

FACULDADE DE ENGENHARIA DA UNIVERSIDADE DO PORTO

Performance characterization of thermal storage tank with phase change material

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Resumo

A eletrificação do setor de aquecimento tem exercido pressão adicional sobre a rede elétrica, gerando desequilíbrios entre oferta e procura de eletricidade em momentos específicos do dia. Estratégias de gestão da procura (DSM, do inglês Demand-Side Management) são essenciais, sendo o armazenamento de energia térmica uma solução comum. O armazenamento de energia térmica latente (LHTES, do inglês Latent Heat Thermal Energy Storage) destaca-se pela elevada densidade energética, maturidade técnica e versatilidade, permitindo a integração com bombas de calor para deslocar a carga térmica através da troca de calor entre o fluido de transferência de calor (HTF) e o material de mudança de fase (PCM).

Este trabalho visa ampliar a investigação nesta área, caracterizando o desempenho de um tanque de LHTES para um edifício residencial estudantil. O PCM escolhido, RT57HC, apresenta um calor latente de fusão de 246,4 kJ/kg e uma temperatura de mudança de fase de 57°C, otimizando a compatibilidade com o sistema de aquecimento.

Um banco de ensaios foi projetado e construído para realizar testes experimentais que determinem indicadores de desempenho como tempos de carga/descarga, caudais volumicos ótimos, temperaturas de entrada e saída, eficiência térmica, perdas e capacidade térmica total e parcial. Os processos de carga e descarga foram testados com diferentes caudais e gamas de temperatura, revelando variações no desempenho. A otimização destes parâmetros pretende facilitar a adoção da tecnologia, com condições de ensaio baseadas em estudos anteriores e na literatura sobre aquecimento através de radiadores na região de Estocolmo, Suécia.

A eficiência de carga aumentou com caudais mais elevados, registando um incremento de 8% na capacidade total de 62°C a 65°C e de 4% de 65°C a 70°C, reduzindo até 37% os tempos para atingir 50%, 75% e 90% da capacidade nas taxas ótimas. A densidade energética cresceu com temperaturas de entrada mais altas, embora a redução do calor latente tenha evidenciado a necessidade de otimização da mudança de fase (50-75% do estado de carga). Nos ensaios de descarga, caudais mais baixos prolongaram a entrega estável de potência, com tempos até 33% maiores a 0,15 m³/h versus 0,5 m³/h, e valores de UA aumentando 193% e 195% em diferentes gamas. A eficiência térmica atingiu 88% e 86%, mas perdas de 5 kWh em 24 horas (13% da melhor descarga) sublinham a importância de melhor isolamento.

O sistema oferece densidade energética, flexibilidade e estabilidade superiores ao armazenamento tradicional de água.

Este estudo estabelece uma base sólida para aplicações futuras de LHTES em aquecimento de água doméstica e residencial, fornecendo dados abrangentes para configurações avançadas e modelos de machine learning que otimizem a sinergia entre tanque e bomba de calor.

Palavras-chave: gestão local da demanda, deslocamento da carga térmica, aquecimento residencial, material de mudança de fase, armazenamento de energia térmica, estudo experimental.

Abstract

The electrification of the heating sector has placed additional strain on the electricity grid by creating an imbalance between electricity supply and demand at specific times of the day. Demand-side management (DSM) strategies are required and thermal energy storage is a common solution to it. Latent heat thermal energy storage (LHTES) has been found to be a promising technology due to its high energy density, technical readiness and versatility. LHTES tanks can be coupled with heat pump systems to realize heat load shifting on demand side, by exchanging heat between the pumped heat transfer fluid (HTF) and the phase change material (PCM).

The present work aims to scale up the research in this topic, by characterizing the performance of a LHTES tank configured for a student residential building. The chosen PCM was RT57HC, with a latent heat of fusion of 246.4kJ/kg and a theoretical phase change temperature of 57°C, for a better integration with the future heating system.

A test bench was designed and built to execute a series of controlled experimental tests to determine key performance indicators (KPIs) such as charging/discharging times, optimal mass flow rates, inlet and outlet temperatures, thermal efficiency and losses, total and partial heat capacity. By refining these parameters, the study aims to facilitate the adoption of LHTES technology. The tank's charge and discharge were tested with varying flow rates and different temperature ranges, revealing significant performance variations. The testing conditions were chosen based on previous experiments performed on the same project and literature on heating temperatures in the Stockholm region.

Charging efficiency improved with higher flow rates, achieving an 8% increase in total capacity from 62°C to 65°C and a 4% from 65°C to 70°C, with the time to reach 50%, 75%, and 90% capacity reduced by up to 37% at optimal rates. Energy density rose incrementally, peaking with higher inlet temperatures, though diminishing latent heat contributions highlighted the need for focused phase change optimization, between 50 and 75% SoC (State of Charge). Discharge tests showed lower flow rates extended stable power delivery, with SoD profiles indicating up to 33% longer discharge times at 0.15 m³/h compared to 0.5 m³/h, while UA values surged by 193% and 195% across temperature ranges. Thermal efficiency reached 88% and 86% for the respective groups, but storage losses of 5 kWh over 24 hours (13% of the best performing discharge delivery) highlights the importance of enhanced insulation.

The LHTES system demonstrates superior energy density, flexibility, and stability compared to traditional water storage.

This study lays a robust foundation for future LHTES applications in domestic hot water and residential space heating, providing a comprehensive dataset for advanced configurations and machine learning models to optimize tank-heat pump synergy.

Keywords: demand side management, heat load shifting, space heating, phase change material, thermal energy storage, experimental setup.

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I dedicate this work to my grandfather David.

Tomás Vieira Rocha

“Panta rhei”

- Heraclitus

Contents

1	Introduction	1
1.1	Institutional Context	1
1.2	Thematic Context	1
1.3	Objectives and Research Questions	5
1.4	Research Methodology and Thesis Outline	6
2	State of Art	7
2.1	Thermal Energy Storage Technologies	7
2.1.1	Sensible Heat Storage	9
2.1.2	Latent Heat Storage	11
2.1.3	Thermochemical Heat Storage	13
2.1.4	Comparison of TES types and KPIs	13
2.1.5	LHTES Systems Coupled with Heat Pumps	16
2.2	Phase Change Materials	19
2.2.1	Types of PCMs and Main Criteria for their Selection	20
2.2.2	Performance of TES Components with PCMs	24
3	Theoretical Analysis	27
3.1	Theoretical Framework	27
3.2	Theoretical Values	33
3.2.1	Total Heat Stored and Released Capacity	33
3.2.2	Heat Losses	33
4	Methodology	35
4.1	Description of the Future Application	36
4.2	Description of the Bench Test Facility	38
4.3	Test Matrix and Testing Procedure	46
4.4	Uncertainty Analysis	48
4.5	Charging and Discharging Strategies	49
5	Results	52
5.1	HTF Charging Behavior	52
5.2	PCM Charging Behavior	54
5.2.1	UA values during melting	56
5.3	Analysis of SoC at 50%, 75%, and 90% and Total Capacity	57
5.4	HTF Discharging Behavior	59
5.5	PCM Discharging Behavior	61
5.5.1	UA values during solidification	62

5.6	Analysis of SoD at 50%, 75%, and 90% and Total Power Delivery	63
5.7	Storage Losses	66
6	Discussion	68
6.1	Interpretation of Results	68
6.1.1	Charge mode	68
6.1.2	Discharge mode	72
6.2	Challenges and Limitations	74
7	Conclusions and Recommendations	77
7.1	Conclusions	77
7.2	Recommendations for future work	78
	References	80
A	Python Scripts used for Theoretical Values Assessment	89
A.1	theoretical_capacity.py	89
A.2	theoretical_discharge_power.py	91

List of Figures

1.1	An example of PLS of a space heating load from on-peak (15:30–19:00) to off-peak hours (19:30 – 07:00) in residential applications. [11]	3
1.2	Example of price fluctuations of electricity during the day in SE3 ¹ (1 Öre equals to 0.01 SEK) [12].	3
2.1	Ragone plot including TES. [25]	8
2.2	Types of TES. [26]	8
2.3	Main differences between real and ideal PCMs. [71]	19
2.4	(a) Simplified schematics of a heat-pump base heating system; (b) profiles of heat curve-controlled radiator forward/return temperatures and bin hours in Stockholm [14].	22
2.5	Sketch of the evolution of temperature during charging (melting) and discharging (solidification) processes of an ideal PCM with subcooling [71].	23
2.6	Macroencapsulation geometries tested in previous experimental work - (a) slab; (b) ellipsoidal; (c) cylindrical - [15, 16, 18].	26
4.1	Simplified schematic of the current AHU system.	37
4.2	Live-in lab current heating set-up.	37
4.3	Slab macroencapsulation by <i>Rubitherm</i> [89].	39
4.4	Schematic of the TES tank RTDs mounting positions.	40
4.5	PCM capsules RTDs mounting positions.	40
4.6	Distribution Plate.	41
4.7	Mesh Plate.	41
4.8	Water Tank' heating and cooling loops.	42
4.9	Test bench configuration.	43
4.10	Overview of components from the test bench setup.	44
4.11	Heat demand along the day compared to electricity price - Constant heat demand scenario [12].	50
4.12	Heat demand along the day compared to electricity price - Single peak scenario [12].	51
4.13	Heat demand along the day compared to electricity price - Two peaks scenario [12].	51
5.1	HTF Charging Behavior.	53
5.2	PCM Charging Behavior with $T_{f,in}$ at 62°C.	55
5.3	PCM Charging Behavior with $T_{f,in}$ at 65°C.	55
5.4	PCM Charging Behavior with $T_{f,in}$ at 70°C.	56
5.5	UA values for different BCs at charge tests.	56
5.6	SoC for different flow rates with $T_{f,in}$ at 62°C.	57
5.7	SoC for different flow rates with $T_{f,in}$ at 65°C.	58
5.8	SoC for different flow rates with $T_{f,in}$ at 70°C.	58

5.9	HTF Discharging Behavior.	60
5.10	PCM discharging behavior for the 25 to 30°C $T_{f,in}$	61
5.11	PCM discharging behavior for the 30 to 35°C $T_{f,in}$	62
5.12	UA values for different BCs at discharge tests.	63
5.13	SoD for different flow rates with $T_{f,in}$ at 25 to 30°C.	63
5.14	SoD for different flow rates with $T_{f,in}$ at 30 to 35°C.	64
5.15	State of Discharge profiles for varying HTF flow rates across $T_{f,in}$ within the cooling range of 25 to 30°C.	65
5.16	State of Discharge profiles for varying HTF flow rates across $T_{f,in}$ within the cooling range of 30 to 35°C.	65
5.17	Self discharge along time and average HTF temperature inside the tank.	67
6.1	Comparison of ED between the current study, a cylindrical encapsulation with C48 and C58 [14] and other SAT-based LHTES prototypes found in literature: SUNAMP [91]; Pure SAT in an immersed HX and SAT mixture in a macro-encapsulated component [92].	71
6.2	Example of leakage on a rubber sealant applied on a connection of the water tank.	75
6.3	Initial Capsule RTD Insulation.	76
6.4	Final Capsule RTD Insulation.	76

List of Tables

2.1	Key Parameters of Energy Storage Systems [20]	9
2.2	Comparison of different thermal energy storage systems, Adapted from [26] . . .	14
2.3	Literature-based KPIs [52]	15
2.4	Summary of other design and LHTES integration effects of heat pump-based heating systems, Adapted from [14]	18
2.5	Comparison between different types of PCMs: advantages and disadvantages [26, 40, 73, 74]	21
3.1	Total TES Heat Stored Capacity for Different Temperature Increases	33
3.2	Total TES Heat Released Capacity for Different Discharge Scenarios	33
4.1	Thermo-physical properties of RT57HC.	38
4.2	Design Parameters of Encapsulated PCM Tank	39
4.3	Labels and functions of test bench components.	45
4.4	Charging Tests Matrix	47
4.5	Discharging Tests Matrix	48
5.1	Performance Metrics for Charging Tests	59
5.2	Performance Metrics for Discharge Tests	66

List of Acronyms

AHU	Air Handling Unit
ASHP	Air Source Heat Pump
CFD	Computational Fluid Dynamics
COP	Coefficient of Performance
CP	Circulation Pump
CSP	Concentrated Solar Power
DCN	District Cooling Network
DSM	Demand-side Management
ED	Energy Density
EP	Electricity Price
EUR/€	Euro
FEUP	<i>Faculdade de Engenharia da Universidade do Porto</i>
FM	Flow Meter
GHG	Green House Gas
HD	Heat Demand
HLS	Heat Load Shifting
HP	Heat Pump
HTF	Heat Transfer Fluid
HX	Heat Exchanger
ID	Identification
KPIs	Key Performance Indicators
KTH	<i>Kungliga Tekniska Högskolan</i>
LHTES	Latent Heat Thermal Energy Storage
MPC	Model Predictive Control
N/A	Non Available
NTU	Number of Transfer Units
OPEX	Operational Expenditure
PCM	Phase Change Material
PLS	Peak Load Shifting
RTD	Resistance Temperature Detector
SCHE	Submerged Coil Heat Exchanger
SEK	Swedish Krona
SHTES	Sensible Heat Thermal Energy Storage
SoC	State of Charge
SoD	State of Discharge
TCES	Thermochemical Energy Storage
TES	Thermal Energy Storage

List of Units and Symbols

α	Packed bed void fraction [-]
Δh_m	Latent heat of fusion per unit of mass [kJ/kg]
ϵ	Instantaneous effectiveness [-]
η_{tank}	Stored energy efficiency of the tank [-]
μ	PCM's viscosity [mPa · s]
ρ_{HTF}	HTF's density [kg/m ³]
$\rho_{PCM,l}$	PCM's density in liquid phase [kg/m ³]
$\rho_{PCM,s}$	PCM's density in solid phase [kg/m ³]
$A_{int,side}$	Tank's internal side area [m ²]
$A_{int,top}$	Tank's internal top/bottom area [m ²]
A_{side}	Tank's external side area [m ²]
$A_{top/bot}$	Tank's top and bottom external area [m ²]
A_{tot}	Total tank's outside area [m ²]
a_m	Fraction of the PCM's mass that's melted [-]
a_r	Extent of conversion [-]
$c_{p,HTF}$	Specific heat of the HTF [kJ/(kg · K)]
$c_{p,PCM,l}$	Specific heat of the PCM in liquid phase [kJ/(kg · K)]
$c_{p,PCM,s}$	Specific heat of the PCM in solid phase [kJ/(kg · K)]
$c_{p,pp}$	Specific heat of the PCM's capsule [kJ/(kg · K)]
$c_{p,solid}$	Specific heat of the PCM's solid phase [kJ/(kg · K)]
$c_{p,liquid}$	Specific heat of the PCM's liquid phase [kJ/(kg · K)]
ED	Energy density [kWh/m ³]
ED_{PCM}	Energy density of the PCM [kWh/m ³]
ED_{TES}	Energy density of the TES tank [kWh/m ³]
h	Tank's height [m]
h_{air}	Convection coefficient of air [W·m ⁻² ·K ⁻¹]
h_{HTF}	Convection coefficient of the HTF [W·m ⁻² ·K ⁻¹]
h_{in}	Enthalpy of the inlet HTF [kJ/kg]
h_{out}	Enthalpy of the outlet HTF [kJ/kg]
k_{ins}	Thermal conductivity of the insulation's material [W·m ⁻¹ ·K ⁻¹]
$k_{PCM,l}$	Thermal conductivity of the PCM in liquid phase [W·m ⁻¹ ·K ⁻¹]
$k_{PCM,s}$	Thermal conductivity of the PCM in solid phase [W·m ⁻¹ ·K ⁻¹]
k_{tank}	Thermal conductivity of the tank's material [W·m ⁻¹ ·K ⁻¹]
m_{pcm}	Mass of PCM [kg]
$N_{capsules}$	Number of capsules [-]
P	Instantaneous power [W]
P_{net}	Net heat collected at each time step [kJ]
Q_{charge}	Charged capacity [kWh]

$Q_{discharge}$	Capacity available for discharge [kWh]
Q_{latent}	Latent heat stored in the PCM [kWh]
$Q_{liq,sensible}$	Sensible heat stored in the liquid phase of the PCM [kWh]
Q_{net}	Net heat collected [kW]
Q_{PCM}	Heat stored in the PCM [kWh]
$Q_{sol,sensible}$	Sensible heat stored in the solid phase of the PCM [kWh]
$Q_{sto,loss}$	Storage losses [kWh]
Q_{TES}	Accumulated capacity [kWh]
$R_{conv,air,side}$	Air convective heat transfer resistance on the side of the tank [$K \cdot m^2 \cdot W^{-1}$]
$R_{conv,air,top}$	Air convective heat transfer resistance on the top of the tank [$K \cdot m^2 \cdot W^{-1}$]
$R_{conv,HTF,side}$	HTF's convective heat transfer resistance on the side of the tank [$K \cdot m^2 \cdot W^{-1}$]
$R_{conv,HTF,top}$	HTF's convective heat transfer resistance on the top of the tank [$K \cdot m^2 \cdot W^{-1}$]
$R_{ins,side}$	Insulation's heat transfer resistance on the side [$K \cdot m^2 \cdot W^{-1}$]
$R_{ins,side}$	Insulation's heat transfer resistance on the top and bottom [$K \cdot m^2 \cdot W^{-1}$]
$R_{tank,side}$	Heat transfer resistance on the wall of the tank [$K \cdot m^2 \cdot W^{-1}$]
$R_{tank,top}$	Heat transfer resistance on the top and bottom of the tank [$K \cdot m^2 \cdot W^{-1}$]
$R_{total,side}$	Tank's side heat transfer resistance [$K \cdot m^2 \cdot W^{-1}$]
$R_{total,top/bot}$	Tank's top and bottom heat transfer resistance [$K \cdot m^2 \cdot W^{-1}$]
$r_{ext,ins}$	Insulation's external radius [m]
$r_{ext,tank}$	Tank's external radius [m]
$r_{int,ins}$	Insulation's internal radius [m]
$r_{int,tank}$	Tank's internal radius [m]
SoC	State of Charge [-]
SoD	State of Discharge [-]
$T_{c,off}$	Lower temperature on the crystallization range [$^{\circ}C$]
$T_{c,on}$	Higher temperature on the crystallization range [$^{\circ}C$]
T_f	PCM's final temperature [$^{\circ}C$]
$T_{f,in}$	HTF's inlet temperature [$^{\circ}C$]
T_{HTF}	HTF's instantaneous temperature [$^{\circ}C$]
$T_{in,final}$	HTF's final inlet temperature [$^{\circ}C$]
$T_{m,off}$	Higher temperature on the melting range [$^{\circ}C$]
$T_{m,on}$	Lower temperature on the melting range [$^{\circ}C$]
T_{out}	HTF's outlet temperature [$^{\circ}C$]
$T_{out,final}$	HTF's final outlet temperature [$^{\circ}C$]
T_{PCM}	PCM's instantaneous temperature [$^{\circ}C$]
T_s	PCM's solidification temperature [$^{\circ}C$]
$T_{TES,ini}$	Initial TES tank temperature [$^{\circ}C$]
T_i	PCM's initial temperature [$^{\circ}C$]
T_m	PCM's melting temperature [$^{\circ}C$]
t_{tot}	Total phase change time [s]
$U_{Q_{net,i}}$	Total uncertainty on the instantaneous thermal power [kW]
U_{tank}	Global heat transfer coefficient [$W \cdot m^{-2} \cdot K^{-1}$]
$U_{Q_{TES}}$	Total uncertainty on the estimated capacity [kWh]
U_{system}	System internal energy [J]
V_{DR}	Volume of the distribution region [m^3]
V_{PB}	Volume of the packed bed [m^3]
$\dot{V}_f$	HTF's flow rate [m^3/s]

Chapter 1

Introduction

1.1 Institutional Context

The present study was conducted utilizing the research infrastructure of the Division of Applied Thermodynamics and Refrigeration at the Royal Institute of Technology (KTH). It was performed as part of the HYSTORE project, an European initiative that aims to promote the development of hybrid services from advanced thermal energy storage systems. The consortium consists of 18 entities from 8 different EU countries, covering the whole value chain of TES, technologies for its interconnection with the grid, and its optimal operation by exploiting flexibility resources [1].

This dissertation was developed within the framework of the MSc program in Mechanical Engineering at Faculdade de Engenharia da Universidade do Porto (FEUP).

1.2 Thematic Context

From a global perspective, green house gas (GHG) emissions have been the main driver of climate change. This reality is pushing the energy market toward more sustainable operations, emphasizing renewable energy sources and energy efficiency. Governments, industries, and consumers are prioritizing low-carbon energy sources like hydropower, solar, and wind, in an effort to minimize reliance on fossil fuels. Additionally, policies such as carbon pricing, emissions trading systems, and subsidies for green energy are being implemented to accelerate the transition to a sustainable energy future. Innovations in electricity and heat storage, hydrogen energy, and smart grid technologies are also playing a crucial role in ensuring a more resilient and cleaner global energy system.

Although the transition to sustainable energy in the industrial sector involves a relatively small number of facilities, the challenge is significantly greater when addressing the homes and offices of more than 450 million EU residents [2]. All of these must be adapted so EU align with the reducing of 90% of the emissions by 2040 and to be climate-neutral by 2050 [3], as they aim to. Failing to do so would make it unlikely for the EU to reduce emissions in accordance with the 1.5°C target set by the Paris agreement [4].

In Europe, one of the main energy use sectors in need of decarbonization is the buildings sector, which accounted for 34% of energy-related greenhouse gas emissions in 2022, [5]. Within this sector, heating and cooling are major contributors to final energy consumption. In 2022, space heating alone represented 63.5% of final energy consumption in the residential sector, while water heating accounted for 14.9%. Together, these uses made up 78.4% of household energy consumption [6]. A significant portion of this can be attributed to the rise in living standards and the growing demands for occupant comfort. In an unsustainable manner, buildings are becoming more reliant on active systems to maintain indoor thermal comfort. This leads to a rise in energy consumption and a corresponding increase in associated GHG. As a result, there is a noticeable increase in the operational costs of buildings. Reducing energy use for air-conditioning and the improvement of the energy conservation are essential steps to boost the energy efficiency and sustainability of buildings.

The electrification of the heating sector with heat pumps is considered an effective strategy to increase the share of renewable heat production while improving energy efficiency. However, this transition may place additional strain on the electricity grid by creating an imbalance between electricity supply and demand at specific times of the day [7]. The grid may struggle to meet demand because certain energy sources are non-dispatchable, meaning they cannot generate electricity on demand request, as is the case with solar and wind power. One effective approach to addressing these challenges and improving grid flexibility is to introduce demand-side management strategies within the electrified heating sector.

Thermal Energy Storage (TES) is a common solution to achieve DSM in residential buildings, as they can be coupled with heat sources. Latent heat thermal energy storage (LHTES), thermo-chemical energy storage (TCES), and sensible heat thermal energy storage (SHTES) are the three categorized TES technologies. Among these, LHTES has been regarded as a promising TES technique. It offers an effective balance of high energy density and technical readiness, particularly when spatial constraints for installation are a significant issue [8].

TES technologies can be divided into passive and active systems. The passive systems rely on the building thermal mass as the heat storage material and are often used to stabilize temperature fluctuations and improve energy efficiency in buildings. Active systems use fans or pumps for charging and discharging TES devices with flowing HTF, often allowing for more precise control over energy storage and release. Recent studies compare both these technologies and present their main differences that lead to different appliances. Passive systems are being implemented on higher scale projects like buildings and communities. In contrast, active TES systems are commonly implemented in residential settings, primarily in the form of buffer tanks designed to store domestic hot water in solar-assisted heating setups or as integrated components of heat pump configurations [9, 10].

Heat Load Shifting (HLS) refers to the practice of shifting heating demand to periods when the electricity grid has excess capacity, typically during times of low demand or when renewable energy sources are generating surplus electricity, improving their overall efficiency - Figure 1.1. This strategy can be implemented using thermal energy storage technologies, which store heat

for later use, helping to balance supply and demand more effectively. Resorting to HLS, the heat pump can be switched off during peak hours to avoid purchasing expensive electricity and to reduce strain on the grid, or even work at an hour which the outside temperature allow its COP to be higher. This matter can also be seen as a way to reduce GHG emissions because otherwise, thermal power plants, much of them based on fossil fuels, would be used to fulfill the energy demands, or also as a way to reduce the operating costs for heating system owners if the consumer adopts a time-of-use electricity pricing contract. As seen in Figure 1.2, electricity prices fluctuate during the day, so enabling the possibility to produce heat in a time where the prices are lower, may lead to lower energy bills. When regulated with the power requirements, LHTES has proven to be a meaningful technology to decrease GHG emissions, grid overload and also household electricity bills.

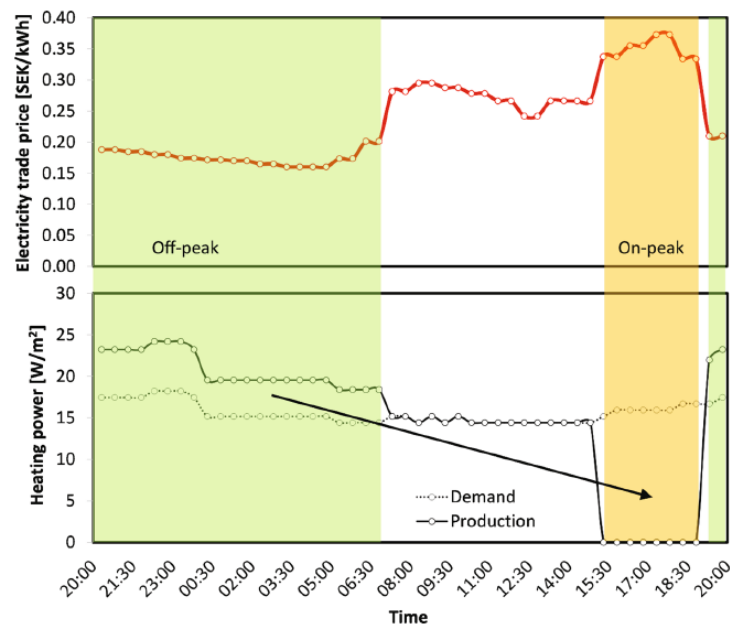


Figure 1.1: An example of PLS of a space heating load from on-peak (15:30–19:00) to off-peak hours (19:30 – 07:00) in residential applications. [11]

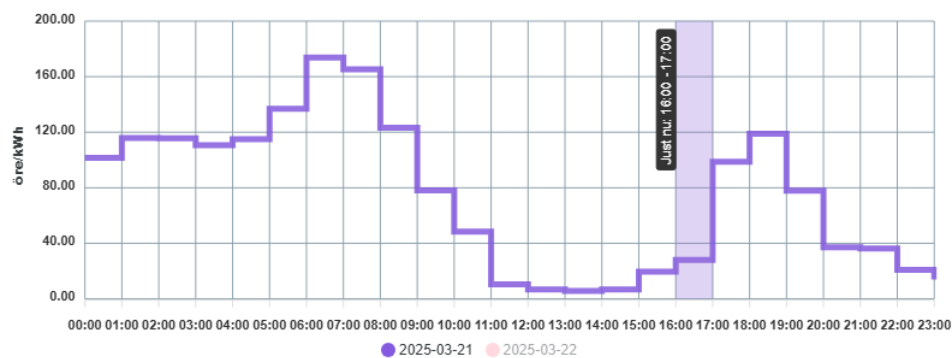


Figure 1.2: Example of price fluctuations of electricity during the day in SE3¹ (1 Öre equals to 0.01 SEK) [12].

The present work aims to build upon the foundational research conducted by Tianhao Xu et al., extending the experiments to the characterization of the storage tank system when coupled with an heat pump. The previous developments on this project led to today's bench test configuration, which includes:

- Material: selection, characterization and optimization of PCMs to maximize the efficiency of the heat transfer and time-efficient phase transition ([13, 14]);
- Components: Design and evaluation of LHTES components under varying conditions to optimize the efficiency of charging and discharging processes for practical applications ([14–16]);
- System: design and optimization of LHTES-integrated energy systems, as well as control strategies, to use storage options to achieve economically and environmentally favorable performance. ([11, 14, 17–19]).

In order to expand green heating options and help achieve the sustainable development objective of providing "affordable and clean energy" to the world's population, this thesis seeks to accelerate the adoption of heating-oriented LHTES technology by presenting a comprehensive analysis of its design, optimization, and integration into existing energy systems.

The bench test rig constructed for this study is a sophisticated setup designed to simulate the thermal energy storage tank coupled with a desuperheater, marking a significant evolution from the previous condenser-based configuration. The mounted test rig will simulate the TES tank connected with a desuperheater, instead of a condenser as it was done previously. This modification aims to take advantage of the higher temperature available in the desuperheater, potentially improving the thermal storage efficiency and enhancing the overall system performance, decreasing the condensation temperature. Additionally, this new configuration may lead to better heat recovery and more effective utilization of the stored thermal energy. Also, as mentioned before, a different PCM will be used - RT57HC. From an post-experiment point of view, the analysis will also differ in a way that partial charge and discharge will be looked at as possible scenarios.

This test rig integrates a precisely controlled environment to replicate operational conditions, including variable inlet HTF temperatures, PCM temperature profiles, and flow dynamics. It features advanced instrumentation, such as temperature sensors and flow meters strategically placed to monitor real-time performance metrics. At the system level, this setup optimizes the project by enabling the evaluation of integrated energy system dynamics, including the interaction between the TES tank and the heat pump. Moreover, the rig facilitates the testing of scalable system designs, providing data to refine control algorithms and assess economic viability under diverse operational scenarios, thus aligning with the project's goal of achieving environmentally and economically favorable performance.

¹The Stockholm metropolitan area and surrounding regions, including Stockholm, Uppsala and Gotland.

1.3 Objectives and Research Questions

LHTES systems have been researched for residential, industrial, and commercial applications, particularly in domestic hot water (DHW) storage, space heating, and waste heat recovery [20]. When integrated with heat pumps, LHTES can enhance overall thermal system efficiency by enabling load shifting, reducing peak energy demand, and optimizing operational strategies [21].

The combination of a heat pump and an LHTES tank allows for effective energy management, leveraging the high coefficient of performance (COP) of heat pumps to charge the storage tank during off-peak hours and efficiently discharge stored thermal energy when demand peaks. However, despite these advantages, the performance characterization of LHTES tanks under dynamic operating conditions remains an essential area of research. Critical factors such as charging and discharging dynamics, thermal stratification, heat exchanger efficiency, and system integration with heat pumps require further experimental and numerical investigation [22].

Considering this research gap, the primary objective of this research is to experimentally characterize the performance of a thermal energy storage tank incorporating PCMs, specifically within the context of a heat pump-based system for space heating. The study aims to analyze and optimize key performance parameters, including:

- Charging and discharging times: Evaluating the rate at which the TES tank absorbs and releases thermal energy to determine system responsiveness and efficiency;
- Optimal mass flow rates: Identifying the most effective heat transfer fluid (HTF) flow rates to ensure efficient heat exchange - UA values - and maintain operational stability;
- Inlet and outlet temperatures: Assessing temperature variations to optimize heat exchange and improve overall system performance;
- Thermal efficiency and losses: Quantifying energy efficiency and identifying potential heat losses in the system to improve thermal storage effectiveness;
- Total heat capacity: Measuring the total amount of energy that the TES tank can store and deliver under specified operating conditions;
- Partial charge and discharge: Evaluating the possibility to charge and discharge partially to work simultaneously with the heat pump or perform under more than one peak.

To achieve this, a bench test was designed, built, and commissioned to ensure optimal functionality and performance. A series of controlled experimental tests were performed in the bench test setup to measure and analyze the thermal response of the TES tank. The findings from this research will contribute to a deeper understanding of LHTES tank performance characteristics, supporting the development of more efficient and compact thermal storage solutions for residential applications. By refining system parameters and addressing performance limitations, this study aims to facilitate broader adoption of LHTES technology, ultimately contributing to a more sustainable energy future.

1.4 Research Methodology and Thesis Outline

The physical boundaries and interactions of the material elements that "flow" through a typical thermodynamic system, such as matter, energy, work, heat, or entropy, are what define it. The problems examined in this thesis are mainly concerned with assessing heat transfer across the boundaries of the energy systems being studied and with increasing the systems' capacity to store heat by employing the latent heat associated with PCM phase-transition processes.

Based on literature and on previous research work on the same project, test matrices were developed to match real life situations. After, a test-rig was built to simulate the demand and the charging. With precision and numerous simulations, the results were obtained and compared between each other. Also, as there are results from different experiments with similar boundary conditions but with different equipments and components, the main performance indicators were compared.

The data resultant from these experiments will be used for future reference on other experiments for the same project.

To document all the process, this thesis is divided in seven chapters.

Chapter 1 introduces the role of Latent Heat Thermal Energy Storage with PCMs in demand-side management and outlines the research objectives, focusing on the performance characterization of a TES tank integrated with a heat pump.

Chapter 2 offers a comprehensive review of TES technologies, exploring recent advancements in PCM selection, storage designs, and LHTES integration with heat pumps, and identifying gaps in current research that this study addresses.

Chapter 3 describes the equations used to access the results from the data extracted from the experimental work.

Chapter 4, the experimental setup is described including the bench test facility and the test matrix for charging and discharging scenarios. It outlines the measurement techniques, RTD placements, and uncertainty analysis procedures.

Chapter 5 presents experimental findings on temperature distributions, charging/discharging times, thermal efficiency and heat transfer rates. This chapter also analyzes the influence of HTF flow rates and inlet temperatures on charging and discharging rates.

Chapter 6 shows the interpretation from the results, addressing all the challenges. It also proposes design optimizations for enhancing TES-heat pump integration and discusses limitations of the current study.

Chapter 7 summarizes key findings and their implications for residential heating applications. This chapter offers recommendations for future research, including scale-up studies and advanced control strategies.

Chapter 2

State of Art

This chapter provides an overview of TES technologies and their comparison with key performance indicators, with a focus on the recent developments in PCM selection criteria, challenges in TES design, and full-scale LHTES component development. Additionally, the integration of TES with heat pump systems is explored, highlighting the potential for efficiency improvements and energy savings in residential applications.

2.1 Thermal Energy Storage Technologies

The concept of storing energy has existed for centuries and is continuously evolving through advancements in technology to reach its current state, where many storage methods have achieved a high level of maturity. Various energy storage solutions exist, each classified based on different criteria.

One of the most widely used methods for comparing energy storage technologies is the Ragone plot, which visually represents the relationship between energy density and power density. This tool is particularly useful for assessing the suitability of different storage technologies for specific applications, as it highlights their capabilities in terms of energy capacity and discharge performance, [23]. Energy density refers to the amount of energy stored per unit mass or volume, while power density describes the rate at which energy can be delivered per unit mass or volume.

When energy must be retained over extended periods, storage systems with high energy density become essential, as they can hold substantial amounts of energy efficiently. On the other hand, for applications where energy must be supplied rapidly over short intervals, high power density technologies are preferred. Another common classification of energy storage systems is based on the duration for which energy is stored. Short-term storage typically spans from a few hours to several days, whereas long-term storage extends over months or even seasonal cycles. Although the Ragone plot is traditionally used to analyze electrochemical storage solutions such as batteries, supercapacitors and fuel cells, recent studies ([24, 25]) have expanded its application to other storage methods, including thermal energy storage, to facilitate broader comparisons, as seen in Figure 2.1.

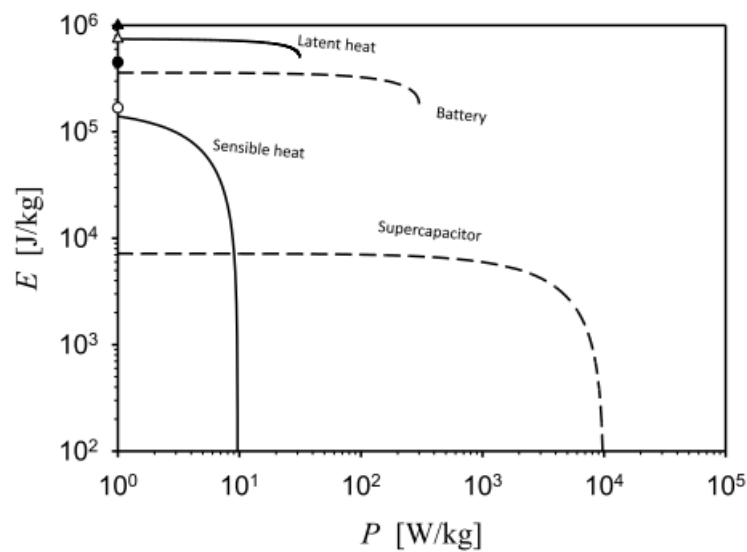


Figure 2.1: Ragone plot including TES. [25]

Energy storage plays a crucial role in balancing supply and demand, improving energy efficiency, and enabling the large-scale integration of renewable sources. Over time, various storage technologies have been developed, each suited to different applications depending on factors such as energy density, power capacity, and discharge duration. While electrochemical and mechanical storage systems have garnered significant attention, thermal energy storage stands out as a highly efficient solution, particularly in heating and cooling applications.

TES systems store energy in the form of heat, either as sensible heat, latent heat, or through thermochemical reactions - Figure 2.2. Among these, LHTES has gained interest due to its ability to achieve high energy densities while maintaining compact system designs [22].

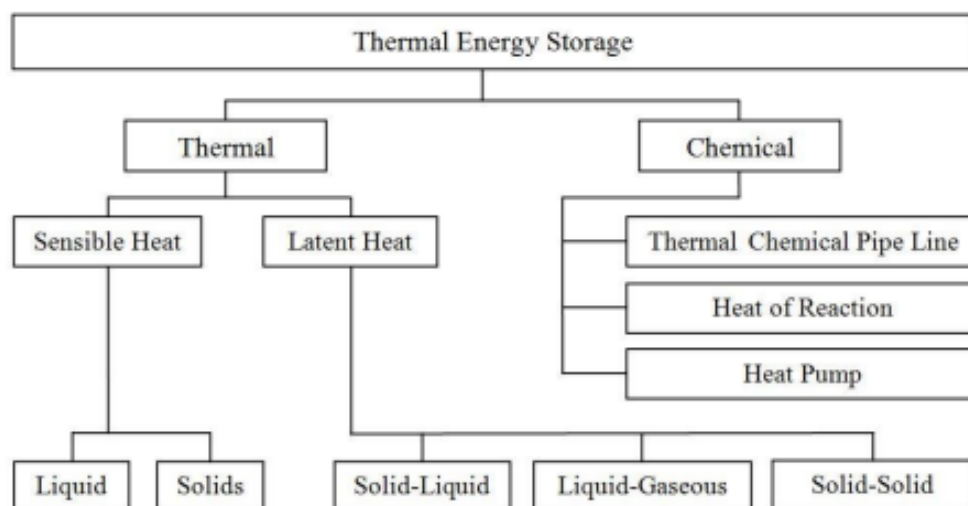


Figure 2.2: Types of TES. [26]

A detailed examination of different thermal energy storage technologies is provided, with a particular emphasis on active latent heat storage (LHS) technologies. Additionally, key performance indicators such as energy density, power output, and other technical parameters are analyzed to highlight the advantages and limitations of various storage solutions.

An energy storage system can be defined based on the following attributes.

Table 2.1: Key Parameters of Energy Storage Systems [20]

Parameter	Description
Capacity	The total amount of energy that can be stored in the system, depending on the storage method, material used, and system dimensions.
Power	The rate at which stored energy can be released (or charged) within the system.
Efficiency	The proportion of energy successfully provided to the user compared to the energy needed to charge the storage system, taking into account losses incurred during charging, discharging, and storage.
Storage Duration	The length of time energy remains stored, ranging from hours to several months (e.g., short-term storage for hours or days and seasonal storage for weeks or months).
Cost	Expressed in terms of capacity (€/kWh) or power (€/kW), influenced by various economic factors such as material costs and system efficiency.
Overall System Cost	Determined by capital expenditures, operational expenses, and the system's lifespan, including the number of charge-discharge cycles it can sustain.

2.1.1 Sensible Heat Storage

SHTES materials store thermal energy within their specific heat capacity, denoted as C_p . The thermal energy retained through sensible heat can be represented by the following equation:

$$Q = \int_{T_i}^{T_f} mC_p dT = m\overline{C_p}(T_f - T_i) \quad (2.1)$$

Where m represents the mass (kg), $\overline{C_p}$ is the average specific heat capacity of the material assessed within the operational temperature range ($\text{kJ.kg}^{-1}.\text{K}^{-1}$) and T_i and T_f are the initial and final temperature of the process respectively. In this type of TES, there is no phase change so the material experiences only a temperature increase within the range required for the storage application [27].

A wide range of materials are used for sensible heat storage, with water being one of the most resorted to. Because water circulates freely, it can be used in active systems as a medium for thermal energy storage as well as a HTF. Its benefits include non-toxicity, availability, affordability, and a high specific heat capacity ($4.184 \text{ kJ.kg}^{-1}.\text{K}^{-1}$). There are three different states of water:

ice, liquid, and steam. While the liquid phase works well for low-temperature thermal energy storage below 100°C, ice is frequently employed for cold storage.

Water is safe for human consumption since it is plentiful, non-toxic, and non-flammable. For applications including hot water supply, cold storage food preservation, and home heating, this makes it the perfect thermal energy storage medium. In contrast, the steam phase is used to store heat energy at high temperatures.

Other materials are used such as thermal oils, molten salts, liquid metals and earth materials. Compared to water, thermal oils maintain a liquid state at elevated temperatures, reaching up to 250°C at atmospheric pressure. These oils operate within a temperature range of 12°C to 400°C, allowing for a greater ΔT in equation 2.1. Also, they have lower vapor pressure, for example DOWTHERM thermal oil, when at 374 °C, has a vapor pressure of 7.6 bar, where as water has a vapor pressure of 221 bar. This reduced pressure minimizes stress on container and pipe walls, resulting in thinner designs and reduced costs for containers and piping [9]. Similarly to water, they can be used as HFT or TES material due to their low viscosity which leads to lower pumping power. However, thermal oils have mediocre heat transfer characteristics, like lower specific heat ($\approx 2 \text{ kJ.kg}^{-1}.\text{K}^{-1}$), and are also more expensive [28].

When the temperature of the system is above the previously mentioned 400°C, molten salts are the chosen heat transfer fluid and heat storage medium. For molten salts, the maximum operating temperature is around 565°C [29], and they're now, the most popular materials for TES in CSP plants. These materials have high boiling temperatures, remarkable thermal stability, and a large volumetric heat capacity. Their vapor pressure is near zero and negligible.

Molten salts do present certain difficulties, however. When a heat source, like solar radiation, is not available at night, their high melting points—which frequently surpass 200°C—can cause solidification in pipes. To enable the HTF to operate across a wide temperature range, the melting point should ideally be close to room temperature and the boiling point should be considerably higher. Commonly, for sensible heat storage in molten salt form, pure salts or eutectic salt combinations with melting temperatures below 250°C are recommended. Additionally, poor heat transfer and flow properties of molten salts raise pumping costs [9].

Liquid metals have a high boiling point and low melting points, frequently at or below room temperature, in contrast to molten salts. Since these metals do not freeze and maintain almost zero vapor pressure even at high temperatures, they are appropriate to use as HTF in high-temperature TES systems. These materials are a popular choice for users because of their large sensible heat storage capacity and wide operating temperature range, which are provided by the significant amplitude between their melting and boiling points.

Because earth materials are readily accessible, affordable, non-flammable, and non-toxic, they are well-suited for use as fillers in single tank thermocline thermal energy storage systems. They can be used as a storage medium in addition to a heat transmission surface. Through direct contact, the HTF exchanges heat as it moves through the packed bed. In addition to reducing the need for costly heat exchangers, direct contact increases the fluid-thermal storage medium surface area of interaction. Selecting sand or natural rocks as filler material can decrease the amount of HTF

needed for charging and discharging thermal energy by as much as 80% in the tank which leads to lower pumping consumptions and faster charging/discharging [30].

In conclusion, SHTES, demonstrates excellent thermal stability at elevated temperatures, making it the most widely utilized TES material for high-temperature applications. Sensible heat storage materials' primary drawback is their inability to maintain a constant temperature during the discharge process. As heat is released, the exit temperature of the fluid steadily lowers over time. The specific heat capacity of these materials is 50–100 times lower than that of latent heat storage, resulting in a lower density of thermal energy storage. However, sensible heat storage materials can still attain a considerable energy storage capacity due to their wide operational temperature range and high material density.

2.1.2 Latent Heat Storage

This section mainly focuses on latent heat storage, which, along with sensible heat storage are more widely used in the residential heating sector [31].

In the context of climate change and the increasing demand for sustainable energy solutions, LHTES technologies are gaining traction as viable options for thermal energy management. Their ability to provide stable thermal energy storage with minimal temperature variation makes them suitable for applications ranging from industrial processes to residential heating and cooling systems [32].

Furthermore, ongoing research continues to explore the potential of PCMs in low-temperature applications, such as refrigeration and air conditioning, where they can significantly improve energy efficiency and reduce reliance on traditional energy sources [33].

Latent heat thermal energy storage has emerged as a critical technology for increasing energy efficiency and integrating renewable energy sources. Phase transitions can be solid-solid, solid-liquid, solid-gas, or liquid-gas. This technique uses phase change materials to store and release thermal energy during these transitions. Transitions from solid to liquid are a financially appealing choice for thermal energy storage systems. In actuality, these changes only include a little volume change (of roughly 10% or less) and have a somewhat lower latent heat than liquid-gas conversions [26]. With a solid-liquid process, the LHTES system's storage capacity is determined by:

$$Q = m_{PCM} \left[\int_{T_i}^{T_m} c_{p,solid} dT + \Delta h_m + \int_{T_m}^{T_f} c_{p,liquid} dT \right] \quad (2.2)$$

The sensible heat of the solid phase is the second member's first term, followed by the sensible heat of the liquid phase in the third term, and the latent heat of fusion in kJ/kg, denoted by Δh_m . In the equation 3.4, T_m represents the material's melting temperature, whereas $c_{p,solid}$ and $c_{p,liquid}$ represent the specific heat of the solid and liquid phases, respectively.

By utilizing phase change materials, LHTES systems can effectively store and release thermal energy during phase transitions, making them well-suited for integration with heat pumps and other thermal management applications. High latent heat, high specific heat, high thermal conductivity, and a desirable melting point for the specified thermal application are all necessary

for LHTES materials to have the right thermophysical characteristics. Low supercooling, low cost, easy availability, low volume change, nonetoxic, low vapor pressure, congruent melting, low flammability, and thermal and chemical stability are further desired characteristics of thermal energy storage materials [34]. Therefore, recent developments in this area have concentrated on improving PCMs' thermal performance.

In a study of a PCM based portable box for cold chain applications, 1 wt.% of graphene was added leading to the thermal conductivity of composite increase by 55.4% [35]. Other recent studies have also highlighted the integration of advanced materials and nanotechnology to enhance the thermal conductivity of PCMs. For example, the incorporation of MXene into organic phase change materials has shown promising results in improving thermal conductivity while maintaining latent heat storage capabilities [36]. Additionally, the use of metal additive manufacturing techniques has been explored to create structures that optimize heat conduction within PCMs, thereby addressing the limitations associated with low thermal conductivity [37]. These innovations are essential to improve the charging and discharging rates of LHTES systems, making them more efficient for rapid energy storage and retrieval.

The high energy storage density of LHTES devices, which enables the storage of substantial amounts of thermal energy in a very compact container, is one of their main advantages. For example, under the same volume conditions, molten salts that undergo solid-liquid phase transitions can store 5–14 times as much energy as sensible heat storage devices [38].

An advantage over sensible heat storage materials is that the HTF temperature stays constant over time due to the storage medium's constant temperature during the discharging process. This makes LHTES attractive for use in solar thermal energy storage, where it is essential to maintain a nearly constant temperature during energy release [39]. The temperature difference between storing and releasing heat is less for latent heat storage materials [40]. In comparison to specific heat, the material's latent heat of fusion will be extremely high. For instance, the specific heat of sodium nitrate salt is $1.1 \text{ kJ kg}^{-1} \text{ K}^{-1}$, but its "latent heat of fusion" is approximately 172 kJ kg^{-1} . The latent heat storage materials benefit from a high energy storage density because of this kind of significant variation. This lowers the TES vessels' volume, which minimizes heat loss and decreases the surface area of their outside walls [34]. These systems can efficiently balance energy supply and demand, particularly in renewable energy contexts, thanks to their capacity to absorb and release significant amounts of latent heat [41].

The applications of LHTES can be broadly categorized based on their operational temperature ranges. PCMs can be classified into four main temperature ranges: low temperature (-20°C to $+5^\circ\text{C}$), medium temperature (5°C to 100°C), high temperature (100°C to 200°C), and very high temperature (above 200°C) [42]. Each of these ranges presents unique challenges and opportunities for thermal energy storage. For instance, low-temperature applications often focus on refrigeration and air conditioning, where PCMs such as paraffin wax or salt hydrates are commonly employed due to their favorable melting points and thermal properties. In contrast, high-temperature applications, such as concentrated solar power systems, typically utilize molten salts or metal alloys that can withstand higher thermal loads and provide efficient energy storage [42].

The power output of LHTES systems is another critical factor that varies between applications. For example, in domestic hot water systems, the thermal power can range from a few kilowatts to several hundred kilowatts, depending on the size of the system and the specific PCM used [43]. In industrial applications, such as those involving concentrated solar power, latent heat storage systems can achieve power outputs in the megawatt range, facilitating large-scale energy generation and storage [44]. This capability is essential for balancing the intermittent nature of renewable energy sources, allowing energy to be stored during peak production periods and released during high demand.

2.1.3 Thermochemical Heat Storage

In thermochemical heat storage, heat is stored during the endothermic phase of a reversible reaction and released during the exothermic phase [45]. An endothermic reaction is used to convert a chemical reactant into its products during the charging phase by absorbing heat energy. After then, these products are kept apart until the energy is needed. The stored products are recombined and go through an exothermic reaction that regenerates the initial reactant during the discharging phase. An energy source is created from the heat that is released during this procedure [46].

The heat stored in thermochemical heat storage is determined by:

$$Q = a_r \cdot m \cdot \Delta h_r \quad (2.3)$$

where a_r is the extent of conversion, m is the amount of storage material, in kg, and Δh_r is the endothermic heat of reaction, in kJ/kg.

The energy density of this kind of TES is five to ten times higher than that of sensible and latent heat storage systems, respectively, despite the fact that research and development is still ongoing. Compared to latent and sensible TES, thermochemical TES devices can offer more compact energy storage due to their higher energy density. This feature is especially helpful in situations where the TES's space is precious or restricted [47]. Additionally, since the reaction products can be stored at room temperature, there is no thermal loss during storage, therefore theoretically, their storage period and transportation are approximately limitless [48–50].

The carbonation process, ammonia breakdown, metal oxidation reactions, and sulfur cycles are the primary reactions utilized in these systems, which are used for storage at temperatures over 400°C [51].

2.1.4 Comparison of TES types and KPIs

Table 2.2 summarizes the main differences between TES types considering also different materials and applications, and a range of relevant performance parameters and factors. The range of overall TES technology is large enough to adapt to almost any situation.

Table 2.2: Comparison of different thermal energy storage systems, Adapted from [26]

	Sensible TES	Latent TES	Thermochemical TES
Temperature range	50 to 1200°C	-100 to 900°C	180 to 2500°C
Energy density	Volumetric: Small (~50 kWh/m ³) Gravimetric: Small (~0.02-0.03 kWh/kg)	Volumetric: Medium (~100 kWh/m ³) Gravimetric: Medium (~0.05-1 kWh/kg)	Volumetric: High (~500 kWh/m ³) Gravimetric: High (~0.5-1 kWh/kg)
Storage period	Limited (thermal losses)	Limited (thermal losses)	Theoretically unlimited
Lifetime	Long	Often limited due to storage material cycling	Depends on reactant degradation
Technology status	Industrial scale	Pilot scale	Laboratory scale
Advantages	Low cost Reliable Simple application with available materials	Medium storage density Small volumes Short distance transport possibility	High storage density Low heat losses Long distance transport possibility
Disadvantages	Significant heat loss over time (depending on level of insulation) Large volume needed	Low heat conductivity Corrosivity of some materials Significant heat loss (depending on insulation)	High capital costs Technically complex

Sensible heat storage, typically operates within a range of 20°C to 400°C, with water-based systems effective up to 100°C and molten salts extending to higher temperatures for concentrated solar power (CSP) applications. Latent heat storage, covers a broader spectrum depending on the material, generally ranging from -20°C to 120°C for organic PCMs (e.g., paraffin) and up to 150°C to 900°C for inorganic PCMs (e.g., salts or metals), enabling applications from building heating to industrial processes. Thermochemical storage, can achieve higher temperature ranges, typically from 100°C to 1000°C or more, depending on the reaction pair (e.g., metal hydrides or carbonate looping), making it suitable for long-term storage and high-temperature industrial use.

Table 2.2 presents Key Performance Indicators (KPIs) to compare different technologies to store thermal energy, such as energy density, storage period and operational lifetime.

In addition to the indicators presented, Table 2.3 compiles further KPIs reported in the literature, offering a broader view of the parameters commonly evaluated in TES studies.

Among all the parameters presented, the approach of this research was based only on quantitative KPIs, which allowed to obtain a measurable comparison under varying boundary conditions, in a way to optimize the storage and the whole process, while characterizing the TES tank.

An additional KPI not included in Table 2.3 is the comparative factor relative to water-based sensible heat storage. This indicator is calculated as the ratio between the stored energy in the

tested configuration and the theoretical capacity of a water-filled tank of the same volume, serving as a benchmark for performance evaluation.

Table 2.3: Literature-based KPIs [52]

KPI	Description	Source
Storage capacity [Wh]	The amount of stored energy in the system under design conditions	[53]
Recharging energy [Wh]	The energy required to recharge the storage for a specific period	
Max. charge/discharge power [kW]	–	
Depth of Discharge	The percentage of maximum storage capacity that can be discharged to provide usable energy	
Durability	Number of working cycles in which the TES can release 75% of its designed capacity	
Specific cost of storage [€ kWh ⁻¹]	Total storage cost, normalized by the total amount of energy it can deliver	
Self-Discharge rate	The portion of energy dissipated during a certain standby period	
Storage size [m ³] and weight [kg]	–	
Energy storage factor	The fraction of the total energy demand that can be covered by storage	
Generated energy and cost saving	–	
Discharging efficiency	Ratio between the depth of discharge and the maximum storage capacity	
Total charging/discharging efficiency	Ratio between the useful capacity and the recharged energy	
Energy density [kWh kg ⁻¹ , kWh m ⁻³]	–	
Specific cost of stored energy [€ kWh ⁻¹]	The cost per kWh released from storage	[54]
CAPEX [€ kWh ⁻¹]	–	
Storage efficiency	–	
Maintenance cost [€ kWh ⁻¹]	–	
Delivered energy cost [€ kWh ⁻¹]	–	
Heat losses [W]	–	
Lifetime [y]	–	
Thermal efficiency	Energy extracted over energy injected	[55]
Exergy efficiency	Exergy extracted over exergy injected	[56]
Energy charged [kWh]	–	
Average power [kW]	–	
Peak power-Energy ratio [h ⁻¹]	Comparison between the 5-min peak power and the energy size of storage	
Charging/discharging temperatures [°C]	–	[57]
Energy consumption by auxiliary units [kWh]	–	
Thermal efficiency of storage	Ratio between the design power, storage time, and storage capacity	
Specific cost of storage [€ kWh ⁻¹]	Ratio between total cost (materials, installation, etc.) and storage capacity	

2.1.5 LHTES Systems Coupled with Heat Pumps

Literature on LHTES systems for air conditioning and DHW is typically categorized based on the heat source. Most studies focus on solar-assisted systems and heat pumps as heat sources, although applications of LHTES for space heating remain limited.

The majority of research on HP heating systems has concentrated on integrating a water tank to address peak load shifting. The great sensitivity of LHTES units' charging and discharging performance to PCM design factors was the primary focus of some earlier research on the LHTES technology. Finding PCM design parameters for heating systems with load shifting constraints, on the other hand, has not been the focus of many studies. Finding the right PCM that fits the heating device well has only been somewhat studied up to this point. Additionally, there has been little research done on how storage architecture affects the combined heating system's total effectiveness.

T. Xu et al. [11] analyzed various configurations of LHTES coupled with heat pumps. The study focused on comparing four different methods for charging the LHTES unit: connecting it to the heat pump's condenser (Case 1), utilizing the de-superheater of the main heat pump (Case 2), integrating the condenser of a cascaded booster heat pump cycle (Case 3), and a combined approach using both the de-superheater and the booster cycle (Case 4). A numerical model was created to evaluate performance parameters which were benchmarked against a baseline heating system without storage (Case 0) and also with Case 1, with a context of a multi-family house in Stockholm. During the evaluation period of Week 1, 2019, two load shifting strategies were simulated: 1) charging during off-peak hours (8 pm to 6 am) and 2) charging during daytime hours (10 am to 7 pm). As demonstrated by the simulation results of the off-peak charging method, Cases 2–4 have a heating performance factor that is 22–26 percent greater than Case 1 and an operational expenditure that is 2–5% lower than Case 0. Furthermore, compared to the off-peak charging approach, the main advantage of using the midday charging method is the notable decrease in CO₂ emissions, which can be as much as 14% lower. According to simulation results, using the desuperheater instead of the condenser to produce hot water at 65 °C can increase the heating coefficient of performance by 38% with the heat pump modelled in this work. However, because of the limited sensible heat capacity of the superheated refrigerant, the desuperheater's available heating capacity is only 24% of the other case. Assigning partial heating capacity to the desuperheater on the heat sink side can also increase the heat transfer area for condensation in the condenser, which may lower the condensation temperature and raise the heating coefficient of performance, per a heat exchanger analysis with a similar desuperheater installed heat pump. Additionally, an investment analysis was conducted, enabling the conclusion that the LHTES's capital expenditure was warranted given the energy savings. This comparison made it possible to comprehend how the coupling location affected the performance metrics, even if it was a numerical analysis.

In a case study conducted in China [58], peak-load shifting was proven using an electric boiler-based heating system integrated with a 2 MWh LHTES system during the course of a continuous

22-day operation. By utilizing the lower power prices during off-peak hours in peak load shifting operation, the unique system managed to achieve a 45% reduction in operating costs by heating, despite some losses in effective storage capacity due to inadequate charging. The LHTES system in this instance can be repaid in two years.

Wang et al. [59] proposed a CO₂ ASHP heating system integrated with TES and carried out experiments to analyze its performance in cold regions, comparing performance indicators with a base system which was composed only by a CO₂ heat pump. Their findings indicated that the system COP increased by 4%, while throttle losses decreased due to the low return water temperature, the system heating capacity decreased by 21% and the energy consumption decreased by 24% during the entire heating period, which was the month of January.

In order to assess the performance metrics associated with the integration of LHTES technology with space heating, case studies of real-world applications were created. These case studies also included comprehensive assessments on the reasons for malfunctions in these systems. According to a 25-month study conducted in a commercial building, the PCM largely stayed inactive since the phase change temperature was incorrectly chosen for the charging process to be started with supplied ambient air [60]. Thus, during summer operation, only 15% of the design storage capacity was used.

The risk of phase separation in a large-scale commercial salt hydrate-based PCM was discovered by Tan et al. [61]. To mitigate this issue, they modified the PCM formulation by incorporating thickening agents. This adaptation was applied to a 275 kWh LHTES component integrated into the air-conditioning system of a commercial building in Sweden. However, because the charging rate was overestimated by 75% during the design phase, only 36% of the theoretical storage capacity was used during the daily charging procedure.

According to previous studies, salt hydrates have an inherent issue with phase separation and supercooling since they melt congruently [62]. Although there are other common problem associated with PCMs used in LHTES for low temperature applications, like low thermal conductivity which can result in inactive core with macro-encapsulated solutions and dead PCM zone with submerged solutions [63]. In Table 2.4, more examples of *in-site* tests are presented.

As observed in the previously mentioned field investigations, LHTES installations have demonstrated both promising outcomes and operational challenges. In successful implementations, a thorough understanding of the thermal behavior of PCMs and the thermal performance of LHTES components was achieved through comprehensive material and component characterization analyses. This allowed for the early identification and mitigation of undesired phenomena, such as phase separation in PCMs or insufficient thermal power from LHTES units, before large-scale deployment.

In this project, particular attention was given to these challenges. To ensure the reliability and efficiency of the LHTES system, experiments were conducted with different PCMs, where their thermal and mechanical properties were carefully evaluated and measured. These tests provided valuable insights into the material performance under various conditions, helping to optimize the system's operation.

By addressing these critical factors, the reliability and efficiency of LHTES technology can be significantly improved, leading to more effective thermal energy storage solutions.

Table 2.4: Summary of other design and LHTES integration effects of heat pump-based heating systems, Adapted from [14]

Heating System Description	HP	PCM	PCM melting point	LHTES Size	Performance Indicators ^{a,b}			Ref.
					P_{el}	$OPEX_{el}$	E_{CO_2}	
Radiator heating system in UK	ASHP	Commercial salt hydrate	48°C	500 L	161%	129%	162%	[64]
Underfloor heating in Scotland	ASHP	$Na_2S_2O_3 \cdot 5H_2O$	48°C	200 L	108%	80%	101%	[65]
Hypothetic residential heating system	GSHP	Heneicosane	40°C	Up to 4 m ³	126%	N/A	N/A	[66]
Small-scale office building in Netherlands	in ASHP	$CaCl_2 \cdot 6H_2O$	29°C	0.5 m ³ ; four units	100% ^d	94%	N/A	[67]
Radiator heating in rural house in Beijing, China	in ASHP	Paraffin wax	47°C	N/A	96%	N/A	N/A	[59]
Residential building air heating in UK	in ASHP	RT58	58°C	1.2 m-long cop-per cylinder filled with 93 kg PCM	N/A	N/A	N/A	[68]
Swimming pool heating system in Hong Kong, China	in ASHP	SAT	58°C	up to 135.8 m ³	27% ^c	19% ^c	N/A	[69]
Swimming pool heating system in Hong Kong, China	in ASHP	mixture of 38% urea and 62% acetamide	53°C	65.2 m ³	N/A	69%	N/A	[70]
		SAT	58°C	54.2 m ³	N/A	N/A	N/A	

^aPercentages indicate ratios on the three performance indicators between load shifted heating system operation and baseline operation.

^bTerms: P_{el} -Electricity input; $OPEX_{el}$ -operational expense for heating from electricity tariffs; E_{CO_2} -equivalent carbon dioxide emissions.

^cIn this reference only, comparison is made against an electric heater-based system (instead of heat pump-based systems as elsewhere).

^dCOP unchanged (COP=4.5) with LHTES integration.

2.2 Phase Change Materials

Phase Change Materials (PCMs) have gained significant traction in recent years as a promising solution for improving energy efficiency and thermal management in various applications, particularly in space heating systems. PCMs are either organic, inorganic or eutectic substances capable of capturing and retaining significant amounts of thermal energy during phase transitions, such as shifting from solid to liquid or vice versa.

A key characteristic of PCMs is their ability to maintain and store thermal energy across multiple phase transition cycles. It is important to note that common PCMs behave somewhat differently than ideal PCMs. Figure 2.3 illustrates how a phase-change melting temperature range takes the place of the ideal melting-peak temperature of the isothermal phase-change process of pure PCMs. Also, the temperature doesn't remain constant during the melting process.

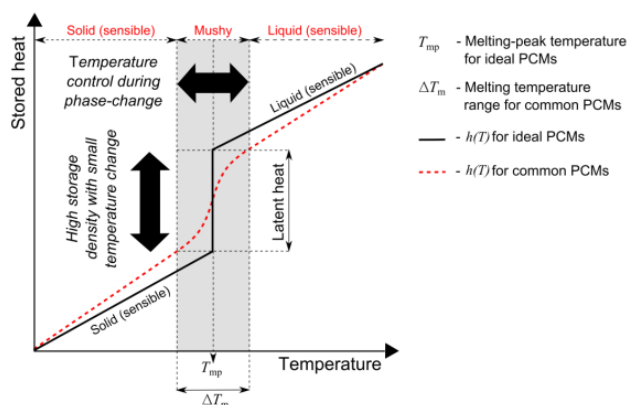


Figure 2.3: Main differences between real and ideal PCMs. [71]

In nature, matter typically exists in one of three states: solid, liquid, or gas. When transitioning between these states, it either absorbs or emits heat, known as "latent heat". Heat is retained during the transition from one crystal to another in solid–solid transitions. Compared to solid–liquid transitions, these transitions often have smaller volume changes and latent heat. Although the latent heat of vaporization is larger for solid–gas and liquid–gas transitions, their use in TES is limited by their significant volume change during phase transitions, which are linked to containment issues. Solid–liquid transitions only require a tiny volume change (of order of 10% or less), but they have a relatively lower latent heat than liquid–gas transformations. For this reason, solid–liquid transitions are considered the most economically attractive option to be applied in TES systems [26].

PCMs have the ability to change their state at a specific temperature range which is what makes them suitable for different applications [72]. The PCM phase change temperature should be proportional to the system's operating temperature for heating or cooling [73]. These unique properties allows PCMs to effectively store thermal energy, making them integral components in thermal energy storage systems, especially when integrated with renewable energy sources such as solar thermal and heat pump technologies [74].

2.2.1 Types of PCMs and Main Criteria for their Selection

PCM materials can be divided in three groups: organic, inorganic and eutetic. Inside the organic type, they can be paraffin compounds or non-paraffin compounds. The inorganic PCMs can be metallics or salt hydrate. The eutetic type PCMs can be a mixture of organic and inorganic PCMs.

Paraffins and non-paraffins are used to further characterize organic PCMs. Congruent melting without phase separation is a property of organic compounds. However, due to their generally low thermal conductivity (0.1-0.35 W/m K), a larger surface area is required to increase the rate of heat transfer [75]. These PCMs haven't been investigated for high temperature activities because of their low melting point.

Paraffin wax is the most widely used organic PCM, composed primarily of a blend of straight-chain n-alkanes $\text{CH}_3\text{-(CH}_2\text{)-CH}_3$. In general, an increase in the average length of the hydrocarbon chain results in a higher latent heat of fusion and an elevated melting temperature [75]. Paraffin based PCMs are known for being safe, cost-effective, non-corrosive, and highly reliable. Additional advantages include their chemical stability, resistance to phase segregation and supercooling, effective nucleation properties, and consistent thermo-physical characteristics. Even after 1000–2000 cycles commercial grade paraffin waxes and other pure paraffin have stable properties and good thermal reliability [34, 75, 76]. However, it has some disadvantages such as: low thermal conductivity, high temperature transition range and they're non compatible with plastic containers. These disadvantages can be contoured by using materials that increase thermal conductivity, promoting homogeneous phase change with macroencapsulation technologies and using metallic tanks, which will be explained in following chapters. In this thesis' experimental work, a commercial paraffin wax will be used.

Fatty acids, esters, alcohols, and glycols are examples of non-paraffin compounds. Due to their flammability, these materials should not be subjected to oxidizing chemicals, flames, or extremely high temperatures. The thermophysical characteristics of fatty acids make them the perfect material for latent heat storage at low temperatures. In contrast to paraffins, fatty acids have relatively high melting and boiling temperatures, and saturated fatty acids show minimal phase transition volume changes. They are more expensive than technical grade paraffins (around 2-2.5 times [26]), mildly corrosive, unstable at high temperature and possess a disagreeable odor [34]. Esters are generally applied to low temperature situations due to low melting point. Glycols have lower heat of fusion when compared with esters but have higher melting temperature making them more suitable for higher temperature applications. Finally, alcohols present the highest melting temperatures and heat of fusion among the organic PCMs, but haven't received much attention by researchers due to problems with subcooling [26, 34].

Salt hydrate and metallic material are the two kinds of inorganic materials utilized as PCM. Metals have melting points that are too high for passive construction applications, and they span a broad temperature range.

When two or more substances, such as organic-organic, organic-inorganic, or inorganic-inorganic

compounds, solidify from the liquid phase at the lowest freezing temperature, referred to as the eutectic point, they form a eutectic mixture. Creating eutectic mixtures of organic and inorganic materials with a high thermal capacity, an ideal melting point, and little to no supercooling effects has long been the subject of research. While organic eutectics are better suited for low-temperature storage, such as building temperature regulation, inorganic eutectic mixes are especially well-suited for high-temperature thermal storage applications, such as concentrated solar power (CSP) systems [77]. Additionally, they are about twice as thermally conductive as paraffins. The primary disadvantage is incongruent melting, which causes irreversible melting and freezing of salt hydrates, resulting in low cycling stability [26].

The following table resumes the advantages and disadvantages of each PCM type:

Table 2.5: Comparison between different types of PCMs: advantages and disadvantages [26, 40, 73, 74]

Classification	Advantages	Disadvantages
Organic: Paraffins and non-paraffins	Availability in a large temperature range; high latent heat of fusion; solidification with little or no subcooling; congruent phase-change; self-nucleation properties; no segregation and good nucleation rate; thermally and chemically stable; low vapour pressure in the melted form; non-reactive and non-corrosive; compatibility with conventional construction materials; recyclable.	Low thermal conductivity; lower volumetric latent heat storage capacity; lower phase-change enthalpy; lower density; flammable; non-compatibility with plastic containers; more expensive; relatively large volume change.
Inorganic: Hydrated salts	Higher volumetric latent heat storage capacity; higher latent heat of fusion; low cost and readily available; sharper phase-change; higher thermal conductivity; non-flammable; lower volume change; compatible with plastics.	Poor nucleating properties and subcooling problems; incongruent melting and dehydration in the process of thermal cycling; phase segregation during transition and thermal stability problems; their application could require the use of some nucleating and thickening agents; decomposition and phase separation; non-compatible with some construction materials; corrosive to most metals and slightly toxic.
Eutectic	Sharp melting temperature; volumetric thermal storage density slightly above organic compounds; no segregation and congruent phase-change-	Limited data are available on their thermophysical properties; some fatty eutectics have quite strong odour.

The following properties should be taken into consideration when choosing PCMs to adapt LHTES: kinetics properties, such as high nucleation rate, minimal or no supercooling of the liquid phase, and high crystallization rate; chemical properties, such as fully reversible melting/solidification cycles, long-term chemical stability, compatibility with system materials (non-corrosive), non-toxic, and non-flammable; physical properties, such as optimal phase balance, high density, minimal volume variation during phase change, low vapor pressure at operating temperature, and renewable phase change capability; thermal properties, which include high latent heat per unit mass, high thermal conductivity, high specific heat, high density, congruent melting, and long-term thermal stability; and kinetics; economical properties such as, availability and cost-effectiveness; and lastly, environmental properties, like high number of cycles, minimal environmental impact, non-polluting, and recyclable [74].

The type of heat transfer medium (air-based or hydronic), the type of heating terminal units (radiators, underfloor heating systems, or fan coil units), and the heat generation components (heat pumps, electric heaters, or solar thermal collectors, among others) all affect the operating temperature of a space heating system. In the context of space heating applications, research indicates that for effective integration with heat pumps, PCMs should ideally have a melting point in the range of 45°C to 60°C. This interval of temperatures is related to experimental tests and numerical analysis performed by Tianhao Xu, [14]. As seen in Figure 2.4, the radiator forward temperatures vary from 55 to 30°C, being the most commonly found along the year, 40 to 25°C. To attend this temperature range, the LHTES system must detain a PCM with a phase change temperature above these values.

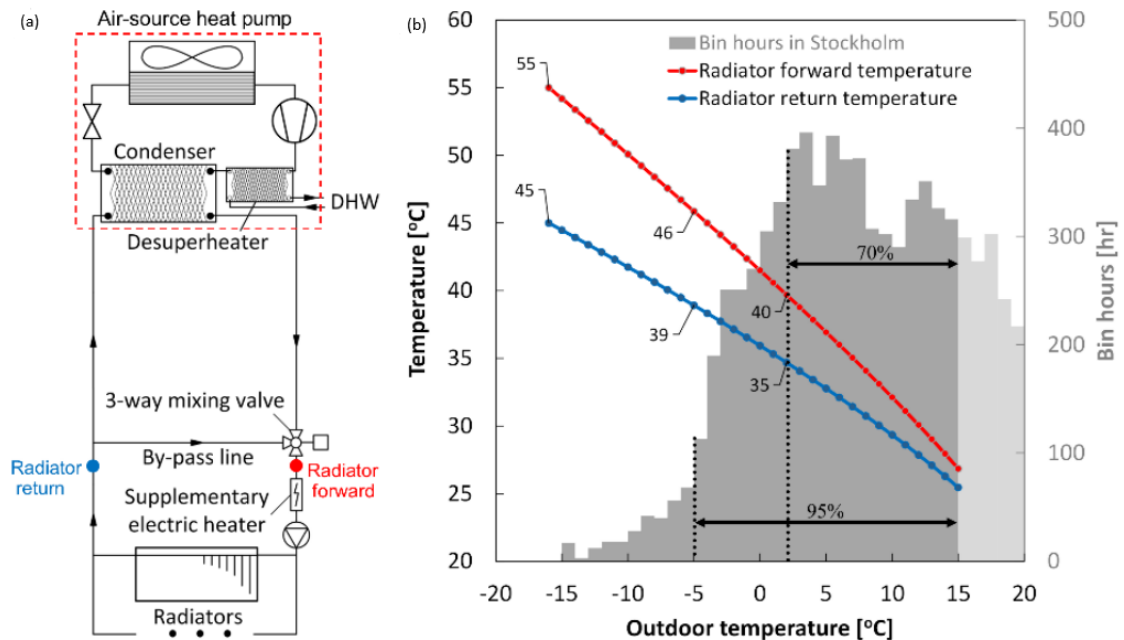


Figure 2.4: (a) Simplified schematics of a heat-pump base heating system; (b) profiles of heat curve-controlled radiator forward/return temperatures and bin hours in Stockholm [14].

The latent heat of fusion per unit volume and thermal conductivity, which enhances the storage and release of latent heat in a given volume of the material in a short amount of time, are thermal properties that need to be precisely measured to be able to create a good testing set and also, numerical support. Therefore, for commercial PCMs with incomplete specifications according to the manufacturers' datasheet, it is imperative to conduct in-house property measurements.

T. Xu previously examined three distinct PCMs—C48, C58, and ATP 60—in relation to this issue [14]. Both C48 and C58, which had similar base materials but differed by roughly 10K in phase-change temperature, and C58 and ATP60, which had similar phase-change temperatures but different material categories and primary thermophysical characteristics, were separated into two groups for the comparison studies. ATP 60 was analyzed using Differential Scanning Calorimetry (DSC), and the phase-change temperature range and enthalpy-temperature profile for C48 and C58 were assessed using the T-history approach. All three PCMs' solid-liquid phase transitions were confirmed by the DSC and T-history experiments. Although it was not mentioned in the manufacturer's specifications, supercooling was found in C48 and C58. The T-history tests revealed supercooling effects in these two SAT-based PCMs, and the DSC results for ATP 60 indicated the occurrence of solid-solid phase transitions. Furthermore, the Transient Plane Source (TPS) method was used to measure the thermal conductivities of ATP 60 [13, 15, 16].

The present work utilizes a different PCM, RT57HC, whose properties were also accessed before the testing was performed.

Accurately measuring these properties is crucial for enhancing the precision of numerical simulations of heat transfer in LHTES components. After determining the thermo-physical properties of the PCMs under analysis, bench tests were conducted under various boundary conditions and with different types of macro-encapsulation. These tests evaluated kinetic properties such as supercooling, crystallization, and nucleation rate, factors that are particularly challenging to replicate in simulations.

The $h(T)$ curve was also determined for the utilized PCMs. During the reversible charging (melting) and discharging (solidification) cycles, the enthalpy-temperature curve $h(T)$ should ideally be constant. However, subcooling, hysteresis, and cycling stability are some of the phenomena that can impact the $h(T)$ curve. The temperature change during the melting and solidification of an ideal PCM displaying subcooling is depicted in Figure 2.5.

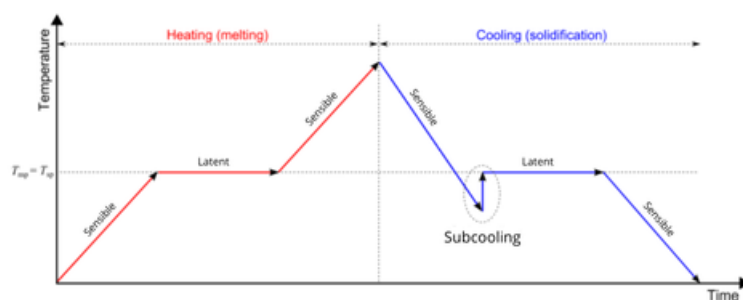


Figure 2.5: Sketch of the evolution of temperature during charging (melting) and discharging (solidification) processes of an ideal PCM with subcooling [71].

In order to start crystallization and release the latent heat that has been stored within, the material must subcool to a temperature that is much lower than its solidification point. The temperature will return to the PCM's solidification temperature, T_s , which should ideally be equal to the melting temperature, T_m , if the heat released during solidification surpasses the sensible heat lost as a result of subcooling. However, if this condition is not met, or if the rate of heat loss to the surroundings surpasses the rate of heat released during crystallization, the temperature will not return to the solidification temperature, resulting in true hysteresis caused by subcooling.

Data from charging and discharging trials usually differ because of hysteresis. As a result, both heating and cooling experiment findings must be given. The impact of natural convection in the molten PCM is another crucial component to take into account when assessing heat transfer during the solid-liquid phase shift, since it has a major influence on the phase transition process [78]. After repeated cycling, a PCM exhibiting phase separation will exhibit a decrease in melting enthalpy [71]. Consequently, subcooling can have negative effects during dynamic experiments and may pose significant challenges in technical applications of PCMs.

2.2.2 Performance of TES Components with PCMs

LHTES tanks can be designed with different storage configurations, including bulk storage, macro-encapsulated, or micro-encapsulated PCMs. By modifying the tank configuration and the operating boundary conditions, the LHTES component characterisation aims to explore potential ways to speed up the charging/discharging process.

Recent advancements in encapsulation technologies, particularly macroencapsulation, have further enhanced the applicability of PCMs in domestic hot water systems. Macroencapsulation involves enclosing PCMs in larger containers, typically greater than 10 mm in diameter, which allows for improved thermal stability and performance. This method can mitigate issues such as leakage and degradation, which are critical for maintaining the integrity of the PCM and TES tank over time.

Studies have shown that macroencapsulated PCMs can achieve higher energy densities compared to their microencapsulated counterparts, thus improving the overall thermal energy storage capacity of the system and also, can solve endogenous problems associated with some PCMs [74].

Bulk storage, involving direct storage of the PCM in tanks without encapsulation, offers notable benefits concerning simplicity and cost-effectiveness. However, this method poses challenges such as the risk of PCM leakage, and materials' mechanical degradation over time can lead to structural failures if not managed properly [79]. Also, traditional bulk PCM storage suffers from low heat transfer rates, which can lead to incomplete melting/solidification and thermal stratification [14].

The second type of PCM containment uses a submerged heat exchanger, to perform the phase change in the material [80]. Various CFD studies have been developed regarding the performance of these heat exchangers. Hu et. al [18] built a CFD model to compare the performance of bulk storage with a submerged coil heat exchanger with slab macroencapsulation. Bulk storage revealed larger energy storage density, the ratio of PCM capacity to the total storage energy capacity

(Q_{PCM}/Q_{TES}) reaches 98% in the SCHE (submerged coil heat exchanger) based unit, 26% in encapsulated PCM unit. On the other hand, the encapsulated PCM situation demonstrated faster discharging capacity. When the SCHE-based unit could only reach 87% of its overall capacity in the same time period, it was able to discharge completely. Different heat exchanger surface areas and PCM quantities cause the heat transfer rate difference, which in turn causes the storage capacity discrepancy. A 1 m² heat exchanger stores and extracts heat from 2.53 kg of PCM in the encapsulated PCM unit; in the SCHE-based unit, this figure rises to 54.4 kg.

Macroencapsulation design has been one of the main priorities when it comes to the development of LHTES systems, specially optimizing the surface area-to-volume ratio [74]. In this project, several numerical and experimental studies have been performed to test the applicability of numerous geometries of capsules. T. Xu et al. [16], experimentally analyzed a 0.38 m³ LHTES unit containing cylindrically macro-encapsulated - Figure 2.6 (c) - phase change materials and evaluated its performance with different boundary conditions. Using the same tank but with different boundary conditions and encapsulation design, ellipsoidal macro-encapsulation, - Figure 2.6 (b) - T. Xu et al. [15] performed a experimental and numerical investigation.

This served as a comparison between both geometries, where the ellipsoidal capsule shows 60% less completion time of discharge but 23% lower storage capacity. This statistic resulted from a highly deviated shape from a sphere, which contributed to the high void fraction when compared with the cylindrical encapsulation.

The ellipsoidal encapsulation of ATP 60 was initially promising, as it allowed for better thermal conductivity and improved heat transfer rates, reducing the time required for complete charging and discharging cycles, although reducing the thermal storage capacity, in comparison with the cylindrical encapsulation of C58 [14, 15, 17]. The following tests, revealed that the slab encapsulation with ATP 60, had even better results when it comes to charging and discharging times, although having less thermal storage capacity overall [18]. At an HTF flow rate of 0.25m³/h, the fully discharged time was 2.8h, which is lower than the 3 h for ellipsoidal encapsulation [15]. The optimized design effectively reduced the void fraction to a level that maximizes PCM loading while maintaining efficient HTF flow. These improvements translated into significantly better performance during both the charging and discharging cycles, as evidenced by reduced cycle times and increased energy storage capacity. The consistency between numerical simulations and experimental results reinforces the conclusion that slab encapsulation is the optimal configuration for practical applications [17].

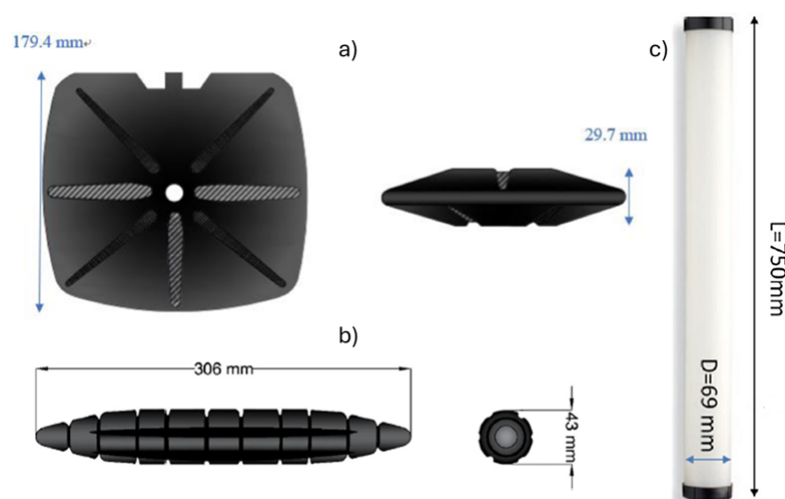


Figure 2.6: Macroencapsulation geometries tested in previous experimental work - (a) slab; (b) ellipsoidal; (c) cylindrical - [15, 16, 18].

The comparison between bulk storage, slab and ellipsoidal macro-encapsulation was also performed numerically, for a cogeneration application [17]. Initially, the impact of the HTF flow rate in the completion time of charge was analyzed, for both geometries. The two simulated real-size macro-encapsulated units generally exhibited similar behavior. The slab capsules, Figure 2.6 (a), can, on average, complete the charge and discharge process in 4% less time than the ellipsoid capsules, largely due to their higher effective heat transfer coefficient. Comparing the three designs' performance reveals that, although having a theoretical storage capacity that is less than that of the bulk storage system, the slab capsule design has the highest accumulated storage capacity (710 kWh) and state of charge (40%) after three hours of charging. After analyzing the storage power cost, the slab capsule design was found to be the most cost-effective, being 6–12% less expensive than the spiral coil design and 2–3% less expensive than the ellipsoid capsule design. It was concluded that the slab encapsulation geometry was the one that performed the better in all parameters compared with the ellipsoidal capsule.

Using lower storage capacities, Hu et al. [18], performed an experimental comparison between the slab macroencapsulation and bulk storage (SCHE). The same approach was used, as the impact of the flow rate on the charging and discharging times was initially studied. Increasing the flow rate led to faster charging and discharging times. When compared to constant flowrates, the enclosed PCM unit discharges with a longer constant thermal output and T_{out} above 57°C when a variable increasing flowrate is used. Additionally, the UA values were extracted for various flow rates in both scenarios. Discharge takes about three hours longer than charging for the SCHE-based storage unit under the testing conditions. When charging, the mean UA value is 69% higher than when discharging. Although the SCHE-based storage unit had a high energy density (57kWh/m³), it took a long time to fully discharge. Partial discharge with a slight trade-off in storage capacity can increase time efficiency. As the other experiments, the macroencapsulated configuration resulted in faster charging/discharging times but lower energy density.

Chapter 3

Theoretical Analysis

3.1 Theoretical Framework

In recent years, researchers have conducted more in-depth analyses of tank performance, exploring various boundary conditions and different PCM storage configurations. Key parameters include the total stored energy, which is essential for efficient utilization. This energy depends not only on the overall characteristics of the tank, including the PCM, but also on the specifications of the heating system.

Stored energy represents the amount of heat transferred by the heat transfer fluid to or from the system. If the control volume encompasses the entire storage tank, the resulting equation from the first law is:

$$\dot{m} \cdot h_{in} = \dot{m} \cdot h_{out} + \frac{dU_{system}}{dt} \quad (3.1)$$

Here, $\dot{m}$ represents the mass flow rate of the heat transfer fluid, h_{in} and h_{out} are the specific enthalpies at the inlet and outlet, respectively, and $\frac{dU_{system}}{dt}$ denotes the rate of change of the internal energy of the system (the tank, including the phase change material (PCM), polypropylene, and water). This equation accounts for the energy balance over the entire system, where the change in internal energy (ΔU_{system}) accumulates as the HTF transfers heat to or from the tank.

We can equate the heat transfer rate $\dot{Q}_{TES}$ to the rate of change of the system's internal energy:

$$\dot{Q}_{TES} = \frac{dU_{system}}{dt} \quad (3.2)$$

Conversely, if the control volume is defined solely around the water (HTF) flowing through the tank, the equation simplifies to:

$$\dot{m} \cdot h_{in} = \dot{m} \cdot h_{out} + \dot{Q}_{TES} \quad (3.3)$$

This derivation justifies the use of $\dot{Q}_{TES}$ rather than ΔU in the context of the experimental setup. The term $\dot{Q}_{TES}$ is preferred because it directly quantifies the heat transfer rate between the

HTF and the tank, aligning with the practical measurement of energy exchange in the charging and discharging processes.

The determination of the capacity is crucial for an optimized LHTES integration, according to the system necessities. For this, Equations 3.4, 3.5 and 3.6 were used to calculate an estimation of the total heat storage capacity of a TES tank.

Equation 3.4 determines the total latent heat that the PCM is able to store.

$$Q_{latent} = N_{capsules} \times m_{PCM} \times \Delta h_m \quad (3.4)$$

Where $N_{capsules}$ represents the total number of capsules inside the tank, m_{PCM} the mass of PCM inside a single capsule in kg, and Δh_m , the latent heat of fusion in kJ/kg.

Equations 3.5 and 3.6 represent the solid and liquid state sensible heat stored in the PCM, respectively.

$$Q_{sol,sensible} = m_{PCM} \times c_{p,solid} \times \Delta T \quad (3.5)$$

Where $c_{p,solid}$ represents the specific heat of the PCM in solid state in kJ/(kg · K) and ΔT , the difference between the initial temperature of the PCM and the temperature of fusion of the PCM.

$$Q_{liq,sensible} = m_{PCM} \times c_{p,liquid} \times \Delta T \quad (3.6)$$

Where $c_{p,liquid}$ represents the specific heat of the PCM in liquid state in kJ/(kg · K) and ΔT , represents the difference between the temperature of fusion of the PCM and the final temperature of the PCM, which is similar to the temperature of the HTF inlet temperature.

Equation 3.7 determines the total latent and sensible heat that can be stored inside the PCM capsules. It results from the combination of equations 3.4, 3.5 and 3.6. Equations 3.8 and 3.9 presents the sensible heat stored by the HTF and by the capsule's material, respectively.

$$Q_{PCM} = m_{PCM} [c_{p,solid} (T_m - T_i) + a_m \Delta h_m + c_{p,liquid} (T_f - T_m)] \quad (3.7)$$

Where Q_{PCM} represents the total heat added or removed from the material, m_{PCM} the mass of the PCM, T_i the initial temperature of the material, T_f the final temperature of the material, T_m the melting temperature of the material, $c_{p,solid}$ the average specific heat capacity of the solid phase between T_i and T_f , $c_{p,liquid}$ the average specific heat capacity of the liquid phase between T_m and T_f , a_m the fraction of the mass that melts ($0 \leq a_m \leq 1$) and Δh_m the latent heat of fusion per unit mass at T_m . The values employed were derived from laboratory experiments conducted prior to the initiation of the testing phase. The thermal properties of the PCM were obtained from the thesis by Songtao and Lu Yu [81], where they identified the scanning rate yielding the best results, minimizing noise interference while maintaining a sufficiently low rate (0.05 K/min) to approach near-thermodynamic equilibrium conditions. These values are detailed in Table 4.1.

The sensible heat stored in the HTF inside the tank must also be considered, which is calculated with Equation 3.8.

$$Q_{HTF} = (\alpha \times V_{PB} + V_{DR}) \times \rho_{HTF} \times c_{p,HTF} \times \Delta T \quad (3.8)$$

Where α represents the packed bed void fraction, V_{PB} the volume of the TES tank packed bed, V_{DR} the volume of the TES tank distribution region, ρ_{HTF} the density of the HTF, $c_{p,HTF}$ the specific heat of the HTF in kJ/(kg · K) and ΔT , the difference between the initial and final temperature of the HTF.

Finally, the heat stored in the capsule material must be accounted. The capsule is made of polypropylene (PP) and the total heat stored by it can be determined by Equation 3.9.

$$Q_{PP} = m_{PP} \times c_{p,PP} \times \Delta T \quad (3.9)$$

Where m_{PP} represents the total mass of polypropylene, $c_{p,PP}$ the specific heat of the material in kJ/(kg · K) and ΔT the difference between its initial and final temperatures.

Another crucial parameter is the heat transfer rate and its temporal evolution, particularly in relation to the amount of stored energy available at any given moment. The HTF temperature at the tank outlet, or the temperature differential between the inlet and outlet, is also critical. The outlet temperature is particularly significant, as it directly affects the heating performance of the building. The heat carried by the HTF is then dissipated by terminal units such as fan coil units, underfloor heating, or radiators. The effectiveness of these units depends on both indoor air conditions and the temperature of the HTF supplied to them. Therefore, monitoring the HTF temperature at the tank outlet over time and for different levels of stored energy is essential. This allows for determining whether the HTF can be used directly in the terminal units or if additional heating support from a heat pump or electric heater is required to increase the temperature.

The temperature difference of the HTF temperature ($T_{in} - T_{out}$), as indicated by Equation 3.10, is used to compute the transient thermal performance of charging/discharging in the LHTES system. Equation 3.11 then shows the net transient state power, P_{net} , after subtracting the heat loss, P_{loss} .

$$P = \rho_{HTF} \times \dot{V}_{HTF} \times c_{p,HTF} \times (T_{in} - T_{out}) \quad (3.10)$$

$$P_{loss} = \rho_{HTF} \times \dot{V}_{HTF} \times c_{p,HTF} \times (T_{in,final} - T_{out,final}) \quad (3.11)$$

$$P_{net} = |P - P_{loss}| \quad (3.12)$$

Equation 3.13 illustrates how P_{net} integrated throughout the time step (dt) yields the net heat collected (Q_{net}) at each time step (i). The final accumulated capacity in the LHTES system (Q_{TES}) is then calculated by adding the net thermal energy from each time step, as indicated in Equation 3.14, where n is the number of timesteps.

$$Q_{net}^i = P_{net}^i \times dt \quad (3.13)$$

$$Q_{TES} = \sum_{i=1}^n Q_{net}^i \quad (3.14)$$

With the capacity of the LHTES tank calculated, it's possible to determine the energy density (ED, kWh/m³). To be able to achieve a high ED while also avoiding issues such as subcooling or "dead zones", it's crucial to select an optimized encapsulation geometry with a high value of the quotient between superficial area and volume. This way, it's possible to increase heat transfer between the PCM and HTF and the capacity of the tank, by including more capsules. Energy densities are compared on both the component level (ED_{TES}) and the encapsulated PCM level (ED_{PCM}) to enable generic comparison between various LHTES designs.

$$ED_{TES} = \frac{Q_{TES,tot}}{V_{TES}} \quad (3.15)$$

$$ED_{PCM} = \frac{Q_{PCM,tot}}{V_{PCM}} \quad (3.16)$$

Upon integrating over the full duration of the charging/discharging cycle (t_{comp}), the total energy storage capacity ($Q_{TES,tot}$) is obtained. The state of charge (SoC) at any given time is defined as the ratio between the cumulative stored energy and the total storage capacity, as shown in Equation 3.17.

$$SoC = \frac{Q_{TES}}{Q_{TES,tot}} = \frac{\int_0^t q_{TES} dt}{\int_0^{t_{comp}} q_{TES} dt} \quad (3.17)$$

The state of discharge (SoD) is also a parameter that must be taken into account when developing discharge strategies. By the same logic, it represents the ratio between the cumulative released energy and the total storage capacity capable of being released on discharge, as shown in Equation 3.18.

$$SoD = \frac{Q_{TES}}{Q_{TES,DIS,tot}} = \frac{\int_0^t q_{TES} dt}{\int_0^{t_{comp}} q_{TES} dt} \quad (3.18)$$

Along with the above mentioned parameters, the charging and discharging times of the tank are crucial for understanding the system's behavior [18]. In this work, to obtain these time values, it was assumed that the LHTES was fully charged/discharged when T_{out} attained a steady temperature, which means when T_{out} and T_{in} converge with a certain difference related with the heat transfer resistance.

Korti et al. [82] experimentally built an LHTES unit using a spiral coil heat exchanger (SCHE) and found that the heat transfer fluid (HTF) flow rate significantly impacts the charging and discharging times. Their results indicated that increasing the HTF flow rate accelerates the process, while a larger HTF temperature difference shortens the completion time. However, reducing the HTF temperature difference helps maintain a more uniform temperature distribution in the PCM

during charging and discharging. Hu et al. [18] reinforced these findings by experimentally studying the effect of HTF flow rate in both a bulk PCM storage tank and a tank filled with macro-encapsulated PCM. Their results showed that increasing the HTF flow rate improved the overall heat transfer coefficient (UA), with a more pronounced effect in the macro-encapsulated case. Numerical studies have also supported these conclusions, showing that heat convection plays a dominant role at the beginning of the charging process, while heat conduction governs PCM melting in the later stages [83].

The stored energy efficiency of the tank must also be evaluated. It determines the amount of heat that can be used after the tank was charged with an external heat source. It has to do not only with the ability of the PCM to hold the heat but also the thermal insulation of the tank. This parameter is affected by the time between the charge and the discharge, so it's crucial to determine the overall heat transfer coefficient of the tank walls. It can be derived as shown in Equation 3.19.

$$\eta_{\text{tank}} = \frac{Q_{\text{discharge}}}{Q_{\text{charge}}} \quad (3.19)$$

Another important parameter is the latent heat transfer performance between the HTF and the PCM. The presented method is called effectiveness-NTU method, also used in other researches [18, 84, 85]. Although the evolution of the UA has been evaluated, the average value will be again determined for different flow rates, due to some differences in the test bench. Equation 3.20 provides the instantaneous effectiveness (ε) that is used to assess the PCM's heat storage/release performance during phase transition.

$$\varepsilon = \frac{Q_{\text{actual}}}{Q_{\text{max}}} = \frac{T_{\text{in}} - T_{\text{HTF}}}{T_{\text{in}} - T_{\text{PCM}}} \quad (3.20)$$

The HTF inlet temperature is represented by T_{in} , while the PCM's instantaneous temperature at the phase transition stage is denoted by T_{PCM} . The HTF's instantaneous outlet temperature is denoted by T_{HTF} . Following the determination of (ε), Equations 3.21 and 3.22 yield the instantaneous overall heat transfer coefficient (UA) and NTU (number of transfer units), respectively.

$$NTU = -\ln(1 - \varepsilon) \quad (3.21)$$

$$UA = NTU * \rho_{\text{HTF}} \times \dot{V}_{\text{HTF}} \times c_{p,\text{HTF}} \quad (3.22)$$

Equation 3.23 finally displays the integral average of the UA throughout the complete phase change time (t_{tot}), in $\text{W} \cdot \text{K}^{-1}$.

$$\overline{UA} = \frac{\int_0^t UA dt}{t_{\text{tot}}} \quad (3.23)$$

UA values serve as a way to evaluate heat transfer between the PCM capsules and the HTF during charge and discharge situations.

During storage periods, where the TES tank isn't neither charging or discharging, it's crucial to determine what are the losses to the external space where the tank is located. For this matter, Equation 3.24 can be used.

$$Q_{sto,loss} = U_{tank}A_{tot} \times \Delta T \quad (3.24)$$

Where U_{tank} represents the global heat transfer coefficient in $W \cdot m^{-2} \cdot K^{-1}$, A_{total} the outer surface of the insulation, in m^2 , and ΔT the temperature difference between the HTF and the outside air.

To determine $U_{tank}A_{total}$, Equation 3.25 was formulated.

$$U_{tank}A_{total} = \left[\frac{A_{side}}{R_{total,side}} + \frac{2 \times A_{top/bot}}{R_{total,top/bot}} \right] \quad (3.25)$$

The values of $R_{total,side}$ and $R_{total,top/bot}$ were determined by the following Equations 3.26 and 3.27, respectively.

$$R_{total,side} = R_{conv,HTF,side} + R_{tank,side} + R_{ins,side} + R_{conv,air,side} \quad (3.26)$$

$$R_{total,top/bot} = R_{conv,HTF,top} + R_{tank,top} + R_{ins,top} + R_{conv,air,top} \quad (3.27)$$

Equations 3.26 and 3.27 have been further elaborated to Equations 3.28 and 3.29.

$$R_{total,side} = \frac{1}{h_{HTF} \cdot A_{int,side}} + \frac{\ln\left(\frac{r_{ext,tank}}{r_{int,tank}}\right)}{2\pi k_{tank} h} + \frac{\ln\left(\frac{r_{ext,ins}}{r_{int,ins}}\right)}{2\pi k_{ins} h} + \frac{1}{h_{air} \cdot A_{side}} \quad (3.28)$$

$$R_{total,top/bot} = \frac{1}{h_{HTF} \cdot A_{int,top}} + \frac{t_{tank}}{k_{tank} \cdot A_{int,top}} + \frac{t_{ins}}{k_{ins} \cdot A_{ext,top}} + \frac{1}{h_{air} \cdot A_{top}} \quad (3.29)$$

Where h_{HTF} and h_{air} represent the convection coefficient from the HTF and outside air, respectively, in $W \cdot m^{-2} \cdot K^{-1}$. A_{int} and A_{ext} represent the area of the tank that is exposed to the HTF and the area of the tank that is exposed to the outside air, respectively, in m^2 . t_{tank} and t_{ins} represent the thickness of the tank wall and insulation, respectively, in m. $r_{ext,tank}$, $r_{int,tank}$ and $r_{ext,ins}$ represent the radius of the external and internal wall of the tank and the external wall of the insulation, respectively, in m. k_{tank} and k_{ins} represent the thermal conductivity of the tank wall and insulation, respectively, in $W \cdot m^{-1} \cdot K^{-1}$. The areas mentioned, $A_{int,side}$, A_{side} , $A_{int,top}$ and A_{top} represent the internal area on the side of the tank, the external area on the side of the tank, the internal area on the top of the tank and the external area on the top of the tank, in m^2 , respectively.

3.2 Theoretical Values

3.2.1 Total Heat Stored and Released Capacity

Theoretical values were determined to predict some of the previously mentioned parameters. To determine the total heat that could, theoretically, be stored, it was considered the sensible and latent heat on the PCM and the sensible heat on the HTF and the capsules' material. All the other forms of heat were not considered but we'll be later discussed. Losses during the charging and discharging are also disconsidered for the determination of the theoretical values.

To optimize the calculus and determination of the theoretical values for the total storage and power output on different charge and discharge scenarios, respectively, a *Python* sheet was developed. It can be found in Appendix A.

Table 3.1 shows how different boundary conditions can impact the theoretical capacity values. This values will further be useful when the SoC is evaluated along the time, creating a reference value to compare different scenarios, as shown in equation 3.17.

Table 3.1: Total TES Heat Stored Capacity for Different Temperature Increases

Temperature Increase [°C]	Total Storage (kJ)	Total Storage (kWh)
25 to 62	169619.06	47.12
25 to 65	182539.30	50.71
25 to 70	199131.13	55.31

It was considered that the discharge started from the end of the charge test and ended by the start of the next charge test. So, Q_{HTF} and Q_{PP} remain the same as in Equations 3.8 and 3.9.

Table 3.2 shows the theoretical power outputs in different discharge tests and how boundary conditions can impact the total power release.

Table 3.2: Total TES Heat Released Capacity for Different Discharge Scenarios

Temperature Decrease [°C]	Total Storage (kJ)	Total Storage (kWh)
65 to 30	129690.46	36.03
65 to 25	147834.73	41.07

3.2.2 Heat Losses

For the $Q_{sto,loss}$ estimation, it was considered that the tank was fully charged and the HTF at 65°C. Equation 3.25 was used to determine the global heat transfer coefficient and the result is shown in Equation 3.32. For the value of h_{HTF} and h_{ext} , *CoolProp* was used to get a approximate value to the situation conditions. The values obtained were 912.71 and 66.41 W·m⁻²·K⁻¹, respectively. The thermal conductivity coefficient of the insulation was selected from the data given in the datasheet, being 0.043 W·m⁻¹·K⁻¹, and for the TES tank wall, it was assumed to be around 15 W·m⁻¹·K⁻¹, as it is stainless steel.

After accessing the geometry and physical-thermodynamical properties of the tank and insulation, these heat transfer resistances were determined, as shown in Equations 3.30 and 3.31.

$$R_{\text{total,side}} = \frac{1}{912.71 \cdot 0.714 \cdot \pi \cdot 1.515} + \frac{\ln\left(\frac{0.367}{0.357}\right)}{2 \cdot \pi \cdot 15 \cdot 1.515} + \frac{\ln\left(\frac{0.387}{0.367}\right)}{2 \cdot \pi \cdot 0.043 \cdot 1.515} + \frac{1}{66.41 \cdot 0.774 \cdot \pi \cdot 1.515} \approx 0.1354 \text{ K} \cdot \text{m}^2 \cdot \text{W}^{-1} \quad (3.30)$$

$$R_{\text{total,top/bot}} = \frac{1}{912.71 \cdot (\pi \cdot (0.357)^2)} + \frac{0.01}{15 \cdot (\pi \cdot (0.357)^2)} + \frac{0.02}{0.043 \cdot (\pi \cdot (0.387)^2)} + \frac{1}{66.41 \cdot (\pi \cdot (0.387)^2)} \approx 1.025 \text{ K} \cdot \text{m}^2 \cdot \text{W}^{-1} \quad (3.31)$$

$$U_{\text{tank}} A_{\text{total}} = \left[\frac{\pi \cdot 0.774 \cdot 1.515}{0.1354} + \frac{2 \cdot (\pi \cdot 0.387^2)}{1.025} \right] \approx 28.132 \text{ W} \cdot \text{K}^{-1} \quad (3.32)$$

$$Q_{\text{sto,loss}} = U_{\text{tank}} A_{\text{tot}} \cdot \Delta T = 28.132 \cdot (65 - 26) \approx 1097.15 \text{ W} \quad (3.33)$$

Equation 3.33 shows the power loss in the tank when it's fully charged at 65°C and the outside air is at 26°C, which is the average value of the temperature inside the laboratory where the tank is being tested. This would result in a loss equivalent to 3949.73kJ/h, which in 10 hours is 39497.32kJ or 10.97 kWh, representing around 21.64% of the theoretical total storage capacity on a charge from 25 to 65°C.

Chapter 4

Methodology

In the Department of Energy Technology laboratory at KTH, a specialized test rig was designed, built, and put into use to study thermal energy storage devices. Specifically, the performance of latent heat thermal storage was experimentally investigated using a precise selection of test rig components.

The future application of the storage system will be in a student housing building, where it will be coupled with a heat pump to improve demand-side management.

The storage tank was designed with a thermal capacity sufficient to cover the peak heat demand of the building for a period of around 3 hours, depending on the discharging strategies. According to NordPool's [12] observations of the hourly electricity trade prices in Sweden, the morning peaks (usually 6 am to 9 am) and evening peaks (usually 5 pm to 8 pm) have higher prices than other hours during heat seasons. Accordingly, the thesis considers three to four hours to be the duration of the daily peak times. Additionally, research indicates that moving the whole heating load from the two daily peak periods (morning and evening peaks, which last roughly three hours each) to off-peak hours is one of the most cost-effective load shifting solutions for Swedish single-family homes [86].

The system's role is to store thermal energy produced by the heat pump during off-peak hours, allowing for flexible energy management and optimized performance under dynamic load conditions. This will contribute to load shifting strategies, aligning energy consumption with periods of lower electricity prices or increased renewable energy availability.

A previous test rig iteration was developed and utilized to conduct experiments focusing on several key parameters, including phase change material selection by considering melting temperature, thermal conductivity, and long-term stability; heat transfer characterization by analyzing the impact of heat exchanger geometries and configurations on charging and discharging rates as well as heat losses from the tank while charged; encapsulation methods by evaluating different macro-encapsulation approaches to enhance thermal performance and durability; and overall system efficiency by benchmarking against existing storage technologies and assessing the system's potential integration in real applications.

Based on the findings from previous research, the most effective components and materials

were selected for the latest test rig iteration, aiming to maximize heat exchange efficiency and storage capacity while maintaining system flexibility.

One of the key innovations in this new experimental setup is the ability to compare two distinct coupling strategies for integrating the storage tank with the heat pump. This configuration will allow for an in-depth assessment of how different hydraulic layouts influence thermal performance and overall system operation. Other differences will be the PCM used for storage and also the performed analysis on charge and discharge strategies.

Additionally, the test rig is designed to reject excess heat into a district cooling network, enabling a more efficient energy utilization strategy and providing a realistic representation of how the system could function in an urban energy infrastructure. This approach aligns with the concept of sector coupling, where waste heat from cooling applications can be repurposed to enhance energy efficiency and reduce overall environmental impact.

The experimental campaign will include multiple testing scenarios, assessing the dynamic behavior of the storage system under varying operational conditions. Since only PCMs for heating applications were examined, water was utilized as the HTF in the experiments described in this thesis.

Key performance indicators such as charging and discharging rates, system response time, energy efficiency, and thermal stability will be analyzed to establish a comprehensive performance mapping of the system. The results obtained will serve as a foundation for future real-scale implementations, bridging the gap between laboratory testing and full-scale applications in residential energy systems.

4.1 Description of the Future Application

The future application of the TES tank will be in a space heating system powered by a heat pump. For this, the student housing facilities in KTH will be made available to perform the future testing [87]. To prepare these tests, the following experiments were performed to evaluate the PCM tank performance, following the real case scenario values for the outlet and return temperatures on the AHU (Air Handling Unit).

The student housing building uses floor venting connected with a AHU to provide space heating. The AHU, after mixing the exhaust air, that comes from the apartments, with outside air, heats the mixture upon the space heating demands. This is achieved using an air-water heat exchanger, where the water is heated by an heat pump system.

A simplified schematic of the current heating system is shown in Figure 4.1. The heat pump is responsible for heating the water that will exchange heat with the air mixture. Connecting the TES tank will make it the supplier of hot water for the same purpose but only when charged, and if the discharging strategy aligns with this. To prevent not having hot water during high demand times, there should be a bypass valve that allows the HP to work without having to charge the TES tank first, or even an electric heater, if the demand exceeds the thermal storage capacity of the tank.

The demand of the student housing building was measured during the last year, before the dimensioning of the tank and the highest peak recorded was around 10 kW for 5 hours. This led to the final dimension of the tank which resulted in approximately 60% of the highest total peak in kWh. This capacity would be enough to cover more than 80% of the year peaks, with the rest of the days needing the heat pump to fulfill the demand.

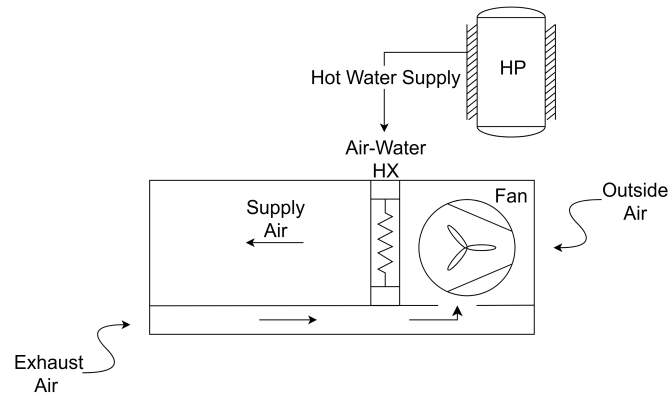


Figure 4.1: Simplified schematic of the current AHU system.

In Figure 4.2, it is possible to see the current Live-in Lab heating set-up. It is composed by a shunt group (1), designed to regulate and divert the flow of the fluid. The air-water heat exchanger (2) performs the transfer of heat between the fluid and the air. The return air duct (3) plays a vital role by channeling a portion of the heated air back into the system after warming the building, promoting heat recovery and reducing energy waste. The supply air duct (4) ensures the distribution of conditioned air throughout the space, while the water tank (5) serves as a thermal storage reservoir, stabilizing temperature fluctuations and supporting sustained heating. Finally, the heat pump (6) acts as the core component, driving the heating cycle by extracting and upgrading heat from the environment or the existing borehole. This setup reflects a holistic approach to heating, balancing performance, energy conservation, and system reliability.

As mentioned before, the presented TES tank will be coupled in this system to allow PLS.

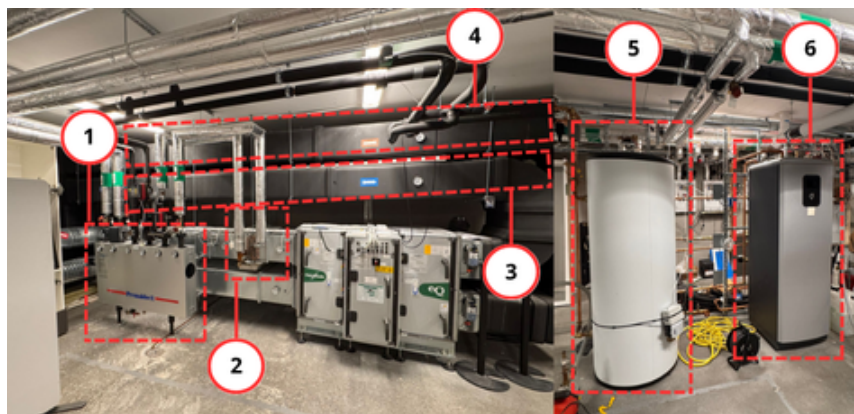


Figure 4.2: Live-in lab current heating set-up.

4.2 Description of the Bench Test Facility

Based on the previous literature review in Chapter 2, the phase-change temperature of the selected PCM candidates for active space heating applications generally falls within the range of 45°C to 60°C. In this study, RT57HC, an organic paraffin based PCM by *Rubitherm*, was used and its properties were measured in-house at specific temperatures to minimize the discrepancy between the future numerical and experimental results, and also, to choose the inlet temperatures of the HTF in the charging and discharging scenarios, as shown in Table 4.1. As long as the HTF temperature was over the highest temperature in the melting range, the latent heat charging was assumed as complete, although other criteria was also used to assure full charge as shown further.

Table 4.1: Thermo-physical properties of RT57HC.

Parameter	Measurement [81]	Datasheet [88]
Density, solid ($\rho_{PCM,s}$) [kg/m ³]	883±28.7 ^a	900 ^h
Density, liquid ($\rho_{PCM,l}$) [kg/m ³]	770-760 ^b	800 ⁱ
Thermal conductivity, solid ($k_{PCM,s}$) [W/(m·K)]	0.25-0.27	0.2
Thermal conductivity, liquid ($k_{PCM,l}$) [W/(m·K)]	0.18	0.2
Specific heat, solid ($c_{p,PCM,s}$) [kJ/(kg·K)]	2.11 ^c	2
Specific heat, liquid ($c_{p,PCM,l}$) [kJ/(kg·K)]	2.72 ^d	2
Viscosity (μ) [mPa·s]	10.1 ^e -5.78 ^f	NA
Melting temperature range ($T_{m,on} - T_{m,off}$) [°C]	57.1-59.4	55 – 58 ^j
Crystallization temperature range ($T_{c,off} - T_{c,on}$) [°C]	51.9-52.4	53 – 57 ^j
Latent heat of fusion per unit of mass (Δh_m) [kJ/kg]	246.35 ^g	240 ± 7.5%

^a Measured at 20°C; ^b Measured from 65°C to 68°C; ^c Measured at 40°C; ^d Measured at 65°C; ^e Measured at 61.5°C; ^f Measured at 80°C; ^g With the optimal scan rate; ^h Measured at 20°C; ⁱ Measured at 60°C; ^j Main peak is 57°C.

The storage design of the PCM was also one of the previous research priorities. After numerous experimental and numerical analysis, the off-the-shelf slab macroencapsulation was selected, shown in Figure 4.3. This capsule will be responsible for the storage of the PCM and was developed to minimize frequent issues related with the phase transition of the material, also enabling fast charge and discharge by having a well balanced surface area-volume ratio.

Inside the tank, it was possible to fit 515 capsules which resulted in a higher theoretical capacity than in previous tests. To fit the most capsules possible, a swirl fitting technique was used to ensure a better distribution and accommodation of capsules within the tank. This was done with a extended mechanical arm to reach the deepest parts of the tank, and manually when it reached a more accessible height.



Figure 4.3: Slab macroencapsulation by Rubitherm [89].

The TES tank configuration is presented in Table 4.2. It was developed to have a thermal energy capacity of around 40 kWh, by previous researchers. The equations used to obtain this thermal capacity are presented in Chapter 3. The schematic of the TES tank is presented in Figure 4.4.

Table 4.2: Design Parameters of Encapsulated PCM Tank

Parameter	Encapsulated PCM tank
Tank volume [m ³]	0.607
Packed bed volume [m ³]	0.480
Distribution region volume [m ³]	0.127
HEX heat transfer area [m ²]	43.775
No. of encapsulated PCM [-]	515 ^{a,b}
Total volume of PCM capsules [m ³]	0.251
Total mass of PCM [kg]	133.385
Packed bed void fraction [-]	0.477
Theoretical Capacity [kWh]	41.07 ^c

^a The capsules are partially filled, around 93%.

^b The number of capsules reduced to 512 after 3 of them leaked.

^c Considering a 25-65°C charging scenario.

The tank volume was dimensioned to deliver partially the peak heat demand of the building. It was originally 0.6m in diameter and 1.1m in height, not considering the distribution regions. These dimensions changed so the tank would be able to meet a thermal energy peak requirement of around 10kWh during at least three to four hours.

Figure 4.4 shows the positions where the RTDs were mounted. RTD 2, 4 and 6 don't reach the center of the tank but the center of the radius of the tank, enabling to measure the temperature of the HTF between the center and the tank wall. RTD 1, 3, 5, 7 and 8 do reach the center of the tank and are used to describe the temperature stratification inside the tank. To ensure that the phase transition is happening according to the expected, RTDs were inserted inside 5 capsules, labeled as $T_{PCM,1}$ through $T_{PCM,5}$. This also allowed to measure the temperature of the PCM at different heights, which was important to comprehend if the stratification would interfere with the PCM

phase change and temperature along the charge and discharge - Figure 4.5. These cables were extracted from the upper cap and sealed.

The top of the capsules was carefully drilled while the PCM was in solid phase, to fit the RTD inside them. After this process, they were taken to the oven for around 90 minutes at 80°C and the PCM melted. The RTD was then carefully inserted while being held in a diagonal position to reach the core of the capsule, until the PCM solidified. Afterwards, the capsule's surface was sanded down, wiped clean with acetone, and treated with a primer application. A silicone paste was then applied around the hole and let to solidify. Finally, a self-vulcanizing tape was placed around the area where the paste was applied. This helped on keeping it in place and on adding an extra layer of sealant.

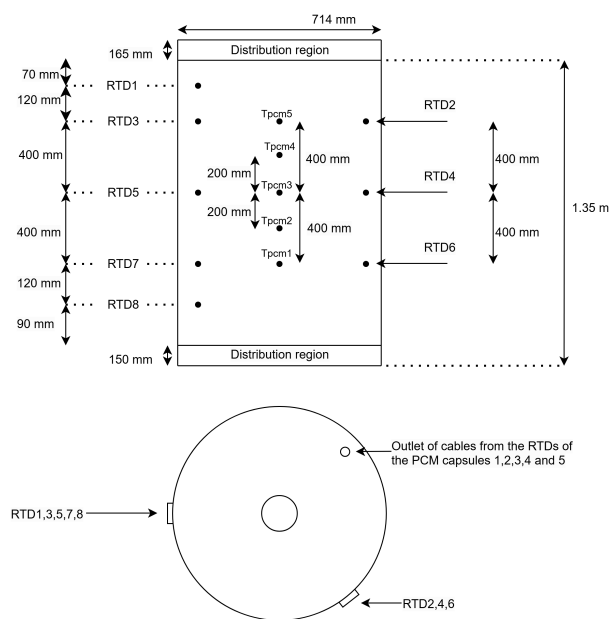


Figure 4.4: Schematic of the TES tank RTDs mounting positions.

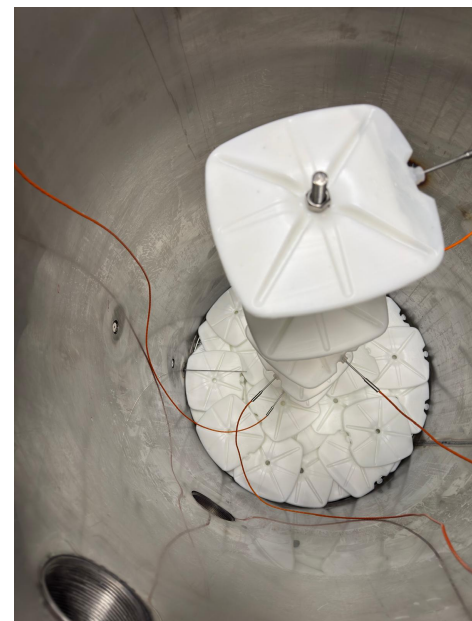


Figure 4.5: PCM capsules RTDs mounting positions.

As seen in Figure 4.4, the distribution regions are located both on the top and lower part of the tank to create a more uniform flow before entering the proper storage region. These zones were created to enhance homogeneous heat transfer in the tank. The distribution plate can be seen in Figure 4.6. To the bottom distribution plate, a mesh plate, with a certain height, was added to prevent the obstruction of the holes by the PCM capsules, as seen in Figure 4.7.

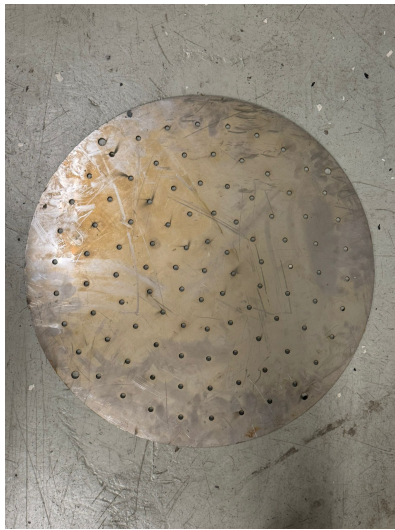


Figure 4.6: Distribution Plate.



Figure 4.7: Mesh Plate.

The schematic of the final configuration of the bench test is presented in Figure 4.9. In the inlet and outlet of the tank, two RTDs were mounted to evaluate the total energy stored and removed during charge and discharge, respectively. The electric heater and the district cooling network (DCN) loops are coupled to the water tank. The first will be responsible for the HTF heating before the charging and the second, for simulating the discharge scenario. An extra electric heater was added outside this loop. This 3kW electric heater served to lower the temperature on the water tank as the HTF could be heated after leaving the tank. Before starting the testing, different flow rates were tested to verify how the electric heater reacted and how much power it could effectively deliver. Its thermostat was set to the desired inlet temperature during discharge.

RTDs were also inserted inside the water tank, to allow the monitoring before and during the tests; between the wall of the TES tank and the insulation; and outside layer of the insulation of the TES tank.

Circulation pumps - CP1 and CP2 - were added to ensure that the HTF was flowing and exchanging heat with the DCN and receiving heat from the electric heater. The outlet of the electric heater circuit was mounted at the top of the water tank and the inlet at the bottom, to counter the stratification effect created by natural convection - Figure 4.8. The cooling was obtained using the district cooling network and a plate heat exchanger, as seen in Figure 4.8. To prevent the freezing of brine, the district cooling network temperature was adjusted to 5°C. It was also added a safety condition, which dictated that the system should stop exchanging heat with the DCN if the return temperature was higher than a fixed value. For this, valves were added before the heat exchanger.

The flow meter, labeled "FM", was utilized to measure the flow rate of the HTF prior to initiating the charging and discharging processes. By adjusting the pump frequency, the HTF flow rate was fine-tuned to meet the required values specified in the test matrix.

The hose mounted on the water tank, referred to as the "Overpressure Water Release Hose", functions as an expansion tank. This became necessary because, when the connections to the

TES tank are closed, there is no link to the existing expansion tank. As the temperature of the HTF inside the water tank increases, the volume of the fluid expands to a point where excessive pressure buildup occurs. To address this issue, an overpressure release hose was added, extending to a height above the entire system, ensuring it does not drain under normal operating conditions. The drain line was added to fasten the switch between charge and discharge mode. When the charge process is finished, it's faster to drain the tank partially and fill the rest of it with mains water, than to cool it until a much lower temperature.

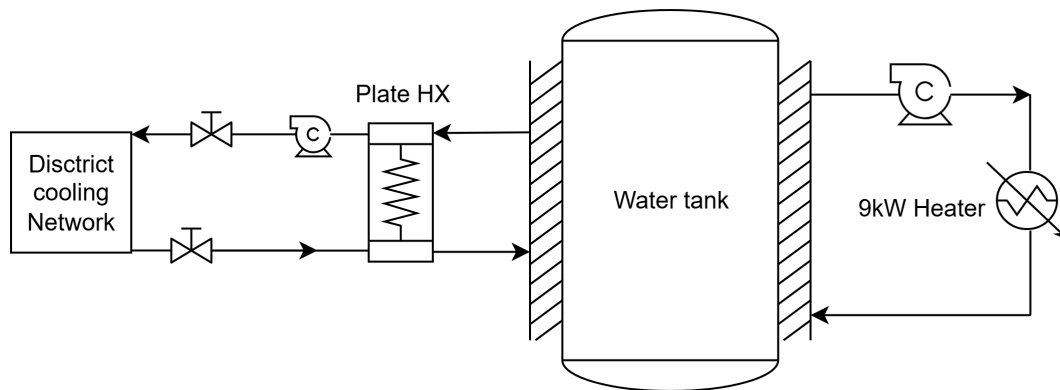


Figure 4.8: Water Tank' heating and cooling loops.

To compensate the pressure increase in the rest of the system, an expansion tank was mounted. It was considered that the water would increase from 20°C to 70°C inside the system, but a safety factor was added. Also, a pressure relief valve was mounted and regulated to 1.75 bar using an analog gauge, to ensure that the maximum pressure that the pipes and components could hold wasn't reached.

To prevent the existence of air inside the system, at the highest point of the rig, a air release valve was mounted. It was calibrated during the filling of the rig with the HTF, by opening it until water was coming out from it.

To capture and stream all measured values from the RTD sensors and flow meter, a data logger, connected to a desktop computer and a monitor, was mounted on the side of the rig. This setup enabled the verification of initial and final test conditions, as well as real-time monitoring and control of the values throughout the testing process.

The cables were securely connected to the cards in the data logger, and subsequently, the device was programmed to convert the measured voltage (in the case of the RTDs, further converted to resistance) into temperature and flow rate values, providing accurate readings for subsequent analysis and system monitoring.

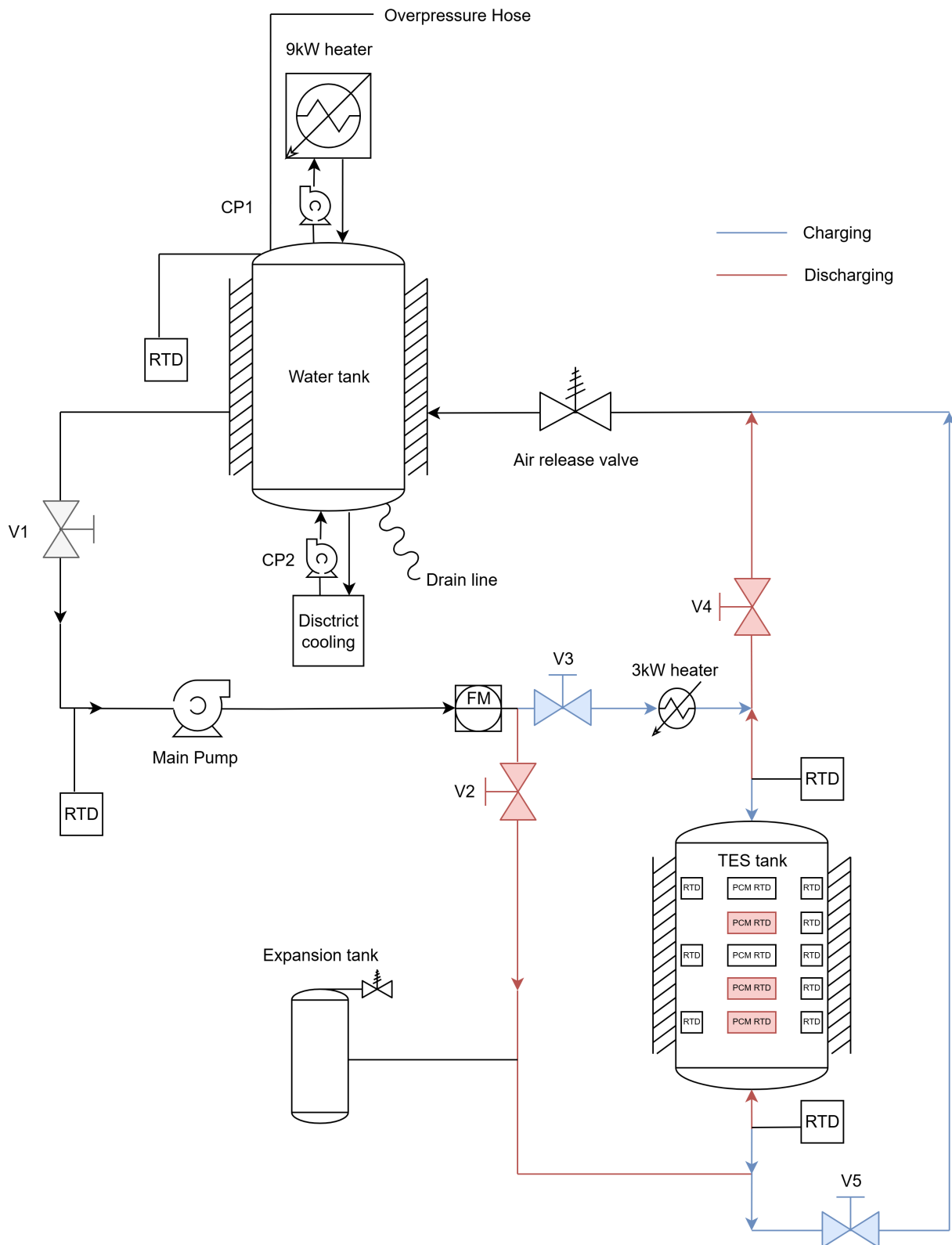
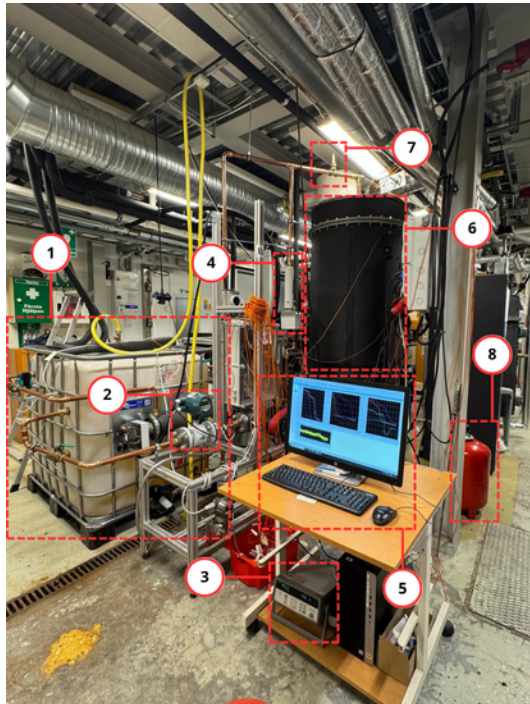
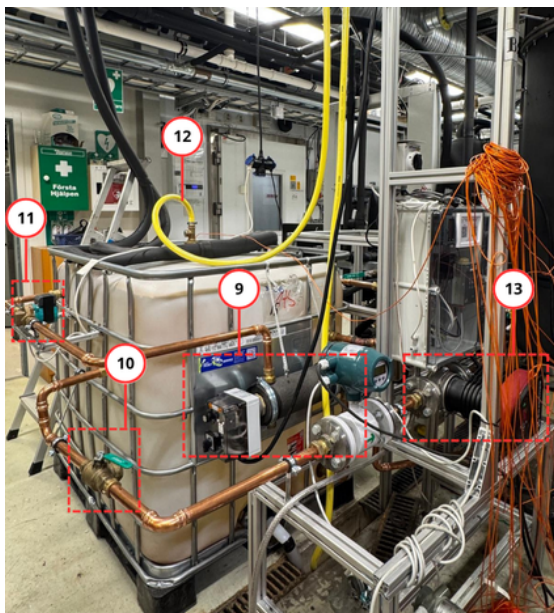


Figure 4.9: Test bench configuration.

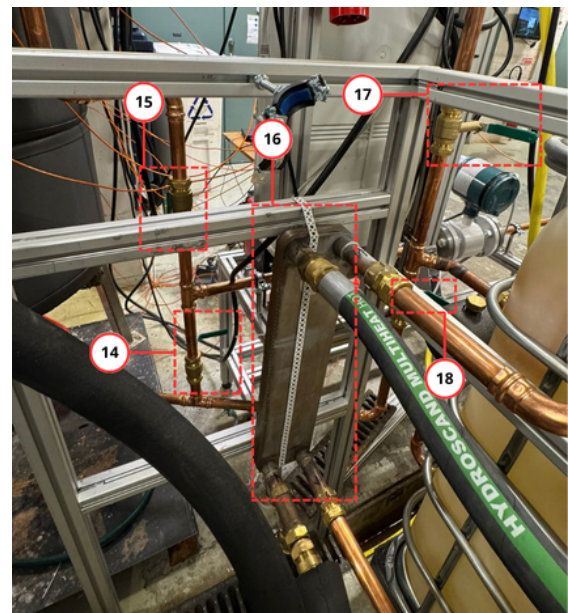
Figures 4.10a, 4.10b and 4.10c present the physical bench test and labels its components. Table 4.3 labels each component and refers its location and function, based on these figures.



(a) Overall view of the test rig including TES tank, monitoring station and pressure-safety equipments.



(b) Part of the equipments that simulate the charging and discharging.



(c) Other equipments that perform charge and discharge, and valves.

Figure 4.10: Overview of components from the test bench setup.

Table 4.3: Labels and functions of test bench components.

Label	Component	Location	Function
1	Water Tank	Between TES tank and filling station	Inertia tank
2	Flow Meter	Between "V1" and Main Pump	Measures whole system flow rate in L/min
3	Data Logger	Below Monitoring Station	Converts electrical signals into measurable temperature and flow rate readings
4	3kW Electrical Heater	Between "V3" and TES tank inlet	Adjusts TES tank inlet temperature
5	Monitoring Station	Desk	Allows to observe live plotting of data
6	TES tank	-	Stores thermal energy
7	Air-Release Valve	Top of the system	Removes existing air inside the hydraulic system
8	Expansion Tank	Between "V2" and TES tank outlet	Depressurizes the system if it reaches a certain pressure
9	9kW Electrical Heater	Water tank heating loop	Increases the HTF temperature until it reaches $T_{f,in}$, for charging scenario
10	"V1" - Main Valve	Between water tank and flow meter	Prevents/Allows the HTF to flow from the water tank to the TES tank
11	"CP1" - Circulation Pump	Water Tank Heating Loop	Circulates the HTF through the heating loop
12	Hose	Cap of the water tank	Doesn't allow overpressure in the water tank when all valves are closed and the heating loop is on
13	Main Pump	Between flow meter and TES tank	Maintains the preset flow rate on the system
14	"V2" - Discharge Valve	Between TES tank and water tank	Operates discharge scenario
15	"V3" - Charge Valve	Between 3kW electric heater and tank inlet	Operates charge scenario
16	Plate Heat Exchanger	Water tank cooling loop	Decreases the HTF temperature until it reaches $T_{f,in}$, for discharging scenario
17	"V4" - Discharge Valve	Between TES tank and water tank	Operates discharge scenario
18	"V5" - Charge Valve	Between TES tank and water tank	Operates charge scenario

4.3 Test Matrix and Testing Procedure

The purpose of evaluating the performance of the LHTES tank is to investigate possible methods for accelerating the charging and discharging procedures by altering the operating boundary conditions and the tank design.

The heat transfer between the HTF and the PCM capsules is the main concern when evaluating these methods. From a theoretical standpoint, the total thermal resistance (R_{tot}) can be thought of as consisting of three primary segments when examining heat transmission within the encapsulated PCM using a quasi-steady state approach: 1) heat conduction in the polypropylene shell ($R_{cond,sh}$); 2) heat transfer in the encapsulated PCM (R_{pcm}) for both solidified and molten sections, including conduction and natural convection in molten PCM; and 3) convection at the capsule surface (R_{conv}).

Therefore, it is expected that the following elements will affect both the overall charging or discharging power and the total thermal resistance.

- HTF inlet temperature ($T_{f,in}$): The primary driver of heat transfer and phase change is the temperature of the HTF at the PCM tank's inlet;
- The PCM tank's initial temperature ($T_{TES,ini}$): In the first step of a charging/discharging process, the temperature differential between HTF and PCM affects the charging/discharging thermal power.
- HTF inlet flowrate ($\dot{V}_f$): Affects convective heat transfer coefficients at the capsule surface. Must take into consideration that having a very high flow rate could result in low heat transfer, so finding a well balanced value is important to enhance the overall system performance.

The test matrix presented in Tables 4.4 and 4.5 was elaborated based on previous experiments, results mentioned on Chapter 2 and also based on Figure 2.4 which presented the most common radiator forward and return temperatures in the Stockholm area. The initial temperature inside the tank was maintained through all tests as it was found to be negligible on the PCM phase-transition time and the completion time of charge and discharge, when the HTF inlet temperature and flow rate were maintained at the same level.

To know that the operating boundary conditions ($T_{f,in}$, $T_{TES,ini}$ and $\dot{V}_f$) were accordingly with the test matrix, it was used temperature and flow rate measuring devices, shown in Figures 4.4 and 4.9. Also, the thermal stratification was verified inside the tank by measuring the temperature of the HTF and PCM on different vertical positions. As shown in Figure 4.4, RTDs were inserted inside 5 PCMs capsules at different height positions in the tank. This served as a way to verify the temperature inside the capsule, making sure that the PCM was fully melted or solidified, depending if the test was for charging or discharging the TES tank.

The HTF and PCM temperatures in the storage tank were kept uniformly at $T_{TES,ini}$ prior to the the start of each test, and $T_{f,in}$ and $\dot{V}_f$ were held at a specified value throughout each test. For instance, if the initial tank temperature ($T_{f,in}$) deviated from the intended value, the rig was operated until the desired temperature was attained, with the valves that lead to the PCM tank

closed. The temperature of the HTF that was already inside the PCM tank couldn't be heated up because otherwise the charging would start from that moment. For this matter, the whole system was flushed before the charging started. The procedure of the testing is shown further in this document.

The absolute temperature difference between $T_{TES,ini}$ and $T_{f,in}$ for each whole charging/dis-charging process is used to compute the operating temperature range (ΔT). The phase-change temperature range established by the T-history test [81] is covered by every measured operational temperature range.

On both charge and discharge tests, the measurements to record were the water tank temperature, the HTF temperatures inside of the tank, the inlet and outlet HTF temperatures, the PCM core temperatures, the tank wall temperature, the insulation temperature and the flow rate.

Table 4.4: Charging Tests Matrix

Test ID	$T_{f,in}$ [°C]	$T_{TES,ini}$ [°C]	ΔT [°C]	$\dot{V}_f$ [m ³ /h]
CH-1	62	25	37	0.5
CH-2	62	25	37	0.6
CH-3	62	25	37	0.75
CH-4	65	25	40	0.25
CH-5	65	25	40	0.35
CH-6	65	25	40	0.50
CH-7	65	25	40	0.6
CH-8	70	25	45	0.25
CH-9	70	25	45	0.35

The test procedure to simulate the charging process involves a series of carefully executed steps. Initially, all valves within the system are closed to isolate the components. Next, the 9 kW electric heater and the circulation pump “CP1” are activated to initiate the heating process. The system then awaits the heat transfer fluid (HTF) to reach the desired inlet temperature ($T_{f,in}$). Once this temperature is achieved, the initial tank temperature ($T_{TES,ini}$) is set to the predetermined value. Subsequently, valves “V3” and “V5” are opened to allow fluid flow, followed by the opening of valve “V1” to complete the circuit. The main pump is then turned on, operating at a pre-adjusted flow rate, while the 3 kW electric heater is activated, and its thermostat is fine-tuned to maintain the desired inlet temperature ($T_{f,in}$). The process continues until a quasi-steady state is attained, characterized by minimal temperature variation over a 30-minute period, with this stable data later truncated in Excel for analysis. Finally, the main pump is turned off, followed by the deactivation of the 9 kW electric heater and circulation pump “CP1”, and the closure of valves “V5”, “V3”, and “V1” to conclude the procedure.

Table 4.5: Discharging Tests Matrix

Test ID	$T_{f,in}$ [°C]	$T_{TES,ini}$ [°C]	ΔT^a [°C]	$\dot{V}_f$ [m ³ /h]
DIS-1	25-30	65	37.5	0.15
DIS-2	25-30	65	37.5	0.25
DIS-3	25-30	65	37.5	0.35
DIS-4	25-30	65	37.5	0.50
DIS-5	30-35	65	32.5	0.15
DIS-6	30-35	65	32.5	0.25
DIS-7	30-35	65	32.5	0.35
DIS-8	30-35	65	32.5	0.50
DIS-9	30-35	65	32.5	0.60

^a The average $T_{TES,ini}$ was used to determine the temperature difference.

The test procedure to simulate the discharging process involves a series of meticulously executed steps. Initially, the initial tank temperature ($T_{TES,ini}$) is set to the desired value to establish the starting condition. The heat transfer fluid (HTF) tank is then partially drained to prepare for the subsequent phase, followed by refilling until the fluid reaches the target inlet temperature ($T_{f,in}$). Next, the District Cooling Network valves are opened, and the circulation pump “CP2” is activated to initiate fluid circulation. Subsequently, valves “V2” and “V4” are opened to facilitate flow through the system, followed by the opening of valve “V1” to complete the circuit. The main pump is then turned on, operating at a pre-adjusted flow rate, and the process continues until a quasi-steady state is achieved, marked by minimal temperature variation over a 30-minute period, with this stable data later truncated in Excel for analysis. Finally, the main pump is turned off, followed by the deactivation of the circulation pump “CP2” and the closure of the District Cooling Network valves, and the shutting of valves “V1”, “V2”, and “V4” to conclude the procedure.

4.4 Uncertainty Analysis

The errors in the inlet and outlet temperature measurements (± 0.05 K) and HTF flow rate ($\pm 0.5\%$ of reading) contribute to the inaccuracies in derived parameters, including the thermal power (Q_{net}).

Over time, these uncertainties in thermal power accumulate in the computed storage capacity through integration. The total uncertainties in the instantaneous thermal power are evaluated using Equation 4.1.

$$U_{Q_{net,i}} = \sqrt{\left(\frac{\partial Q_{net}}{\partial T_{f,in}} U_{T_{f,in}}\right)^2 + \left(\frac{\partial Q_{net}}{\partial T_{f,out}} U_{T_{f,out}}\right)^2 + \left(\frac{\partial Q_{net}}{\partial \dot{V}_f} U_{\dot{V}_f}\right)^2} \quad (4.1)$$

The aggregated uncertainties in the determined $Q_{net,i}$ are denoted by U_{qTES} , which is computed by analyzing the measurement uncertainties. These consider $U_{T_{f,in}}$ for the inlet temperature, $U_{T_{f,out}}$ for the outlet temperature, and $U_{\dot{V}_f}$ for the flow rate. This method ensures a thorough consideration of all possible error sources during measurement, delivering a detailed evaluation of the total uncertainty in the calculated thermal power.

The total uncertainty on the estimated capacity is given by Equation 4.2.

$$U_{QTES} = \sqrt{\sum_{i=1}^n (Q_{net,i})^2} \quad (4.2)$$

4.5 Charging and Discharging Strategies

The purpose of the experiments is to simulate the "Live-in Lab" application by creating similar heating demand conditions and also equipments to mimic the real situation [87]. This also involves creating different discharging scenarios, for example having a partial support where only a percentage of the total demand is shifted to the tank before the peak hits, while having the heat pump working for the rest of the heat demand (Case 0); full support of one peak per day where the tank is fully discharged during only one of the peaks, allowing the HP to be turned off during the high heat demand time intervals (Case 1); alternating the daily peak coverage which, in certain cases can be a good solution for when the charging time is bigger than the time span between peaks (Case 2); dual-peak partial coverage, where half of the capacity supports partially each peak (Case 3); full coverage on both peaks, which is feasible when the charging time is smaller than the time span between peaks and the tank capacity allows to sustain all the house heating demand (Case 4).

These strategies aim to maximize the efficiency and flexibility of the LHTES-HP system by adapting the operation to varying boundary conditions such as electricity price fluctuations, peak load patterns, and available charging times. The purpose is not only to reduce peak electricity consumption but also to enhance the system's cost-effectiveness and energy self-sufficiency. To explain this further, the heat demand from the student housing building was monitored for the last year to, first dimension the storage tank and second, to prepare charging and discharging strategies before the implementation.

Figure 4.11 shows a situation where Case 0 would suit, but in different scenarios. In both days, the heat demand remains constant for the whole day which allows the electricity prices to determine when the charging and discharging should occur. As it peaks during small time intervals, this would be ideal to discharge or charge during long periods of time to allow the heat pump to work at a low power. The curve that relates to the 27th of February has a lower heat demand than the 6th of January curve. This fact would slightly change the operation of the system LHTES tank + heat pump. In the first case, the tank could operate for the most amount of hours, if the heat pump worked at the same power on both scenarios. The second case is the perfect example for when both should operate at the same time, having a power distribution such as the tank delivering half of the power for 6 hours and then the heat pump covering the other half during this same

period of time. The heat demand during the rest of the day could be covered by the heat pump. Case 0 would fit any scenario where the heat demand can't be predicted, due to its versatility, although predicting the electricity prices would be essential for its economical efficiency.

Case 1 would suit a scenario where either there's one or two peaks during the day. This would flat the heat production curve, by shifting the heat production to before the peak. This way, the heat pump could cover the rest of the day with ease. Figures 4.12 and 4.13 would both be good situations where applying this strategy could improve the overall heat production curve, by flattening it.

Case 2, although similar to Case 1, has some practical differences. This strategy only fits cases where the time span between the second peak of the day and the first peak of the next day isn't long enough to charge the whole tank. This would vary with how fast it's possible to charge the tank, but Figure 4.13 shows a possible case - 13th of February.

Case 3 is ideal for when there are two peaks along the day with similar heat demand and duration, or that end up taking the same capacity of the tank. This would allow the heat pump to deliver the rest of the power during the peaks and the heat demand during the rest of the day. Case 4, the ideal case, would fit a scenario where there are two peaks during the day and the charge can be made in less time than the time span between those two peaks. Case 4 is likely to achieve the highest storage utilization and electricity cost savings, as the full tank capacity is used strategically during both peaks while charging occurs during low-price hours. Figure 4.13 shows a scenario where these last two cases would fit.

In practice, the optimal strategy must be selected dynamically, based on real-time forecasts of both electricity prices and building heat demand. A control algorithm or model predictive control (MPC) strategy may be required to adapt the charging and discharging patterns accordingly. Flexibility in the heat pump's modulation and accurate measurements from the RTDs inside the tank will play a crucial role in implementing these strategies effectively.

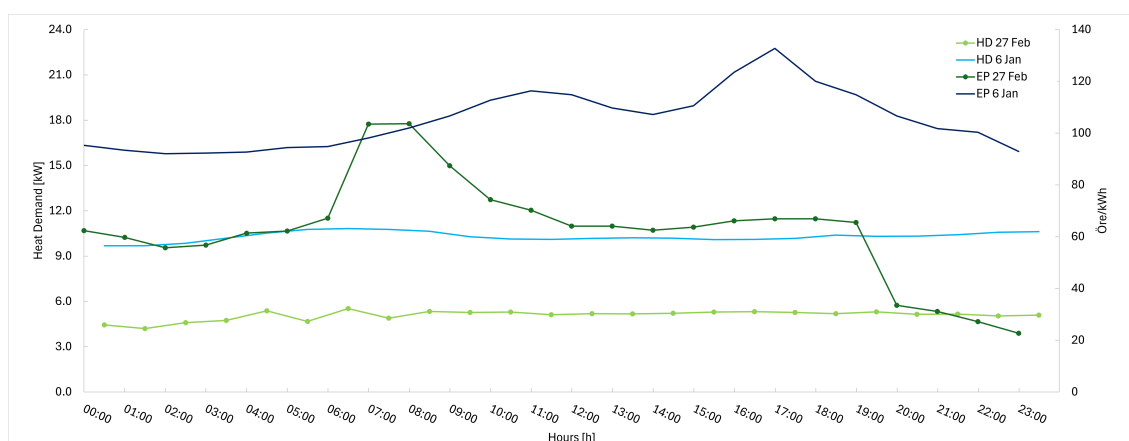


Figure 4.11: Heat demand along the day compared to electricity price - Constant heat demand scenario [12].

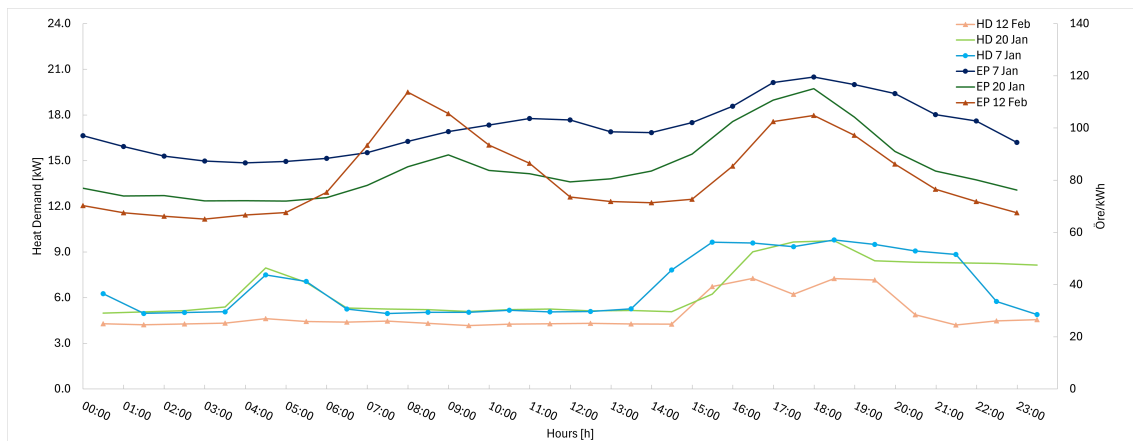


Figure 4.12: Heat demand along the day compared to electricity price - Single peak scenario [12].

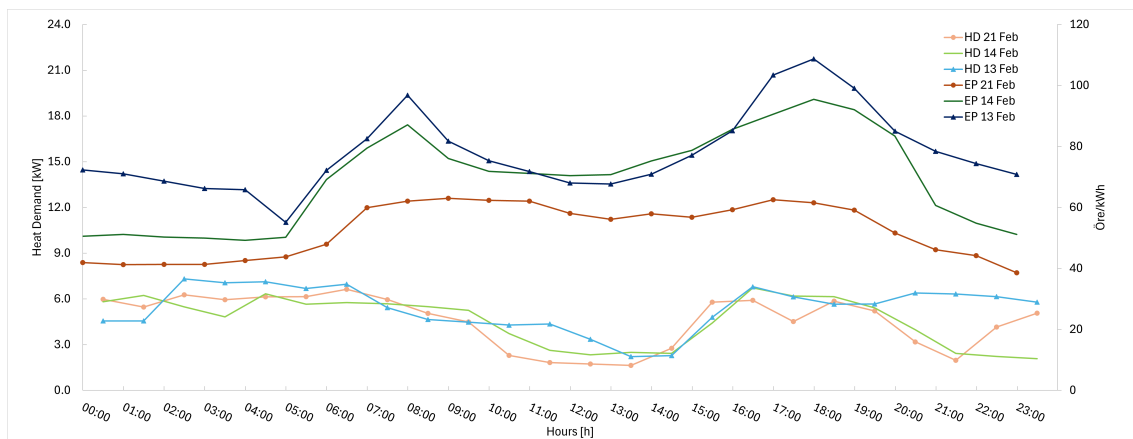


Figure 4.13: Heat demand along the day compared to electricity price - Two peaks scenario [12].

Chapter 5

Results

This chapter presents the outcomes of the experimental investigation into the performance of the latent heat thermal energy storage system integrated with a heat pump, building upon the design and optimization efforts outlined in previous sections. The analysis focuses on evaluating the system's behavior under various operating conditions.

The data from this research will be a foundation to future applications on control and system development.

5.1 HTF Charging Behavior

The analysis of the HTF charging behavior is crucial for determining the optimal conditions under which the PCM capsules are charged effectively. This evaluation provides insights into the thermal environment surrounding the capsules, which was meticulously monitored during every test. Such monitoring enabled the identification of the precise moment when the HTF ceased transferring heat to the PCM, marking the transition from active charging to a stabilization phase.

The results also allowed to verify the stratification that was created during the whole latent heat charging and near the end of the test. They're shown in Figure 5.1.

The low void fraction within the TES tank plays a significant role in influencing this stratification. The compact geometry of the capsules allowed for a high density of units to be packed into the tank, reducing the unoccupied space and thereby limiting the convective mixing of the HTF.

From the graphs, it's noticeable that the higher the flow rate, the faster the initial ramp-up behavior disappears. Also, the stratification is more evident and, takes more time in the graph, on the smaller flow rate scenarios, which inevitably makes the charging time higher, as the HTF around the capsules on lower positions has a lower temperature than the upper capsules.

Although higher flow rates might result in smaller charging times, this affirmation holds true only if the $T_{f,in}$ is the same, for comparison. When the HTF inlet temperature varies, there's another criteria of influence to look at, as it impacts on the instantaneous heat that's being transferred. So, firstly the tests will be analyzed in groups of the same $T_{f,in}$ and different $\dot{V}_f$ and after, particular comparisons between the same $\dot{V}_f$ and different $T_{f,in}$.

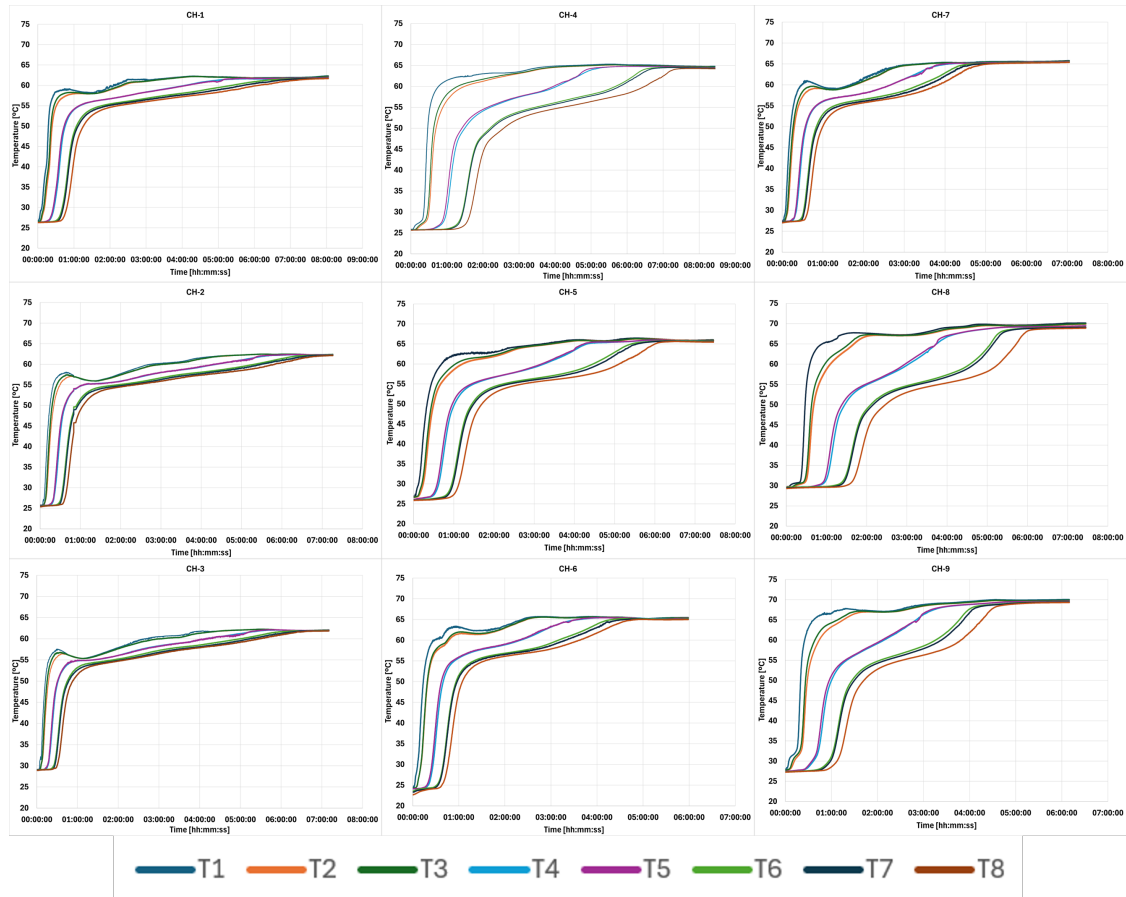


Figure 5.1: HTF Charging Behavior.

CH-1 through CH-3, have the lowest HTF inlet temperature on the matrix, it's visible on the three graphs that all the temperatures follow the same profile on similar times. This has to do essentially with the fact that the temperature is similar to the phase change temperature of the PCM, leading to a lower sensible heat increase after the latent heat transfer, which allows for the "next height" of capsules to be heated. This phenomena doesn't appear on the next tests due to the fact that they have more sensible heat storage after the phase change. CH-1, with the lower flow rate, has a smoother transition from the initial ramp-up to the decline from the latent heat transfer. CH-2 and CH-3 have similar profiles, which will be discussed further with the rest of the data provided.

CH-4 through CH-7 were the main tests due to the chosen $T_{f,in}$, which is the future design aim temperature. CH-4 and CH-5 have flow rates below $0.5\text{m}^3/\text{h}$, so they were considered tests with low flow rates. This lead to a very smooth transition after the ramp-up temperature increase until the stabilization, which allowed to understand with precision when were the different heights on the tank being charged. Both the tests have similar profiles. CH-6 profile is a combination between these last two and CH-7, as it has a smoother curve than CH-7 but with a faster increase of the temperature in the beginning when compared with CH-4 or CH-5. The stratification on CH-6 and CH-7 is less noticeable. All these tests differentiate themselves from the ones that have the lower

$T_{f,in}$ in the phase transition temperature interval. It's visible that the temperature in this phase of the charging process increases with a lower slope due to the charging of the capsules.

Lastly, CH-8 and CH-9, the tests with the highest $T_{f,in}$ presented very similar response from tests CH-4 and CH-5, respectively. The cause can be the fact that they've the same flow rate and also, the temperatures on both scenarios are much higher than the phase transition temperature which leads to a diminished influence of the latent heat contribution. As the inlet temperature exceeds the PCM's melting point of around 57°C significantly (reaching 70°C), the heat transfer process becomes dominated by sensible heat exchange, reducing the relative impact of the phase change on the overall thermal response.

5.2 PCM Charging Behavior

The PCM charging behavior was evaluated through a series of tests designed to assess heat absorption efficiency under varying inlet temperatures and flow rates and also, specific phenomena like crystallization and the solid-liquid phase change. For this, different HTF flow rates and inlet temperatures were used and the values of T_{PCM3} and T_5 were extracted. These values were chosen because they represented the temperatures on the capsule and on the HTF in the middle of the tank, which could be considered an average of the whole system, simplifying the analysis.

Figures 5.2, 5.3 and 5.4 illustrate the temperature profiles within the PCM capsules during the charging process, showing a clear change in the slope when the temperature of the HTF around the capsule reaches the crystallization point and after, the solid-liquid phase change temperature. The crystallization point represents a solid-solid phase change, storing heat during this process. Then, the solid-liquid phase change is well noticed on the graph as it starts after the crystallization and ends before a different slope comes in, representing the sensible heat.

At 62°C, the flow rate doesn't impact much on the time that T_{PCM3} takes to reach the latent heat region, but at the other two temperatures, there's a significant difference on the impact of the flow rates. As seen before with the HTF, also the PCM capsules have an ramp-up in the beginning and then, the profile is very similar with the HTF, being more evident on the higher flow rates. This happens because there's not much hysteresis on the heat transfer between the HTF and the PCM capsule, which brings the different temperatures closer on the graph.

There's an evident difference on the sensible and latent heat charging. The slope of the temperatures of the PCMs is greater on the sensible region, which explains that it's harder to charge the latent heat on the capsules, also making it harder to lose afterwards. This is also supported by the fact that the existing HTF is being replaced by the HTF with the charging temperature.

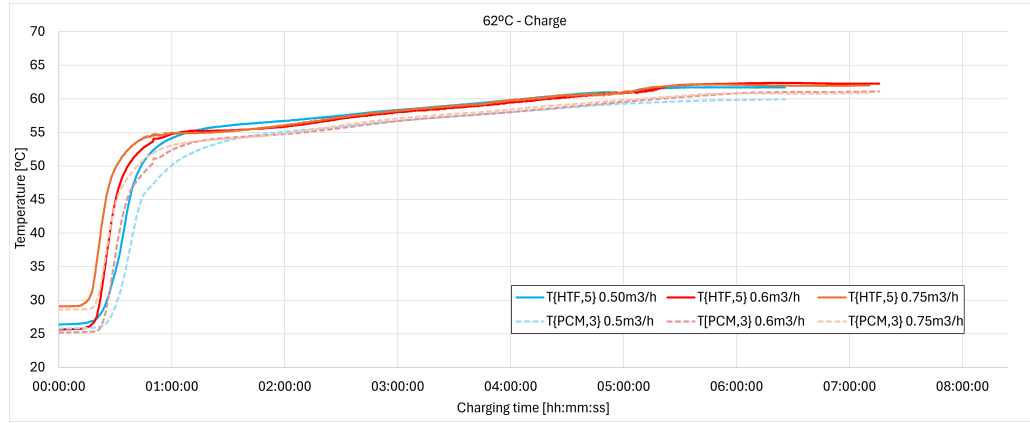


Figure 5.2: PCM Charging Behavior with $T_{f,in}$ at 62°C.

In Figure 5.3, it's visible that the two higher flow rates detain a very similar profile both on the HTF and PCM temperatures. The moment the HTF hits the phase change temperature, the PCM and HTF temperature lines go up with almost the same slope. The two lowest flow rates aren't as similar but after the PCM phase change they follow closer with each other until the end of the test.

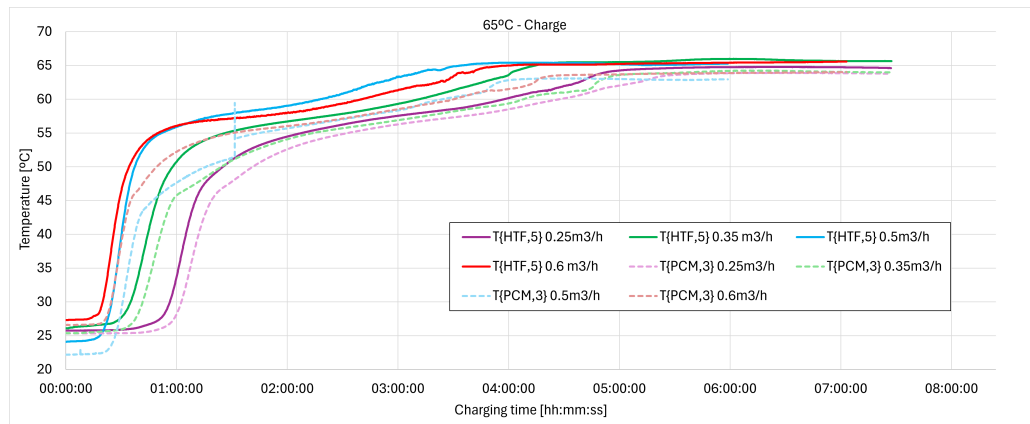


Figure 5.3: PCM Charging Behavior with $T_{f,in}$ at 65°C.

In Figure 5.4, it's noticeable the impact of the smaller flow rates on the time where the temperature increase starts. On the case of 0.25m³/h, it only starts ramping-up at approximately 1 hour after the start of the test. As there is greater discrepancy between these two tests, it's also visible the existing difference in both temperatures, which can be explained by the resistance of heat transfer by the HTF, the capsule material and the PCM itself.

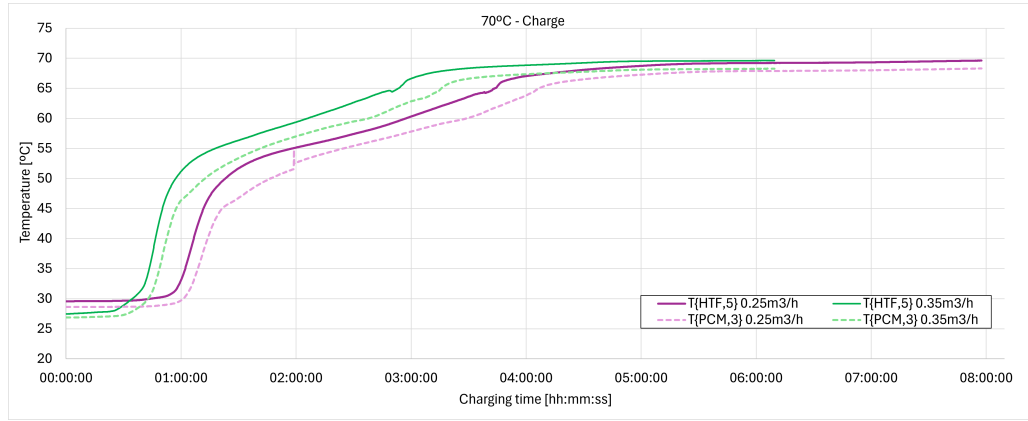


Figure 5.4: PCM Charging Behavior with $T_{f,in}$ at 70°C.

5.2.1 UA values during melting

The average global heat transfer coefficient between the HTF and the PCM during the phase change was determined for different flow rates and temperatures. For this calculation, the phase change range determined experimentally in the in-house labs was used [81].

The UA value, which represents the product of the overall heat transfer coefficient and the effective heat transfer area, quantifies the system's thermal performance and is crucial for optimizing energy transfer efficiency in LHTES applications, and it was determined using the NTU method presented in Chapter 3. To temperatures used in the equation 3.20 where the $T_{HTF,4}$ and $T_{PCM,3}$ as they were in the middle of the tank and at the same height. As explained in Chapter 4, the RTD at $T_{HTF,4}$ measures the temperature at the center of the tank's radius, a location unaffected by the temperature of the PCM capsule, unlike $T_{HTF,5}$. This characteristic justified its selection over the RTD positioned closer to the capsule.

Figure 5.5 show the results from these calculations, performed for all charge scenarios.

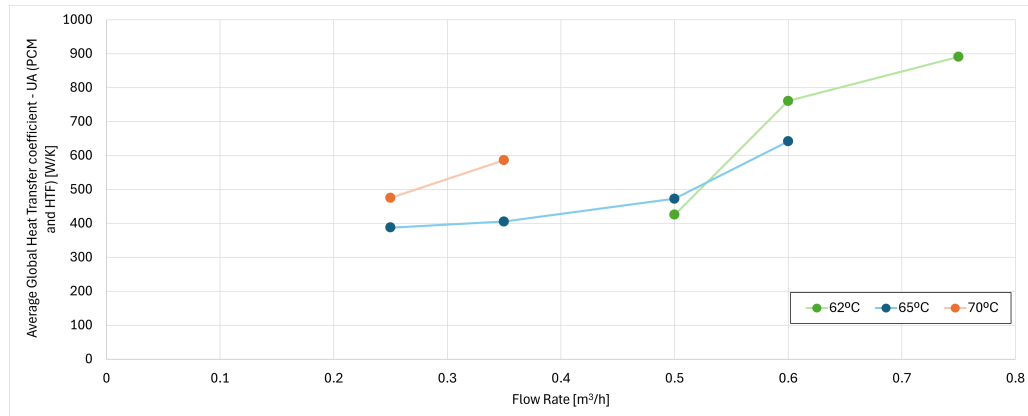


Figure 5.5: UA values for different BCs at charge tests.

In all instances, an increase in the average global heat transfer coefficient is observed as the flow rate escalates. This specific phase of the charging process, which is dedicated to supplying

latent heat to the PCM capsules, exhibits a behavior that aligns with predictable thermodynamic principles. The enhanced heat transfer can be attributed to the improved convective exchange between the HTF and the PCM, driven by higher flow velocities, which facilitate more efficient energy distribution across the capsule surfaces. This trend is consistent with established models of latent heat storage systems, where increased fluid dynamics typically amplify the rate of phase change, thereby optimizing the thermal performance during this critical stage.

5.3 Analysis of SoC at 50%, 75%, and 90% and Total Capacity

The state of charge of the experiments was analyzed and at 50%, 75% and 90%, these values were extracted to be compared among them and ahead, with previous experiments. Also, to create a reference value to be able to compare equally all tests, the theoretical capacity with different boundary conditions was determined. These values can be seen in Chapter 3.

Figures 5.6, 5.7 and 5.8 show the values of partial and full charged capacity for different $T_{f,in}$ and $\dot{V}_f$.

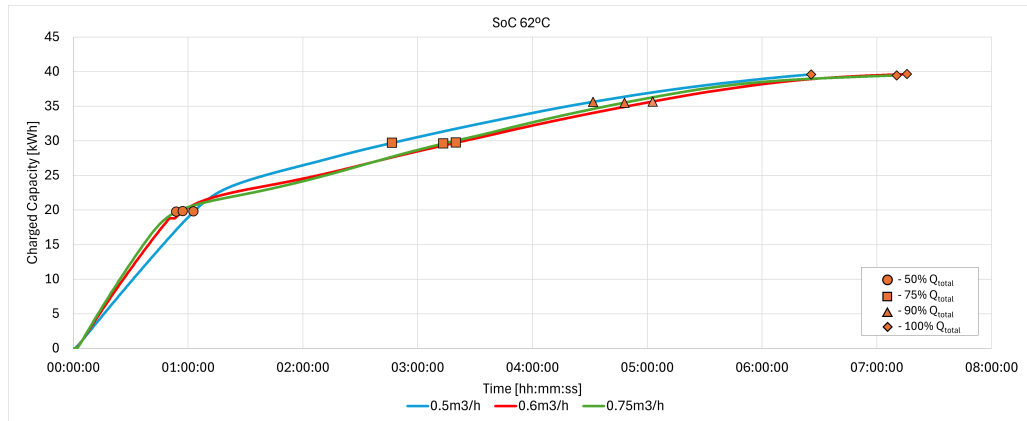


Figure 5.6: SoC for different flow rates with $T_{f,in}$ at 62°C.

In Figure 5.6, all three profiles follow the same behavior, despite having different flow rates. This has to do with the fact that with a lower temperature, there's a lower temperature difference and so, also a smaller heat exchange. This tests were the ones with the higher flow rates due to their low temperature, but in the end they came up very similar.

As expected, the total capacity on all three is very similar, as well as the partial charges. With such flow rates, the tank was 50% charged with only one hour after starting. It needed two more hours to charge up to 75% and another two to 90%, which is past the latent heat charge on all capsules.

The sudden change of slope in the profiles is due to the start of the charging of the capsules that create a bigger resistance to the heat transfer than the sensible heat. Later on the test, the curves stabilize, reaching almost a zero slope behavior, which is when the temperature difference is smaller.

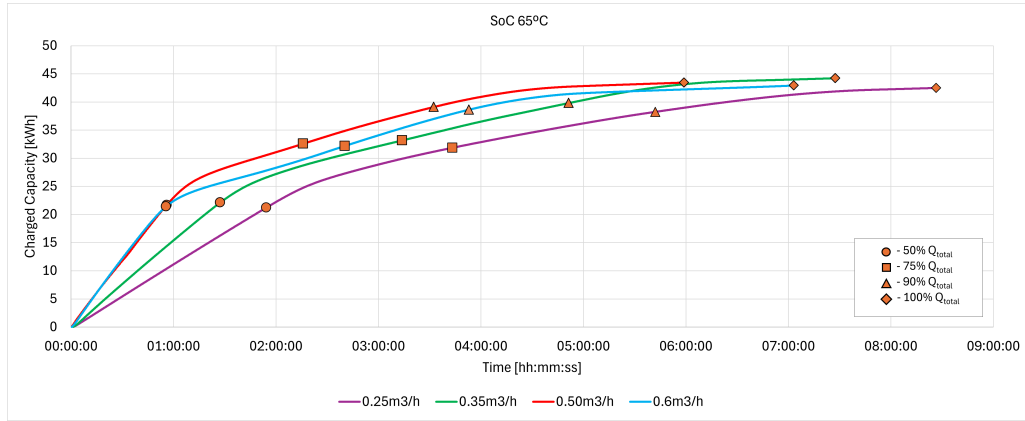


Figure 5.7: SoC for different flow rates with $T_{f,in}$ at 65°C.

In these cases, the behavior is more scattered than in the previous analysis.

The two higher flow rates follow a similar profile until the first hour of charge and then split until the end of the test. They're able to charge 50% of the tank in this first hour. The test with 0.5 m³/h HTF flow rate later charges faster than the test with 0.6 m³/h. It reaches faster the 75%, 90% and the full charge than the highest flow rate. This proves that increasing the flow rate without analyzing the charged capacity along the test might lead to lost pumping power, with no increase in heat transfer efficiency.

The two lower flow rates behave as expected, with the lower one being the slower at partially and fully charging the tank.

As seen in the 62°C $T_{f,in}$ case, there's also a change in the slope of the charged capacity. In this case, the slope is steeper, which is due to the fact that this test has a higher $T_{f,in}$, increase the heat transfer rate. The latent heat is being delivered faster and gradually, obeying the stratification effect.

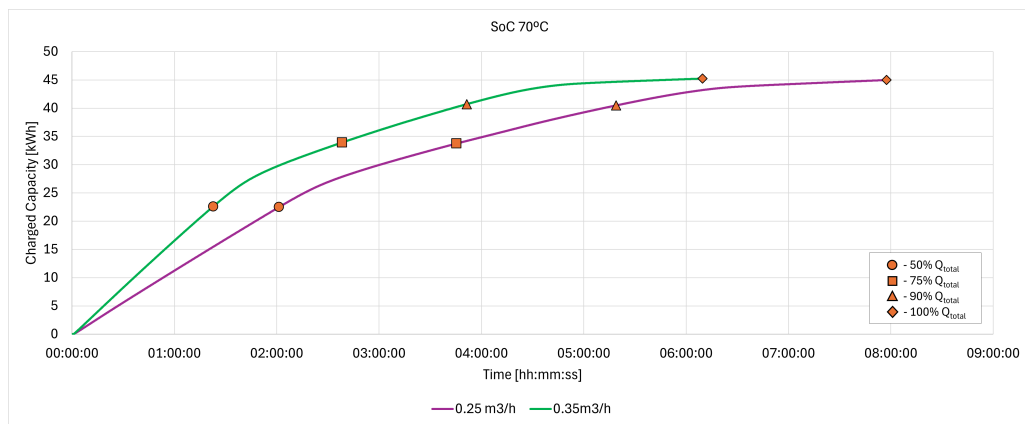


Figure 5.8: SoC for different flow rates with $T_{f,in}$ at 70°C.

Lastly, Figure 5.8 shows a similar pattern as the two previous analysis. There's also a change on the slope explained by the start and end of the latent heat charge. Here, the lower HTF flow rate takes more time to partially and fully charge the tank.

As expected, the tests with the same $T_{f,in}$ have all similar total capacities as they were exposed to the same temperature increase.

Table 5.1 resumes the data extracted from the tests referring to the capacity of the tank at different boundary conditions. KPIs such as energy density and the time to charge 90% and 100% are also presented. Also, β represents the relative difference in heat storage capacity between the TES system (Q_{TES}) and a water tank that stores SHTES, normalized by the heat stored in the water tank.

Table 5.1: Performance Metrics for Charging Tests

Test ID	Experimental Capacity ^a [kWh]	Theoretical Capacity ^b [kWh]	ED_{TES}^c [kWh/m ³]	β [%]	Time 90% ^a [hh:mm:ss]	Time 100% ^a [hh:mm:ss]
CH-1	39.60 ± 6.12	47.12	65.24	51.72	04:31:42	06:25:42
CH-2	39.65 ± 6.49	47.12	65.32	51.92	05:02:53	07:15:44
CH-3	39.47 ± 8.87	47.12	65.02	51.23	04:48:05	07:10:26
CH-4	42.50 ± 4.76	50.71	70.02	50.71	05:42:04	08:26:25
CH-5	44.23 ± 5.86	50.71	72.87	56.84	04:51:20	07:27:24
CH-6	43.45 ± 5.93	50.71	71.58	54.08	03:32:14	05:58:48
CH-7	42.92 ± 5.72	50.71	70.71	52.20	03:52:51	07:03:02
CH-8	44.99 ± 4.96	55.31	74.19	41.48	05:18:55	07:57:30
CH-9	45.25 ± 6.02	55.31	74.55	42.30	03:51:27	06:09:35

^a Experimental capacity (Q_{TES}) and time values are derived from test data.

^b Theoretical capacity assumes ideal conditions.

^c ED calculated based on tank volume of 0.607m³, considering the distribution regions and for experimental capacity.

^d β allows to compare with a SHTES tank with water as the storing material and with the same volume, considering the same temperature increase.

5.4 HTF Discharging Behavior

The analysis conducted for the charging tests was also extended to the discharging tests, which are particularly critical for refining boundary conditions to support control strategies and machine learning data collection.

The HTF discharging behavior was monitored using eight RTDs positioned within the tank at five different heights and two distinct radial locations, with five sensors placed at the tank's center and three at the mid-radius position. This allowed for a better visualization on the stratification during the test and the impact of the discharge of the capsules at different heights.

Figure 5.9 shows the results for the nine discharge tests performed.

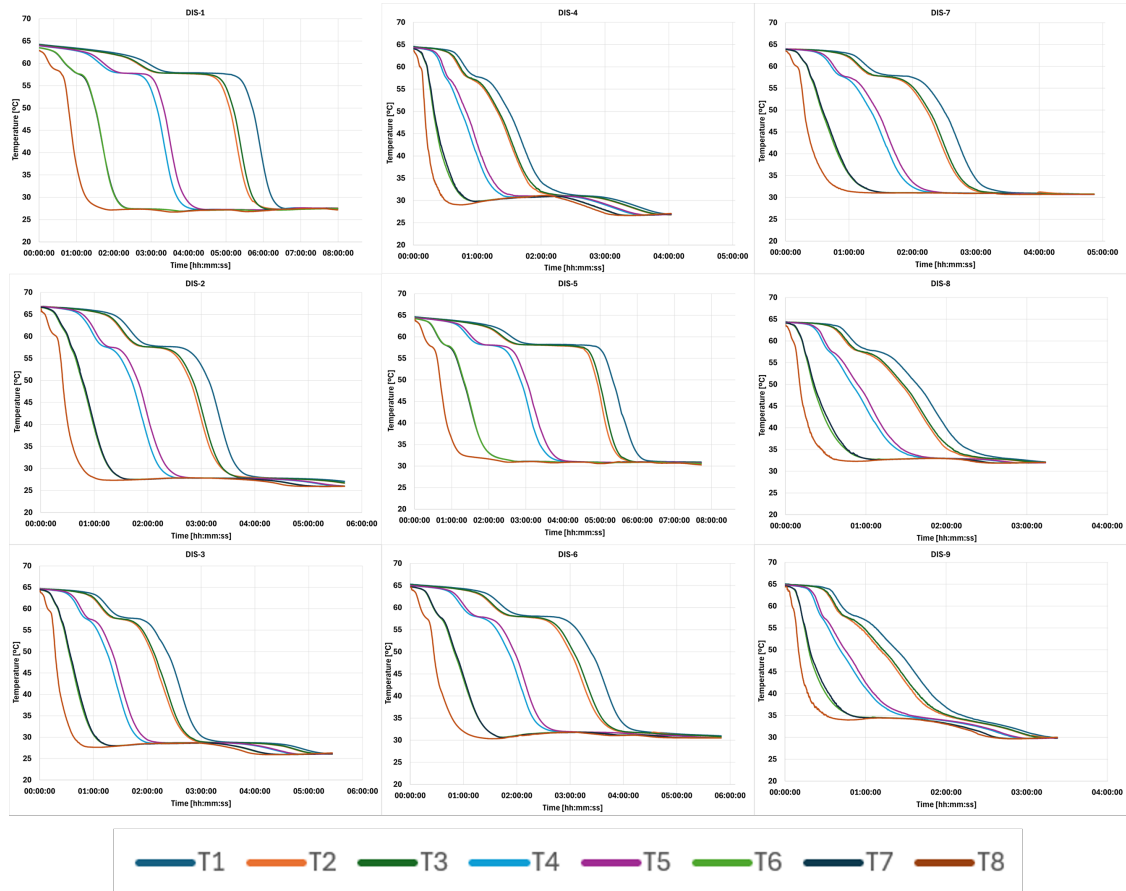


Figure 5.9: HTF Discharging Behavior.

The first four discharge tests were performed with an $T_{f,in}$ stabilized between 25 and 30°C and the other five, between 30 and 35°C.

Through DIS-1 to DIS-4, the flow rates of the HTF go from 0.15 to 0.5m³/h. On the first image in Figure 5.9, DIS-1 shows a severe stratification since the beginning of the test, explained by the small flow rate. The aim on the smaller flow rates for the discharge scenarios is to extend the power delivery along the time, exchanging smaller amounts of energy with the PCM. This, will create a steadier output temperature with a short amount of power to support the demand. One of the phenomena that's transparent in this test is the PCM solid-liquid phase change. As seen in the figure, all the temperature profiles have a steady region at the melting temperature range.

As the flow rate increases, this stratification gets less visible due to the quicker heat exchange between the PCM capsules and the moving HTF. The steady temperature regions also get smaller with the increase of the flow rate, which can impact the time that there's a constant outlet temperature. The fact that there are multiple capsules per height allows the tank to output this constant temperature for a longer period of time, as the capsules get discharged from the bottom to the top. For this matter, a bigger stratification is ideal for longer periods of discharge with smaller power deliveries, while a faster discharge allows for a bigger power output.

It's also noticeable, on DIS-2, DIS-3 and DIS-4 that the steady regions at the top of the tank

don't follow the same profile as the ones at the bottom where the discharge starts. This can be explained by the fact that when the HTF reaches the top, already warmer, it combines both the heat that is extracting from the upper capsules with the heat coming from the bottom capsules. On smaller flow rates, this doesn't happen, creating a constant profile between all heights.

Through DIS-5 to DIS-9, the flow rates go from 0.15 to 0.6 m³/h. These experiments, when compared with the same flow rate ones from the previous tests, have very similar profiles despite having a different $T_{f,in}$ range. The main difference is the total time of discharge, as it can output all the power retained faster than the first four tests.

5.5 PCM Discharging Behavior

In the context of discharge tests, the assessment of the PCM discharging behavior, coupled with the output temperature, represents the most critical analysis. This evaluation elucidates the duration during which specific test conditions can maintain a stable temperature output and deliver a consistent power output, providing key insights into the system's operational reliability and energy delivery capacity.

The analysis was divided in two, the 25 to 30°C and the 30 to 35°C range. Figure 5.10 shows the first four tests and Figure 5.11 shows the other five. For these analysis, $T_{PCM,3}$ was chosen due to its positioning in the middle of the tank, allowing for a good overview of the functioning of the capsules during the discharge. Along with it, the $T_{f,out}$ is also the aim of the analysis as they're correlated.

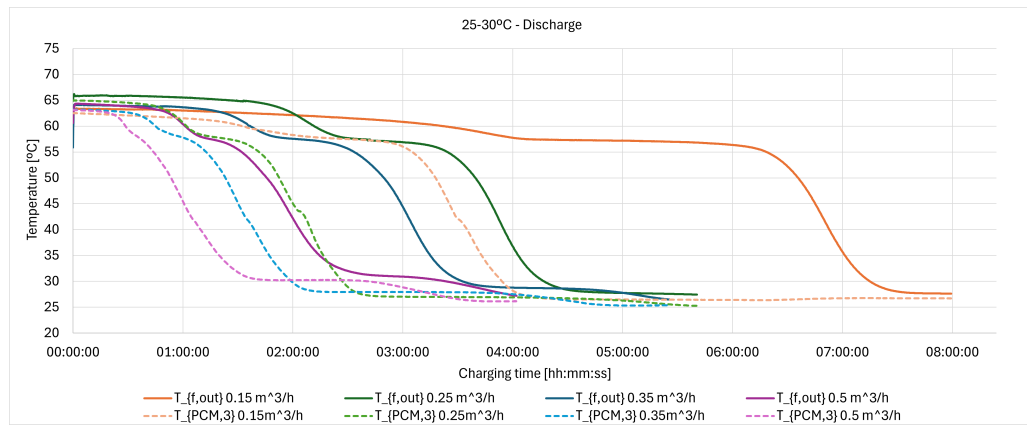


Figure 5.10: PCM discharging behavior for the 25 to 30°C $T_{f,in}$.

In Figure 5.10 it is noticeable that the slower the HTF flow rate, the longer the tank can deliver a close to constant output temperature. The stored sensible heat above the melting range is responsible for the initial constant temperature profile in the discharge. This sensible heat is not only on the HTF inside the tank but also in the PCM capsule and the material itself.

After, along with the PCM profile, the output stabilizes again but now at the mentioned melting range, extracting the latent heat stored inside the PCM capsules. It's visible that the output has a certain delay over the PCM curve which can be explained by the positioning of the RTD sensor

that measures $T_{PCM,3}$. It is anyway influenced by the discharge of the capsules in this region, as in any other. After the steady outlet, as soon as the latent heat gets extracted, there's a ramp-down temperature decrease. This can be explained by the fact that the material's conductivity increases with the solidification and the heat gets dissipated easier.

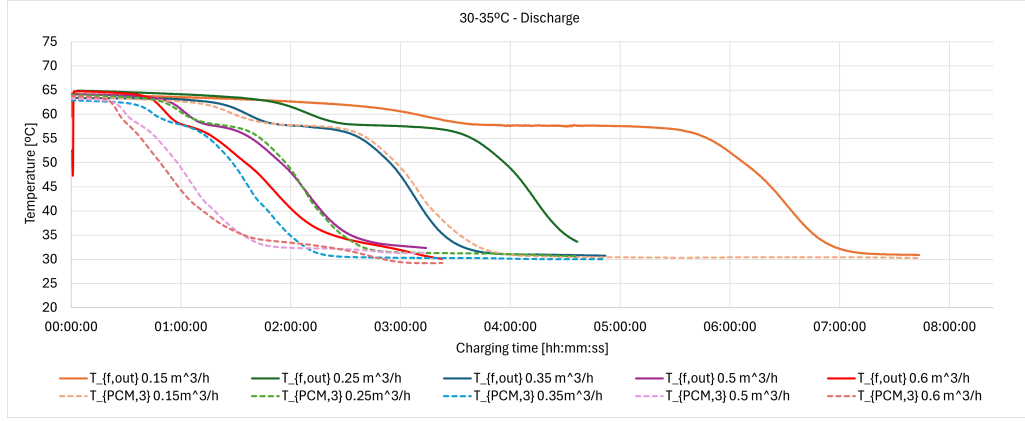


Figure 5.11: PCM discharging behavior for the 30 to 35°C $T_{f,in}$.

Figure 5.11 profiles come very close the ones shown previously in Figure 5.10. Despite of being similar, there's a difference that makes them quite distinguishable. It has to do with the time needed to release all the latent heat inside the PCM capsules. With the same flow rate, due to different temperature differences, which leads to different heat rates, there's a contrast on the prolongation of the profiles within the melting range.

5.5.1 UA values during solidification

The mean overall heat transfer coefficient between the HTF and the PCM during the phase transition was assessed across various flow rates and temperature settings for the discharge tests. This evaluation relied on the experimentally determined phase change range obtained in the in-house laboratory facilities [81].

The UA value, indicative of the product of the global heat transfer coefficient and the effective heat exchange area, serves as a key indicator of the system's thermal efficiency and plays a vital role in enhancing energy transfer performance in LHTES systems. This parameter was computed using the effectiveness-NTU method detailed in Chapter 3.

The outcomes of these computations are depicted in Figure 5.12, illustrating the UA values derived from all discharge test conditions.

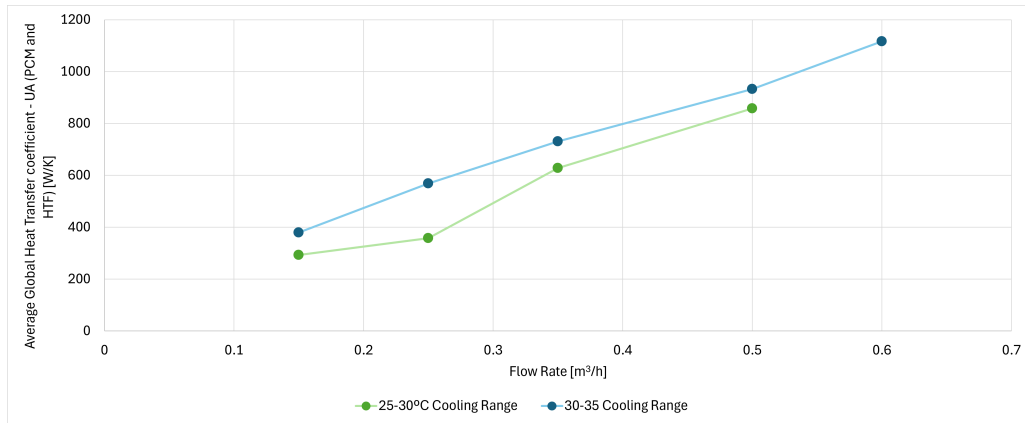


Figure 5.12: UA values for different BCs at discharge tests.

A noticeable increase in the average global heat transfer coefficient is observed as the flow rate rises during the discharge process, mirroring the behavior seen during the charging phase. Initially, in the 25 to 30°C cooling range, as the flow rate increases the increment on the average global heat transfer coefficient isn't as high as when the flow rate goes above 0.25m³/h.

On the 30 to 35°C cooling range data, it forms a function with an almost constant slope, which doesn't necessarily mean that increasing the flow rate would fasten the whole discharging process, as it only considers the phase-change process.

5.6 Analysis of SoD at 50%, 75%, and 90% and Total Power Delivery

To create a reference value to be able to compare equally all tests, the theoretical power output with different boundary conditions was determined. These values will be presented in this chapter to compare with the experimental values which include partial charges at 50, 75 and 90% as well as full charges.

Figures 5.13 and 5.14 show the values of partial and fully charged power output for different $T_{f,in}$ and $\dot{V}_f$.

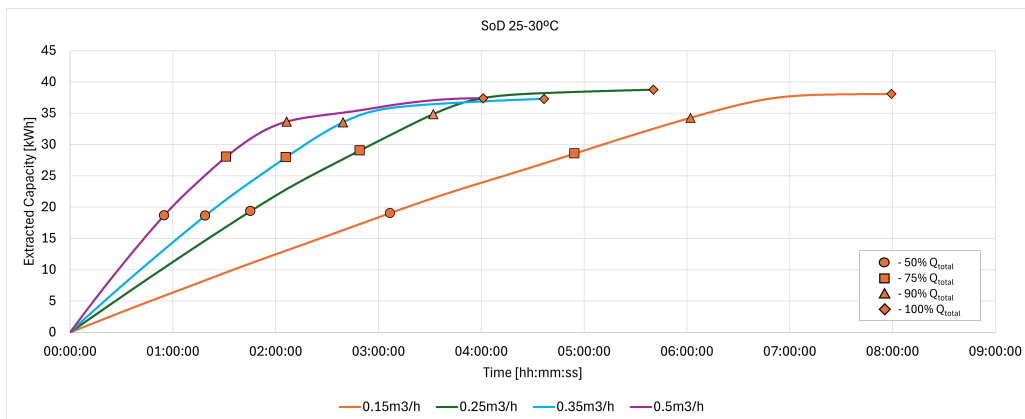


Figure 5.13: SoD for different flow rates with $T_{f,in}$ at 25 to 30°C.

As expected, the tests with the same temperature difference have similar total capacities. Also, the lower the HTF flow rates, the more the time to reach the mentioned partial discharge percentages. One of the analysis that is most important is for the 90% partial charge due to the fact that it's for sure past the latent heat discharge and doesn't take into consideration the time that it takes to reach to the full discharge after it. The profile that differs the most from the collection of data of Figure 5.13 is the $0.15\text{m}^3/\text{h}$ curve. It is the one that takes the most time to reach the selected points and in the previous analysis shown in Figure 5.10, it explains why the output temperature remains that much time constant.

The other three curves follow a similar profile but shifted in time.

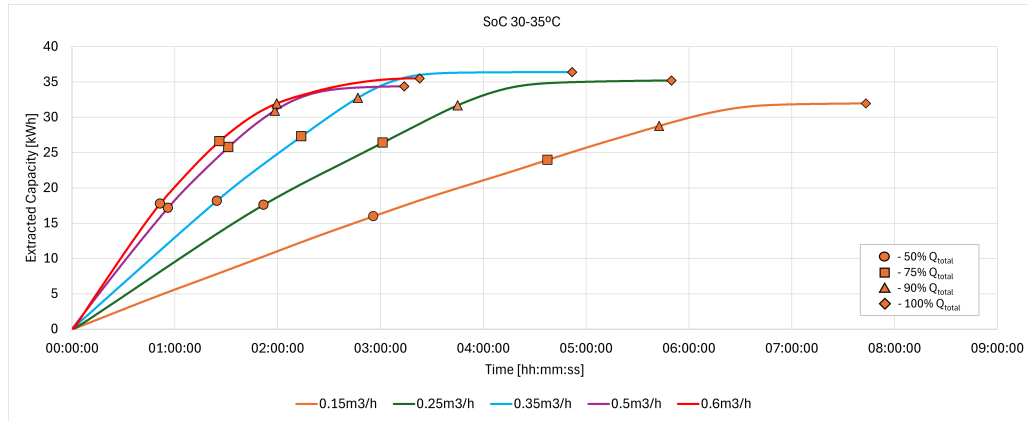


Figure 5.14: SoD for different flow rates with $T_{f,in}$ at 30 to 35°C .

In Figure 5.14 the profiles that stand the most are the two bigger flow rates and the smaller one. The curve that relates to the $0.15\text{m}^3/\text{h}$ test has the same behavior as the previous one from Figure 5.13. The curves that relate to the $0.5\text{m}^3/\text{h}$ and $0.6\text{m}^3/\text{h}$ tests almost intercept each other, which can lead to conclude that increasing the flow rate above $0.5\text{m}^3/\text{h}$ might be not worth it as it increases pumping power without necessarily increasing the power output.

Figures 5.15 and 5.16 show a different perspective from the SoD, allowing a deep analysis on the power that's being extracted in kW. These graphs allow also to understand for how long is the tank able to output a constant power for different BCs.

As mentioned earlier, the starting temperature of the tank was the same for all the discharge tests and so, the first group of experiments has the same operating temperature range among them, just as the second group of experiments. For this reason, the total discharge powers should be the same among the group of tests, although the following graphs have different profiles.

Figure 5.15 allows to understand deeper what was discussed when Figure 5.10 was analyzed. The profile of the output temperature and the discharge power have similar behaviors because they're correlated. The lower flow rate extends the constant power for a longer time than the other three tests. In contrast, the higher flow rate, has the highest peak of power extraction, also finishing the discharge the fastest.

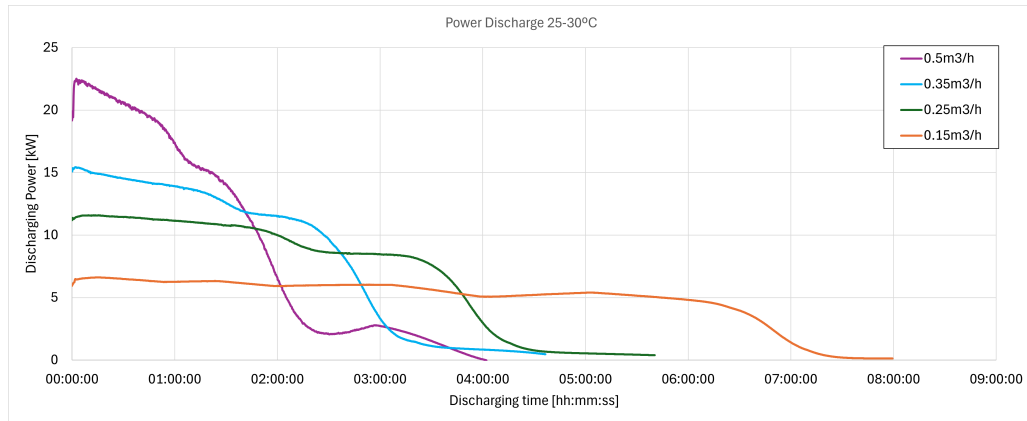


Figure 5.15: State of Discharge profiles for varying HTF flow rates across $T_{f,in}$ within the cooling range of 25 to 30°C.

The three lowest flow rates in Figure 5.16 behave similarly with the lowest from the previous group of tests, with some minor differences on the time to complete the discharge. The two highest HTF flow rates, 0.5m³/h and 0.6m³/h bring again their similarity. Although the highest flow rate has the bigger peak, it discharges in a higher amount of time. The two profiles meet approximately around the one hour of test, which is when they almost become a single profile, in the end the 0.5m³/h being a bit faster.

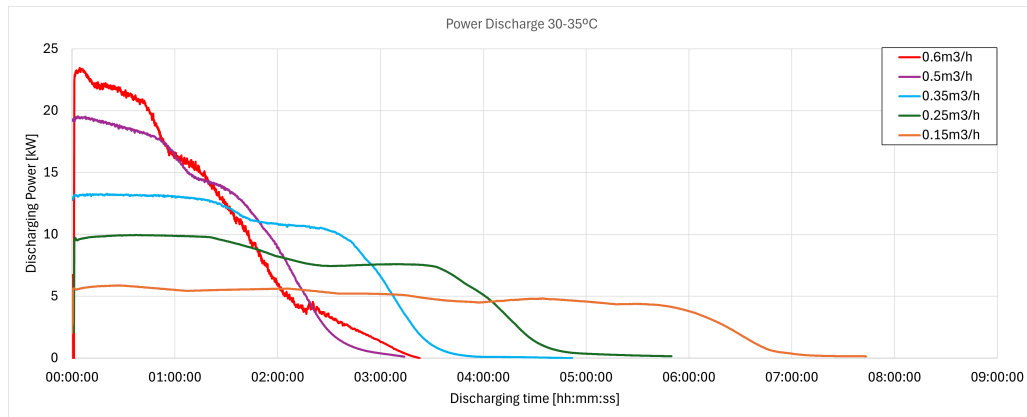


Figure 5.16: State of Discharge profiles for varying HTF flow rates across $T_{f,in}$ within the cooling range of 30 to 35°C.

Table 5.2 resumes the data extracted from the tests referring to the total power delivery of the tank, the discharge efficiency and the time it took to reach certain percentages of discharge, at different boundary conditions.

Table 5.2: Performance Metrics for Discharge Tests

Test ID	Total Power Delivery ^a [kW]	Theoretical Delivery ^b [kW]	η^c [%]	Power Delivered in ^a [kW]			Time 90% ^a [hh:mm:ss]	Time 100% ^a [hh:mm:ss]
				2h	3h	4h		
DIS-1	38.09 ± 3.55	41.07	88.02	12.47	18.37	23.91	06:02:00	07:59:23
DIS-2	38.77 ± 4.75	41.07	89.59	21.87	30.58	37.39	03:31:59	05:40:30
DIS-3	37.30 ± 5.77	41.07	86.2	26.82	35.52	36.89	02:39:20	04:36:37
DIS-4	37.41 ± 6.49	41.07	86.45	33.08	35.96	37.41	02:06:26	04:01:10
DIS-5	31.97 ± 3.15	36.03	79.09	11.05	16.33	21.08	05:42:40	07:43:21
DIS-6	35.22 ± 4.34	36.03	87.14	18.73	26.25	33.12	03:45:00	05:49:49
DIS-7	36.41 ± 5.27	36.03	90.08	24.85	34.44	36.36	02:46:44	04:51:53
DIS-8	34.39 ± 5.60	36.03	85.08	31.20	34.33	-	01:58:21	03:13:55
DIS-9	35.52 ± 6.64	36.03	87.88	32.01	35.30	-	01:59:31	03:22:51

^a Total power delivery and time values are derived from discharge test data.

^b Theoretical delivery assumes ideal conditions, with no heat and pressure losses.

^c Efficiency (η) is calculated as the ratio of total power delivery to the average of all 65°C charge tests power capacity.

Since all discharge tests started with an initial tank temperature ($T_{TES,ini}$) of 65°C, the efficiency (η) was calculated using the average power capacity previously determined from charge tests CH-4 through CH-7, which corresponded to a temperature range of 25°C to 30°C for discharge tests DIS-1 through DIS-4. For discharge tests DIS-5 through DIS-9, a dedicated charge test was conducted with an initial tank temperature of 30°C, designed to reflect an equivalent temperature differential, ensuring consistency in the efficiency assessment across the test series. This test had a total capacity of 40.42kWh.

5.7 Storage Losses

A key aspect of this study is the evaluation of heat losses over a 24-hour period, with the tank initially charged to a temperature of 65°C, a condition that simulates a realistic operational scenario following a charging cycle.

Central to this analysis is a comprehensive graph that illustrates the temporal evolution of these losses, providing a visual representation of energy dissipation rates. Additionally, the graph incorporates the average internal temperature of the heat transfer fluid, highlighting the interplay between temperature gradients and energy retention. By examining these metrics, this chapter aims to identify the primary factors contributing to storage losses, such as conduction through the tank walls, natural convection, and ambient heat exchange. Further, this analysis will be a foundation to propose strategies for their mitigation, thereby enhancing the overall efficiency of the LHTES system.

Figure 5.17 shows the result from the experiment, where the tank was previously charged up to 65°C and left untouched for 24 hours straight. During this test, the average temperature inside the testing room was around 26°C.

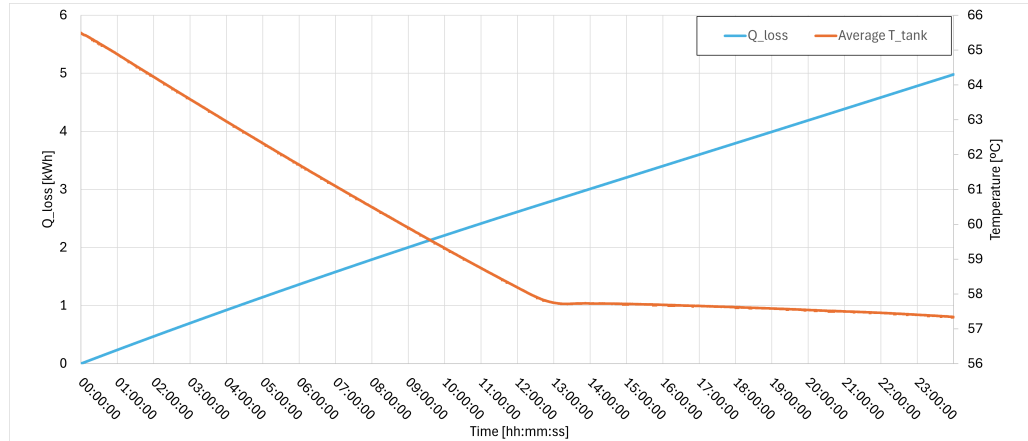


Figure 5.17: Self discharge along time and average HTF temperature inside the tank.

The tank loses approximately 5kWh during these 24 hours, without the HTF average temperature inside it, ever dropping below the phase change interval. Although this happens, there's a reason why this temperature doesn't go below the phase change interval and it's due to the fact that there are capsules, in the colder spots of the tank (middle to bottom), that are releasing latent heat and heating the HTF while it loses heat to the outside, through the tank walls and cap. For this reason, if the point is to store latent heat, there should be a time limit between a charge and a discharge.

Chapter 6

Discussion

6.1 Interpretation of Results

The evaluation of any thermal energy storage system hinges on a thorough interpretation of its performance metrics, which provide critical insights into its efficiency, reliability, and practical applicability.

This section is dedicated to the detailed analysis and interpretation of the experimental results obtained from the charging and discharging tests conducted on the latent heat thermal energy storage system. By synthesizing data from key performance indicators—such as experimental and theoretical capacities, energy density, discharge efficiency, and temporal dynamics—the discussion aims to elucidate the underlying thermal behaviors and system responses under varying operational conditions. The results, presented through tables and graphical representations, reveal the influence of factors such as initial tank temperature, flow rate, and phase change material characteristics on the system's effectiveness. Furthermore, this section explores the implications of observed trends, including storage losses and heat transfer variations, to assess the system's alignment with design expectations and its potential for real-world deployment. Through this interpretive lens, strengths will be identified, limitations highlighted, and avenues for optimization proposed, laying the groundwork for informed enhancements to the LHTES technology. When possible, this analysis will also consider previous tests for comparisons.

6.1.1 Charge mode

The charge mode was divided in three different temperatures - 62°C, 65°C and 70°C - with various HTF flow rates. Figure 5.1 already suggest that, in general, increasing the flow rate leads to a faster charge with a smaller stratification. A larger HTF flow rate led to a higher HTF displacement rate, thus creating a greater temperature difference between HTF and PCM.

Previous research on the same project have used different PCM geometries and different tank sizes which lead to different void fractions. Tianhao et. al [15] used a 0.382 m³ tank filled with 285 ellipsoid slabs and resulted in a void fraction of 0.63, which is higher than that of typical storage devices with a void fraction of 0.3-0.4 [90]. The current tank has a void fraction of around

0.48 and so, it shows a more stratified combination of HTF temperatures along the test due to the reduced volume of unoccupied space within the tank, which limits the convective mixing of the heat transfer fluid and increases the stability of temperature gradients. A numerical study about void fraction in LHTES applications with encapsulated PCMs was performed and concluded that a smaller void fraction was associated to a smaller size of capsule and lead to a lower time for complete charge and discharge [19].

In Figures 5.3 and 5.4 it's noticeable how much increasing HTF inlet flow rate accelerated the charging process by increasing the HTF-PCM temperature difference through a faster replacement of HTF volume in the tank. Figure 5.2 highlights a slower heat transfer rate due to the lower inlet temperature, resulting in a smaller temperature difference throughout the test, which delays the energy exchange between the HTF and PCM. Notably, the total charging time remains relatively consistent across the 62°C $T_{f,in}$ tests, suggesting a saturation effect at this temperature. In contrast, the 65°C and 70°C tests exhibit a pronounced impact of HTF flow rate on both partial and total charging times, indicating a stronger dependence on convective heat transfer at higher temperatures.

Figure 5.5 provides a clear evaluation of how HTF flow rate increase influence the overall heat transfer performance, particularly during the PCM melting phase. This trend is not uniform across all test phases, as will be elaborated later. The 62°C test, with a smaller increment on the flow rate revealed a higher increment on the average global heat transfer coefficient. From 0.5 to 0.6 m³/h there's a higher slope than from 0.6 to 0.75 m³/h, increasing 79% and then only 17%. This diminishing return suggests the system may be approaching a point of flow rate optimization or even a potential behavioral inversion. On the other hand, the 65°C experiments revealed a gradual increase, with the UA values increasing 66% from the lowest to the highest HTF flow rate. The 70°C tests, limited to two data points, indicate a UA value increase of 23%, though the small sample size precludes a definitive trend. This variability underscores the need to assess flow rate effects across a broader range of conditions.

The charging time analysis, as mentioned before, is one of the most important to do, as it allows for a better decision when it comes to the selection of charging strategies.

From the first group of experiments, we've accessed that they've a similar charging profile, shown in Figure 5.2, and so, as expected, there's also a similar time for partial and full charge between them, which translates correspondingly in Figure 5.6. Also, the charging of the capsules changes the slope on the capacity profile. In these experiments, it changes around the first hour of testing, stopping near the end as the last moments serve to charge the rest of the sensible heat up to the temperature above the phase change interval.

CH-1 revealed to be the fastest test among them, despite having the smaller flow rate. The lower HTF flow rate has a 7% slower 50% partial charging time when compared with the higher flow rate one. When reaching the 75% and the 90% partial charges, CH-1 charges faster 16% and 6%, respectively. The total time for charging is also smaller, in this case 12%. This phenomena may arise because the lower flow rate allows prolonged heat exchange with the PCM capsules, especially after the 50% mark, which coincides with the slope change. This suggests that optimal

flow rates might vary depending on the charging stage.

The SoC on the 65°C $T_{f,in}$ differentiates itself from the previous group as the HTF flow rate impacts on all tests, both on the total and partial charge times. CH-7 and CH-6 follow the same profile until the first hour of testing which is also when the capsules start getting charged, and coincides with the 50% partial charge. After that, CH-6, although charging with a smaller flow rate, reaches 75% partial charge 17% faster and 90% partial charge 11% faster, completing the charge in 18% less time, when compared with CH-7.

CH-5, when compared with CH-6, takes 37% more time to charge up 50%, 32% to 75% and 28% to 90% partial charge, completing the charge in 25% more time. Lastly, CH-4, the test with the lowest flow rate in the 65°C $T_{f,in}$ spectrum, presents a 39% higher time to reach 75% partial charge and 38% higher time to reach 90% partial charge, when compared with CH-5. Charging fully with an HTF flow rate of 0.25m³/h at 65°C $T_{f,in}$ takes 13% and 29% longer than with 0.35m³/h and 0.5m³/h, respectively.

The 70°C $T_{f,in}$ group demonstrates the most significant charge time increases with minimal flow rate changes. The higher flow rate (0.35 m³/h) achieves 50%, 75%, 90%, and full charges 32%, 30%, 27%, and 23% faster, respectively, than the 0.25 m³/h test. The increasing time difference with higher charge levels emphasizes how flow rate has a greater impact at higher temperatures, most likely as a result of improved convective heat transfer.

This time analysis identifies CH-1, CH-6, and CH-9 as the fastest tests within their respective groups, with CH-6 emerging as the overall quickest. These findings suggest that an optimal flow rate exists for each temperature range.

When comparing the total experimental capacity between series of tests, divided by their temperature increase, the average for each spectrum was determined. Between the first and second group of experiments, there was an 8% increase on the total capacity, and between the second and the third, a 4% increase was registered.

For the energy density comparison, also the average values among the same $T_{f,in}$ were determined. There was a 9% increase from the first to the second group of tests, and a 4% increase from the second to the third. These increments align with the total capacity trends, attributable to the reduced density of sensible heat beyond the latent phase. This lower density is less advantageous for volume-optimized reservoirs, necessitating a focus on maximizing latent heat utilization, where the PCM offers superior energy density. For this reason, the design of the TES system must prioritize the effective utilization of the latent heat region, where the phase change material provides a higher energy density, to maximize performance in compact applications. Tianhao [14] studied several tank configurations with different void fraction, which resulted in different energy densities. None of these experiments had the same PCM and also, the temperature increase didn't match with the current scenario. Although this is the case, Figure 6.1 plots the comparison between the current and the already performed researches. As seen on the graph, the current study reaches the top values but with a higher operating temperature range. This is due to the fact that the analysis faced a different approach which led to a higher difference between the initial and the final TES tank temperatures.

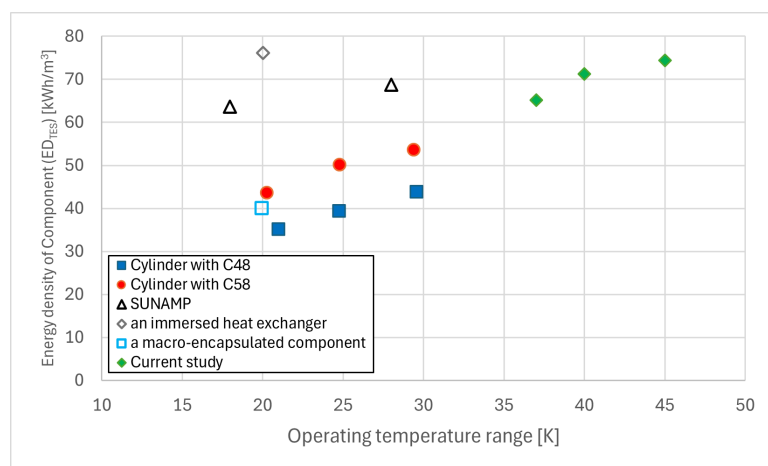


Figure 6.1: Comparison of ED between the current study, a cylindrical encapsulation with C48 and C58 [14] and other SAT-based LHTES prototypes found in literature: SUNAMP [91]; Pure SAT in an immersed HX and SAT mixture in a macro-encapsulated component [92].

Another KPI was studied, and it involves the comparison between the projected LHTES tank and a water SHTES. As the factor should, theoretically, be equal for the same temperature increase, the average capacity enhancement factor, β , was determined. There was a 4% increase from the first to the second group of tests and a 22% decrease from the second to the third group of tests. This goes, initially, against the theory that the capacity enhancement factor should decrease if the charge temperature also increases. From the second to the third group of tests, it decreases, meaning that there's clearly a loss of influence of the latent heat in the total capacity. Theoretically, β should decrease slightly from the first to the second group of tests and more from the second to the third, as the temperature difference between them is even higher (3 to 5K).

The primary cause of the decline in the capacity enhancement factor as the operating temperature range widens is the fact that the heat transfer fluid, which is water, exhibits a higher sensible specific heat capacity in both its solid and liquid phases compared to the three phase change materials. As the operational temperature range extends beyond the phase-change intervals of the PCMs, water increasingly surpasses the three PCMs as an effective heat storage medium. Consequently, expanding the temperature range progressively diminishes the benefit of employing PCMs to achieve high energy storage density. So, for this reason, the only aspect that might lead to increase the HTF inlet temperature, in terms of optimization, would be the time for partially charge the TES tank.

If there would be a scenario where the application needed a higher peak support, there should be a redesign of the tank and not a HTF inlet temperature increase, as this leads to lower latent heat storage percentage and higher losses to the outside. This tank was dimensioned for the exact application that it will be applied to, so for that reason, KPIs such as β and the time for partial and full charge should be the main criteria for optimal BCs decision.

6.1.2 Discharge mode

In performance analysis, TES tanks should be examined as an element of a building heating system, rather than just a standalone unit. This chapter discusses performance indicators that might impact the behavior of the tank upon future real case applications.

The storage discharge behavior had a similar division from the charge mode. The analysis is divided in two: the initial set of tests, which span an $T_{f,in}$ range from 25°C to 30°C, and a subsequent set of tests covering a $T_{f,in}$ range from 30°C to 35°C, incorporating four and five distinct flow rates, respectively.

The HTF discharge behavior acts similar with the charging tests, presenting a higher stratification for lower HTF flow rates. This stratification explains why the latent heat extraction time along the different heights of the tank gets more expanded as we go higher. The gradual increase of the HTF inside the tank due to the latent heat extraction from the bottom to the top, makes the heat transfer rate lower as we go up. This creates a more stable region, seen as a constant line in Figure 5.9. This phenomena, enhanced by a lower flow rate, allows for a more constant power output and outlet temperature, which will be observed ahead.

The analysis on the PCM discharging behavior and the outlet temperature are two of the most important. Figure 5.10 shows the relation between them for the first group of tests. The lower the HTF flow rate, the longer the outlet temperature was kept above or equal the phase change temperature. This is due to the fact that there's a faster exchange of the cold HTF coming in, fastening the discharge by enhancing the heat transfer. The 0.5m³/h discharge was only able to keep the $T_{f,out}$ above the solidification temperature for around 1.3 hours, the 0.35m³/h discharge for 2.2 hours, the 0.25m³/h discharge for 3.2 hours and the lowest flow rate, 0.15m³/h, for 5.9 hours. It resulted in an increase of 84% when compared with the second lowest and 354% with the highest flow rate.

If there's a direct application of the TES tank to provide heating to radiators in an integrated system, the discharged heat with a rapidly decreased temperature is deemed less useful. Scenarios where there's a double daily peak, the two middle flow rates would perform the best while if there was a single very extended peak, the slowest flow rate would fit the control strategy.

The deviation of $T_{PCM,5}$ and $T_{f,out}$ results from the existence of capsules above the capsule labeled as 5. They have an even lower heat transfer rate as the temperature difference between them and the HTF at that height and at that stage of discharge, is very low, leading to a slower power extraction.

Figure 5.11 shows the same analysis, now performed on the 30 to 35°C cooling range. The behavior of the experiments is around the same as the previous group of tests. The highest flow rate, 0.6m³/h was able to keep the $T_{f,out}$ above the solidification temperature for 1 hour, 0.5m³/h for 1.5 hours, 0.35m³/h for 2.4 hours, 0.25m³/h for 3.4 hours and 0.15m³/h for 5.5 hours. From the lowest flow rate to the highest, there was an increment of 450% in the time that the outlet temperature was kept above the PCM solidification temperature.

The UA values, derived from the phase change process, are graphically represented in Figure

5.12, revealing a clear upward trend as the HTF flow rate increases. From the lowest to the highest flow rate, the first group of tests experienced an increase of 193% and in the second 195%. This pattern is particularly pronounced during the melting or solidification phase, reflecting enhanced convective heat transfer between the HTF and the PCM as flow velocity rises. Unlike other stages of the tests, this behavior is not observed uniformly across the discharge cycle. Notably, the fastest discharge rates are not consistently associated with the highest flow rates, suggesting that the observed UA increase is specific to the phase change region. This discrepancy may arise due to the dominance of latent heat exchange during this phase, where higher flow rates facilitate more efficient energy transfer within the PCM's melting or solidifying interval.

The SoD graphs, presented in Figures 5.13 and 5.14, allow to observe the tendency of the discharged capacity along the time, with variable flow rates. The first group of tests have a uniform tendency which deviates the state of discharge from the different flow rates. DIS-1, with the lowest flow rate presents the longest time for partial and full discharge, having a 33% higher time to reach 100% when compared with DIS-2. It takes 76%, 77% and 71% more to reach 50, 75 and 90%, respectively. DIS-2 takes 26%, 25%, 25% and 19% longer to reach 50, 75, 90 and 100%, respectively, when compared with DIS-3. Finally, DIS-4, the faster discharge on this group of experiments, provides 50, 75, 90 and 100% of its capacity in 27%, 28%, 20% and 13% less than DIS-3, respectively.

Figure 5.15 shows a different perspective of the discharge, now plotting the discharge power along the time. The different flow rate profiles follow the same grading as mentioned before. DIS-4 shows a ramp-down behavior, having the highest power peak and fastest discharge. DIS-2 and DIS-3 plot very similarly, with the slower flow rate, having a longer close-to-constant power delivery. DIS-1 discharges almost a constant power for 6 hours straight. In a real case application, DIS-4 would fit a single peak situation with full support, DIS-2 and DIS-3 would fit a partial double peak support, saving half of the capacity for the second peak and DIS-1 would serve as a partial long peak support, due to its low power delivery but long discharge time.

The second group of tests show a similar tendency from DIS-5 to DIS-8. DIS-8 and DIS-9, as mentioned before are tests that revealed that increasing the flow rate past $0.5\text{m}^3/\text{h}$ doesn't improve the heat exchange between the PCM and the HTF in the sensible heat phase, which can be counter-productive as there's higher energy spending on the pumping power. When comparing both of them, DIS-9 discharges only 9% and 7% faster 50 and 75% of the total capacity, and slower 1% 90% of the capacity. While DIS-8 discharges the the full capacity 4% faster than DIS-9. DIS-5 is another test that stood out, but due to its duration. It took 56%, 55%, 61% and 32% more to discharge 50, 75, 90 and 100% of its capacity than DIS-6, which is only $0.1\text{m}^3/\text{h}$ higher.

Figure 5.16 allowed to observe the power delivery profiles of the second group of tests. DIS-8 and DIS-9 showed again a similar profile, while DIS-9 having a 20% higher peak but 6% shorter completion time. These two tests would fit a single peak scenario where initially they could support the entire demand, while after they would probably require the heat pump to power the rest of it. DIS-8 would be preferable as the capacity is more evenly distributed than DIS-9. DIS-7 has a higher peak than DIS-6 but it stands above the 7.5kW discharge power value for 20% less

time. DIS-7 curve stands above the 10kW for about 2.5 hours, while DIS-8 stands above the 7.5kW for about 3.5 hours, making them suitable for days where there are double peaks which can be partially supported. DIS-5 has an average of 5kW discharging power for 6 hours straight making it an unique test. When compared with DIS-6, it has a 70% smaller peak but 32% higher completion time.

As expected, due to the operating temperature range difference, the first group of tests averaged an 9% higher total capacity, when compared with the second group. The two tests from each group that had the lower discharging time, were 20% apart from each other, being DIS-4 slower than DIS-8. DIS-8, although, had a 9% smaller total power delivery.

This results allowed to conclude that the optimal solution for discharging would be adopting a variable flow rate solution, where the flow rate would adjust along the test in response of the operating temperature range decreasing. Depending on the discharge strategy, which relies on the demand side, the value of the flow rate would decrease from the beginning, stabilizing in the phase change range and, decrease again until the end of the test, allowing for a more constant power output. If there was a very high initial power peak like in DIS-8 or DIS-9, this strategy wouldn't work. To keep up with such high power demand during a whole peak, the tank capacity should be higher. For this reason, a partial support would be suggested as it assures a decrease on the heat pump usage.

When it comes to discharge efficiency, the first group of tests has a 2% higher value. The main reason why the value of the efficiency is distant to the maximum efficiency is the fact that the latent heat of fusion is higher on the melting than on the solidification, leading to a higher input of power to extract a smaller amount of heat. The other factor are the thermal losses that occur during the discharge.

As demonstrated in Chapter 5, a storage loss test was conducted to evaluate the amount of heat transferred to the external environment with the heat transfer fluid stationary within the tank. The results indicated that over a 24-hour period, the tank lost approximately 5 kWh of stored capacity, equivalent to 13% of the total power delivery recorded for DIS-4. This loss is significant, particularly for scenarios where the tank remains idle following a charging cycle, which may occur on days of low heating demand. To ensure optimal performance and energy retention, enhancing the tank's insulation is crucial to minimize heat dissipation during these inactive periods, thereby preserving the stored energy for future use and improving overall system efficiency.

Figure 5.17 also allows to predict which is the time where the capsules start to discharge the latent heat previously stored. This transition is observed around the 13th hour, a critical moment that should be avoided to maximize the utilization of the tank's high-density latent heat storage.

6.2 Challenges and Limitations

The experimental phase of the Thermal Energy Storage tank testing presented a series of significant challenges that influenced the reliability and accuracy of the performance data collected.

This chapter, titled "Experimental Problems and Rig Iterations", outlines the primary issues encountered during the testing process, the innovative solutions proposed to address these challenges, and the evolution of the test bench layouts undertaken to enhance system functionality.

The difficulties experienced included maintaining a consistent inlet temperature during both the charging and discharging phases, managing water leakages at specific points within the system, and addressing inaccuracies in temperature measurements due to water flowing into the RTD sensors within the PCM capsules, by capillary effects. These problems not only tested the resilience of the experimental setup but also provided valuable insights that drove iterative improvements to the rig design. This section will detail the nature of these issues, the strategic interventions implemented to mitigate them, and the various configurations of the test bench explored throughout the project, reflecting a comprehensive approach to optimizing the TES tank's performance under real-world conditions.

The first problem found was water leakage in many points of the system, like RTD nuts, threaded connections and rubber sealants. These leakages were solved by using sealant rings and applying filler for the threads. This process was time consuming as the procedure required that the whole system was fully filled with water and whenever there was a leakage, the filling was stopped to correct the problem. Due to constant expansion and contraction of the inertia tank, some of the rubber sealants wore out quickly and needed repair or replacement - Figure 6.2.



Figure 6.2: Example of leakage on a rubber sealant applied on a connection of the water tank.

One of the most critical problems encountered was that, with the initial bench test configuration, it was hard to maintain a constant inlet temperature. This was due to the fact that the difference between the inlet and outlet temperatures was so high that the inertia tank was not big enough - or the electric heater not powerful enough - to assure a constant temperature on the inlet of the TES tank. This problem was not mitigated but the situation was improved by decreasing the temperature on the inertia tank and adding an extra electric heater just before the inlet of the tank. This allowed to have more precision as we didn't rely on the inertia tank and could regulate the temperature by hand on the thermostat found on the additional electric heater.

Another time-consuming challenge was the infiltration affecting the RTD cables. As depicted in Figure 4.9 and elaborated upon in this chapter, the initial setup included the installation of five capsules equipped with temperature monitoring inside the tank.

During the initial tests, it was observed that several of the sensor outputs exhibited significant measurement noise, which was traced back to water infiltration into the cables. This led to the damage of the RTDs, either by causing short circuits or by altering the resistance measurements along the cables.

These five RTDs were strategically placed at varying heights within the tank, and the visibly damaged units were identified as the first, second, fourth and the fifth, counting from the bottom to the top, as shown in Figure 4.9. This type of problem has emerged in previous tests performed in the same project. It never got resolved until this experiment. With the time we had left, we were able to extract one of the damaged PCM capsules (the one at the top) and replace it with a better insulating approach. The whole process of insulation was explained in Chapter 4.2.

Figures 6.3 and 6.4 show the difference between the initial and the second insulation method to protect the RTD cables. The first approach was with a drilled rubber cork that held the RTD in place and insulated the hole in the capsule. Around this cork, a small amount of silicone glue was added to prevent any leakages. This wasn't meant to insulate the RTD but the capsule, which resulted in the damaging of the cables of the RTD. On the second approach, the cables insulation was also taken into account and it worked during the whole testing procedure, as the temperatures reading was without any noise and aligned with which was expected.



Figure 6.3: Initial Capsule RTD Insulation.



Figure 6.4: Final Capsule RTD Insulation.

To conclude, the initial RTDs 1, 2, 4 and 5 got damaged by water infiltration. Extracting all capsules would be highly time-consuming, requiring a complete system flush and meticulous manual removal. Given the time constraints, it was determined that only the first capsule (associated with RTD 5) would be replaced with a better-insulated version. So, only T_{PCM5} and T_{PCM3} were measuring properly, therefore, they were the ones used for the calculations and graphs.

Chapter 7

Conclusions and Recommendations

7.1 Conclusions

The primary objective of this thesis was to characterize the performance of a latent heat thermal energy storage tank integrated with heating system that simulated a heat pump, focusing on its potential for sustainable heating applications and peak load shifting. Through a comprehensive experimental investigation, performed on a bench test that was built in the study period, this work extended previous research by evaluating the system's behavior under varying boundary conditions, including inlet temperatures and flow rates, while addressing key challenges in the test rig setup. Nine charges and discharges were performed and analyzed to extract KPIs that allowed for a deep understanding of the TES tank behavior.

The results demonstrated that the LHTES tank, utilizing RT57HC as the PCM, exhibits robust performance in both charging and discharging modes. During charging, higher HTF flow rates accelerated the process and reduced stratification, with an 8% increase in total capacity from 62°C to 65°C inlet temperatures and a 4% increase from 65°C to 70°C. The UA values showed a positive correlation with flow rate during the phase change, highlighting the importance of convective heat transfer optimization. Partial charge times at 50%, 75%, and 90% levels varied significantly with flow rate, with lower rates allowing for prolonged heat exchange but extending overall charging duration. Energy density comparisons revealed incremental improvements with higher temperatures, though the diminishing returns in latent heat contribution beyond the phase transition underscore the need to prioritize this region for volume-optimized designs. The average capacity enhancement factor analysis confirmed the LHTES superiority over water-based SHTES, with positive values indicating enhanced storage capacity, although decreasing with wider temperature ranges.

In discharge mode, similar trends were observed, with lower flow rates yielding more constant power output and outlet temperatures, ideal for consistent energy delivery. The SoD profiles at 50%, 75%, and 90% levels showed that higher flow rates speed up discharge but reduce the duration of stable power, while lower rates extend this period, suitable for prolonged peak support. UA values during solidification increased with flow rate by approximately 193% and 195% in

the 25-30°C and 30-35°C ranges, respectively, though not uniformly across phases. Efficiency remained high at 88% and 86% on the first and second group of tests, respectively, but the slight asymmetry in latent heat (melting vs solidification) contributed to minor losses. Storage losses over 24 hours at 65°C were approximately 5 kWh, or 13% of DIS-4's delivery, emphasizing the need for enhanced insulation to minimize dissipation.

The experimental rig iterations addressed critical issues, such as water leakage solved with sealant rings replacement and fillers, inlet temperature stability improved by adding a 3 kW heater just before the inlet of the TES tank and RTD infiltration mitigated by enhanced insulation, ensuring reliable data. These challenges highlighted the importance of robust design in real-world applications.

Overall, the LHTES system outperforms traditional water storage in energy density, flexibility and stability, with optimal strategies like variable flow rates for charging/discharging enabling effective peak load shifting. The findings validate the desuperheater integration and RT57HC PCM selection, offering economic and environmental benefits through reduced energy costs and emissions.

The current study establishes a robust foundation for future investigations into latent heat thermal energy storage systems applied to domestic hot water and residential space heating. By providing detailed performance metrics such as charging/discharging efficiencies, energy densities, UA values, and partial state-of-charge/discharge behaviors, this work offers a comprehensive dataset that can guide the development of more advanced LHTES configurations tailored to household needs.

7.2 Recommendations for future work

Notably, the extracted data, including real-time temperature profiles and flow dynamics, provide valuable inputs for developing machine learning models, such as neural networks or regression models, to optimize and control the synergy between the storage tank and heat sources like heat pumps. A prioritized focus could be predicting optimal charging times based on electricity pricing, forecasting heat demand, estimating energy losses during idle periods, and dynamically adjusting HTF flow rates to enhance peak load shifting and optimized power delivery. This could reduce operational costs and improve system reliability in real-world residential settings. Future work should include data pre-processing, on the flow rate and HTF and PCM temperatures, to ensure model accuracy.

A future experimental campaign is recommended to conduct charging/discharging testing with transient HTF flow rate control. It would be interesting if, in the future application of the TES tank, the live-in lab, this strategy was tested and maybe adopted. To improve accuracy and reduce variability, these experiments should be repeated multiple times under identical boundary conditions, enabling statistical analysis of the results.

To mitigate fluctuations in $T_{f,in}$ during discharge tests, a cooling loop with active control or an additional water tank could be incorporated. This would ensure that the inlet temperature of the

tank aligns with the values specified in the test matrix throughout the discharge process, either by increasing the thermal inertia of the cooling system or by establishing a controlled loop.

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Appendix A

Python Scripts used for Theoretical Values Assessment

A.1 theoretical_capacity.py

```
1 import numpy as np
2 import matplotlib.pyplot as plt
3 import CoolProp.CoolProp as CP
4
5 #TR
6
7 N_capsules = 515
8 m_pcm = 133.385
9 hf_pcm = 246.35
10 hc_pcm = 242.3
11 T_m = 57
12
13 cp_s_PCM = [1.73, 1.85, 1.95, 2.11, 2.29]
14 cp_l_PCM = [2.7, 2.72, 2.72, 2.75]
15 T_s_pcm = [10, 20, 30, 40, 55]
16 T_l_pcm = [62, 65, 70, 75]
17
18 m_pp = N_capsules * 0.324
19 cp_pp = 1900 / 1000
20
21 p = 101325
22
23 def interpolate_cp(temps, cp_values, target_temp):
24     if target_temp <= temps[0]:
25         return cp_values[0]
26     if target_temp >= temps[-1]:
27         return cp_values[-1]
```

```

28     for i in range(len(temps) - 1):
29         if temps[i] <= target_temp <= temps[i + 1]:
30             w = (target_temp - temps[i]) / (temps[i + 1] - temps[i])
31             return cp_values[i] + w * (cp_values[i + 1] - cp_values[i])
32
33 t_initial = np.arange(30, 65.5, 0.5)
34 results_kj = []
35 results_kwh = []
36 cumulative_kwh = []
37
38 for t_ini in t_initial:
39     t_fin = t_ini + 0.5
40     delta_t = t_fin - t_ini
41     t_mean = (t_ini + t_fin) / 2
42
43     rho_water = CP.PropsSI('D', 'T', t_mean + 273.15, 'P', p, 'Water')
44     cp_water = CP.PropsSI('C', 'T', t_mean + 273.15, 'P', p, 'Water') /
45         1000
46
47     m_water = (0.477 * 0.480 + (0.15 + 0.165) * 0.714**2 * np.pi) *
48         rho_water
49
50     if t_mean < T_m:
51         cp_pcm = interpolate_cp(T_s_pcm, cp_s_PCM, t_mean)
52     else:
53         cp_pcm = interpolate_cp(T_l_pcm, cp_l_PCM, t_mean)
54
55     q_pcm_sensible = m_pcm * cp_pcm * delta_t
56     q_pcm_latent = m_pcm * hf_pcm if not latent_added and t_ini < T_m and
57         t_fin >= T_m else 0
58     if q_pcm_latent > 0:
59         latent_added = True
60     q_pcm = q_pcm_sensible + q_pcm_latent
61
62     q_pp = m_pp * cp_pp * delta_t
63
64     q_water = m_water * cp_water * delta_t
65
66     q_total = q_pcm + q_pp + q_water
67     q_total_kwh = q_total / 3600
68     results_kj.append(q_total)
69     results_kwh.append(q_total_kwh)
70     cumulative_kwh.append(sum(results_kwh))

```

```

69     print(f"T_ini={t_ini} C , T_fin={t_fin} C , cp_pcm={cp_pcm:.2f}
      kJ/kg K , cp_water={cp_water:.2f} kJ/kg K , "
70         f"rho_water={rho_water:.2f} kg/m , Q_total={q_total:.2f} kJ,
      Q_total={q_total_kwh:.4f} kWh")
71
72 total_capacity_kj = sum(results_kj)
73 total_capacity_kwh = sum(results_kwh)
74
75 print(f"\nCapacidade Total Acumulada: {total_capacity_kj:.2f} kJ")
76 print(f"Capacidade Total Acumulada: {total_capacity_kwh:.4f} kWh")
77
78 plt.plot(t_initial, results_kwh, label='Q_total (kWh) per step',
      marker='o')
79 plt.plot(t_initial, cumulative_kwh, label='Q_total Accumulated (kWh)',
      linestyle='--', marker='s')
80 plt.axhline(y=total_capacity_kwh, color='r', linestyle=':',
      label=f'Total: {total_capacity_kwh:.4f} kWh')
81 plt.xlabel('Temperature ( C )')
82 plt.ylabel('Thermal Capacity (kWh)')
83 plt.title('Thermal Capacity per step (0.5 C ) and Accumulated')
84 plt.legend()
85 plt.grid(True)
86 plt.yticks(np.arange(0, max(cumulative_kwh) + 5, 5))
87 plt.show()

```

A.2 theoretical_discharge_power.py

```

1 import numpy as np
2 import matplotlib.pyplot as plt
3 import CoolProp.CoolProp as CP
4
5 #TR
6
7 N_capsules = 515
8 m_pcm = 133.385
9 hf_pcm = 246.35
10 hc_pcm = 242.3
11 T_m = 57
12
13 cp_s_PCM = [1.73, 1.85, 1.95, 2.11, 2.29]
14 cp_l_PCM = [2.7, 2.72, 2.72, 2.75]
15 T_s_pcm = [10, 20, 30, 40, 55]
16 T_l_pcm = [62, 65, 70, 75]
17

```

```

18 m_pp = N_capsules * 0.324
19 cp_pp = 1900 / 1000
20
21 p = 101325
22
23 def interpolate_cp(temps, cp_values, target_temp):
24     if target_temp <= temps[0]:
25         return cp_values[0]
26     if target_temp >= temps[-1]:
27         return cp_values[-1]
28     for i in range(len(temps) - 1):
29         if temps[i] <= target_temp <= temps[i + 1]:
30             w = (target_temp - temps[i]) / (temps[i + 1] - temps[i])
31             return cp_values[i] + w * (cp_values[i + 1] - cp_values[i])
32
33 t_initial = np.arange(65, 29.5, -0.5)
34 results_kj = []
35 results_kwh = []
36 cumulative_kwh = []
37
38 for t_ini in t_initial:
39     t_fin = t_ini + 0.5
40     delta_t = t_fin - t_ini
41     t_mean = (t_ini + t_fin) / 2
42
43     rho_water = CP.PropsSI('D', 'T', t_mean + 273.15, 'P', p, 'Water')
44     cp_water = CP.PropsSI('C', 'T', t_mean + 273.15, 'P', p, 'Water') /
45         1000
46
47     m_water = (0.477 * 0.480 + (0.15 + 0.165) * 0.714**2 * np.pi) *
48         rho_water
49
50     if t_mean <= T_m:
51         cp_pcm = interpolate_cp(T_s_pcm, cp_s_PCM, t_mean)
52     elif T_m < t_mean <= 59.4:
53         cp_pcm = interpolate_cp(T_l_pcm, cp_l_PCM, t_mean)
54     else:
55         cp_pcm = interpolate_cp(T_l_pcm, cp_l_PCM, t_mean)
56
57     q_pcm_sensible = m_pcm * cp_pcm * delta_t
58     q_pcm_latent = 0
59     if 57.1 <= t_ini and t_fin >= 57.1 and t_ini <= 59.4:
60         fraction = min(1.0, max(0.0, (59.4 - t_fin) / (59.4 - 57.1)))
61         q_pcm_latent = m_pcm * hc_pcm * fraction
62     q_pcm = q_pcm_sensible + q_pcm_latent

```

```

61
62     q_pp = m_pp * cp_pp * delta_t
63
64     q_water = m_water * cp_water * delta_t
65
66     q_total = q_pcm + q_pp + q_water
67     q_total_kwh = q_total / 3600
68     results_kj.append(q_total)
69     results_kwh.append(q_total_kwh)
70     cumulative_kwh.append(sum(results_kwh))
71
72     print(f"T_ini={t_ini} C , T_fin={t_fin} C , cp_pcm={cp_pcm:.2f}
73           kJ/kg K , cp_water={cp_water:.2f} kJ/kg K , "
74           f"rho_water={rho_water:.2f} kg/m , Q_total={q_total:.2f} kJ,
75           Q_total={q_total_kwh:.4f} kWh")
76
77 total_capacity_kj = sum(results_kj)
78 total_capacity_kwh = sum(results_kwh)
79
80
81 print(f"\nCapacidade Total Liberada: {total_capacity_kj:.2f} kJ")
82 print(f"Capacidade Total Liberada: {total_capacity_kwh:.4f} kWh")
83
84 plt.plot(t_initial, results_kwh, label='Q_total (kWh) per step',
85          marker='o')
86 plt.plot(t_initial, cumulative_kwh, label='Q_total Accumulated (kWh)',
87          linestyle='--', marker='s')
88 plt.axhline(y=total_capacity_kwh, color='r', linestyle=':',
89             label=f'Total: {total_capacity_kwh:.4f} kWh')
90 plt.xlabel('Temperature ( C )')
91 plt.ylabel('Thermal Capacity (kWh)')
92 plt.title('Thermal Capacity per step (0.5 C ) and Accumulated -
93           Discharge')
94 plt.legend()
95 plt.grid(True)
96 plt.yticks(np.arange(min(cumulative_kwh) - 5, 0, 5))
97 plt.show()

```