvantages for every system. These results are summarized in Table 7.3 below:

<table>
<thead>
<tr>
<th></th>
<th>Two tank salt storage</th>
<th>Concrete-PCM-</th>
<th>salt-PCM-</th>
</tr>
</thead>
<tbody>
<tr>
<td>Constant output</td>
<td>++</td>
<td>--</td>
<td>+</td>
</tr>
<tr>
<td>Controllability</td>
<td>++</td>
<td>--</td>
<td>++</td>
</tr>
<tr>
<td>Achieve prescribed output requirements</td>
<td>-</td>
<td>++</td>
<td>++</td>
</tr>
<tr>
<td>Efficiency</td>
<td>-</td>
<td>+</td>
<td>+</td>
</tr>
</tbody>
</table>

Table 7.3: Results of the performance of the three thermal energy storage techniques

Based on these performance results, the best applicable storage technique can be selected. An important note here is that the comparison of the techniques is not completely fair. The salt-PCM-salt storage is more optimized than the other three storage techniques. For example, a buffer tank is used in the salt-PCM-salt storage to increase the performance and controllability. At the two tank salt storage it is also possible to add storage tanks to the system. The problem is that two storage tanks are needed but as a result the prescribed output requirements would be met. A second example is the applied optimization of the PCM for the salt-PCM-salt storage. In the same way the concrete-PCM-concrete storage could be more optimized. The reason to justify the chosen approach is that it has no influence on the efficiency of the system.

The advantages and disadvantages of the different storage techniques in Table 7.3 show that the third model, the Salt-PCM-Salt storage, is the most promising technique. In Chapter 6 the results clearly showed that the demanded storage output requirements can be reached and a moderate efficiency of 75.7% can be obtained. It is very important that the chosen storage technique will be easy to control. In case the TES will implemented in the ELSTA plant, it has to be able to adapt to sudden changes. The salt-PCM-salt storage is able to do this while maintaining a good efficiency. The concrete-PCM-concrete storage has the drawbacks that it is very difficult to control and complex to predict the performance. The two tank salt storage is a very interesting technique but is not well applicable in combination with steam. The pinch point limits its efficiency. The salt-PCM-salt storage shows an good overall performance and is therefore selected as the best applicable storage technique.

### 7.2 Discussion

This part of the chapter will examine the possible improvements and determine the losses of the salt-PCM-salt storage that are not taken into account in the numerical analysis. First, some improvements are suggested, secondly the energy losses through the insulation are estimated, then the pressure drop in the tubes is calculated and finally the pump power is determined. This paragraph will finally determine a new efficiency of the storage system that is more realistic.

#### 7.2.1 Improvements

The determined efficiency of the Salt-PCM-Salt storage is 75.7% but improvements can be made. The biggest loss in the Salt-PCM-Salt technique is caused by the not fully cooled input steam. The input steam is cooled down to an average temperature of 201°C. This is caused by the limitations of the liquid salt material. The Hitec heat transfer salt that is used has a freezing point of 142°C. In
order to maintain a reliable storage system, the minimum allowed temperature of the cold salt is assumed to 165°C. In case a different storage material with a lower freezing point will be used, the storage would be able to achieve a higher efficiency. Another possibility is to use the not fully cooled input steam for different applications. In case this hot water could be used in for example district heating, the efficiency of the system would also go up.

7.2.2 Energy losses

The energy losses of the salt-PCM-salt storage consist of heat losses from the storage tanks and heat losses from the heat exchangers. The total storage system contains four storage tanks: One for the hot salt, one for the cold salt, a buffer tank and the PCM tank. In Chapter 3 the overall heat losses through the insulation of the tank were assumed to be 0.22 W/m²K. Based on these data, an ambient temperature of 20°C and the dimensions of the storage tanks, the total heat losses are determined to be 13.44 GJ over one cycle. The two applied heat exchangers also contribute to the total heat losses. The outer tube of the heat exchangers will transfer heat to the environment. Assuming 0.5cm glass wool insulation with a conductivity of 0.023 W/mK, the overall heat transfer coefficient is also 0.22 W/m²K. Based on this coefficient, the surface area of the outer tube and the ambient temperature, the heat losses are determined to be 9.00 GJ over one cycle. Summarizing, the total losses through the insulation are 22.44GJ. The formulas and detailed calculations can be found in Appendix I.

7.2.3 Pressure drop

During discharging of the storage, the generated steam has to fulfill some prescribed requirements. The severe restrictions do not allow large pressure deviations. When steam is generated by the TES, pressure losses occur and have to be taken into account. The pressure drop can be determined with the Darcy-Weisbach equation and the friction factor is based on the Colebrook equation. The formulas can be found in Appendix I. The next results are found:

<table>
<thead>
<tr>
<th>Pressure drop [bar]</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Superheated section</td>
<td>12</td>
</tr>
<tr>
<td>Evaporation section</td>
<td>Negligible</td>
</tr>
<tr>
<td>Preheat section</td>
<td>0.11</td>
</tr>
</tbody>
</table>

Table 7.4: Pressure drop Salt-PCM-Salt model per section

The negligible pressure drop in the evaporation section is the result of the high amount of parallel tubes. Steam in the superheated section experiences high pressure losses. Therefore it is advised to use more tubes in this section to split the mass flow in smaller flows. For example a shell and tube heat exchanger could be used but then the storage needs to be redesigned. Splitting the total mass flow in multiple mass flows results in different heat transfer coefficients. The pressure drop will decrease but the costs for the heat exchanger will go up. The tube diameter can also be decreased to increase the heat transfer. The new design will only influence the amount of needed contact surface area while the efficiency remains unchanged.

7.2.4 Pump losses

The TES system needs at least two pumps to run the storage: First a boiler feed pump to pressurize the water that is needed for discharging and to pump the pressurized water through the system and
second a positive displacement pump to pump the salt through the system. Both pumps will need approximately an engine power of 100kW. It is assumed that during charging and discharging both pumps operate at full power. The resulting energy consumption over 16 hours is 9.22 GJ.

7.3 Conclusion

The total summarized energy losses are 31.7 GJ. The efficiency can be recalculated by adding these losses to the input of equation 6.2 of Chapter 6. The resulting efficiency of the salt-PCM-salt system is 74.6%. The energy losses contribute to a decrease of the efficiency of 1.1%. On the other hand, the efficiency can be increased by using a different storage material or find an application for the hot water at the outlet of the charging cycle.
Chapter 8 Conclusions and recommendations

The objective of this research was to develop an appropriate model of high temperature thermal energy storage for a combined heat and power plant. Based on the results of the model, conclusions can be drawn whether the flexibility and the efficiency of the ELSTA plant can be increased. The conclusions of this research and recommendations for further research are summarized in this chapter.

Conclusions

The fundamental motivation of this research is the search for increasing the flexibility and the efficiency of the combined heat and power plant ELSTA. This report started with a data analysis of the ELSTA plant and showed the potential improvements. First of all, a suitable TES system can decrease the steam demand for the ELSTA plant by 18% and subsequently increase the flexibility of the plant. Secondly, it is possible to increase the gas turbine efficiency from 27% to 33% during the night. Thirdly, the total efficiency of the plant can be increased in case the storage efficiency is higher than the steam turbine efficiency of 35%.

The requirements of a suitable TES system are also based on the data analysis of the ELSTA plant. The storage system needs to be charged with superheated 90 bar input steam of 500°C. In order to utilize most of the input energy, the input steam has to be cooled down to the lowest possible temperature during charging. During discharging of the system, 20.28 kg/s of superheated 35 bar steam of 350°C needs to be generated. Pressurized water of 60°C will be used for the inlet. Both charging and discharging time is eight hours. The charging mass flow of the input will be minimized to reach the above formulated output. In that case the efficiency of the TES is the highest.

A literature study examined the possible TES techniques. Based on a multi criteria analysis, the three most interesting systems are chosen for further numerical analysis. The first technique is the two tank salt storage, the second the concrete-PCM-concrete storage and the third the salt-PCM-salt storage.

At first, the two tank salt storage is examined. This storage system uses an advanced liquid salt which has the drawback of a relatively high freezing temperature. It is not possible to cool the salt to a lower temperature than the freezing point because then the system will clog. As a result the minimum temperature of the cooled input steam is above the freezing salt temperature and not all energy can be extracted. The results of the numerical model show that during discharging a constant steam outlet temperature is maintained. Two drawbacks are that the required temperature of 350°C cannot be reached and a low efficiency of 62.4% is achieved. An auxiliary heater is needed to reach the demanded output temperature and the low efficiency is caused by the pinch point problem. During evaporation and condensation of the steam, the temperature is constant while the temperature of the salt is changing. The temperature of the salt will approach the steam temperature and as a result the heat transfer goes to zero. Not all heat can be stored in the salt and causes a low efficiency.
The second analyzed technique is the concrete-PCM-concrete storage and is a more complex system than the two tank salt storage. The initial conditions change every cycle and it takes nine cycles until the solution stabilizes. Charging/discharging the system is a time dependent process and therefore the power output is not constant. The conductivity of the material is the most important parameter of the storage. It has a major influence on the performance and the amount of needed storage material. The efficiency of the storage system is 68.7%. The losses are caused by the not fully cooled input steam and the temperature of the generated steam that is too high. The system is difficult to control because the performance depends on how long the storage has been used.

The third and last system is the salt-PCM-salt storage. This combination of salt from the first storage technique and PCM from the second storage system has the advantages of both the techniques. It is able to generate a relatively constant output and the prescribed requirements are met. An efficiency of 75.7% is reached and the system is easy to control. Based on these results this technique is selected as the best applicable storage technique for the ELSTA plant.

In the numerical model of the salt-PCM-salt storage assumptions are made that are incorrect. No energy losses were taken into account, pressure drop was neglected and the needed pump power was excluded from the efficiency. Including these losses in the efficiency, the overall efficiency of the storage will decrease to a more realistic value of 74.6%.

The overall conclusion of this investigation is that high temperature TES is recommended for the ELSTA plant. This investigation was purely based on the thermal performance of the storage system. It decreases the steam demand of the plant but it is not possible to temporarily stop the power production. Nevertheless the increasing future demand for more flexible power generation of ELSTA is fulfilled. Furthermore, the efficiency will be increased significantly. The TES is able to supply enough steam during the night to shut down a gas turbine. As a result the turbine efficiency will increase from 27% to 33%. The salt-PCM-salt storage shows the best performance in combination with the plant. The steam supply from the TES has an efficiency of 74.6% which is considerably higher than the steam turbine efficiency of 30%.

Recommendations

Numerical simulations of a salt-PCM-salt storage system showed the best results for a TES system in the ELSTA plant. The liquid salt part is modeled as a two-pipe heat exchanger but this will lead to a large pressure drop in the superheated section. It is recommended to redesign the liquid salt heat exchanger as a shell and tube heat exchanger and compare the results. Nevertheless the liquid salt storage system is commercially available and the technology has already been proven.

The PCM part of the storage is based on an average heat transfer coefficient of the steam during evaporation and condensation. For a more accurate model of this storage technique, the two-phase steam flow should be worked out more in detail with a more accurate heat transfer coefficient. Another important advice is to do a chemical analysis to find the best applicable phase change material. The numerical model assumes a non-existing material with a high latent heat. Problems in the stability of the PCM material can occur and need to be taken into account as well.
The PCM is stored in a tank together with heat exchanging tubes. Since the conductivity of PCM is low, it is assumed that the PCM is impregnated into a high conductive graphite structure. This structure increases the conductivity of the storage material but has only been proven on small scale. Special attention for this assumption is advised.

With the model recommendations in mind, some economical advice is also needed. The salt-PCM-salt model is optimized for the best thermal performance of the system. The cost implications are not taken into account. For example, doubling the number of heat exchanging tubes in the PCM results in a more constant output while the extra costs are not calculated. It is strongly recommended to find a balance between costs and performance. The salt-PCM-salt storage is recommended as a TES system in the ELSTA plant based on the thermal performance. It is recommended to compare the investment costs of the system with the extra annual revenues and decide whether the implementation is commercially interesting.

The next challenge for further research is the real-time implementation of the model. Based on the electricity price, the model has to determine whether the TES is switched on and at what workload. Mass flows can be varied to fulfill the changing demands and to operate optimally.
## Nomenclature

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Appendix A Argumentation for MCA, normalization and material properties

Argumentation multi criteria analysis

Two tank salt
Based on experience with solar plant Andasol I and the best applicable salt called HITEC Heat Transfer Salt [18], the following values for the criteria are determined. The high heat capacity of 1350 J/kg·K, the good average temperature dependent density of 1750 kg/m³ and the temperature range of 250°C result in a high energy density of 472.5 MJ/m³. The conductivity of the salt is 0.57 W/m·K which is moderate but because the liquid can be pumped through a heat exchanger, the heat transfer capacity can be very high. This technique is highly suitable with steam because with separated heat exchangers it is possible to produce the steam per phase in an easy way. Another important advantage is that this system is already proven. There currently are a lot of solar power plants operating with the two tank salt storage (Solar Two, Solar Tres, Andasol 1&2, Gemasolar). The investment costs of the salt storage are predicted to be 50 dollar/kWh in 2015 [19]. A rough estimation for the storage of 462 MW is therefore 25 million dollar what is very good. The salt is non-flammable, non-toxic but corrosive so advanced materials in the set-up are necessary. This is why it is chemically moderate compatible. Molten salts are also used in the chemical and metals industry so a lot of experience is available [7]. The biggest drawback is the high-risk of solidification [20]. The salt has a high freezing temperature of 142°C which makes the need for more storage material necessary. The superheated steam cannot be cooled to a lower temperature than the freezing temperature. It is also necessary to prevent a lower temperature than 142°C in the system. This is the reason why the stability is marked as moderate. The control is proven to be easy.

One tank salt
The one tank storage system has some similarity with the two tank system because the storage material is the same. Therefore the heat capacity, the heat transfer capacity, the density, the chemical compatibility and the applicability are the same. The total investment costs are predicted to be 35% lower when a combination with natural rock is used because of the lower costs for the filler material and only one storage tank is needed [13]. The literature does not mention costs for a one tank salt storage because this technique is not developed well enough. The costs are predicted to be lower because only one storage tank is needed. A lot of research is going on about this technique but it is not proven on large scale yet. The difficulty is its stability and keeping the thermoclyne at the same position all the time. Thermal ratcheting is the critical phenomenon that occurs in the tank [14] and makes the control difficult.

Steam vessel
Direct steam storage in a pressure vessel can be very interesting on small scale and for short storage [21]. The specific heat capacity is very high but the density of the pressurized steam is very low. Because of the direct storage there is no temperature range available to increase the energy density. To be able to determine the energy density, it is assumed that the 90 bar steam is first stored in a steam vessel at saturation temperature (304°C). The steam is released with a thermal expansion valve at a pressure of 35 bar. Since the process is isenthalpic, the released steam has a temperature of
244°C. With the difference in density it can be determined that using this system 84.36 MJ/m³ can be stored. This results in a minimum storage volume of almost 2.00E05 m³. It is impossible to apply this volume in one tank so a lot of smaller tanks are needed. This steam accumulator can only produce 35 bar steam of 244°C at the output. That is why an extra burner is needed to superheat the steam to the superheated level. This technique is proven on small scale but not on scale needed for this research. This is a real drawback because the pressurized vessels will be very expensive too. The heat transfer capacity is infinite because this is a direct storage technique. The applicability with steam good but not perfect because an auxiliary heater needed. The system is moderate stable because the steam is stored inside a pressurized and insulated tank but the reversible is not that good because during discharging the pressure goes down. It is chemically very compatible because steam is relatively harmless. The control of all the vessels will not be easy.

Concrete

High temperature concrete has a moderate specific heat capacity at 370°C of 1100 J/kg·K, a good density of 2250 kg/m³ but the allowed temperature difference is only 40°C. This small temperature range is due to the small allowed thermal expansion during charging and discharging. This results in a low energy density of 99.0 MJ/m³. It has a moderate conductivity of 1.30 W/mK but because the concrete is a fixed solid material it is not that easy to charge. A lot of heat exchangers inside the material are needed to transfer the heat in an effective way. This still results in a good heat transfer capacity. These numbers are based on recent research [22]. Adding the tubing register to the concrete makes the technique applicable for steam storage and generation. The costs for these heat exchangers are the most expensive part of the system because of the high amount of needed tubes. On the other hand the concrete is really inexpensive. The total costs of the system result in a low price of the storage. The predicted investment costs for the system are 20 million, based on estimation for the costs for such a system for the Andasol I power plant [2]. A drawback of the system is the fact that it just recently has been proven for temperatures up to 500°C but only on small scale. The system is moderate stable. A problem could be the risks for cracks in the concrete but former research already found good ways of preventing this. The decreasing power output during discharging is a real drawback. The system is chemically very compatible but the control is not that easy because of the time dependent temperature of the concrete. Other advantages are the ease of handling the material and the low degradation of the heat transfer [15].

Ceramics

The advantage of ceramics over concrete is that the conductivity is a little bit higher and the density too. The fact that ceramics are solid is again not favorable for the heat transfer capacity. The specific heat capacity is relatively low and the possible temperature difference is also 40°C because of the small allowed expansion of the material. The biggest disadvantage is the high costs for the material. It has not been applied in combination with steam yet. Ceramics are used in blast furnaces and these bricks are exposed to hot air. Changing the material and implementing tubes in the ceramics will be a very expensive solution. The advantages of ceramics are the chemical stability and the good chemical compatibility. A disadvantage is the instability during discharging because the generated power will decrease. Because of the severe restrictions of the output this is a real issue. The risk for cracks in the material is very low [23]. Because of the lack of experience with this technique it is difficult to evaluate the control so it is assumed to be moderate.

Natural stones
The heat capacity of for example fused silica pebble is 745 J/kg·K and its density is 2800 kg/m³ [24]. During charging stratification in the tank will arise. The difference with concrete and ceramics is that the thermal expansion difference between the rock and the heat exchanger tubes has influence on the performance. The temperature difference between charging and discharging is therefore estimated on 200°C and leads to a high energy density of 417.3 MJ/m³. The conductivity is very high, 2.00 W/mK, but this number is not representative when this system is used as a storage material. When the natural stones storage is implemented in a CHP plant, it will only have the mentioned conductivity during charging. The superheated steam will condensate by flowing through the pebble and exchanging its heat to the rocks. During discharging, cold water flows through tubes that are implemented in the storage tank but experiences much less of the conductivity with the pebble. Therefore the heat transfer capacity is evaluated as moderate. This is also the reason why this technique is not well applicable with steam. The investment costs will be very high because a special tank is needed that reduces the back pressure which is build up by charging the system. Also adapting the heat exchanger pipes into the complex tank is new and therefore likely to be expensive [9]. The technology is very interesting in combination with hot air but less with superheated steam. This is probably the reason why the technique is not proven yet in combination with superheated steam. The advantage of the system is the chemical compatibility. The stability is not optimal because of the difference in conductivity between charging and discharging. The amount of power also reduces during discharging, which is another disadvantage. The control is estimated to be moderate.

### Phase Change Materials

The Phase Change Materials, especially with the solid-liquid phase change, are very promising. Their biggest advantage is that they have an enormous heat capacity. For example NaN₃ has a heat of fusion of 175 KJ/kg during phase change but also has a high density. The specific heat capacity is 1500 J/kg·K in solid state and 1700 J/kg·K in liquid state around the melting temperature. Assuming a temperature difference from 10°C below melting temperature to 10°C above melting temperature, the energy density is 395.4 MJ/m³. The condensation temperature of the input steam during charging is 304°C and the evaporation temperature of the output steam during discharging is 244°C. For an optimal heat transfer, the melting temperature of the PCM should be exactly between these temperatures. A PCM with a melting temperature of 274°C is therefore needed. The biggest drawback is the low conductivity that leads to a low heat transfer capacity during charging and discharging. This can be overcome by adapting a so called sandwich-concept within the heat exchanger material. The tubes where the HTF flows through contain graphite plated fins, mounted to the vertical axis. The fins are flat plates that are mounted on the tube, next to each other [25]. This structure will increase the conductivity from 0.50 W/mK to approximately 3.00 W/mK [26]. The heat transfer capacity is therefore rated as high. The PCM will not move what makes the need for a lot of heat exchanging tubes high. The costs for the PCM are 9.5 $/kWh what leads to the relatively low investments costs of 4.5 million dollar for the storage material [27]. The costs for the storage tank, the heat exchangers and the sandwich structure are relatively high. The total investment costs are therefore estimated as moderate. The advantage of using PCM is that during charging or discharging the temperature of the storage material is constant. Using this property, a constant power output can be created. The only instability is caused by the chance for phase separation. The stability of the material is therefore evaluated as moderate. Another drawback is that the PCM material can
be corrosive. That is why the chemical compatibility is marked as moderate. The set-up is easily reversible because changing from charging to discharging is done by just changing the direction of the flow. The controllability is difficult to predict but assumed to be moderate.

**Thermo Chemical Materials**

Thermo chemical storage is a relatively new technique where a lot of research is going on about. It has the promise to be a very effective storage material because of its very high energy density. Literature mentions experimental performances up to 709.2 MJ/m³ [28] but calculations of the theoretical performance even show that energy densities far above one GJ/m³ are possible with carbonates and hydroxides. The conductivity of the TCM is relatively low but can be increased when a good heat exchanger is used. Increasing the porosity of the material also results in a better heat transfer. This is why it is evaluated with a high heat transfer capacity. Since there is not much known about large scale storage with TCM, the applicability for steam storage and steam production is evaluated almost the same as phase change materials. The principle of storing the heat is comparable. Both have a constant temperature at which the energy is stored. This is an important advantage for the applicability with steam. It is evaluated as moderate because this is all based on an assumption. The investment costs would be very high. The costs for further development and the complex system would cost most of the money. Another advantage of this technique is that there are no storage losses. If the reaction products of the endothermic reaction are separated and stored, almost no energy is lost until they are combined again. A drawback is the instability of the material. This results in an moderate evaluation of the stability. The chemical properties depend on the material that is used but a lot of them are corrosive so the chemical compatibility is evaluated as moderate [12]. The control of this system is easy because unwanted side reactions are not possible.

**Salt-rock thermocline**

The salt-rock thermocline has the advantage over for example the one tank molten-salt system that a part of the filler material is cheap natural rock [5]. This will make the total cost price for the system cheaper while the principle is the same. The density and specific heat capacity depend on the ratio between the salt and the rock. It is clear that both are high. The biggest drawback is that two times heat is exchanged for charging but also for discharging; first from the steam to the salt, then from the salt to the rock. This will reduce the efficiency because this influences the heat transfer capacity. This is probably the reason, according to the literature, why this system is not used in combination with steam. The system also needs an extra storage for the molten-salt because steam is used as the HTF in the CHP plant. This results in extra costs. The thermal behavior and efficiency of these systems under different operating conditions is not well understood. Also the temperature in the tank needs to be controlled well to maintain good stratification in the tank. This requires a controlled charging and discharging procedure [29].

**Concrete-steam vessel**

This technique stores the energy of the superheated phase of steam in the concrete. The remaining saturated steam will be stored in a steam accumulator at a pressure of 35 bar, at a temperature of 244°C. Compared to solar power plant PS10, 23 times more steam accumulators are needed. This equates to a total volume of 1.38E04 m³, so 92 separated tanks. Pressurized tanks are very expensive but the concrete is not. The stored saturated steam has a high heat capacity of 4798 J/kg K at a pressure of 35 bar but the density of 807 kg/m³ is relatively low. The steam is directly stored until the pressure and temperature reach 35 bar and 244°C. The energy density is based on the enthalpy
under these conditions and therefore no temperature range is available for this storage. Only a small part of the energy is stored in the concrete so that heat capacity and density is less important. The resulting energy density is therefore evaluated as low. This technique has only been proven numerically [15]. It is very well applicable with steam. The heat transfer of direct storage is perfect and of steam to concrete good so the heat transfer capacity is marked as very good. The stability is low because it is not possible to produce a constant output during discharging with the steam accumulator but the chemical compatibility is very good. The control is difficult because of the decreasing power output during discharging.

**Salt-PCM-salt**

The salt-PCM-salt uses a combination of the two tank salt storage and a PCM. The PCM stores the largest part of the energy so the energy density of the system is the same as for the PCM. The heat transfer capacity is marked as very high because both the two tank salt and the PCM have a good heat transfer capacity. This technique is specially designed to store and produce steam so it is very applicable for this research. The two tank salt system is commercially available and the PCM performance has been proven on small scale [25]. Therefore the criterion of proven technology is marked as good. The total costs for the system are moderate because the two tank salt storage is relatively cheap and the PCM expensive. The stability of the system is moderate because the salt part will generate a constant output while the PCM has a decreasing power outlet. The chemical compatibility is moderate and the controllability very good, based on the averages of the two tank salt and the PCM.

**Concrete-PCM-concrete**

This combination uses two different storage techniques but the energy density can be assumed to be the same as PCM because that part saves the biggest part of the energy. The same holds for the heat transfer capacity so it is assumed to be high. This technique is also specially designed to store and produce steam so the applicability to this problem is very good. The performance of the needed PCM is proven on small scale [25]. A big advantage is that the operating temperature of the PCM will be at saturation temperature of the 90 bar steam; 304°C. The investment costs will be moderate because the PCM system is expensive but the concrete is not. The stability is moderate because the power of the concrete reduces during discharging. Since most of the energy is stored in the PCM, the effect is less compared to just concrete. The chemical compatibility is assumed to be good too because of the use of the harmless concrete in combination with possible corrosive PCM. The test set-up from former research has proved the easy control.

**Normalization criteria**

The plus and minus signs are evaluated as follows: ++ is 1, + is 0.8, + - is 0.6, - is 0.4 and - - is 0.2. For the energy density the maximum value of 7.092E+08 J/m³ is the maximum value and evaluated as 1. The minimum is zero and also evaluated as 0. The rest of the values can be found by interpolation. For the investment costs it works the other way around. The lower the costs, the better it is. This is why the lowest investment costs of 20 Mio are evaluated as 1. The maximum costs are set at 80 Mio € and evaluated as 0.2. The other values can be found by interpolation again. In this normalization it is not possible to be evaluated as zero. Only the multiplication with the weighting factor can result in a value of zero. This is only the case when a criterion is not important at all and does not have to be taken into account.
### Material parameters

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Appendix B Properties of Hitec heat transfer salt

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*Figure B.1: General parameters Hitec salt*

Dynamic viscosity

*Figure B.2: Viscosity over temperature Hitec salt*

Density

*Figure B.3: Density per temperature of Hitec salt*
Appendix C Tank dimensions

In order to minimize the heat losses, the shape of the tank has to be optimized. Figure C.1 shows the graph of a storage tank with a volume of $4400\,m^3$.

![Graph of storage tank](image)

**Figure C.1:** Ratio between area and volume of the storage tank over the diameter/height ratio

When the diameter is smaller than the height, the area of the tank is much larger. The same holds for the case when the diameter is larger than the height. To minimize the losses through the insulation of the tank, the diameter has to equal the height. This results in a diameter and height of 17.8m.
Appendix D Analytical approach dimensions concrete

The surface area of concrete around the tube is:

$$A_c = \frac{\pi}{4} (D_{\text{out}}^2 - D_{\text{in,out}}^2) \quad (D.1)$$

The volume of concrete per tube is:

$$V_{c,\text{tube}} = A_c \cdot L \quad (D.2)$$

The output demand is known. Therefore the concrete storage will be designed to fulfill the demand for output steam. For the superheated steam section, the parameters can be determined with the next equation for the total amount of energy:

$$t \cdot \dot{m}_{\text{steam,out}} (H_{\text{steam,350C}} - H_{\text{water,244C}}) = V_c \cdot \rho_c \cdot c_{p,c} \cdot \Delta T \quad (D.3)$$

The number of needed tubes can be determined with:

$$n_{\text{tubes}} = \frac{V_c}{V_{c,\text{tube}}} = \frac{t \cdot \dot{m}_{\text{steam,out}} (H_{\text{steam,350C}} - H_{\text{water,244C}})}{\rho_c \cdot c_{p,c} \cdot \Delta T} \cdot \frac{4}{\pi \cdot (D_{\text{out}}^2 - D_{\text{in,out}}^2) \cdot L} \quad (D.4)$$

It is important that it is possible to exchange all the heat of the steam to the concrete. The amount of power that is needed is:

$$q = \frac{\dot{m}_{\text{steam,out}} (H_{\text{steam,350C}} - H_{\text{water,244C}})}{n_{\text{tubes}}} = U_{\text{total}} \cdot A_{\text{tube}} \cdot (T_{\text{steam}} - T_c)_{\min} \quad (D.5)$$

where $A_{\text{tube}} = \pi \cdot D_{\text{in}} \cdot L$ and $(T_{\text{steam}} - T_c)_{\min} = 3$

For the overall heat transfer coefficient, the steam transfers its heat to the first layer of concrete around the tube. The overall heat transfer coefficient can be expressed as:

$$U_{\text{total}} \cdot A_{\text{layer1}} = \frac{1}{\frac{1}{h_{\text{in}}} A_{\text{tube}} + \frac{R_1}{A_{\text{tube}}} + \frac{\ln \left( \frac{D_{\text{in,out}}}{D_{\text{in}}} \right)}{2 \pi k_{\text{tube}} L} + \frac{\ln \left( \frac{D_{\text{layer1}}}{D_{\text{in,out}}} \right)}{2 \pi k_c L}}$$

where $A_{\text{layer1}} = \pi \cdot D_{\text{layer1}} \cdot L$

Filling in the surface areas results in:
$$U_{total} = \frac{1}{\frac{D_{layer_1}}{h_{in}D_{tube}} + \frac{D_{layer_1} R_1}{D_{tube}} + \ln\left(\frac{D_{in}_{out}}{D_{in}}\right) \frac{D_{layer_1}}{2 k_{tube}} + \ln\left(\frac{D_{layer_1}}{D_{in}_{out}}\right) \frac{D_{layer_1}}{2 k_c}}$$ (D.7)

This results in:

$$\rho_c \cdot c_{pc} \cdot \Delta T \cdot \left(\frac{D_{out}^2 - D_{in}_{out}^2}{h_{in}D_{tube}} + \frac{D_{layer_1} R_1}{D_{tube}} + \ln\left(\frac{D_{in}_{out}}{D_{in}}\right) \frac{D_{layer_1}}{2 k_{tube}} + \ln\left(\frac{D_{layer_1}}{D_{in}_{out}}\right) \frac{D_{layer_1}}{2 k_c}\right) = (T_{steam} - T_{c})_{min}$$ (D.8)

The unknown L has fallen out of the equation and the only unknowns left are the diameters ($D_{out}$, $D_{in}_{out}$ and $D_{in}$), the temperature difference of concrete between charging and discharging ($\Delta T$) and the minimum temperature difference between steam and concrete ($T_{steam} - T_{c})_{min}$). The dimensions of the inner diameter of the tube and its wall thickness can be picked from the literature. If the temperature difference of the concrete between charging and discharging is known, the outer diameter of the concrete can be determined. The next graph shows the requirements of the storage system. During charging /discharging the temperature of the concrete will increase/decrease during time. The more mass flow of steam that passed the concrete, the higher or lower the temperature will get. It is important that the demands for steam are met. Steam from 500°C has to be cooled down to 304°C during charging and steam from 244°C had to be superheated to 350°C during discharging. Since the temperature of the concrete is not constant during time, the steam temperature outputs are also changing. The first assumption is made that at the end of the discharge cycle and at the end of the charge cycle, the above mentioned demands are met. This means that during charging for example, the superheated steam is cooled down further than 304°C and in the beginning of charging, superheated steam of more than 350°C is produced. The second assumption is made that the minimum temperature difference between concrete and steam ($T_{steam} - T_{c})_{min}$ is 10°C. The last assumption is made that the temperature difference between charging and discharging of the concrete ($\Delta T$), is 40°C. The next graph shows the requirements at the end of the cycle, assuming constant temperature slopes:
This will result in the amount of needed concrete to realize all the heat transfer with a certain minimum temperature difference between steam and concrete.

The assumed inner tube diameter from literature is 0.02m with a wall thickness of 0.005m for the superheated section. Then the total volume of concrete needed is 1763m³. This is equal to a total tube length needed of 675030m. In literature, concrete blocks with a length of 23m are used so this would result in 29346 tubes. Assuming a single concrete storage module of 4 by 2.6m would result in 12 concrete storage modules. Because the tubes and the concrete around the tubes are round and the concrete storage module is a square block, some concrete is not taken into account in the analysis.

The analysis above is done analytically. It is based on looking at the end of the charging and the end of the discharging cycle. It is important that at the end of the cycle it is still possible to produce steam. In this analytical analysis it is not easy to take the whole situation into account. Therefore the numerical model is built. Based on the above design parameters, the concrete storage for the superheated section can be analyzed by a numerical model. The results can be found in the main text.
Appendix E Example temperature distribution concrete for discharging

Figure E.1: Temperature distribution concrete over time during discharging. Upper figure shows the result after 1 hour discharging and the lower figure after 8 hours discharging.
Appendix F Results concrete-PCM-concrete with different conductivities

Results of charging and discharging concrete with different conductivities

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Sign</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner tube radius</td>
<td>$R_{in}$</td>
<td>0.015</td>
<td>m</td>
</tr>
<tr>
<td>Radial thickness concrete</td>
<td>$t_i$</td>
<td>0.05</td>
<td>m</td>
</tr>
<tr>
<td>Length tube</td>
<td>$L$</td>
<td>26</td>
<td>m</td>
</tr>
<tr>
<td>Number of tubes</td>
<td>$n_{tubes}$</td>
<td>11000</td>
<td>-</td>
</tr>
<tr>
<td>Specific heat capacity</td>
<td>$C_p$</td>
<td>1100</td>
<td>J/kgK</td>
</tr>
<tr>
<td>Density</td>
<td>$\rho$</td>
<td>2250</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>$k$</td>
<td>Differs, see graphs</td>
<td>W/mK</td>
</tr>
<tr>
<td>Initial temperature</td>
<td>$T_i$</td>
<td>300</td>
<td>°C</td>
</tr>
<tr>
<td>Nodes in radial direction</td>
<td>$m$</td>
<td>21</td>
<td>-</td>
</tr>
<tr>
<td>Nodes in axial direction</td>
<td>$n$</td>
<td>51</td>
<td>-</td>
</tr>
</tbody>
</table>

Table F.1: Dimensions and parameters of concrete used for the next graphs

Figure F.1: Temperature concrete after eight hours charging. The upper figure shows the result with a thermal conductivity of the concrete of 0.5 W/mK and the lower figure with a conductivity of 1.0 W/mK
Figure F.2: Temperature concrete after eight hours discharging. The upper figure shows the result with a thermal conductivity of the concrete of 0.5 W/mK and the lower figure with a conductivity of 1.0 W/mK

Results of charging PCM with different conductivities

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Sign</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inner tube radius</td>
<td>( R_in )</td>
<td>0.015</td>
<td>m</td>
</tr>
<tr>
<td>Radial thickness concrete</td>
<td>( t_i )</td>
<td>0.05</td>
<td>m</td>
</tr>
<tr>
<td>Length tube</td>
<td>( L )</td>
<td>70</td>
<td>m</td>
</tr>
<tr>
<td>Number of tubes</td>
<td>( n_{tubes} )</td>
<td>5000</td>
<td>-</td>
</tr>
<tr>
<td>Specific heat capacity</td>
<td>( C_p )</td>
<td>1470</td>
<td>J/kgK</td>
</tr>
<tr>
<td>Density</td>
<td>( \rho )</td>
<td>2907</td>
<td>kg/m³</td>
</tr>
<tr>
<td>Thermal conductivity</td>
<td>( k )</td>
<td>Differs, see graphs</td>
<td>W/mK</td>
</tr>
<tr>
<td>Initial temperature</td>
<td>( T_i )</td>
<td>280</td>
<td>°C</td>
</tr>
<tr>
<td>Nodes in radial direction</td>
<td>( m )</td>
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<tr>
<td>Nodes in axial direction</td>
<td>( n )</td>
<td>121</td>
<td>-</td>
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</table>

Table F.2: Dimensions and parameters of PCM used for the next graphs
Figure F.3: Temperature PCM after eight hours charging. The upper figure shows the result with a thermal conductivity of the PCM of 0.5 W/mK and the lower figure with a conductivity of 1.5 W/mK.
Appendix G Instable results concrete-PCM-concrete model

The next graph shows the solution of charging the concrete-PCM-concrete model with a mass flow of 23 kg/s:

A mass flow of 23 kg/s is too low to achieve the requirements. The outlet temperature of the 35 bar steam during discharging is not able to reach 350°C. Figure G.1 shows what happens with the temperature of the generated steam. The numbers indicate a new cycle. Numbers 1, 2 and 3 show that in the beginning the prescribed requirements are met but this changes when more cycles are performed. Therefore a minimum mass flow of 24.4 kg/s is needed to run the storage model appropriate.

Figure G.1: Result of discharging cycle of the concrete-PCM-concrete model. Increasing the number of cycles results in an instable solution.
Appendix H Salt temperature over time in salt-PCM-salt model

Figure H.1: Exit temperature of heated hot salt over time during charging in the left graph and exit temperature of cooled salt over time during discharging in the right graph.

The graphs show that the temperature of the salt goes above 450°C and is cooled to a temperature lower than 175°C.
Appendix I Detailed calculations heat losses and pressure drop

The total energy losses can be determined with the next formula:

\[ E_{\text{loss}} = U \cdot A \cdot (T - T_a) \quad (J.1) \]

The overall heat transfer coefficient \( U \) can be found in equation 4.11. For the storage tanks it is assumed that during the eight hours of discharging, the average contact surface area is half the wall plus the area of the bottom. The next values for the surface area temperature are used:

<table>
<thead>
<tr>
<th></th>
<th>HE1 liquid section</th>
<th>HE2 superheated section</th>
<th>Tank 1 cold salt</th>
<th>Tank 2 hot salt</th>
<th>Buffer tank</th>
<th>PCM tank</th>
</tr>
</thead>
<tbody>
<tr>
<td>Surface area [m²]</td>
<td>920</td>
<td>1433</td>
<td>398</td>
<td>245</td>
<td>245</td>
<td>1995</td>
</tr>
</tbody>
</table>

Table 1.1: Surface area for components in salt-PCM-salt storage system

The next values for the average salt and PCM temperature during charging are used:

<table>
<thead>
<tr>
<th></th>
<th>Tank 1 cold salt</th>
<th>HE1 liquid section</th>
<th>Buffer tank</th>
<th>HE2 superheated section</th>
<th>Tank 2 hot salt</th>
<th>PCM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature [°C]</td>
<td>170</td>
<td>238</td>
<td>248</td>
<td>350</td>
<td>450</td>
<td>280</td>
</tr>
</tbody>
</table>

Table 1.2: Average temperature during charging in all components in salt-PCM-salt storage system

The next values for the average salt temperature during discharging are used:

<table>
<thead>
<tr>
<th></th>
<th>Tank 2 hot salt</th>
<th>HE2 superheated section</th>
<th>Buffer tank</th>
<th>HE1 liquid section</th>
<th>Tank 1 cold salt</th>
<th>PCM</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature [°C]</td>
<td>450</td>
<td>400</td>
<td>355</td>
<td>240</td>
<td>170</td>
<td>280</td>
</tr>
</tbody>
</table>

Table 1.3: Average temperature during discharging in all components in salt-PCM-salt storage system

The pressure drop is determined with the Darcy-Weisbach equation:

\[ \Delta p = f \frac{L}{D} \frac{V^2}{2} \quad (J.2) \]

where \( \rho \) is the density, \( L \) the length of the tube, \( V \) the flow speed and \( f \) the friction factor. The friction factor for a turbulent flow is given by the Colebrook equation:

\[ \frac{1}{\sqrt{f}} = -2 \cdot \log \left( \frac{e}{3.7} + 2.51 \frac{Re}{f} \right) \quad (J.3) \]

The roughness coefficient \( e \) for carbon steel can be found in tables with material properties and the Reynolds number is already mentioned. This equation is solved numerically.