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Component Design of a Hybrid Thermal Energy Storage for Increased Flexibility

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Kurzfassung

Die Reduktion der Treibhausgasemissionen und damit die Bekämpfung des Klimawandels gewinnt in den letzten Jahren immer mehr an Bedeutung. Die Europäische Union hat daher Klimaziele definiert die darauf abzielen die Treibhausgasemissionen um 80% bis 95% bis 2050 gegenüber dem Jahr 1990 zu reduzieren. Um den Anteil erneuerbarer Energiequellen zu steigern und damit die Treibhausgasemissionen in industriellen Prozessen zu reduzieren, müssen diese Prozesse flexibler auf die Verfügbarkeit von erneuerbaren Energiequellen reagieren. Der thermische Energiespeicher ist eine Technologie, welche unter anderem eine solche Flexibilitätssteigerung bewirken kann, indem das zeitliche Ungleichgewicht zwischen Energieproduktion und Energieverbrauch kompensiert wird.

Der Ruths-Dampfspeicher ist ein thermischer Energiespeichertyp der bereits in der Industrie eingesetzt wird, wie zum Beispiel in der Papier- oder Stahlindustrie. Dieser Speichertyp speichert Wasserdampf in einem Druckbehälter, welcher zum Großteil mit flüssigem Wasser gefüllt ist. Der größte Anteil des zugeführten Wasserdampfes kondensiert innerhalb des Druckbehälters. Auf Grund des konstanten Volumens steigt der Druck und die Temperatur bei Dampfzufuhr und sinkt beim Dampfentnahme. Mit einem Ruths-Dampfspeicher können hohe Be- und Entlademassenströme realisiert werden. Nachteilig ist jedoch, dass für eine hohe Speicherkapazität große Behältervolumina oder Temperaturänderungen benötigt werden. Der Latentwärmespeicher ist ein thermischer Energiespeichertyp bei dem hohe Speicherkapazitäten mit geringeren Behältervolumina realisiert werden können. Dieser nutzt Phasenwechselmaterialien, welche während ihres Phasenwechsels bei annähernd konstanter Temperatur eine große Menge an Energie aufnehmen bzw. abgeben. Jedoch ist die Leistung dieses Speichertyps auf Grund der niedrigen Wärmeleitfähigkeit der meisten Phasenwechselmaterialien begrenzt.

In dieser Arbeit wird ein Hybridspeicherkonzept, welches die Kombination eines Ruths-Dampfspeichers mit Phasenwechselmaterial vorsieht, untersucht. Ziel ist es, die hohe Speicherkapazität des Phasenwechselmaterials mit der hohen Leistungsfähigkeit des Ruths-Dampfspeichers zu kombinieren. Bei dem untersuchten Hybridspeicherkonzept wird das Phasenwechselmaterial an der Außenseite des Druckbehälters angeordnet, wobei auch unterschiedliche Phasenwechselmaterialien gleichzeitig verwendet werden können, um den Pha-

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senwechsel auf einen breiteren Temperaturbereich auszudehnen. Weiters, ist die Integration von elektrischen Heizelementen oder Wärmeübertragern im Phasenwechselmaterial berücksichtigt um eine Power-to-Heat Maßnahme im Speicher umzusetzen oder im Falle der Integration von Wärmeübertragern, Abwärme von anderen Prozessmedien speichern zu können. Drei Patentanmeldungen, welche das Hybridspeicherkonzept betreffen wurden vonseiten der Austrian Institute of Technology GmbH durchgeführt.

Im Zuge dieser Arbeit wurden fünf wissenschaftliche Publikationen veröffentlicht, darunter ein Konferenz- und vier Journalbeiträge. In diesen Beiträgen werden Hybridspeichermodelle und verschiedene Analysen, welche mit diesen Modellen durchgeführt wurden, vorgestellt. Die Analysen dienten dazu das Verständnis über das Zusammenspiel der unterschiedlichen Komponenten zu verbessern und die Vor- und Nachteile des Konzeptes zu evaluieren. Mit dem Hybridspeicherkonzept ist es möglich die Speicherkapazität eines herkömmlichen Ruths-Dampfspeichers bei gleichem Gesamtvolumen zu erhöhen, jedoch ist dies stark von den Prozessanforderungen abhängig. Die niedrige Wärmeleitfähigkeit des Phasenwechselmaterials und der niedrige Wärmeübergangskoeffizient zwischen dem Dampf und der umgebender Druckbehälterwand limitieren die Performance des Hybridspeichers stark.

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Abstract

The reduction of greenhouse gas emissions and the fight against climate change have become increasingly important in recent years. The European Union has defined climate targets, which seek to reduce greenhouse gas emissions by 80% to 95% from 1990 levels by 2050. To increase the share of renewable energy and thus reduce the greenhouse gas emissions in industrial processes, these processes need to react more flexibly to the availability of renewable energy sources. Thermal energy storage is a technology that, among other things, can provide such flexibility by compensating the temporal imbalance between energy production and energy consumption.

The Ruths steam storage is a thermal energy storage type that is already used in industry, such as in the paper and steel industry. This storage type stores steam in a pressure vessel that is mainly filled with liquid water. Most of the steam supplied condenses inside the pressure vessel. Due to the constant volume, the pressure and temperature increase during steam injection and decrease during steam removal. With a Ruths steam storage, high charging and discharging mass flows can be realized. However, a disadvantage is that, for high storage capacities, large vessel volumes or temperature changes are needed. The latent heat thermal energy storage is a thermal energy storage type, where for high storage capacities smaller vessel volumes can be realized. It uses phase change materials that absorb or release a large amount of energy during their phase change at almost constant temperature. However, the performance of this storage type is limited due to the low thermal conductivity of most phase change materials.

In this thesis, a hybrid storage concept, which combines a Ruths steam storage with phase change material, is investigated. The aim is to utilize the high storage capacity of the phase change material in combination with the high performance of the Ruths steam storage. In the hybrid storage concept under investigation, the phase change material is arranged on the outside of the pressure vessel, whereby different phase change materials can also be used simultaneously to extend the phase change to a wider temperature range. Furthermore, the integration of electrical heating elements or heat exchangers in the phase change material is also taken into account to enable a power-to-heat measure to be implemented in the storage or, in the case of heat exchanger integration, waste heat from other process media to be

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stored. Three patent applications concerning the hybrid storage concept have been filed by the Austrian Institute of Technology GmbH.

In the course of this thesis, five scientific contributions were published, including one conference paper and four journal papers. In these papers, hybrid storage models and various analyses carried out with the models, were presented. The analyses served to enhance the understanding of the interaction of the different components and to evaluate the advantages and disadvantages of the concept. With the hybrid storage concept, it is possible to increase the storage capacity of a conventional Ruths steam storage with the same total volume, but this is strongly dependent on the process requirements. The low thermal conductivity of the phase change material and the low heat transfer coefficient between steam and the surrounding pressure vessel wall significantly limit the performance of the hybrid storage.

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Nomenclature

Abbreviations

$AHCM$	Apparent heat capacity model
ITM	Interface tracking model
$LHTES$	Latent heat thermal energy storage
PCM	Phase change material
RSS	Ruths steam storage
TES	Thermal energy storage

Roman Symbols

$\dot{m}$	Mass flow, kgs^{-1}
$\dot{Q}_{sl}$	Thermal energy flow between the steam and liquid water phase, Js^{-1}
$c(x)$	Function of variable x
c_p	Specific heat capacity at constant pressure, $Jkg^{-1}K^{-1}$
$c_{p,eff}$	Specific heat capacity used in the apparent heat capacity method, $Jkg^{-1}K^{-1}$
D	Diameter, m
F	Filling height, m
f	Fraction of phase change enthalpy, 1
f_T	Temperature difference fraction, 1
$g(x)$	Function of variable x
H	Enthalpy, J
h	Specific enthalpy, Jkg^{-1}
h_{pc}	Phase change enthalpy, Jkg^{-1}
$j(x)$	Function of variable x
k	Overall heat transfer coefficient $Wm^{-2}K^{-1}$
L	Length, m
m	Mass, kg

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n	Parameter
o	Parameter
p	Pressure, bar
Q	Thermal Energy, J
q	Parameter
r	Radius, m
s	Layer thickness, m
T	Temperature, K
t	Time, s
V	Volume, m^3

Greek Symbols

α	Heat transfer coefficient, $Wm^{-2}K^{-1}$
Δ	Difference
γ	Angle, rad
λ	Thermal conductivity, kgm^{-3}
ρ	Density, kgm^{-3}
τ	Relaxation time, s

Subscripts

c	Condensation
e	Evaporation
in	Incoming
l	Liquid
max	Maximum
min	Minimum
out	Outgoing
pc	Phase change
ref	Reference
s	Steam
w	Wall

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Superscripts

$'$	Properties of saturated water
$''$	Properties of saturated steam

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List of Figures

1.1 Austrian's greenhouse gas emissions for different sectors in the year 2017 [1]. 1

1.2 (a) Share of primary energy consumption for various energy sources [2]; (b) Share of renewable energy consumption for various renewable energy sources [2]. 2

2.1 Temperature as a function of stored energy for sensible heat TESs and LHTESs, following [3]. 8

2.2 Solidifying of a PCM layer, following [3, 4]. 12

2.3 (a) Enthalpy depending on temperature (enthalpy method), following [3]; (b) Heat capacity depending on temperature (apparent heat capacity method), following [3]. 14

2.4 Ruths steam storage, following [5, 6]. 15

2.5 Example of load balancing through the integration of steam accumulators. . 16

3.1 Hybrid storage concept. 22

4.1 Calculation of angle γ, which describes the actual filling height. 26

4.2 Simulation results in terms of inner temperature of the LHTES cylinder for the ITM and AHCM with different numbers of PCM volumes in the radial direction (number after the slash). 27

4.3 Simulation results in terms of outer temperature of the LHTES cylinder for the ITM and AHCM with different numbers of PCM volumes in the radial direction (number after the slash). 28

4.4 Simulation results in terms of pressure inside storage vessel for the ITM and AHCM with different numbers of PCM volumes in the radial direction (number after the slash). 29

4.5 Simulation results in terms of stored energy for the ITM and AHCM with different numbers of PCM volumes in the radial direction (number after the slash). 30

5.1 Hybrid storage discretisation in the peripheral direction. 34

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5.2 Mean temperature inside the pressure vessel wall volumes and temperature inside liquid/steam volume. 35
5.3 Heat transfer area between the pressure vessel wall volumes and the steam phase. 36
5.4 Heat flow between the pressure vessel wall volumes in the peripheral direction of one circular cross-section. 37
5.5 Pressure inside the liquid/steam volume for various LHTES volumes. 38
5.6 Stored energy inside the hybrid storage for various LHTES volumes. 38

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Contents

1 Introduction **1**

1.1 Scope of This Work . . . 3

1.2 Publications and Related Patent Applications . . . 5

2 State of the Art **7**

2.1 Sensible Heat Thermal Energy Storage . . . 8

2.2 Latent Heat Thermal Energy Storage . . . 8

2.2.1 Heat Transfer Enhancement . . . 10

2.2.2 Modeling of Latent Heat Thermal Energy Storages . . . 11

2.3 Ruths Steam Storage . . . 15

2.3.1 Modeling of Ruths Steam Storages . . . 17

2.4 Combined Thermal Energy Storages . . . 18

2.4.1 Sensible/Latent Heat Thermal Energy Storages . . . 18

2.4.2 Ruths Steam Storage Extension . . . 19

2.4.3 Electrical Heating Elements Used in Phase Change Materials . . . 20

2.5 Design Optimization . . . 20

3 Hybrid Storage Concept **22**

4 Hybrid Storage Models **24**

4.1 Interface Tracking Model . . . 25

4.2 Apparent Heat Capacity Model . . . 26

5 Hybrid Storage Concept Analysis **31**

5.1 Discrestisation of the Pressure Vessel Wall . . . 34

6 Hybrid Storage Design Optimization **39**

7 Papers **42**

7.1 Paper 1 . . . 42

7.2 Paper 2 . . . 42

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7.3 Paper 3 43
7.4 Paper 4 43
7.5 Paper 5 44

8 Conclusion **45**

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Chapter 1

Introduction

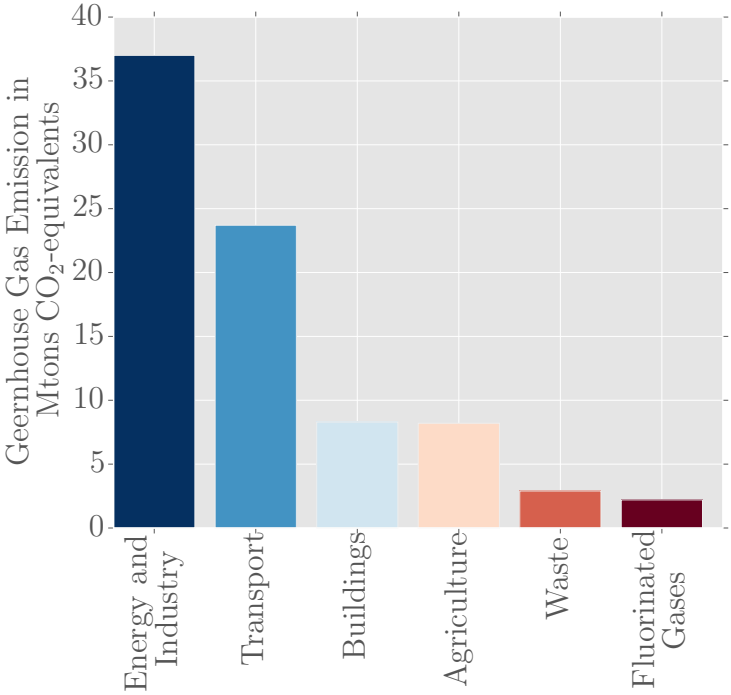

Figure 1.1: Austrian's greenhouse gas emissions for different sectors in the year 2017 [1].

Climate change is an evident problem. In the year 2017, the average annual air temperature in Austria was 7.8°C, which is 0.9°C higher than the average of the years 1981–2010 [7]. To remedy this problem, the European Union aims to reduce the emissions by 80% to 95% below 1990 levels by 2050. To achieve this long-term goal, interim targets have been formulated. By 2020, greenhouse gas emissions are to be reduced by 20% compared with 1990 levels, 20% of total energy consumption is to be generated by renewable energy sources, and energy efficiency is to be increased by 20%. By 2030, greenhouse gas emissions are to be reduced by 40% compared with 1990 levels, 32% of total energy consumption is

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to be generated by renewable energy sources, and energy efficiency is to be increased by 32.5%. [8]

Figure 1.1 shows the greenhouse gas emissions in Austria for different sectors in the year 2017. The energy and industry sector emits the major share of greenhouse gases, with emissions of 37 million tons carbon dioxide equivalents. With 23.7 million tons carbon dioxide equivalents, the transport sector generates the second largest amount of greenhouse gas emissions. In total, the greenhouse gas emissions amounted to 82.3 million tons of carbon dioxide equivalents, 4.6% more than in 1990. [1]

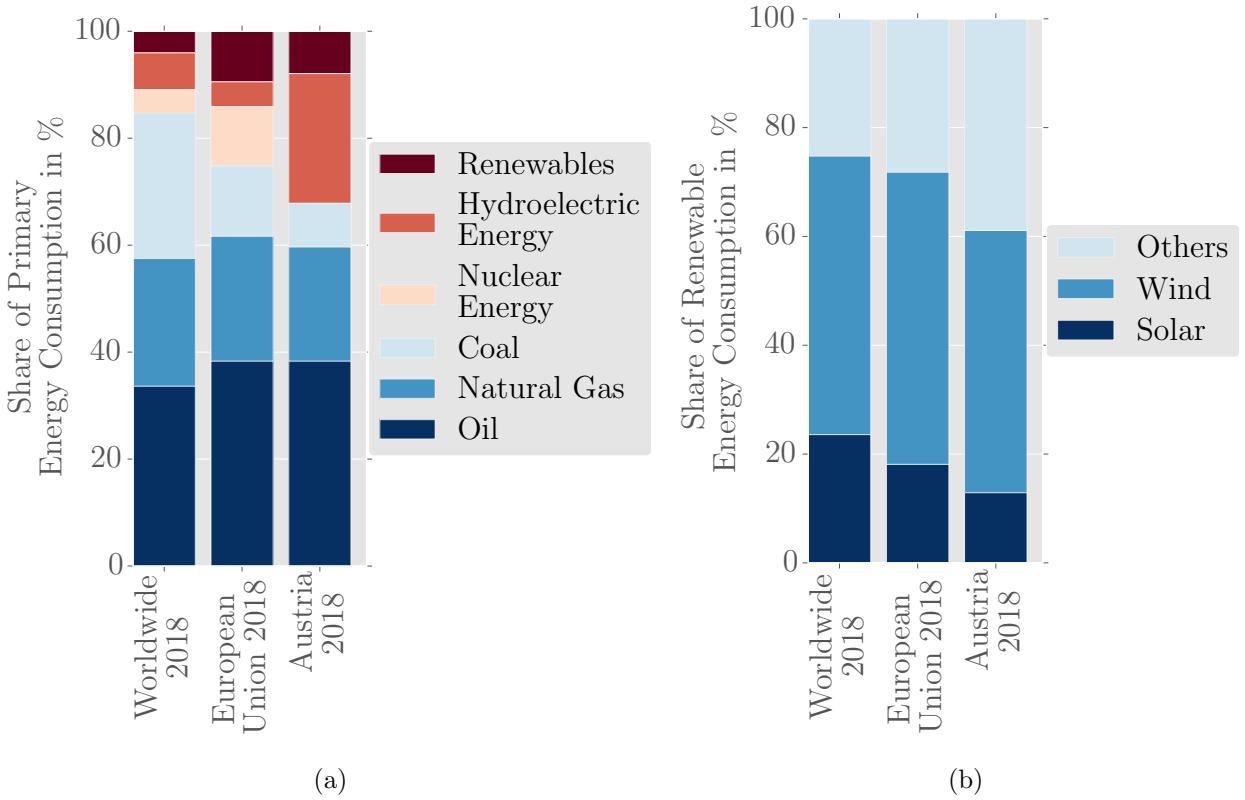

Figure 1.2: (a) Share of primary energy consumption for various energy sources [2]; (b) Share of renewable energy consumption for various renewable energy sources [2].

In general, global primary energy consumption has increased in recent years. Worldwide primary energy consumption reached 580,495 PJ in 2018. In the year 2000, primary energy consumption was 391,759 PJ, which is 33% lower than in 2018 [2]. Figure 1.2(a) illustrates the shares of energy sources in primary energy consumption. This figure depicts the share of hydroelectric energy separately from the other renewable energy sources. In Austria in 2018, the share of primary energy consumption generated by hydroelectric was 24%, and the share of primary energy consumption generated by other renewable energy sources was 8%.

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Oil, natural gas, and coal accounted for the largest share of worldwide primary energy consumption in 2018. By comparison, the share of renewable energy sources was significantly lower (4% to 9%), even taking hydroelectric energy into account (11% to 32%). Figure 1.2(b) shows the consumption of energy generated from renewable energy sources such as wind, solar, and others. Other renewable energy sources are geothermal and biomass. The largest share of consumed renewable energy was generated by wind (48% for Austria in 2018). By comparison, the share of solar energy was low (13% for Austria in 2018).
A characteristic problem which complicates the use of renewable energy sources is that they are not always available when energy is needed. Thermal energy storages (TESs) can be a key technology for increasing the share of renewable sources in energy consumption, as their integration allows energy supply and energy demand to be temporal decoupled. Alva et al. [9] mentioned that, besides temporal differences, also the local differences between the place of thermal energy production and the place of thermal energy consumption can be compensated by using TESs. In processes it can be the case that thermal energy is generated at a time when it cannot be used. This energy is then released into the environment as waste heat. By integrating a TES, this waste heat could be stored and used at a later point in time. Such a measurement would generally reduce the thermal energy demand of the process and increase its efficiency. Moreover, cost advantages can be offered by integrating TESs because it is possible to store energy when it is cheap and to discharge it when it is expensive [9].
In Austria, 99868 TJ of final energy consumption was used for steam generation in 2017 [10]. The energy needed for steam production was 56,840 TJ in the paper, pulp and print industry, 10,937 TJ in the food, tabacco and beverages industry, and 439 TJ in the iron and steel industry [10]. Steam storage systems are already used to increase efficiency in steel and paper production. Other fields applying steam storage systems such as nuclear industry, food manufacturing, biotechnology, sterilization and product testing are highlighted by González et al. [11]. Moreover, steam storages are the only commercially used storage type for solar power plants, where water is used as a heat transfer medium for direct steam generation. The steam generated by solar radiation can be used directly without a heat exchanger to, for example, generate electricity with a turbine. [11]

1.1 Scope of This Work

In this thesis, a hybrid storage concept is investigated that surrounds the Ruths steam storage (RSS) vessel with one or more phase change materials (PCMs). Therefore, this concept combines the high performance of the RSS with the high storage capacity of latent heat thermal energy storages (LHTESs) in one component. The aim of the concept is to

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increase the storage capacity compared with a classic RSS. Electrical heating elements or heat exchangers can be arranged in the PCMs to additionally use a second heat source in the same component. In this thesis, the interaction of the technologies involved will be studied in more detail. The following questions about the hybrid storage concept are answered in this thesis:

- How do the two combined storage types affect each other in terms of storage capacity and application limits?
- How does the arrangement of different PCMs affect the hybrid storage performance?
- How does an additional energy source placed inside the PCM influence the performance of the hybrid storage?
- Can a hybrid storage system offer cost advantages over already-in-use RSS systems?

The aim of answering these questions is to advance the development of the new storage component and to gain knowledge about design, application limits and advantages in terms of storage capacity and costs. First of all, in Chapter 2 TESs are described in general. Then RSSs and LHTESs are discussed in more detail, because these storage types are combined in the hybrid storage concept. Some modeling approaches for these storage types are also summarized. In addition, already published methods for the extending classical storage types by combining them with other storage types or other technologies are presented. Thus, this chapter focuses on the combination of sensible heat TESs and LHTESs, as well as the extension of RSSs. At the end of the chapter, the topic optimization is briefly discussed.
After providing a detailed description of the hybrid storage concept and the goal of this concept in Chapter 3, the development of the hybrid storage models is summarized in Chapter 4. Chapter 5 summarazies the results of the concept analysis presented in the published papers. To assess whether hybrid storage systems can offer cost advantages over conventional RSS systems, Chapter 6 describes the content of the paper that was published on this topic. Chapter 8 presents main conclusions and future steps for the further development of the hybrid storage are proposed. All scientific contributions published in the course of this thesis are listed in the following chapter and are also presented in Chapter 7.

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1.2 Publications and Related Patent Applications

The scientific contributions listed in the following are part of this thesis.

Paper 1:
S. Dusek, R. Hofmann. A Hybrid Storage Concept for Improving Classical Ruth's Type Steam Accumulators. In *Proceedings of the 12th Conference on Sustainable Development of Energy, Water and Environment Systems*, Dubrovnik, Croatia, October 4-8, 2017.

Paper 2:
S. Dusek, R. Hofmann. A Hybrid Energy Storage Concept for Future Application in Industrial Processes. *Thermal Science*, 22(5): 2235-2242, 2018, `https://doi.org/10.2298/TSCI171230270D`.

Paper 3:
S. Dusek, R. Hofmann. Modeling of a Hybrid Steam Storage and Validation with an Industrial Ruths Steam Storage Line. *Energies*, 12(6):1014, 2019, `https://doi.org/10.3390/en12061014`.

Paper 4:
S. Dusek, R. Hofmann, S. Gruber. Design Analysis of a Hybrid Storage Concept Combining Ruths Steam Storage and Latent Thermal Energy Storage, *Applied Energy*, 251: 113364, 2019, `https://doi.org/10.1016/j.apenergy.2019.113364`.

Paper 5:
R. Hofmann, S. Dusek, S. Gruber, G. Drexler-Schmid. Design Optimization of a Hybrid Steam-PCM Thermal Energy Storage for Industrial Applications, *Energies*, 12(5):898, 2019, `https://doi.org/10.3390/en12050898`.

Related Patent applications:
The hybrid storage concept examined in this thesis is based on the following patent application.

Application 1:
AIT Austrian Institute of Technology GmbH. *Dampfspeicher (German) [Steam Storage]*. Number AT518793A1. Inventor: R. Hofmann, C. Zauner, S. Dusek, and F. Hengstberger, Publication Date: 15.01.2018, Status: Submitted.

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Application 2:

AIT Austrian Institute of Technology GmbH. *Dampfspeicher (German) [Steam Storage].* Number AT518828A1. Inventor: R. Hofmann, C. Zauner, S. Dusek, and F. Hengstberger, Publication Date: 15.01.2018, Status: Submitted.

Application 3:

AIT Austrian Institute of Technology GmbH. *Dampfspeicher (German) [Steam Storage].* Number EP3260803A1. Inventor: R. Hofmann, C. Zauner, S. Dusek, and F. Hengstberger, Publication Date: 27.12.2017, Status: Patent granted.

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Chapter 2

State of the Art

Gibb et al. [12] mentioned that there are various TES concepts available. Depending on the requirements of the application, a particular TES concept can be more or less suitable [12]. However, TESs can be rated according to cost, capacity, power, efficiency, storage period, and charging and discharging time [13]. Moreover, there are various means by which to classify TESs [14]:

- Into high and low temperature TESs, according to the temperature range in which the TES system is applied. For example, Zhang et al. [14] defined 200°C as the boundary between the two areas;

- Into sensible heat TESs, LHTESs, and thermo-chemical storages, according to the storage mechanism; and

- Into active, passive, and hybrid TESs, according to the storage concept. For active concepts, the storage medium circulates through the heat transfer components, like a heat exchanger or a steam generator. Therefore, the heat transfer is mainly due to forced convection. For passive concepts, this is not the case, the storage medium does not circulate. Active concepts can be further divided into direct and indirect concepts. In contrast to indirect concepts, in direct concepts, the heat transfer fluid and storage medium are the same and no heat exchanger is required. The combination of active and passive concepts is called a hybrid concept, as the concept presented in this thesis.

In the following, the classification according to the storing mechanism is described in more detail. Chemical TESs are not relevant for this work and are therefore not discussed in further detail.

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2.1 Sensible Heat Thermal Energy Storage

In sensible heat TESs, the temperature increases when energy is stored and decreases when energy is released [15]. The stored thermal energy can be calculated with Equation (2.1) [14].

$$Q_{sensible} = m \int_{T_1}^{T_2} c_p dT \tag{2.1}$$

For temperatures below 100°C water can be used as storage material in sensible heat TESs. Water as a storage material is characterized by low cost, high availability, non-toxicity, and a high specific heat capacity compared with other sensible storage materials. [14, 16]
Li [16] mentioned that water is the most commonly used storage material in buildings and is also used in industrial applications. Moreover, solid materials such as rock, metal, concrete, and sand can also be used in sensible heat TESs. Compared with a water storage system, these materials can be used as a storage material at temperatures above 100°C. A disadvantage of solid storage materials is their low volumetric heat capacity compared with water. [16]
Other sensible storage materials for temperatures above 100°C are, for example, synthetic oils, liquid metals, or molten salts [14].

2.2 Latent Heat Thermal Energy Storage

In an LHTES, the phase change of a material is used to store energy at a nearly constant temperature, compared with a sensible heat TES. The difference between sensible heat TESs and LHTESs is illustrated in Figure 2.1.

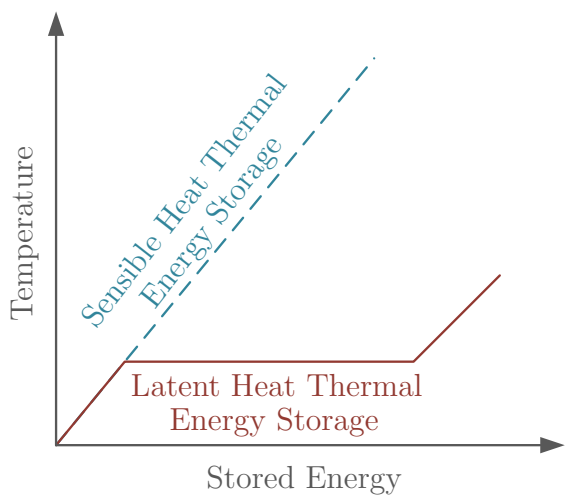

Figure 2.1: Temperature as a function of stored energy for sensible heat TESs and LHTESs, following [3].

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The energy stored or released during phase change is typically called latent heat. Latent heat occurs at the phase change between solid and liquid, liquid and gaseous, or solid and solid phase. The solid–liquid phase change is preferred because the volume changes are small (often less than 10%). Also, during the solid–solid phase change of some materials, the volume changes are small, but the latent heat is often low. High volume changes are to be expected during the liquid-gaseous phase change at constant pressure. If the volume is kept constant during the liquid-gaseous phase change, the pressure and also the phase change temperature increases. In terms of LHTES, commonly only the phase changes between solid and solid, and between liquid and solid are included. The materials used in LHTESs are called PCMs. [3] In this thesis, only the solid–liquid phase change of a PCM is considered for the application in LHTES.
When the pressure and the volume are both nearly constant, the latent stored thermal energy can be calculated with Equation 2.2, where h_{pc} is the specific enthalpy difference between the two phases, also called phase change enthalpy [3]:

$$Q_{latent} = h_{pc} m_{pcm}. \tag{2.2}$$

Many PCMs, with phase change temperatures in various temperature ranges are available. For example, the phase change temperature of salts is between 250 and 1680°C [17]. To select a suitable PCM for the application, not only the economic but also the thermodynamic, kinetic, and chemical properties are relevant [18]. More precisely, the following requirements for PCMs in LHTESs are described by Abhat [18] according to Lorsch et al. [19]:

- Thermodynamic properties: suitable phase change temperature, high latent heat per unit mass, high density, high specific heat capacity, high thermal conductivity, small volume changes, and congruent melting;
- Kinetic properties: low or no subcooling;
- Chemical properties: stability, no decomposition, non-corrosive, non-toxic, non-flammable, and non-explosive; and
- Economic properties: high availability and low costs.

In the list above, congurent melting means that the forming liquid phase has the same composition as the solid phase [18, 19]. Furthermore, subcooling means that the temperature of the PCM must decrease to a temperature below the melting temperature before the PCM starts to solidify. This leads to a different behavior for solidifying than for melting. [3]
One problem with most PCMs is their low thermal conductivity, which affects the performance of LHTESs. Studies have investigated various methods for countering this problem. Some of these methods are presented in the following chapter.

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2.2.1 Heat Transfer Enhancement

As mentioned in the previous section, several methods are available for improving the performance of an LHTES. Following Tao and He [20], they can be divided into the enhancement of thermal conductivity, the expansion of the heat transfer surface, and the improvement of process uniformity.

Thermal conductivity enhancement methods take into account the use of porous media and nanomaterial additives to create composites with the PCM. When a compound of porous material and PCM is used, the porosity and the thermal conductivity of the porous material must be sufficiently high to achieve the required storage capacity and thermal conductivity of the compound. [20]

For example, Li et al. [21] experimentally and numerically investigated paraffin in copper forms with a porosity equal to or greater than 90%. The influence of pore density and porosity was analyzed, and the results showed that the influence of porosity is higher than the influence of pore density. [21]

When a compound of PCM and nanomaterial additives as nanopowders, nanowires, or cabon nanotubes are used, the nanomaterial must have a high thermal conductivity and must be chemically stable to achieve a satisfactory result [20, 22]. Lokesh et al. [23] experimentally studied the phase change time of paraffin improved by a multi-wall carbon nanotube. The measurements showed a reduction of the phase change time up to 42%. [23]

Methods for the aforementioned extension of the heat transfer surface are PCM encapsulation and finned tubes. PCM encapsulation means that the PCM is arranged in several capsules. These capsules are placed in a vessel and the heat transfer fluid flows inside this vessel through the gaps between the capsules. For finned tubes used in shell-and-tube LHTESs, fins of highly thermally conductive material can be used on the tube side where the PCM is attached. They can also be used on the heat transfer fluid side when the heat transfer is too low on this side. [20] Longitudinal and circular fins on the PCM side were experimentally studied by Agyenim et al. [24]. In the analysis, the PCM was arranged around a tube in which the heat transfer fluid flows. The arrangement with longitudinal fins provided the best results. [24]

Another method, called sandwich design, can be used for increasing the heat transfer surface and was presented by Steinmann and Tamme [25]. In this method, a layer arrangement of highly thermally conductive foil, such as graphite foil, and PCM surrounding a pipe was used. An arrangement based on this sandwich design was investigated in **Paper 4** with regard to its application within the hybrid storage concept. The analysis carried out in **Paper 4** was based on the studies of Gruber [26].

The last group of methods to increase the performance of LHTESs, mentioned by Tao and He [20], improves process uniformity. This category can be further subdivided into the

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methods of process optimization, i.e. thermodynamic optimization, and the use of multiple PCMs. The aim of these methods is to realize a spatial and temporal constant temperature difference between the heat transfer fluid and the PCM. Thus, multiple PCMs with different phase change temperatures are arranged in a cascadic form. The arrangement shall allow both, the PCM melting temperature and the heat transfer fluid temperature, decrease in flow direction during charging. During discharging, the solidification temperatures of the PCMs should increase in accordance with the increasing temperature of the heat transfer fluid in flow direction. [20]

Fang and Chen [27] investigated such an arrangement along a pipe. The arrangement of multiple encapsulated PCMs with different phase change temperatures in a container was also discussed in the literature. Ezra et al. [28] analyzed a tube bank, where the tubes were filled with various PCMs. The heat transfer fluid flowed through this bank. An arrangement of multiple PCMs in vertical tubes with parallel heat transfer fluid flow was described by Li et al. [29]. Aceves et al. [30] studied the arrangement of several cells with different PCMs. Spherical capsules filled with different PCMs used in an LHTES were discussed by Aldoss and Rahman [31]. In the patent application [32] an arrangement of different PCMs surrounding each other in a shell-like shape was presented. Energy can be transferred by a heat transfer medium that passes through the different PCMs by heat exchangers. Furthermore, the shell-like shaped PCM containers can be surrounded by an additional heat transfer medium. This can be heated up by an additional energy source, for example, an electrical heating element. [32]

The arrangement of several PCMs with different properties is also considered in the hybrid storage concept and is described in detail in Chapter 3.

2.2.2 Modeling of Latent Heat Thermal Energy Storages

Various approaches for modeling LHTESs, more precisely, the phase change problem, were presented in the literature. Some of these approaches are presented in the following sections, which are divided into analytical solutions and computational methods.

Analytical Solutions

The analytical solution for solidification time t of a plain PCM layer (Stefan problem) with the thickness s (Figure 2.2) is shown in Equation (2.3). As can be seen in Figure 2.2, at the left side, a constant temperature T_0 is assumed. This temperature is lower than the phase change temperature. Moreover, it is assumed that the temperature of the liquid PCM layer is equal to the phase change temperature. Furthermore, the sensible stored energy is neglected. The smaller the Stefan number (ratio between sensible and latent stored energy)

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is, the more this equation corresponds to the exact solution. [3, 4]

$$t = \frac{h_{pc}\rho s^2}{2\lambda\left(T_{pc} - T_0\right)} \tag{2.3}$$

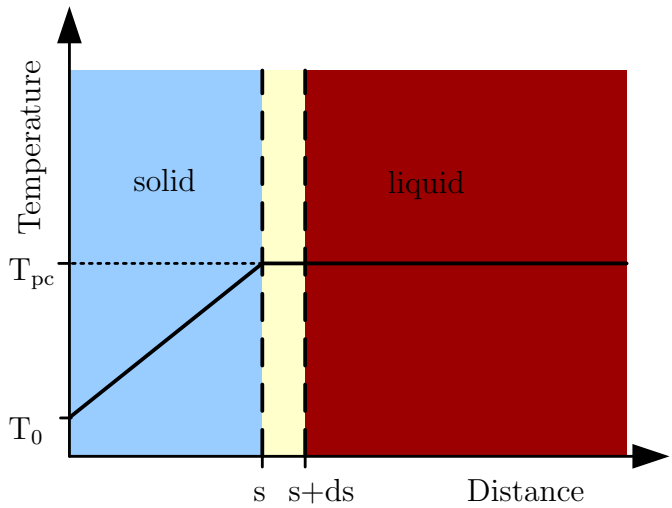

Figure 2.2: Solidifying of a PCM layer, following [3, 4].

If an additional wall is added on the left side of the layer, quasi-stationary approximation can be used to calculate the solidification time, as shown in Equation (2.4). In this equation, k is the overall heat transfer coefficient between the heat transfer fluid with constant temperature and the PCM. In this case, T_0 is the temperature of the heat transfer fluid. [3, 4]

$$t = \frac{h_{pc}\rho s^2}{2\lambda\left(T_{pc} - T_0\right)}\left(1 + 2\frac{\lambda}{ks}\right) \tag{2.4}$$

In the following, the quasi-stationary approximation for a pipe with length L, that is surrounded by a PCM layer (thickness s) is derived under the above described assumptions. This was also presented in Baehr and Stephan [4]. Therefore, the inner pipe radius r and the wall thickness s_w are used. Inside the pipe there is a fluid with a temperature T_0. The thermal energy dQ, which is released during solidifying of a PCM layer with the thickness ds, is calculated using Equation (2.5).

$$dQ = h_{pc}\rho 2\pi\left(r + s_w + s\right)Lds \tag{2.5}$$

This thermal energy is transferred through the solidified PCM layer via the wall to the heat transfer fluid (Equation (2.6)).

$$dQ = \frac{T_{pc} - T_0}{\frac{1}{2\pi\lambda L}ln\left(\frac{r+s_w+s}{r+s_w}\right) + \frac{1}{2\pi\lambda_w L}ln\left(\frac{r+s_w}{r}\right) + \frac{1}{2\pi Lr\alpha}}dt \tag{2.6}$$

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Equation (2.7) results from applying Equation (2.5) to Equation (2.6).

$$dt = \frac{h_{pc}\rho\left(r+s_w+s\right)}{T_{pc}-T_0}\left[\frac{1}{\lambda}ln\left(\frac{r+s_w+s}{r+s_w}\right)+\frac{1}{\lambda_w}ln\left(\frac{r+s_w}{r}\right)+\frac{1}{r\alpha}\right]ds \tag{2.7}$$

At the beginning the solidified layer thickness is equal to zero. Therefore, the integration of Equation (2.7) provides Equation (2.8) for approximating the solidification time.

$$\begin{aligned} t = \frac{h_{pc}\rho s^2}{2\lambda\left(T_{pc}-T_0\right)}\Bigg[&\left(1+\frac{r+s_w}{s}\right)^2 ln\left(1+\frac{s}{r+s_w}\right) \\ &-\left(1+\frac{2\left(r+s_w\right)}{s}\right)\left(\frac{1}{2}-\left(\frac{\lambda}{\lambda_w}ln\left(\frac{r+s_w}{r}\right)+\frac{\lambda}{r\alpha}\right)\right)\Bigg]. \end{aligned} \tag{2.8}$$

If, in contrast to the Stefan problem described above, the liquid PCM phase has a higher temperature than the phase change temperature, this is called the Neumann problem. With the Neumann solution, the material properties of the liquid PCM phase are taken into account, and convection is neglected. [4]

Computational Methods

The computational methods for solving the phase change problem or moving boundary problem can be divided in interface capturing and interface tracking (or front tracking) methods. The two types of methods differ in the manner in which the moving phase interface between the two phases during phase change is considered. In interface tracking methods, the position of the moving phase interface is calculated. By contrast, the interface capturing method calculates the temperature distribution in the PCM. The position of the phase interface can then be deduced based on the temperature distribution. [33, 34, 35]
The enthalpy and the apparent heat capacity method (or effective heat capacity method) are the most common interface capturing methods [33]. The enthalpy method uses the enthalpy as a temperature dependent variable, as is shown in Figure 2.3(a) and Equation (2.9), where the reference temperature T_{ref} and the fraction of phase change enthalpy f are used and an exact phase change temperature is assumed. The volume is completely solid or liquid, if the fraction of phase change enthalpy is zero or one, respectivly. [3]

$$h(T) = \begin{cases} c_{p,solid}\left(T-T_{ref}\right) & T < T_{pc} \\ c_{p,solid}\left(T-T_{ref}\right)+fh_{pc} & T = T_{pc} \\ c_{p,solid}\left(T_{pc}-T_{ref}\right)+h_{pc}+c_{p,liquid}\left(T-T_{pc}\right) & T > T_{pc} \end{cases} \tag{2.9}$$

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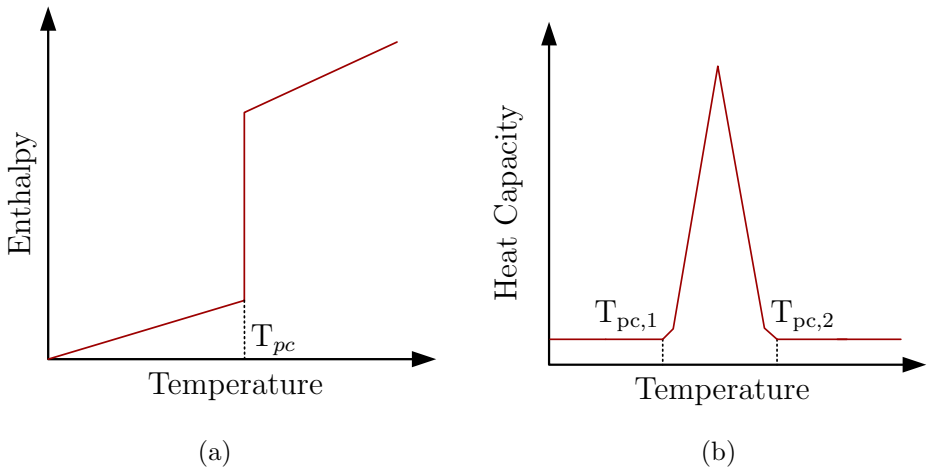

Figure 2.3: (a) Enthalpy depending on temperature (enthalpy method), following [3]; (b) Heat capacity depending on temperature (apparent heat capacity method), following [3].

In the apparent heat capacity method, heat capacity is used to consider the latent stored energy. This is shown in Figure 2.3(b) and Equation (2.10) [36].

$$c_{p,eff} = \begin{cases} c_{p,solid} & T < T_{pc,1} \\ c_{p,solid} + \frac{h_{pc}}{T_{pc,2} - T_{pc,1}} & T_{pc,1} \leq T \leq T_{pc,2} \\ c_{p,liquid} & T > T_{pc,2} \end{cases} \tag{2.10}$$

Each of the two interface capturing methods has disadvantages. For example, the application of the enthalpy method for subcooling or when temperatures are oscillating can be problematic. For small phase change temperature ranges, it is disadvantageous to use the apparent heat capacity method. Moreover, for exact phase change temperatures, it is not possible to use the apparent heat capacity method. [36]
Interface tracking methods are more complex and computationally difficult than the interface capturing methods. Furthermore, the enthalpy and the apparent heat capacity methods are more often used than the interface tracking methods. [33] Klimeš et al. [33] compared the mentioned interface capturing methods with an interface tracking method and concluded that the interface tracking method is more accurate, but its processing is slower. The comparison also included a third interface capturing method the temperature recovery method, but this approach is not discussed in detail in this thesis. Zauner et al. [37] used an interface tracking method and the apparent heat capacity method to model a hybrid sensible–latent heat storage consisting of a shell-and-tube heat exchanger. The PCM was placed inside the tubes and the sensible storage medium was placed on the shell side. The sensible storage medium was also the heat transfer medium. They compared the simulation results of both models with experimental investigations and determined that both modeling approaches provided good results. [37]

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2.3 Ruths Steam Storage

Another TES type for temperatures above 100°C is the steam storage. This storage type, also known as RSS or sliding pressure steam storage, consists of a pressure vessel filled with a two-phase fluid comprised of liquid water and steam. Figure 2.4 depicts the schematic of such an RSS.

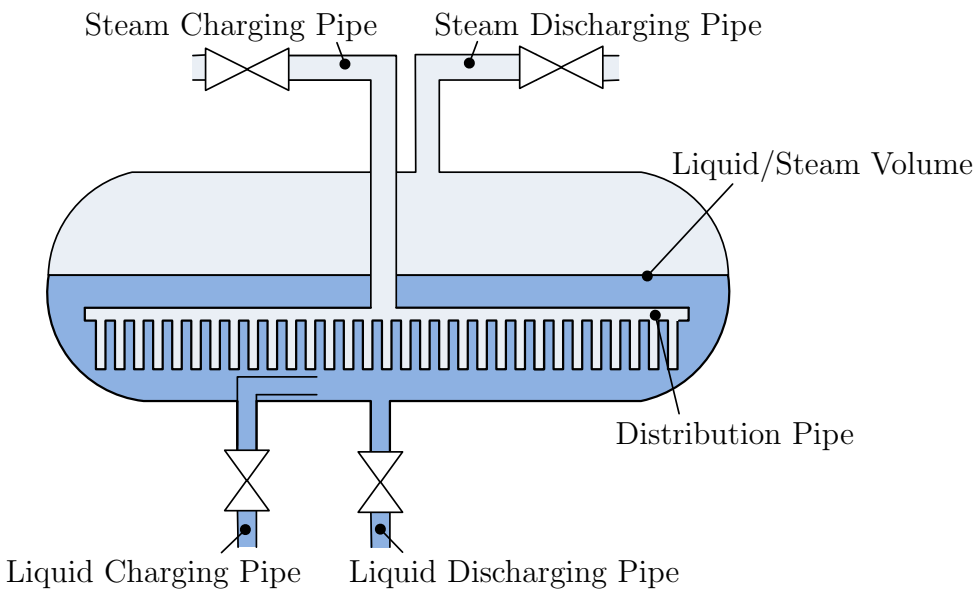

Figure 2.4: Ruths steam storage, following [5, 6].

The pressure vessel is usually designed as a horizontal cylinder, and the heads are hemispherical, elliptical, or semi-elliptical. The charging steam is introduced directly into the liquid contained in the pressure vessel via a distribution pipe with nozzles. To achieve adequate mixing of the charging steam and the liquid, circulation pipes are used around the nozzle bodies. During charging, most of the steam condenses inside the liquid phase. Thus, the pressure and the temperature increase. For discharging, steam is released from the vessel, and the pressure and temperature of the two-phase fluid consequently decrease. The liquid partly evaporates, yielding a thermodynamic equilibrium. [38]
The maximum pressure inside the vessel depends on the maximum pressure of the charging steam mass flow. The process requirements determine the minimum pressure of the RSS. In general, there are three different types of steam consumption [38]:

- uniform steam consumption, which means that the consumer needs a constant capacity without interruption;
- steam consumption with known fluctuations, such as known downtimes; and
- steam consumption with unknown fluctuations.

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Goldstern [38] explained that the integration of storages can be used to change the worst form of steam consumption (steam consumption with unknown fluctuations) into the best one (uniform steam consumption). For example, when an RSS system is used, the steam generator can be operated in such a manner that it constantly generate an average amount of steam [39]. This is shown in Figure 2.5. If excess steam is produced, the steam storage can be charged, and if more than the average steam amount is required, the additional steam mass can be provided by the storage.

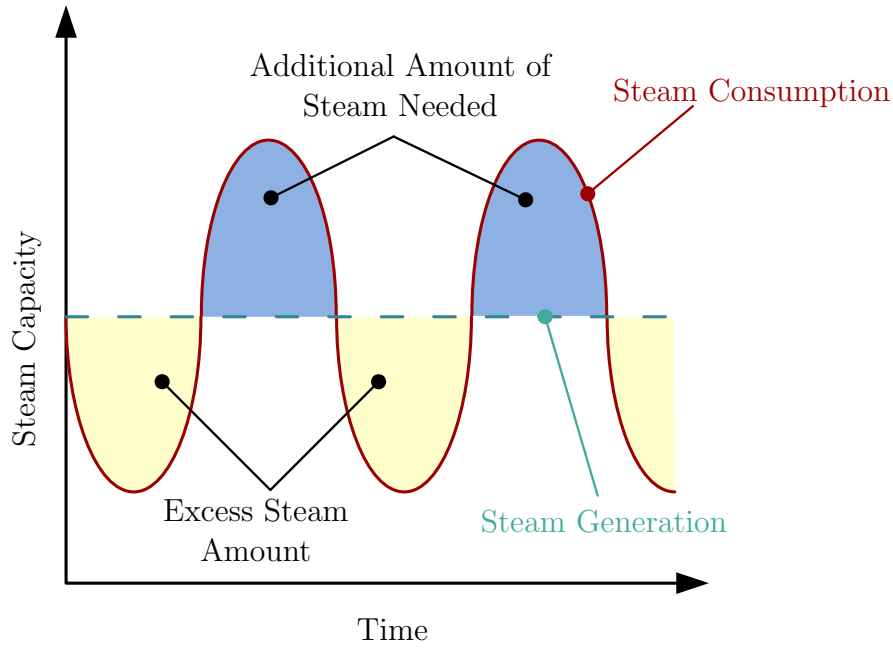

Figure 2.5: Example of load balancing through the integration of steam accumulators.

The application of steam storages in industrial food processes was mentioned by Fabrizio et al. [39]. The integration of an RSS system in the district heating plant Bad Elster was presented by Dittmann et al. [40]. This RSS system consists of nine 180 m^3 storage vessels divided into three storage lines and has been operated without breakdown since 1981 [40]. In the literature, RSSs are also mentioned in combination with direct-steam-generation solar power plants. González-Roubaud et al. [11] discussed using molten salt storages and RSSs in solar power plants. High discharging rates and the short reaction time make the RSS an attractive storage type. In addition to the low volumetric energy density, the decreasing discharging pressure and the wall thickness of the pressure vessel can be disadvantageous by using RSSs. A certain wall thickness is necessary for the vessel to withstand the internal pressure. However, thermomechanical stresses in the vessel wall due to high temperature gradients must also be taken into account. [11]

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2.3.1 Modeling of Ruths Steam Storages

Two approaches to model an RSS are presented in this chapter, the equilibrium and the non-equilibrium approaches. These two modeling approaches were also presented by Stevanovic et al. [5, 6]. For the equilibrium approach, it is assumed that the two phases inside the pressure vessel are in thermodynamic equilibrium and therefore have the same temperatures. This results in one mass (Equation (2.11)) and energy (Equation (2.12)) balance equation for the two-phase fluid [6]. This approach was, for example also, used by Fabrizio et al. [39] and Hofmann et al. [41].

$$\frac{dm}{dt} = \dot{m}_{l,in} - \dot{m}_{l,out} + \dot{m}_{s,in} - \dot{m}_{s,out} \tag{2.11}$$

$$\frac{dH}{dt} = \dot{m}_{l,in}h_{l,in} - \dot{m}_{l,out}h_{l,out} + \dot{m}_{s,in}h_{s,in} - \dot{m}_{s,out}h_{s,out} + V\frac{dp}{dt} \tag{2.12}$$

In the non-equilibrium approach, the two phases can have different temperatures, and for both phases, their own mass and energy balance equations are necessary. Therefore, the non-equilibrium model is based on the following equations [5, 6]:

$$\frac{dm_l}{dt} = \dot{m}_{l,in} - \dot{m}_{l,out} + \dot{m}_c - \dot{m}_e \tag{2.13}$$

$$\frac{dm_s}{dt} = \dot{m}_{s,in} - \dot{m}_{s,out} + \dot{m}_e - \dot{m}_c \tag{2.14}$$

$$\frac{dH_l}{dt} = \dot{m}_{l,in}h_{l,in} - \dot{m}_{l,out}h_{l,out} + (\dot{m}_c - \dot{m}_e)\,h'' + \dot{Q}_{sl} + V_l\frac{dp}{dt} \tag{2.15}$$

$$\frac{dH_s}{dt} = \dot{m}_{s,in}h_{s,in} - \dot{m}_{s,out}h_{s,out} + (\dot{m}_e - \dot{m}_c)\,h'' - \dot{Q}_{sl} + V_s\frac{dp}{dt} \tag{2.16}$$

The mass and energy balance equations of the two phases are connected by the evaporation $\dot{m}_e$ and condensation $\dot{m}_c$ rates [5, 6].

$$\dot{m}_e = \begin{cases} \frac{\rho_l V_l (h_l - h')}{\tau_e (h'' - h')} & h_l > h' \\ 0 & h_l \leq h' \end{cases} \tag{2.17}$$

$$\dot{m}_c = \begin{cases} \frac{\rho_l V_l (h' - h_l)}{\tau_c (h'' - h')} & h_l < h' \\ 0 & h_l \geq h' \end{cases} \tag{2.18}$$

The evaporation and condensation relaxation time τ_e and τ_c, which are used in Equation (2.17) and Equation (2.18), are empirical values or can be calculated, as presented by Stevanovic et al. [6]. A heat flow between the superheated steam and the liquid water is also considered in Equation (2.15) and Equation (2.16). Moreover, the total volume is constant and must be equal to the sum of the volumes of each phase. [5, 6]

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Stevanovic et al. [5] validated the non-equilibrium in terms of pressure in a charging example. Moreover, Stevanovic et al. [6] compared results of the non-equilibrium and the equilibrium model with the measurement results for the charging example presented by Stevanovic et al. [5]. For testing the non-equilibrium model, the condensation relaxation time was calculated in one instance, and a constant empirical value was chosen, in another instance. The non-equilibrium model with a constant empirical value for the condensation relaxation time provided the most accurate result. [6]
Furthermore, Stevanovic et al. [5] compared the results of the non-equilibrium and the equilibrium model in terms of pressure in a charging and discharging example, where differences in the results were recognizable. In addition, a comparison between the equilibrium and the non-equilibrium model in terms of pressure with constant empirical condensation and evaporation relaxation time for charging and discharging was presented by Stevanovic et al. [6]. Three different mass flows for charging and discharging were chosen. The analysis showed that the deviation between the equilibrium and the non-equilibrium model increased with increasing mass flow. [6]

2.4 Combined Thermal Energy Storages

Many approaches, which combining different storage types or extend storages with other technologies have been presented in the literature. Some of them are presented in this section, where the focus is on the combination of sensible storage systems, especially water storages and RSSs with LHTESs or PCMs respectively. Moreover, the combination of PCMs and electric heating elements and RSS extensions with other technologies are discussed in this chapter.

2.4.1 Sensible/Latent Heat Thermal Energy Storages

The application of PCM inside a sensible water storage has been discussed in the literature. Abdelsalam et al. [42] studied the integration of PCM inside a water storage vessel, which was charged by a coil heat exchanger. Numerical simulation indicated that it is possible to increase the energy density two to five times compared with a storage vessel that only contains water. They mentioned that the configuration and geometry of the PCM encapsulation has a high influence on the performance of the storage. The results demonstrated that with a PCM volume fraction from 0.025 to 0.15, the charging rate can be increased by 85%. A further increase in PCM volume fraction did not lead to an improvement in terms of charging rate because of the thermal conductivity of the coil. [42]
Underwood et al. [43] studied a water storage vessel where helical coils filled with a PCM was placed at the internal side. Bony and Citherlet [44] presented a model for simulating a

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water storage with PCM modules inside, where the modules can have different shapes. A stratified water storage vessel with a PCM in the upper range was numerically and experimentally investigated by Mehling et al. [45]. An increased storage density of between 20 to 45% was reached with this arrangement.
In the patent application [46], a water storage vessel, surrounded by a chamber filled with PCM was described. The chamber can be divided in several chambers filled with PCMs with different phase change temperatures. Also, one or more containers filled with PCM arranged inside the storage vessel was a concept mentioned in this patent application.
Other sensible storage types in combination with PCM were mentioned in literature. For example, Zauner et al. [37] used thermal oil both as a sensible storage material and as a heat transport medium and arranged it on the shell side of a shell-and-tube heat exchanger. On the tube side of the shell-and-tube heat exchanger PCM was arranged. Geissbühler et al. [47] investigated a packed bed of rocks, where a steel-encapsulated PCM was arranged on top.

2.4.2 Ruths Steam Storage Extension

Two methods to extend RSSs with PCM were presented by Buschle et al. [48] and Tamme et al. [49]. Both methods can be applied to already-existing RSS vessels. The first method provides the arrangement of encapsulated PCM inside the pressure vessel. For encapsulation tubes were proposed. In the second method, several tubes are surrounded by PCM. The charging steam flows inside these tubes and condenses. During discharging steam is generated by leding liquid water through the pipes. The RSS vessel is used as separator and as reservoir in this arrangement. [48]
Buschle et al. [48] compared both methods. For comparing the wall thickness, a pipe with the same outer diameter was used for both arrangements. The results showed that the necessary wall thickness for the external arrangement was smaller than for the encapsulation. Moreover, the solidification time of PCM arranged at the outside of a tube was shorter than that for the encapsulated PCM, where the same tube geometry and PCM amount were used for both arrangements. [48]
Both, the first and second concept menioned above were also presented in the patent documents [50] and [51]. Another patent application [52] discussed the possibility of storing waste heat from vehicles in a vessel, which is filled with a two-phase fluid and a PCM bed. For thermal energy supply, a heat exchanger is used. Mechanical or thermal energy can be generated with the gaseous medium from the storage vessel. [52] Furthermore, the combination of PCM and a liquid–vapour medium in a container was described in the patent application [53]. During solidification, the PCM transfers energy to the liquid transport medium for evaporation. The energy is then transferred by condensation to a heat ex-

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changer. [53]
Because an RSS only provides saturated steam during discharge, Steinmann and Eck [54] mentioned that an additional storage such as a molten salt or a concrete storage can be connected downstream to obtain superheated steam. A storage system consisting of an RSS and an LHTES was described in the patent application [55]. In this concept, the discharging steam with a subcritical temperature from the RSS is superheated to a supercritical temperature in the following LHTES. Also, the arrangement of PCM inside the liquid volume of the RSS was mentioned. An additional storage volume, filled with a liquid medium with storage temperature, to refill the RSS during discharging was also remarked. [55]
Moreover, a two-stage storage system consisting of an RSS and a concrete storage was discussed by Bai and Xu [56]. Furthermore, Steinmann and Eck [54] presented that, in addition to the application of encapsulated PCM inside the RSS, the usage of an external flash evaporator can achieve nearly constant pressure inside the pressure vessel. Therefore, saturated liquid water from the pressure vessel can be led to the evaporator.

2.4.3 Electrical Heating Elements Used in Phase Change Materials

A storage system with electrical heating elements directly integrated into the PCM was discussed by Nepustil et al. [57]. The electrical energy to be stored can be, for example, that which is produced by photovoltaic plants or wind farms. For discharging, a plate heat exchanger and oil as heat transfer medium were used. Furthermore, to increase the heat transfer from the PCM to the oil, a method using moving blades to reach a constant solidified layer thickness on the heat exchanger surface was presented. [57] The method described in the patent application [58] entailed placement of an electrical heating element inside a container filled with PCM. In the container, a heat exchanger with a heat transfer medium was also placed. [58]

2.5 Design Optimization

Investigations in terms of hybrid storage design optimization were done in the course of this work to identify the cheapest hybrid storage design for specified requirements.
Optimization problems can be divided into static, dynamic and stochastic optimization problems. If the optimization environment is static, it is called a static optimization problem, whereas in dynamic optimization problems, the optimization variables are functions. If the optimization problem contains stochastic procedures, then it is a stochastic optimization problem. In general, the aim of static optimization problems is to solve a problem of the form

$$min\left(g\left(\mathbf{x}\right)\right)\ \mathbf{x} \in \mathbb{R}^{\mathbf{n}}, \tag{2.19}$$

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where the equality constraints

$$\mathbf{c}\left(\mathbf{x}\right) = \mathbf{0}\ \mathbf{c} \in \mathbb{R}^{\mathbf{o}} \tag{2.20}$$

and inequality constraints

$$\mathbf{j}\left(\mathbf{x}\right) \leq \mathbf{0}\ \mathbf{j} \in \mathbb{R}^{\mathbf{q}} \tag{2.21}$$

must be met. Here, g is the objective function and n, o, and q the corresponding dimensions, where o must be smaller than n to be able to perform an optimization. If the number of independent equality constrains is equal to the number of optimization variables, the system is determined and an optimization is not necessary. [59]

Investigations to design optimization of TES have already been presented in the literature. Jian et al. [60] designed solid TESs with minimal costs by using the MATLAB optimization toolbox. Beck and Hofmann [61] presented the optimization of a pressurized water storage. In the static optimization problem presented in **Paper 5**, the aim was to find the optimal design that minimizes the cost of the storage system and meets the defined requirements. The optimization problem was formulated in MATLAB [62]. To solve the problem, the GlobalSearch algorithm provided in the MATLAB Optimization Toolbox [62] was used. GlobalSearch uses the solver fmincon and an algorithm to generate multiple start points. The start points will be evaluated and sorted out if there is no chance that the start point provide a good solution. [63]

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Chapter 3

Hybrid Storage Concept

The hybrid storage concept investigated in this thesis is based on the three patent applications by the Austrian Institute of Technology GmbH, as listed in Chapter 1.2. The concept, which is schematically represented in Figure 3.1, concerns an RSS that is combined with an LHTES to create a new storage component.

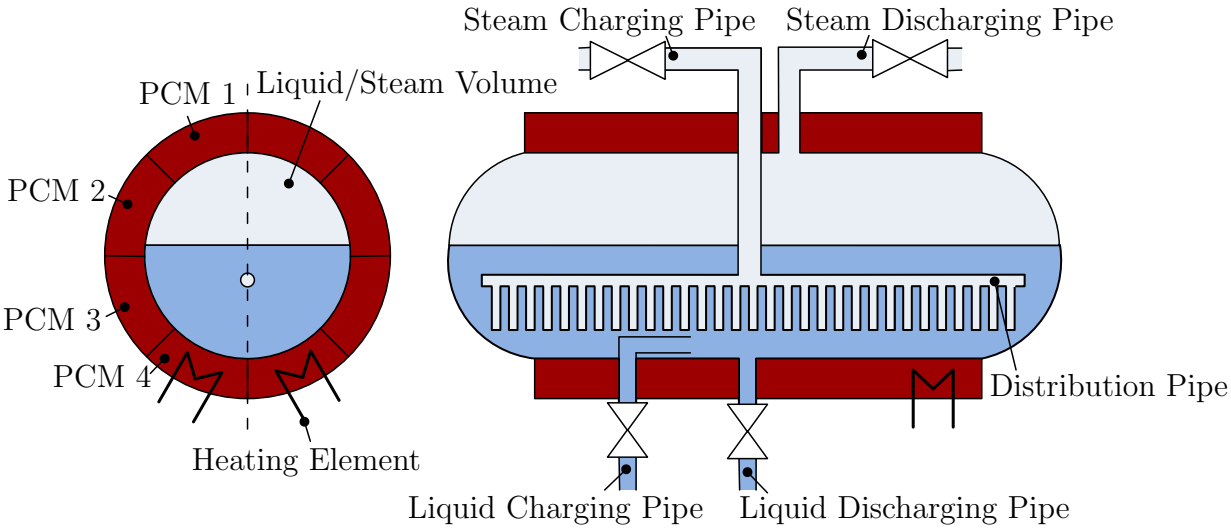

Figure 3.1: Hybrid storage concept.

The pressure vessel of the RSS is surrounded by the LHTES container. More precisely, it is surrounded by the PCM. This results in thermal contact between the two storage materials via the pressure vessel wall. In addition, the LHTES surrounding the RSS vessel shell can be subdivided into several chambers containing PCMs with different properties. In these chambers electrical heating elements or heat exchangers can be integrated. The aim of the hybrid storage concept is to have the following advantages and to compensate the disadvantages of the individual TES types:

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- Higher storage capacity than an RSS, with the same total volume and operating pressure difference. This leads to a smaller pressure vessel volume, which results in lower material costs for the pressure vessel;
- Better performance of the hybrid storage because of the arrangement of PCMs with different properties;
- The possibility of storing electrical energy as thermal energy by using electrical heating elements arranged within the PCM; and
- The possibility of storing waste heat from other process media, by integrating heat exchangers in the PCM.

The basis of all three patent applications is the idea of surrounding an RSS with PCM. In addition, the integration of heating elements in the LHTES is in the foreground in **Application 1** and **Application 3**. However, in **Application 2**, the main topic is the division of the LHTES into several chambers filled with different PCMs. Moreover, the patent applications present the arrangement of elements inside the hybrid storage for increasing the heat transfer between the liquid/steam volume, and the PCM. The patent applications also indicate that the LHTES can be arranged on the shell surface inside the pressure vessel. This type of arrangement is not part of this thesis. Instead, only the arrangement at the outer side of the pressure vessel is discussed. The patent applications also include, a steam discharging pipe passing through the LHTES, but that is also not studied in this work. The hybrid storage concept discussed in this thesis is described, at least briefly, in all the papers listed in Chapter 1.2.

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Chapter 4

Hybrid Storage Models

In the course of all five papers, two different models for analyzing the hybrid storage concept were presented. The models differ from each other in terms of the model approaches that were chosen for modeling the LHTES part of the hybrid storage. In the first hybrid storage model, an interface tracking approach is used, and in the second hybrid storage model, an interface capturing approach, more precisely the apparent heat capacity approach, is used. In this section, the similarities of the two hybrid storage models are briefly described, which is followed by two sections in which the differences of the two hybrid storage models are summarized.

The hybrid storage models were developed in Modelica/Dymola 2015 FD01 [64]. Furthermore, the TIL 3.3 library [65] is used. The entire hybrid storage is modeled as a horizontal cylinder, where the volumes of the dished heads are neglected and the side faces are regarded as adiabatic. Different sub-models (RSS, wall, heat resistor, LHTES) are combined in the hybrid storage model. The sub-models consist of nodes, and the energy flow between these nodes is described using heat conduction equations. In the nodes, mass and energy balance equations have to be fulfilled. The wall sub-model is used to represent the pressure vessel wall, the partition wall between PCM and insulation and the outer panel, and the insulation. Furthermore, the stored sensible energy inside the walls is considered. Therefore, in the wall model, a node is placed at the mean diameter related to mass. The heat resistor model is used to model the resistance between the storage vessel surface and the surrounding air.

The equilibrium model (Chapter 2.3.1) approach is used to model the RSS, and the TILMedia library [65] is used to determine the thermo-physical properties of the two-phase fluid inside the pressure vessel of the RSS. The non-equilibrium approach presented in Chapter 2.3.1 was also considered to model the RSS, but to reduce the complexity, the equilibrium approach was chosen for the investigations concerning the hybrid storage concept. Validation of the RSS sub-model with process measurement data of an operating RSS line including three storage vessels was presented in **Paper 3**. A maximum error of 6.43% in

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terms of pressure and 2% in terms of liquid filling level occurred.
In both modeling approaches for the LHTES part constant and equal liquid/solid PCM properties are assumed and only heat conduction in the radial direction is considered in the LHTES. In addition to the stored latent energy the stored sensible energy in the PCM is also considered.

4.1 Interface Tracking Model

The hybrid storage model, which uses an interface tracking approach, was described in **Paper 1**, **Paper 2**, **Paper 3**, and **Paper 4** and is called interface tracking model (ITM) in the following. In the model shown in **Paper 1** and **Paper 2**, the RSS pressure vessel is surrounded by a homogeneous PCM layer with an exact phase change temperature. For each phase of the PCM, a separate volume was defined. This volumes changes according to the position of the phase interface. To include the stored sensible energy inside the PCM, the mass of each phase is concentrated inside a node placed on the mass mean diameter of the corresponding volume. On the phase interface, another node is placed to include the stored latent energy.
In **Paper 3** and **Paper 4**, an extended model was used, where the LHTES volume can be divided into several chambers with different sizes in the peripheral direction, the partition walls between the chambers are neglected, and the contact between the PCMs is assumed to be adiabatic. These chambers could be filled with different PCMs, and a symmetrical structure related to the vertical axis of symmetry is assumed. To model the electrical heating elements or heat exchangers, in each chamber, an additional heat source is placed at the inner diameter. Depending on the number of LHTES chambers, the included wall volumes are also divided into the same number of partial volumes. The heat transfer by heat conduction in the peripheral direction between the wall volumes is also modeled. Moreover, in the extended version, different heat transfer coefficients between each phase of the two-phase fluid and the pressure vessel wall are included. Furthermore, depending on a positive or negative temperature difference between the temperature at the inner diameter of the pressure vessel wall and the temperature of the two-phase fluid, different heat transfer coefficients can be selected. If the temperature at the inner diameter of the pressure vessel wall is lower than the temperature of the two-phase fluid, the steam phase condenses at the pressure vessel wall and no phase transition of the liquid phase occurs at the inner diameter of the pressure vessel wall. Appropriate heat transfer coefficients can be selected for the two phases of the two-phase fluid, where the heat transfer coefficient between the pressure vessel wall and condensing steam is significantly higher than that between the pressure vessel wall and the liquid phase. If the temperature at the inner diameter of the pressure vessel wall

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is higher than the temperature of the two-phase fluid, the liquid phase evaporates at the pressure vessel wall and no phase transition of the steam phase takes place at the inner diameter of the pressure vessel wall. Additionally, for this case, appropriate heat transfer coefficients can be selected, and the heat transfer coefficient between the steam phase and the pressure vessel wall will be significantly lower than that between the evaporating liquid phase and the pressure vessel wall.

As presented in **Paper 3**, for each wall volume, the heat transfer area of each phase inside the liquid/steam volume is calculated. Thus, for each wall volume, two heat transfer areas are determined, one for the steam and one for the liquid phase. To calculate the heat transfer areas, the actual filling level must be known. Therefore, angle γ, which describes the actual filling height (see Figure 4.1), is determined with Equation (4.1). With the resulting heat transfer areas and the corresponding heat transfer coefficients, a mean heat transfer coefficient can be determined for each pressure vessel wall volume. Therefore the heat flows between the liquid/steam volume and the wall volumes can be calculated.

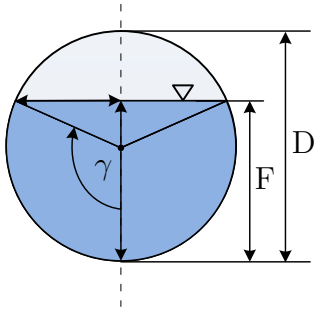

Figure 4.1: Calculation of angle γ, which describes the actual filling height.

$$\gamma = arccos\left(\frac{D-2F}{D}\right) \tag{4.1}$$

In **Paper 3**, the entire extended ITM was described in detail and was presented graphically using a bond graph. More details about bond graph modeling can be found in Thoma and Ould Bouamama [66].

4.2 Apparent Heat Capacity Model

In **Paper 5**, another hybrid storage model was presented. As mentioned before, this model mainly differs from the non-extended ITM with respect to the chosen modeling approach for the phase change procedure. In this model, the apparent heat capacity method (see Chapter 2.2.2 and **Paper 5**) is used. Therefore, in this work, it is called the apparent heat capacity model (AHCM). In the AHCM, the RSS is surrounded by only a single PCM and

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is discretized into several partial volumes in the radial direction. In each volume, a node is placed at the mean diameter to consider the stored energy (sensible and latent stored energy) inside the partial volume. As in the extended ITM, a mean heat transfer coefficient between the heat transfer coefficients of each phase that depends on the actual filling height was applied in this model. Furthermore, for charging and discharging, different heat transfer coefficients are used. An advantage of the AHCM is that it is possible to simulate partial load states.

In the following diagrams, simulation results of both hybrid storage models are compared (non-extended ITM and AHCM). Therefore, the example presented in **Paper 2** is used. In this example, a hybrid storage was charged with a valve for 50 minutes (26 bar_a charging steam pressure and 227°C charging steam temperature) and, after a 20-minutes storing phase, the storage was discharged with a valve for 50 minutes (20 bar_a pressure after the valve). Unlike the ITM, where an exact phase change temperature (220°C [67]) is used, in the AHCM, an insignificantly small temperature range is used (see **Paper 5**). Since the temperature range is very small, in terms of the AHCM, it is also called a phase change temperature instead of a phase change temperature range. In addition, for the comparison, only one heat transfer coefficient for the two-phase fluid (4,700 $Wm^{-2}K^{-1}$ [68]) is applied in the AHCM.

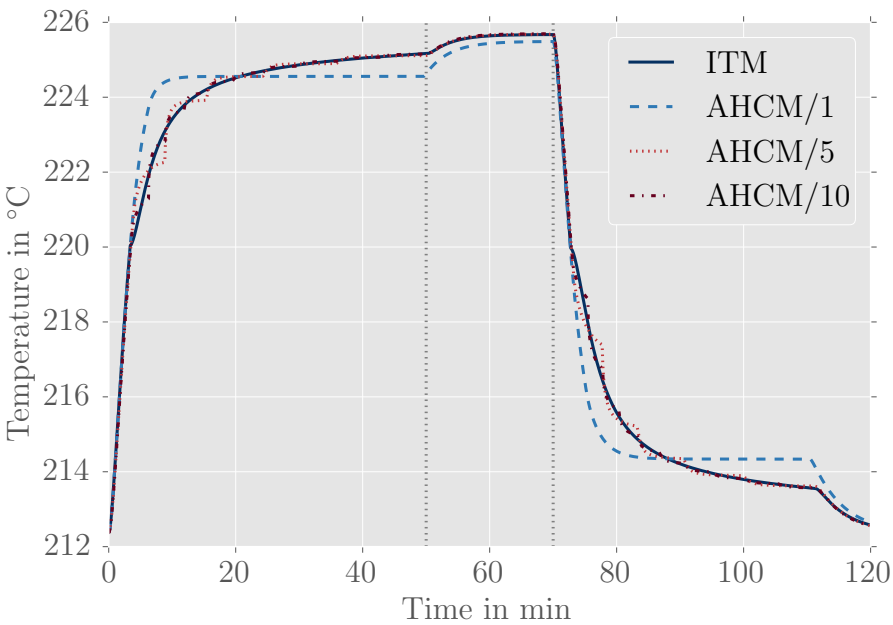

Figure 4.2: Simulation results in terms of inner temperature of the LHTES cylinder for the ITM and AHCM with different numbers of PCM volumes in the radial direction (number after the slash).

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In Figure 4.2 and Figure 4.3, the inner and outer temperature, respectively, of the LHTES cylinder are presented. In Figure 4.2, it can be seen that with the AHCM with one PCM volume in radial direction (AHCM/1), the inner temperature of the LHTES cylinder increases after approximately 4 minutes faster than in the ITM or in the AHCM with five (AHCM/5) and ten partial (AHCM/10) PCM volumes in radial direction. This is because with AHCM/1, the mean temperature of the total PCM volume must first reach the phase change temperature before it starts to melt and thus absorbs latent energy. More precisely, before latent energy can be absorbed, the entire sensible energy to reach the phase change temperature must be absorbed into the PCM. In AHCM/5 and AHCM/10, on the contrary, the total volume is divided into several partial volumes in which only the mean temperature of the partial volume must reach the phase change temperature to start melting. Therefore, the PCM in AHCM/5 and AHCM/10 absorb latent heat earlier. In the ITM model, only the inner temperature of the LHTES cylinders must reach the phase change temperature before the PCM starts melting. This influence can also be seen in the outer temperature of the LHTES cylinders in Figure 4.3. The outer temperature of the LHTES cylinder increases faster after approximately 4 minutes with AHCM/1 than in the other simulations (ITM, AHCM/5, AHCM/10).

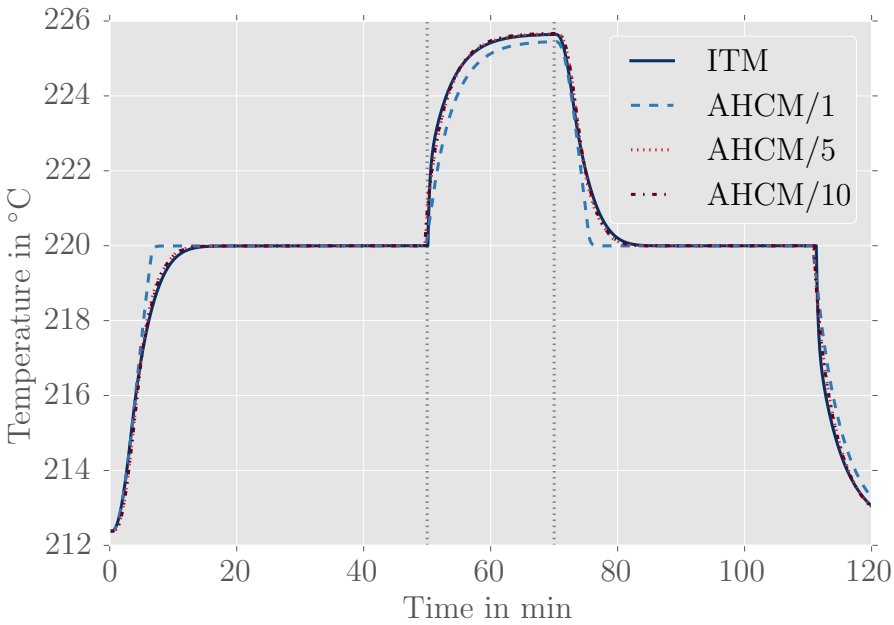

Figure 4.3: Simulation results in terms of outer temperature of the LHTES cylinder for the ITM and AHCM with different numbers of PCM volumes in the radial direction (number after the slash).

Moreover, Figure 4.4 depicts the pressure inside the liquid/steam volume of the hybrid storage. The diagram shows that the pressure of the AHCM/1 is higher between approximately

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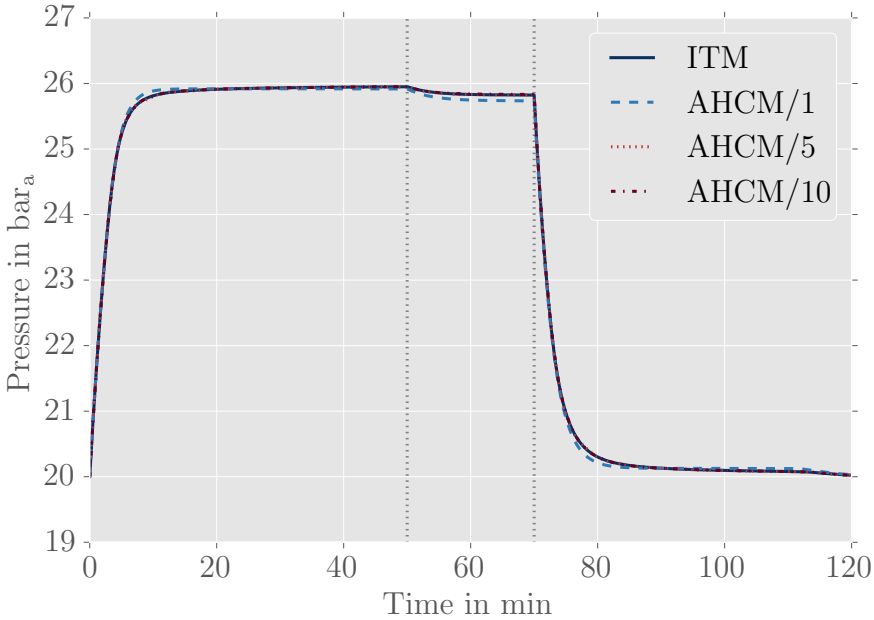

Figure 4.4: Simulation results in terms of pressure inside storage vessel for the ITM and AHCM with different numbers of PCM volumes in the radial direction (number after the slash).

5 minutes and 21 minutes because latent energy is absorbed later in this case. Therefore, the charging mass flow is lower earlier, which results in a lower amount of stored energy in the AHCM/1 than in the AHCM/5, AHCM/10, and ITM. This can be seen in Figure 4.5. Furthermore, the effect results in a constant pressure after 12 minutes in the AHCM/1. The pressure in the AHCM/5, AHCM/10, and ITM exceed this pressure value after 21 minutes. The same effect can be observed in terms of inner LHTES cylinder temperature.

At the end of the charging phase in all simulations, the PCM is nearly completely melted, which can be seen in Figure 4.3, because during the storing phase, the outer temperature of the LHTES cylinder further increases. Moreover, this further increase results in a decrease in pressure during the storing phase. Because of the lower pressure at the end of the charging phase, the pressure at the end of the storing phase also is lower in the AHCM/1 than in the other simulations. The lower pressure results also in a weaker rise in inner and outer LHTES cylinder temperatures during the storing phase. The same behavior as during charging is also noticeable during discharging.

In summary, it can be seen that the results from the AHCM hybrid storage model with an increasing number of partial PCM volumes in the radial direction are more consistent with the ITM. A difference between the results of the AHCM/5 and AHCM/10 is not noticeable in terms of pressure inside the liquid/steam volume, stored energy, or outer LHTES cylinder temperature.

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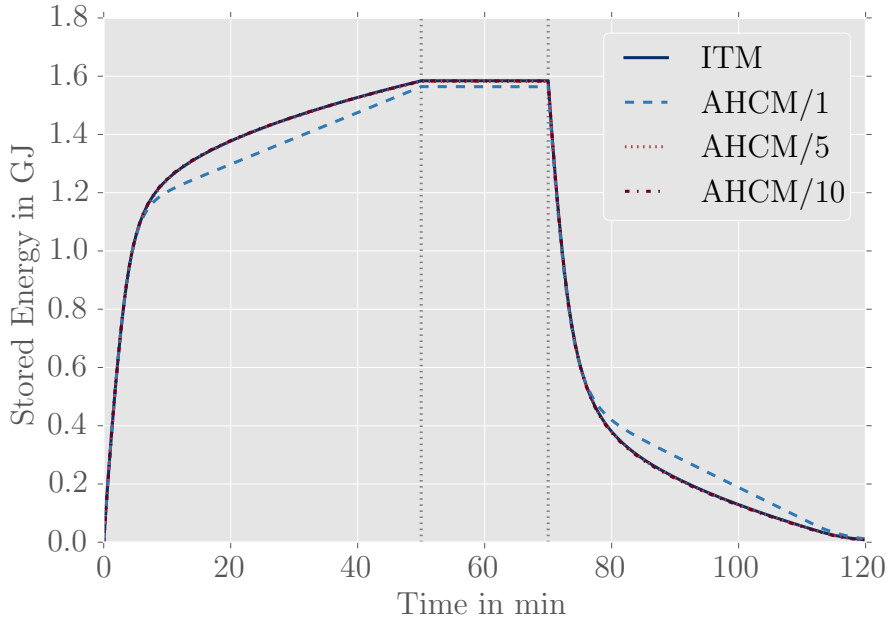

Figure 4.5: Simulation results in terms of stored energy for the ITM and AHCM with different numbers of PCM volumes in the radial direction (number after the slash).

In Figure 4.2, a difference between the simulation results of the AHCM/5 and AHCM/10 can be seen during phase change. More precisely, a step-shaped curve is more clearly recognizable with five partial volume. This step-shaped curve is attributable to the fact that a partial volume must have completed the phase change when reaching the phase change temperature before the temperature of the partial volume continues to rise or fall. Therefore, an increase in the number of partial volumes results in smaller steps and closer agreement with the ITM model.

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Chapter 5

Hybrid Storage Concept Analysis

In **Paper 1**, **Paper 2**, **Paper 3**, and **Paper 4**, the thermodynamic behavior of the hybrid storage concept, was studied using different examples in terms of temperature range and total storage volume. Therefore, the simulation results of the ITM, described in Chapter 4, were used. In all these papers, the simulation results of the hybrid storage were compared with the results of a comparable RSS. In **Paper 1** and **Paper 2**, the simulation results of the hybrid storage were compared with the simulation results of an RSS with the same total volume. For the analyses done in **Paper 3** and **Paper 4**, the results for the hybrid storages were compared with the results for the RSS with the same pressure vessel geometry. Furthermore, in **Paper 1** and **Paper 2**, for the PCMs used in the examples, increased thermal conductivities compared with the real thermal conductivities, were applied. How to achieve the increased thermal conductivities was outside the scope of these papers. However, in **Paper 3** and **Paper 4**, an adaptation in phase change enthalpy and specific heat capacity corresponding to the increase in thermal conductivity was carried out under the assumption of an ideal mixture of highly thermally conductive material and the respective PCM.

In the following, the main results of the papers are summarized. First, the results of **Paper 1** and **Paper 2**, which used simulations of the non-extended version of the ITM, are discussed. Then, the examples and results from simulations of the extended version of the ITM, discussed in **Paper 3** and **Paper 4**, are presented in summary.

In **Paper 1**, a hybrid storage with a total volume of 29 m^3 was charged with a valve for 120 minutes. The pressure and the temperature of the inlet steam mass flow were assumed to be constant (5 bar_a and 152°C). Furthermore, in this example a pressure range from 2 to 5 bar_a was chosen. The mean temperature inside the liquid/steam volume was approximately 136°C, and therefore, for the PCM, high-density polyethylene with a phase change temperature of 135°C and a latent heat of 245 $kJkg^{-1}$ was chosen [69]. An increased thermal conductivity of 5 $Wm^{-1}K^{-1}$ was assumed for this PCM. The example demonstrated the following major results:

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- The hybrid storage (10% PCM volume) stored approximately 30% more energy than the RSS; and
- Most of the time during charging, the pressure in the hybrid storage is lower than in the RSS.

The example presented in **Paper 2** consisted of a 50-minutes charging, 20-minutes storing, and 50-minutes discharging phase. Charging and discharging were accomplished via a valve, where 26 bar_a and 227°C were chosen for the charging steam mass flow. As in **Paper 1**, 29 m^3 was the total volume of the hybrid storage used in the example presented in **Paper 2**. Based on the selected pressure range of 20 to 26 bar_a and an average temperature of approximately 219°C, $NaNO_3/KNO_3$ with a phase change temperature of 220°C [67] and an assumed thermal conductivity of 6 $Wm^{-1}K^{-1}$ was chosen for the PCM. The example exhibited the following major results:

- The hybrid storage (4.6% PCM volume) stored approximately 29% more energy than the RSS;
- Most of the time during charging, the pressure in the hybrid storage is lower than in the RSS;
- Pressure decreased during the storing phase; and
- Most of the time during discharging, the pressure in the hybrid storage is higher than in the RSS.

The hybrid storage in **Paper 2** achieved nearly the same improvement as the hybrid storage in **Paper 1**, but the PCM volume was smaller and the charging time was shorter in **Paper 2** than in **Paper 1**. The major reason for this was the lower storage capacity of the RSS for the pressure range chosen in **Paper 2** than for the pressure range chosen in **Paper 1**.
Unlike in **Paper 1** and **Paper 2**, in the example of **Paper 3**, a constant charging and discharging mass flow were chosen to accomplish the same amount of charged and discharged energy for the hybrid storage and the RSS. Compared with using a valve, a constant mass flow results in a more uniform increase or decrease in pressure and temperature of the liquid/steam volume. In the examples in **Paper 1** and **Paper 2**, the pressure increased fast at the beginning of charging or discharging, and afterwards, the pressure was constant at nearly the maximum or minimum temperature.
In **Paper 3**, the liquid/steam volume of the studied hybrid and RSS was 55 m^3, and for the PCM, a layer thickness of 5 cm was chosen for use in the hybrid storage, where this part was divided into three chambers per side of the symmetrical cross-section in the peripheral direction, resulting in six LHTES chambers around the pressure vessel. The example

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presented in **Paper 3** consists of a 90-minutes charging phase with steam (0.69 kgs^{-1}), 60-minutes charging with additional heat sources (780 kW), 10-minutes break and 150-minutes discharging phase (0.51 kgs^{-1}). For the charging steam conditions, 28 bar_a and 231°C were selected, and the minimum pressure was set to 14 bar_a. Furthermore, for the PCM, a material with a phase change temperature of 220°C [70] was applied. To compare the results, an additional heat source was placed in the liquid/steam volume of the RSS and both storages were charged and discharged with the same amount of energy. The example demonstrated the following major results:

- At the end of charging with steam, lower pressure was reached in the liquid/steam volume of hybrid storage than in the RSS;
- At the end of charging with additional heat sources, a 5 bar_a lower pressure was reached in the liquid/steam volume of the hybrid storage than in the RSS; and
- A limiting effect resulting from the assumed low heat transfer coefficient between steam and the connected wall during discharging was identified.

The arrangement of several PCMs around the pressure vessel were studied in **Paper 4**. Three different operating strategies were investigated, one for charging (60 minutes), one for discharging (60 minutes) and one for charging with additional heat sources (10 minute) after charging with steam (60 minutes). For each example, 29 different PCM arrangements were simulated. The arrangements differed in the number of arranged PCMs (maximum of three different PCMs), where the PCMs had different material properties or only different phase change temperatures. Furthermore, the placement of the different PCMs was investigated. Moreover, for the discharging example, different heat transfer coefficients for steam were tested. For the investigations, a pressure vessel volume of 33 m^3 and a PCM layer thickness of 10 cm were chosen. The LHTES was divided into six chambers per side of the circular cross-section. Furthermore, a pressure range of 4 to 30 bar_a for charging and discharging with steam was chosen. The following findings were made in **Paper 4**:

- A high temperature difference between the temperature in the liquid/steam volume and the phase change temperature is advantageous for the performance of the hybrid storage. However, a maximum temperature difference for both charging and discharging is not possible with only one PCM;
- In addition to the phase change temperature, the storage capacity of the PCMs is also an important factor. The results indicated that it depends on the PCM properties whether an arrangement with several different PCMs improves the performance of the hybrid storage;

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- The influence, in terms of PCM placement, is higher for discharging when a low heat transfer coefficient between the steam phase and the connected wall occurs. A low heat transfer coefficient of steam can significantly limit the power of the hybrid storage; and

- A lower maximum pressure in the liquid/steam volume can be reached when the additional heat sources are placed in the hybrid storage at the inner diameter of the LHTES cylinder than in the liquid/steam volume of an RSS with the same pressure vessel volume.

5.1 Discrestisation of the Pressure Vessel Wall

This chapter discusses the influence of the division of the pressure vessel wall into several volumes in the peripheral direction resulting from the division of the LHTES volume into several chambers. Therefore, the example and simulation results presented in **Paper 3** are used. As mentioned in **Paper 3**, the LHTES volume was divided into three chambers per side of the symmetrical cross-section, as can be seen in Figure 5.1, where in each chamber the same PCM was used. As described in **Paper 3**, four different heat transfer coefficients between the two-phase fluid and the pressure vessel wall were used in this example. For condensing steam, 5000 $Wm^{-2}K^{-1}$, steam 10 $Wm^{-2}K^{-1}$, evaporating liquid water 1000 $Wm^{-2}K^{-1}$, and liquid water 700 $Wm^{-2}K^{-1}$ were used [71].

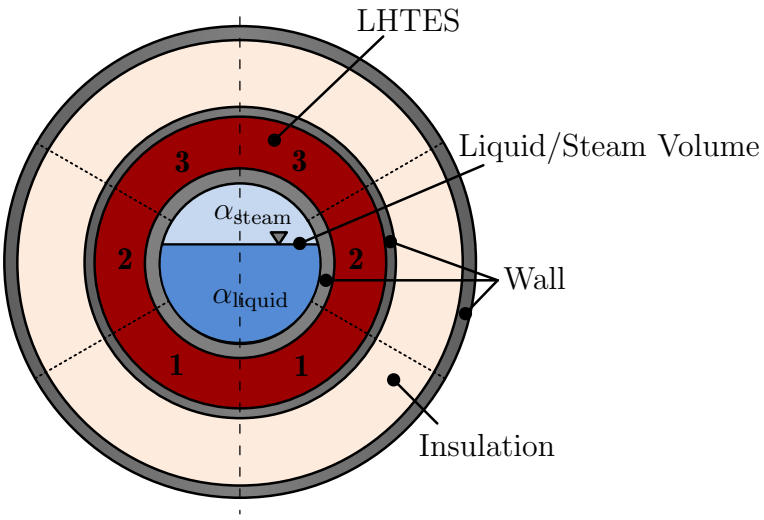

Figure 5.1: Hybrid storage discretisation in the peripheral direction.

In Figure 5.2, the influence of the division in terms of mean temperatures for each pressure vessel wall volume can be seen. The increase or decrease of the mean temperature depends on the position of the wall volume and thus on the heat transfer between the pressure vessel

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wall volume and the two-phase fluid. For better understanding, Figure 5.3 depicts the heat transfer area between the steam phase and the pressure vessel wall volumes. In this figure, it can be seen that the second pressure vessel wall volume is only in contact with the steam phase for a short time and in a very small proportion. Furthermore, the first pressure vessel wall volume is only in contact with the liquid phase. Therefore, no difference is noticeable in terms of mean temperature (Figure 5.2) of the first and second pressure vessel wall volume.

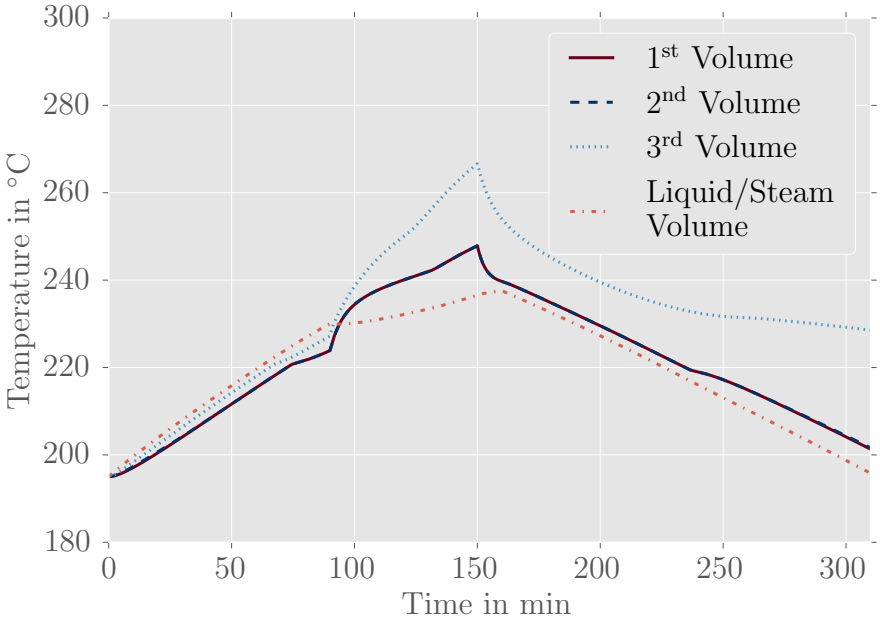

Figure 5.2: Mean temperature inside the pressure vessel wall volumes and temperature inside liquid/steam volume.

As can be seen in Figure 5.2, during charging with steam (first 90 minutes) the temperature of the liquid/steam volume is higher than the temperatures at the inner diameters of the pressure vessel wall volumes. Furthermore, the temperature of the highest pressure vessel wall volume (third volume, see Figure 5.1) increases faster than in the lower pressure vessel wall volumes, due to the fact that the highest pressure vessel wall volume is mainly in contact with the steam phase and the high heat transfer coefficient of condensing steam occurs. The lower pressure vessel volumes are, for the most part, in contact with the liquid phase, which has a lower heat transfer coefficient. Furthermore, shortly after the start of the charging process with additional heat sources at the inner diameter of the LHTES cylinder (after 90 minutes), a larger difference between the mean wall temperatures of the third and the other two pressure vessel volumes can be seen in Figure 5.2. Due to the mean wall temperature increases above the two-phase fluid temperature, the steam does not condense further at the pressure vessel wall and the much lower assumed heat transfer

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coefficient of steam becomes effective. This leads to greater thermal resistance between the third pressure vessel wall volume and the two-phase fluid and thus to a higher temperature increase. This effect can also be seen during discharging (after 150 minutes), where the mean temperature of the third pressure vessel wall volume decreases significantly slower than the mean temperatures of the first and second pressure vessel wall volume.

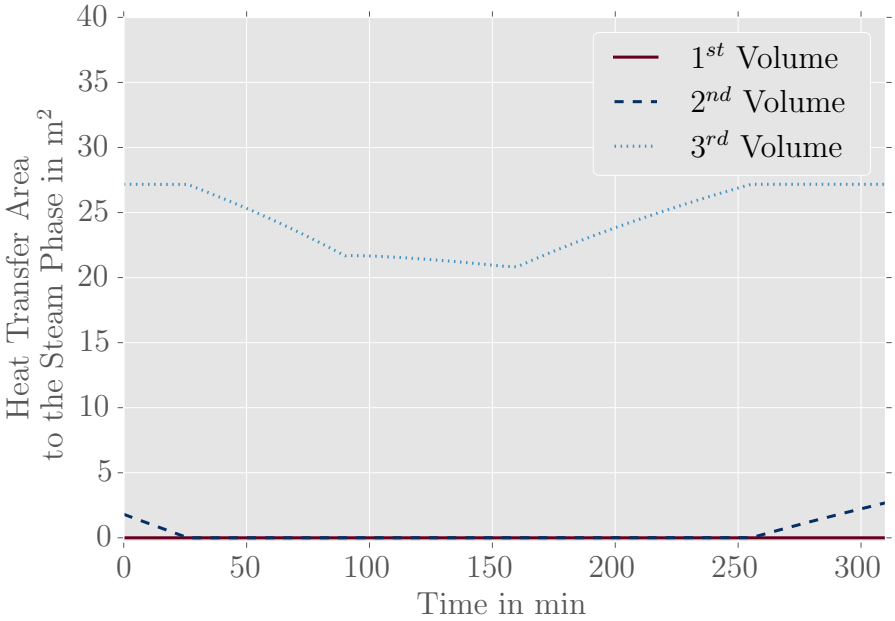

Figure 5.3: Heat transfer area between the pressure vessel wall volumes and the steam phase.

As mentioned in Chapter 4.1, the heat transfer in the peripheral direction between the wall volumes is also modeled, which is described in detail in **Paper 3**. The heat flow in peripheral direction between the first and the second and between the second and third pressure vessel wall volume is presented in Figure 5.4. No heat flow is transferred between the pressure vessel wall volumes bordering the symmetry line because the temperatures in these two volumes are equal. It can be see that the heat flow between the first and second pressure vessel wall volume is approximately zero. This accords with the fact that there is no difference noticeable between the temperatures in the first and second wall volume. On the contrary, there is a heat flow between the second and third wall volume, but this heat flow is very small, also when the temperature difference between these two pressure vessel wall volumes is high. Because of the beginning of the phase change in the three LHTES volumes (in the first and second LHTES volume after 76 minutes and in the third LHTES volume after 69 minutes) and the starting of charging with additional heat sources (after 90 minutes), fluctuations are visible in the heat transfer flow between the second and third wall volume (Figure 5.4). Corresponding changes in the slope of the mean temperature are

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visible in Figure 5.2. Afterwards, the heat flow increases until charging is finished. At the beginning of discharging, the heat flow decreases for a short time, because the temperatures are approaching each other for a short time. The reason is the high temperature difference between the third wall volume and the liquid/steam volume. A further short-term change in the slope of the heat flow is visible after 239 minutes as the PCM begins to solidify in the first and second LHTES volume. In the simulated time span, the PCM in the third LHTES volume did not start to solidify. Figures depicting the discussed example in terms of inner and outer LHTES cylinder temperature and positions of the phase interface for each LHTES volume are presented in **Paper 3**.

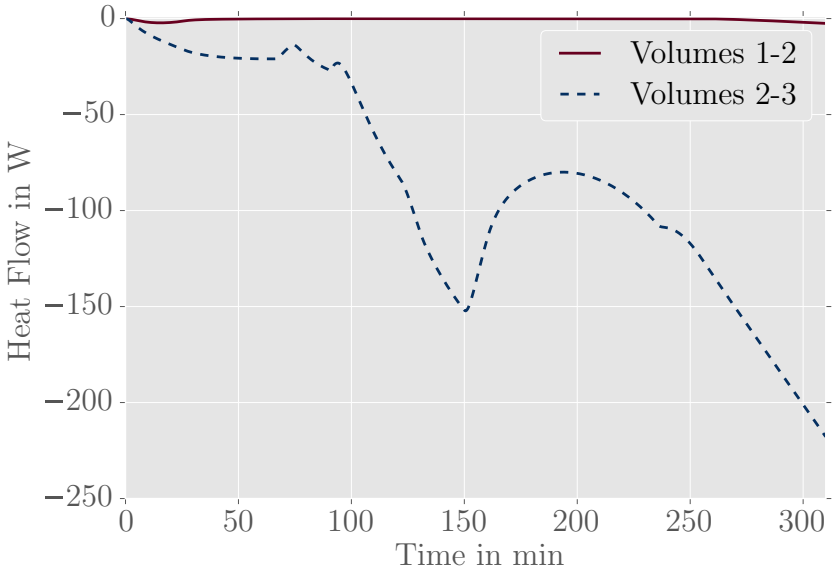

Figure 5.4: Heat flow between the pressure vessel wall volumes in the peripheral direction of one circular cross-section.

For the comparison presented in the following, the example was also simulated without dividing the LHTES volume and therefore without dividing the pressure vessel wall. The results, in terms of pressure inside the liquid/steam volume and stored energy, are shown in Figure 5.5 and Figure 5.6. It can be seen that the pressure curve hardly differs, and in terms of stored energy, no visible difference appears over the entire cycle. Therefore, for this example, no peripheral subdivision is necessary. Furthermore, the investigation leads to the conclusion that, in the wall volumes, the heat transfer in the peripheral direction is negligible.

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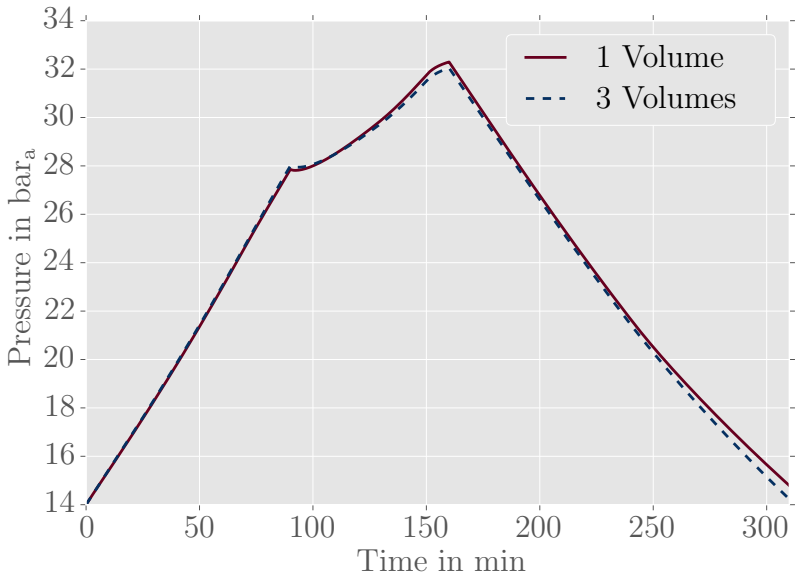

Figure 5.5: Pressure inside the liquid/steam volume for various LHTES volumes.

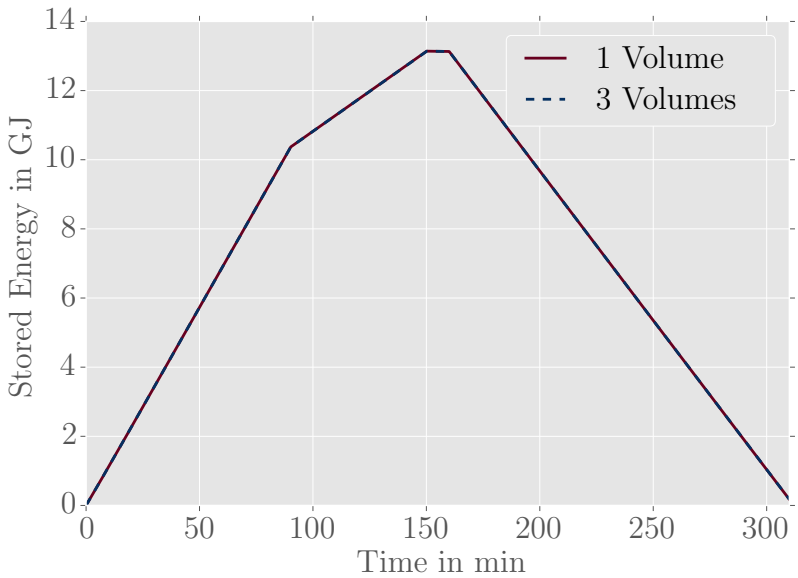

Figure 5.6: Stored energy inside the hybrid storage for various LHTES volumes.

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Chapter 6

Hybrid Storage Design Optimization

The hybrid storage combines the ability to realize high charging and discharging rates of an RSS with the high storage capacity of an LHTES. When designing the hybrid storage the liquid/steam volume and the LHTES volume must be suited to fulfill the process requirements. Attention must be paid to the low thermal conductivity of the PCM, which can limit the charging and discharging rates of the LHTES. Therefore, not only the storage capacity but also the time in which the steam must be provided or stored is an important design factor. For designing a hybrid storage system with the lowest possible investment costs for defined requirements, a design optimization model, created in MATLAB [62], is presented in **Paper 5**.

In the optimization model, a simplified storage geometry, where the pressure vessel is approximated as a cylinder with plane-side faces, is used. The shell surface of the pressure vessel is surrounded by a homogeneous mixture of PCM and a highly thermally conductive material. Around this layer and on the side faces, an insulation layer with a defined thickness is arranged.

Moreover, in **Paper 5**, the results of the hybrid storage design optimization model for two theoretical use cases were discussed. In addition, the results of an RSS systems design optimization for the two use cases, were also presented to compare the investment costs for the two storage system types. For designing the storage systems, it was assumed that the requirements on the storage system are known. This means that the following parameters must be given before the optimization can be started:

- Minimum and maximum pressure inside the hybrid storage pressure vessel;
- Maximum stored energy;
- Limit value for the phase change time of the PCM; and
- Maximum liquid filling level inside the pressure vessel.

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Due to the fact that the maximum and minimum pressure were assumed as given, the properties of the two phases inside the pressure vessel can be determined. Therefore, CoolProp [72], an open-source library for determining thermo-physical properties of pseudo-pure fluids, was used. In the two-phase region, the temperature can be determined at given pressure. Thus, with the given maximum and minimum temperature, the PCM can be chosen. In the use cases presented in **Paper 5**, the PCM was chosen so that the phase change temperature lies in the middle between the maximum and minimum temperature. In addition to the PCM and highly thermally conductive material properties, the properties of the pressure vessel material must be given in order to start the optimization model. Furthermore, a heat transfer coefficient must be selected between the vessel wall and the two phase fluid.
Equality and inequality constraints are used in the optimization model. For example the mass balance equation is formulated as equality constraint and the energy balance equation is formulated as inequality constraint. Furthermore, the analytic solution for the phase change time of a PCM layer surrounding a cylinder is used in the model as an inequality constraint to limit the phase change time of the LHTES part. For the use cases presented in **Paper 5**, the storage system was only designed for charging or discharging depending on which of the two operations had to be faster.
A temperature difference is needed in the analytic solution for the phase change time. For charging, the temperature difference between the maximum temperature and the phase change temperature was used. For discharging, the temperature difference between the minimum temperature and the phase change temperature was applied. During the operation, the temperature inside the RSS changes, and therefore, the temperature difference between the liquid/steam volume and the PCM also changes. Furthermore, the sensible cooling or heating of the PCM before the phase change starts is not considered in this equation. Therefore, the time required for a complete phase change would be underestimated, with the chosen temperature difference. For considering this facts, the temperature differences were reduced by the temperature difference fraction f_T, as it is shown in Equation (6.1). The temperature difference fraction f_T must be greater than zero, and if this factor is equal to 1, the entire temperature difference would be effective.

$$\Delta T = \begin{cases} (T_{max} - T_{pc}) f_T & \text{for charging} \\ (T_{pc} - T_{min}) f_T & \text{for discharging} \end{cases} \tag{6.1}$$

As it was described in **Paper 5**, the results of the optimization for different temperature difference fractions were tested with the AHCM. It was found that with an assumption of a temperature difference fraction of 0.2, the pressure limits are adhered to with sufficient accuracy for the two presented use cases. Furthermore, in the analytic solution for the phase change time, an effective thermal conductivity and an effective phase change enthalpy are

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used in the optimization model to consider the mass of the highly thermally conductive material included in the LHTES cylinder.
Moreover, in the design optimization model it was assumed that the designed hybrid storage system can consist of several storage vessels with the same geometry. Therefore, the number of storage vessels is also calculated in the design optimization. As described in Chapter 2.5 and **Paper 5**, the solver GlobalSearch from the MATLAB Optimization Toolbox [62] was chosen for solving the optimization problem. Thus, the result for the number of storage vessels can not be restricted to an integer value. Therefore, a maximum of three optimization runs were done for the use cases presented in **Paper 5**. The rounded-up and -down values from the first optimization run were used in two further optimization runs to fix the number of storage vessels to an integer value by using upper and lower bounds.
Another important part of the optimization model is the cost function. In **Paper 5**, the cost function included PCM, highly thermally conductive material, the pressure vessel material, and other costs. The pressure vessel costs and other costs were derived from experience values for RSSs. For approximating the pressure vessel costs, a linear function depending on the mass was created. For the other costs, including transport, insulation, and assembly costs, a third-degree polynomial function was used for approximation. In summary, the pressure vessel mass, the PCM mass, the highly thermally conductive material mass, the storage vessel surface, and the number of storage vessels are needed in the cost function presented in **Paper 5**.
The investigations presented in **Paper 5** demonstrated that the hybrid storage system can be advantageous in terms of costs, storage vessel volume, and number of needed storage vessels. However, it must be taken into account that the used-cost function must be adapted in the future because the real implementation of the hybrid storage component is not considered and the other costs are based experience values for RSSs, respectively. Therefore, additional costs can occur. Also, costs for the integration of each storage vessel into the process must be considered in the next step.

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Chapter 7

Papers

7.1 Paper 1

Peer-reviewed conference contribution:
S. Dusek, R. Hofmann. A Hybrid Storage Concept for Improving Classical Ruth's Type Steam Accumulators. In *Proceedings of the 12th Conference on Sustainable Development of Energy, Water and Environment Systems*, Dubrovnik, Croatia, October 4-8, 2017.

In this conference contribution, the hybrid storage concept was presented. Moreover, a hybrid storage model that considers an RSS surrounded by a single PCM was discussed. For the heat transfer coefficient between the two-phase fluid and the surrounding wall, a constant value is used in this model. The interaction between the RSS and the LHTES was studied by charging the hybrid storage component.

7.2 Paper 2

Peer-reviewed journal contribution:
S. Dusek, R. Hofmann. A Hybrid Energy Storage Concept for Future Application in Industrial Processes. *Thermal Science*, 22(5): 2235-2242, 2018, `https://doi.org/10.2298/TSCI171230270D`.

In this paper, the hybrid storage concept and the hybrid storage model were again presented. Moreover, the simulation results for a hybrid storage with the same total volume as in **Paper 1** were shown. Unlike the example presented in **Paper 1**, besides charging, discharging of the storage component was included in the example. Also, a higher pressure level than in **Paper 1** was chosen.

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7.3 Paper 3

Peer-reviewed journal contribution:
S. Dusek, R. Hofmann. Modeling of a Hybrid Steam Storage and Validation with an Industrial Ruths Steam Storage Line. *Energies*, 12(6):1014, 2019, `https://doi.org/10.3390/en12061014`.

In this paper, an extended version of the hybrid storage model presented in **Paper 1** and **Paper 2**, was described in detail. The hybrid storage model discussed in this paper includes different heat transfer coefficients for the two phases inside the pressure vessel, and a distinction in heat transfer coefficients is also made between charging and discharging of the LHTES volume. Moreover, this model allows the LHTES volume to be divided into several chambers in the peripheral direction and different PCM properties to be defined for each chamber. Depending on the division of the LHTES volume, the walls and the insulation layer contained in the hybrid storage vessel are also divided into partial volumes, and heat conduction in the peripheral direction is considered. For a first analysis of the integration of electrical heating elements, additional heat sources are included inside the LHTES chambers.
In the second part, the validation of the RSS model part with the measurement data from an industrial RSS line in operation mode was shown. In the last part, the simulation results of a charging-discharging example were presented. Furthermore, charging by additional heat sources was included in this example.

7.4 Paper 4

Peer-reviewed journal contribution:
S. Dusek, R. Hofmann, S. Gruber. Design Analysis of a Hybrid Storage Concept Combining Ruths Steam Storage and Latent Thermal Energy Storage. *Applied Energy*, 251: 113364, 2019, `https://doi.org/10.1016/j.apenergy.2019.113364`.

In this paper, simulation results of the model presented in **Paper 3** were used to analyze different parts of the hybrid storage concept. The arrangement of a PCM around the pressure vessel, compared with the arrangements of two and three different PCMs was investigated. In first case, the PCMs only differed in terms of phase change temperature, and in the second case, they differed also in terms of density, phase change enthalpy, and specific heat capacity. A charging and a discharging example were used for the analysis of the above described arrangements. Moreover, an example was discussed in which the hybrid storage was also charged by additional heat sources placed in the LHTES volume.

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Investigations on a layer arrangement of a highly thermally conductive material and PCM, based on Gruber [26], were also presented in **Paper 4** but are not discussed in this work.

7.5 Paper 5

Peer-reviewed journal contribution:
R. Hofmann, S. Dusek, S. Gruber, G. Drexler-Schmid. Design Optimization of a Hybrid Steam-PCM Thermal Energy Storage for Industrial Applications. *Energies*, 12(5):898, 2019, `https://doi.org/10.3390/en12050898`.

A hybrid storage system design optimization model created in MATLAB was presented in this paper. The aim of this model is to calculate a hybrid storage system with the lowest possible costs that meets the process requirements. The paper presented the optimization results for two theoretical use cases. The results, were verified by a hybrid storage model created in Modelica/Dymola [64]. Moreover, **Paper 5** discussed a theoretically design concept of a hybrid storage component, which was also presented by Gruber [26], and the results of a study on retrofitting an RSS to a hybrid storage component, but these two topics are not part of this work.

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Chapter 8

Conclusion

In this thesis, a hybrid storage concept, which should be used in industry in the future to support an increase in flexibility and efficiency of processes, was investigated. The investigated hybrid storage is a combination of a quick-reacting RSS and an LHTES with high storage capacity surrounding the pressure vessel. In addition, the integration of electrical heating elements or heat exchangers in the PCM is included in the hybrid storage concept. In the course of completing this work, five scientific contributions were published, including a conference paper and four journal papers.
The aim of this work was to investigate the interaction of the different components that are combined in the hybrid storage concept to obtain more information regarding the advantages and disadvantages and the field of application for the newly developed storage component. To study the hybrid storage concept, models were developed in Modelica/Dymola. The presented models consist of various sub-models, where the RSS and the LHTES sub-model are the most important. The RSS model exhibited close accordance with operational measurement data of an industrially applied RSS line. Furthermore, an optimization model to achieve a minimal cost design was created in MATLAB. The following major conclusions were obtained in the course of the studies on the hybrid storage models:

- The storage capacity of a hybrid storage consisting of an RSS vessel surrounded by PCM can be higher than an RSS with the same total volume, with the PCM volume smaller than the liquid/steam volume. The low thermal conductivity of the PCM is a disadvantage and limits the power and the storage capacity of the hybrid storage. A low heat transfer coefficient of steam during discharging is also disadvantages for the performance of the hybrid storage. The choice of the PCM has an effect on the storage capacity. For charging and discharging, a lower phase change temperature and a higher phase change temperature, respectively, can improve the performance of the hybrid storage.

- The answer to the question whether an arrangement of several PCMs is preferable to

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an arrangement of only one PCM depends on the properties of the PCMs available in the temperature range. Furthermore, the height of the pressure vessel surface at which the PCMs should be arranged depends on the corresponding PCM properties and the overall heat heat transfer coefficient between each water phase inside the pressure vessel and the corresponding PCM volume. In **Paper 4**, the results for the arrangements of several PCM were compared with a arrangements with only one PCM. Improvement through the arrangement of several PCMs could not be achieved in every investigated example. Moreover, the differences in terms of storage capacity improvement were not as significant, but under different operating conditions, the differences could be more significant.

- The arrangement of additional heat sources (electrical heating elements or heat exchangers) on the inner diameter of the LHTES volume can significantly reduce the pressure increase inside the liquid/steam volume compared with placing an additional heat source in the liquid/steam volume of an RSS with the same pressure vessel volume.

- The results of a rough estimation of the investment costs through non-linear optimization for two different use cases showed that the costs for a hybrid storage system can be lower than those for a conventional RSS system. Furthermore, in one use case, the number of storage vessels could be reduced by one in the hybrid storage system.

In conclusion, the hybrid storage presents advantages over conventional RSS. However, the extent of such advantages is strongly dependent on the requirements imposed on the storage. The knowledge and models gained during this thesis can be useful for further development of the hybrid storage component and its subsequent application in industrial processes. As a next step, different practical investigations should be undertaken to validate the hybrid storage models. A constructive design concept must be created and tested to realize the real-world implementation of the hybrid storage concept and also to implement a more realistic cost function in the design optimization model. Moreover, different heat transfer enhancement methods should be investigated and evaluated with regard to costs and reducing phase change time.

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