

Hollowed and perforated fins in latent heat storage units for High-Temperature hybrid thermal energy storage applications

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High-temperature thermal energy storage (TES) is essential for next-generation concentrated solar power (CSP) plants in order to ensure continuous energy supply. Hybridization of latent heat storage (LHS) and sensible heat storage (SHS) enhances energy density, thermal stability, and efficiency by leveraging the high storage capacity of phase change materials (PCMs) while reducing thermal ratcheting for sensible storage. This study focuses on a numerical analysis of a shell-and-tube LHS using sodium as heat transfer fluid (HTF). It examines the impact of hollowed and perforated fins to enhance effective heat exchange. Simulations were conducted in a 3D solution domain using ANSYS Fluent. The results show that fin removal rate and hole placement are crucial design factors. A 20% perforation rate in the *Perforated fin-Middle(full)* configuration maintains high heat transfer efficiency, reduces material costs, and increases PCM storage. In comparison to molten salts as HTFs, liquid metals exhibit effectively lower HTF outlet temperatures, which is vital for LHS-SHS integration. These findings provide valuable insights for optimizing high-temperature TES units in large-scale CSP applications.

1. Introduction

In 2025, electricity generation from renewable sources is expected to overtake that from fossil sources [1]. Among the renewables, competition is evident between solar photovoltaic (PV) and Concentrated Solar Power (CSP) technologies. While the former generates electricity directly and showed strongly decreasing costs in the past, research on the latter is focusing on achieving higher heat-to-power conversion efficiencies, likely through higher operating temperatures, to reach lower electricity generation costs [2].

Nevertheless, neither PV and CSP nor other renewable sources can be considered viable without energy storage, if dispatchability on demand is desired. Thermal energy storage (TES) has the cost advantage over electrochemical batteries and hydrogen conversion processes because it is significantly cheaper, especially at large scale [3]. The integration of TES into CSP plants increases the capacity factor, positively impacting the levelized cost of electricity and storage, while enabling the system to operate continuously 24/7. Currently, the installed or commissioned new plants use single-tank or two-tank molten salt TES systems. However, with the aim of 3rd generation CSP plants, operating temperatures are expected to go beyond 600 °C temperature. These high-temperature

conditions create new challenges, as only few fluids are capable of remaining stable at such high temperatures [4].

1.1. Scope of this study

This numerical study focuses on a shell-and-tube LHS unit for the high-temperature hybrid thermal energy storage (HiHyTES) consisting of an LHS unit and a packed-bed SHS using liquid metals as HTF. As previous studies with other HTFs showed that thermal ratcheting can play a major role with regard to reduced lifetime of packed-bed SHS [5], this study aims to minimize this expected effect by reducing the temperature difference in the SHS through adding an LHS for the high-temperature section as shown in Fig. 1.

For this study, liquid metals (LMs) are proposed as HTF. LMs do not chemically decompose, however, are corrosive at high temperatures. In addition, safety concerns should be mentioned, such as toxicity and molecular weight of Pb or fast reactivity of Na or Mg with oxygen [6]. There are some pilot-scale SHS systems integrated into CSP plants, i.e., liquid Na (two-tank configuration) with $T_{\max} = 530$ °C in Almeria, Spain [7], liquid Na (two-tank configuration) with $T_{\max} = 565$ °C in Jemalong, Australia [8], liquid Pb-Bi eutectic (two-tank configuration) with $T_{\max} = 770$ °C in Australia [8], and liquid Pb (one-tank configuration) with

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Nomenclature

A	coefficient of momentum source term, $\text{kg}/(\text{m}^3 \cdot \text{s})$
b	dummy constant
c_p	specific heat, $\text{kJ}/(\text{kg} \cdot \text{K})$
C	mushy zone constant, $\text{kg}/(\text{m}^3 \cdot \text{s})$
D	diameter, mm
g	gravitational acceleration, m/s^2
h	enthalpy, kJ/kg
H	latent heat capacity, kJ/kg
k	thermal conductivity, $\text{W}/(\text{m} \cdot \text{K})$
L	length, mm
M	material factor
P	pressure, Pa
S_{buoyancy}	buoyancy source term, $\text{kg}/(\text{m}^2 \cdot \text{s}^2)$
S_{ph}	energy source term, W/m^3
t	time, s
T	temperature, $^{\circ}\text{C}$
u, v, w	velocity components, m/s
$\mathbf{u}$	velocity vector
x, y, z	spatial coordinates, m

Greek symbols

μ	dynamic viscosity, Pas
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ρ	density, kg/m^3
β	thermal expansion coefficient, $1/\text{K}$
ε	fin effectiveness

Subscripts

lat	latent
l	liquification point
ref	reference condition
sens	sensible
s	solidification point
tot	total

Abbreviations

CSP	concentrated solar power
HEX	heat exchanger
HiHyTES	high temperature hybrid thermal energy storage
HTF	heat transfer fluid
HPC	high performance computing
LHS	latent heat storage
lf	liquid fraction
PCM	phase change material
SHS	sensible heat storage
TES	thermal energy storage

$T_{\text{max}} = 750^{\circ}\text{C}$ in Brasimone, Italy [8].

In the numerical study of Niedermeier et al. [9], in which liquid sodium and three different molten salts are compared as HTF and storage material, liquid sodium was found to discharge heat slightly more efficiently than molten salts due to differences in thermal conductivity and heat transfer coefficients. In addition, the lower viscosity of liquid Na requires lower pumping power, and its material cost as €/kWh stays in the price range of molten salts. In a comparative study by Laube et al. [4], LMs such as sodium (Na), lead (Pb), and lead-bismuth eutectic (LBE) were preferred over solar salt in packed-bed TES due to their superior thermal conductivity and higher temperature stability. Furthermore, it was indicated that liquid sodium exhibits almost four times higher thermal conductivity than other metals.

In the literature, several hybrid TES systems are proposed. Ahmed et al. [10] explored three different configurations within storage tanks: sensible rods, spherical PCM, and their combination. While PCM encapsulation improves thermal conductivity due to increased heat transfer surface area, it also substantially raises pumping power requirements. In a study by Tehrani et al. [11], which examined various

SHS and LHS techniques in shell-and-tube and packed-bed configurations, while the latter using encapsulated PCM proved to be the least favorable option in terms of techno-economic performance, primarily due to the high cost of encapsulation. On the other hand, Michels and Pitz-Paal [12] proposed the use of cascaded LHS tanks to benefit from the high fusion enthalpy of PCMs during melting and solidification in solar power generation plants. They emphasized the need to improve the conductivity of PCMs to enhance system thermal responsiveness. Liu et al. [13] conducted numerical analyses to compare the thermal performance of SHS, cascaded LHS, and hybrid SHS/LHS storage systems under identical operating conditions. Their findings indicate that the hybrid system demonstrated the best thermal effectiveness. Furthermore, Tehrani et al. [14] investigated shell-and-tube LHS systems using commercially available high-temperature PCMs. They found that LHS tanks required 70 % less volume than SHS tanks.

Moreover, when LMs are being used as HTF, the thermal conductivity of the PCM becomes a critical factor. Riahi et al. [15] compared the exergetic performance of SHS and LHS tanks in CSP plants. They found that while supercritical CO_2 ($k = 0.53 \text{ W}/\text{mK}$) as an HTF requires

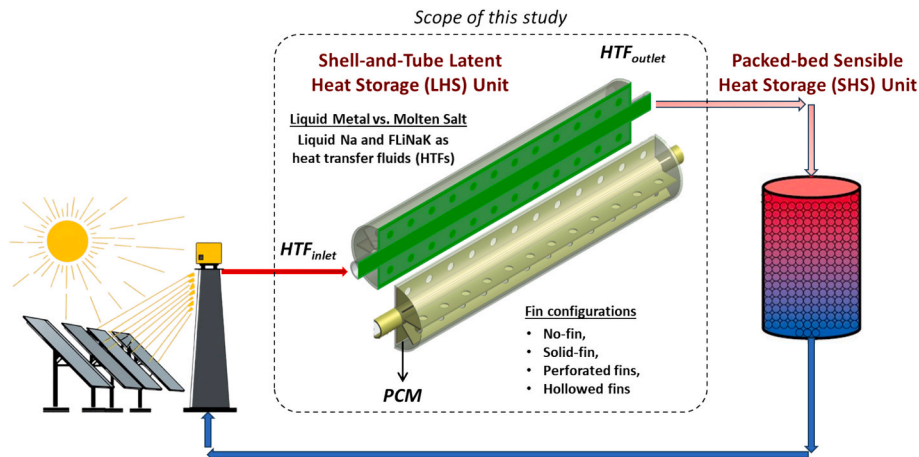


Fig. 1. Simplified schematic of HiHyTES integrated with CSP Plant, dashed line showing the scope of this study.

longer tubes with more residence time, while liquid sodium ($k = 58 \text{ W/mK}$) can achieve the same heat transfer with shorter tubes and less residence time. Although exergy losses in LHS tanks can be minimized through design improvements, cost remains the primary concern. Zauener et al. [16] conducted both numerical and experimental investigations into hybrid sensible-latent heat storage, focusing on economies of scale. Their results indicated that the size of the storage tank has a significant impact on costs per stored thermal energy, with larger tanks offering more favorable economic outcomes.

The literature review summarizes that current developments on behalf of next-generation CSP plants favor the use of LMs as HTF and hybridization of SHS and LHS units, but in separate tanks due to economic aspects. However, LHS units should be able to respond to the high conductivity of LMs, by employing either salt-based high-temperature PCMs enhanced with innovative heat transfer methods or directly employing metallic PCMs.

1.2. Overview of perforated fin performances within Shell-and-Tube LHS units

Excluding metallic PCMs, all other types require the support of heat transfer enhancement methods. These methods typically involve incorporating highly conductive structures made of metals or carbon-based materials, which intrinsically boost the conduction heat transfer capabilities of LHS units. However, these enhancement methods often hinder natural convection heat transfer. For instance, while nano-additives increase the viscosity of melted PCM, porous skeleton structures obstruct natural convection currents. Similarly, extended fins can directly inhibit the rising and sinking movement of molten PCMs, even though this method uses the least conductive material (especially as in perforated [17], dimple [18], or recessed [19] fins) for the same melting performance [20]. Unlike current fin designs—such as longitudinal, radial, tree-shaped, branched, snowflake, or helical fins—it is quite possible to design natural convection-friendly structures by perforating fins. The concept is based on removing certain local sections of the fin surface. Thus, in addition to improving the thermal conduction through the insertion of fins, natural convection in the melting process is improved by allocating gaps for upward/downward circulation flows.

To the best of the authors' knowledge, the first study of perforating fins was conducted by Karami and Kamkari [21]. They experimentally investigated the charging dynamics of a vertically oriented shell-and-tube LHS unit, comparing the performance of solid and perforated radial fins. The authors reported that a 16 % perforation of the fins led to a 30 % increase in the Nusselt number (Nu) due to a minor restriction of natural convection circulation. Despite of this, the reduction in melting time is only 7 %. In a similar vertical-oriented LHS system with radial fins, Li et al. [22] numerically study perforated fin designs and stand out the parameters of hole diameter and location of punctures. The results show that increasing hole diameter benefits natural convection but gradually deteriorates conduction. Besides, holes placed near the tube cause weakened conduction. It indicates that an optimum exists for both hole diameter and the location. The study of Modi et al. [23] supports the recommendation of using smaller diameter holes to avoid comprising conduction. They suggested that employing multiple rows of smaller-diameter holes on fins is preferable to using single rows of larger holes. The authors also noted that perforating upper fins work better due to the stronger presence of natural convection in horizontally oriented LHS units and are more prominent in the second half of the melting process. Wang et al. [24] numerically study how the accordance of heat conduction–convection changes with the involvement of holes in different branches of the tree-shaped longitudinal fin structure. He et al. [25] experimentally investigate the effect of perforation on spiral fins in a vertical arrangement. The temperature profiles demonstrated in the study indicate that the perforated design with a lower removal amount and greater perforation diameter is superior by $\sim 19 \%$ to the solid fin case. Dhaidan et al. [26] conduct an experimental comparison of radial

and longitudinal fins, with and without perforation, during the charging process in a horizontal orientation. Notably, the PCM-to-fin material ratio was kept constant, even with varying perforation rates. The best case is obtained when perforated radial fins are used. In the simultaneous charging/discharging study of Khobragade et al. [27], the maximum exergy efficiency is reached in the solid fin case with horizontal orientation. In contrast, Kalapala et al. [28] reported that vertical orientation yields higher exergetic efficiency, regardless of whether fins are used. They also found that perforated fins do not outperform solid fins at any inclination. Raj et al. [29] attempt to perforate radial fins via hollowing circumferential holes in a vertically oriented LHS unit. The authors claim that once the melted PCM surrounds all the tube and fins, and isolated solid PCM regions are away from the heat transfer surfaces, perforated fins start to act more prominent. This is because natural convection heat transfer becomes more determinant as the thermal conductivity of melted PCM that stays between fins and isolated solid PCM is low during the later stages of the melting process. During the discharge process, where heat transfer is primarily through conduction, perforated fins either underperform [25] or show no significant improvement [2330].

There are few studies about perforated fins applied on shell-and-tube heat exchangers, and all have been recently published. There is a strong need to discuss the performance of perforated fins in terms of melting time reduction relative to fin removal rate. These results are documented in Table 1, some details are gained by self-calculation (marked by asterisk) if there is no information provided in the relevant articles. The first notable observation from the table is that either there is no melting time reduction or perforation causes only a minor time improvement compared to unperforated solid fins. Interestingly, the reduction in melting time occur when fin removal rates are quite low about 2 %-3.5 %, yet Karami and Kamkari obtained 7 % reduction with 16 % fin removal rate. If fin material information of the studies is analyzed comparatively to identify the reason behind the change in melting time or fin removal rate, it does not give a meaningful correlation between any factors in Table 1. Perforated fins help reduce obstruction to natural convection. They also offer practical advantages, such as lower material costs and increased PCM volume, which can boost storage capacity. Moreover, most studies involving perforated fins are conducted within a narrow operating temperature range (mostly using lauric acid as PCM). Almost all lacks diversity in hole shapes, as nearly all holes are circular.

The aim of the present study is to investigate the development of both perforated fin designs and LHS units suitable for CSP operations, specifically HiHyTES applications. Melting/solidification performances of perforated fins are numerically compared with various fin shapes (such as hollowed, fully circular-perforated and semicircular-perforated fins), different fin removal rates (20 %, 40 %), and perforation locations (bottom or middle holes) in a horizontally oriented shell-and-tube LHS tank. The novelty of the present work also lies in the selection of the HTF (liquid metal vs. molten salt) that can withstand high temperatures up to 750°C since the HiHyTES concept is created for satisfying next-generation CSP plants. The effects of change in fin designs are discussed over parameters of liquid fraction, heat flux, stored/released enthalpies, fin effectiveness, and HTF outlet temperature. The results provide valuable insights to examine how to maximize latent heat capacity with minimal fin occupation without significant sacrifices in heat transfer, while also minimizing fin material costs.

2. Numerical model

2.1. Physical domain and thermophysical properties

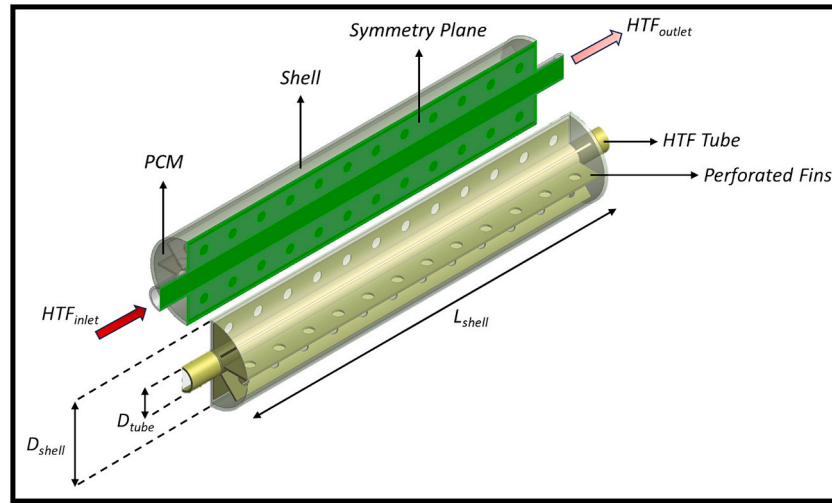
Fig. 2 demonstrates the shell-and-tube TES unit, where high-temperature HTF heated via a CSP receiver circulates inside the tube, while PCM fills the annular gap between shell and HTF pipe. One main function of this LHS unit is to absorb thermal energy carried via HTF and

Table 1

Summary of perforated fin studies for HEX applications in literature.

Studies	Fin Removal Rate	Melting Time Reduction	Hole Shape	Fin Material	T _{melting}
Raj et al. [29]	(0.6–7.2 %) 2 %	5 %	Hollowed	Stainless steel (n.a.)	Lauric Acid (43.46–49.94 °C)
Wang et al. [24]	(2–2.65 %) 2 %	~5.37 %	Circular	Aluminum (202.4 W/mK)	Lauric Acid (42–44 °C)
Li et al. [22]	*(0.5–56 %) 3.5 %	~5.49 %	Circular	Copper (387.6 W/mK)	Paraffin Wax (45–51 °C)
Karami and Kamkari [21]	16 %	7 %	Circular	Copper (387.6 W/mK)	Lauric Acid (43.5–48.2 °C)
Dhaidan et al. [26]	No reduction	15.9 %	Circular	Copper (387.6 W/mK)	RT42 (38–43 °C)
Kalapala et al. [28]	*13.7 %	No reduction	Circular	Stainless steel (n.a.)	Lauric Acid (43.46–49.94 °C)
He et al. [25]	26 %	(n.a.)	Circular	Aluminum (234 W/mK)	RT55 (48.8–57.4 °C)
Khobragade et al. [27]	*2.1 %	No reduction	Circular	Stainless steel (16.27 W/mK)	Lauric Acid (43.46–49.94 °C)
Modi et al. [23]	*~10 %	~4.2 %	Circular	Copper (385 W/mK)	RT60 (55–61 °C)

* Own calculated values.

**Fig. 2.** Schematic representation of shell-and-tube heat exchanger filled with PCM and equipped with perforated fins.

to decrease its temperature as much as possible before conveying it to the thermocline storage tank. That is why tubes are equipped with innovative fins in order to increase heat transfer when low thermal conductive PCMs are employed. As recommended by Modi et al. [31], a shell-to-tube diameter ratio of ~ 4 appears to be optimum for complete melting and solidification processes regarding natural convection effectiveness. All previous studies investigating the influence of orientation on melting and/or solidification processes offer a vertical arrangement of the LHS unit due to the reduced melting time, regardless of whether solid or perforated fins are used [17,23,28]. However, horizontal orientation favors stronger natural convection as depicted in Ref [27]. Therefore, in this study, a horizontal arrangement of the unit was selected in order to be able to analyze the role of natural convection compared to conductive heat transfer, especially when using convection-friendly fin configurations [32–35]. Additionally, the horizontal orientation offers benefits in terms of reduced thermal stress and improved structural integrity compared with vertical arrangements [36]. A concentrically centered tube was selected in this work for simplicity and to primarily assess the effects of fin modifications.

All geometric parameters given in Table 2 are kept constant after alterations in fin designs, except for the resulting changes in the fin-to-total volume ratio. Depending on the extent of fin material removal as more PCM fills the holes in the fins, leading to a higher heat storage

Table 2

General dimensions of HEX and fin geometries of the solid fin as the base case.

HEX Geometry		Fin Geometry	
D _{tube,inner} , mm	30	Fin/Total Volume, %	8.6
D _{tube,outer} , mm	36	Fin Thickness, mm	3
D _{shell,inner} , mm	120	Annular Gap, mm	42
D _{shell,outer} , mm	126	Fin Height, mm	37
D _{shell} /D _{tube}	4	Number of fins	8
L _{shell} , mm	600	L _{fin} , mm	596

capability in the same total storage domain.

Fig. 3 presents an isometric 3D view of the finned shell-and-tube heat exchanger (HEX) concept along with 2D sectional views of the innovative fin designs. In the base fin design (Case 1), eight solid fins occupy 8.6 % of the volume ranging from the tube to the shell. To mitigate the negative effects of extended fins on convection circulations, holes can be introduced to create alternative pathways for the melted PCM to flow. Innovation in fin design was achieved by removing parts of the fins to create hollow or semicircular/circular perforated shapes at various locations along the longitudinal solid fins. In Case 2, a hollowed region measuring 8 mm in height and 560 mm in length was created, leaving a fin-to-tube contact surface of 18 mm at both the entrance and exit

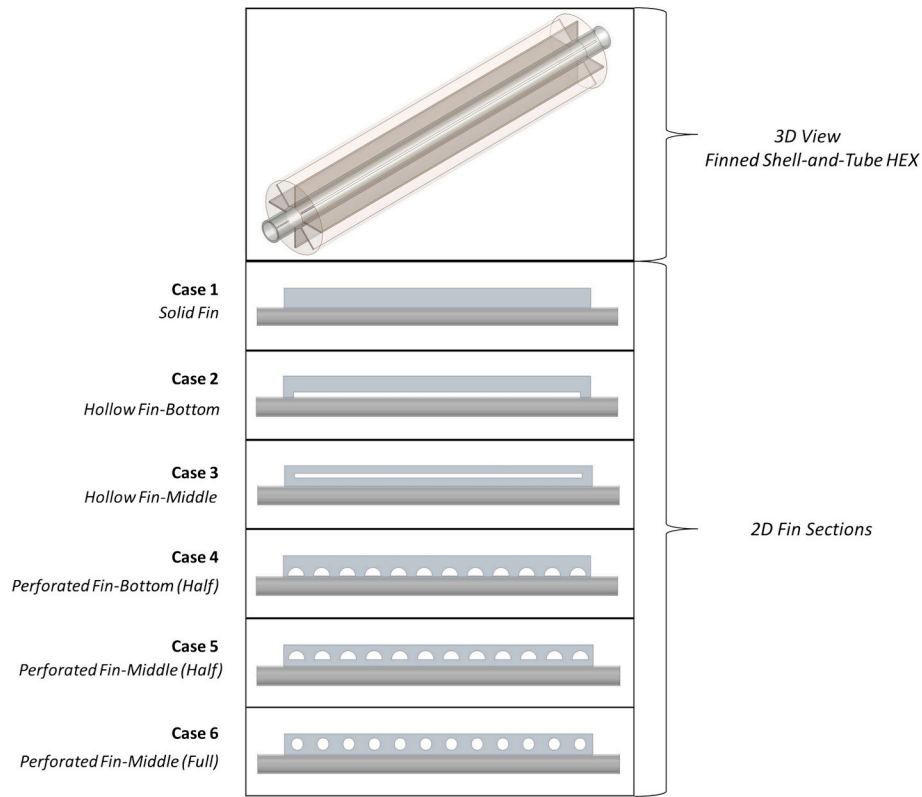


Fig. 3. 3D view of finned shell-and-tube HEX and 2D section views of solid, hollowed, and perforated fins.

regions. When this hollow region is centered along the midline of the solid fin, as shown in Case 3, the fin remains in full contact with the hot tube surface. In Case 4, semicircular perforations were introduced into the fins.

Each fin contains 12 semicircular perforations with radii of 15.3 mm. Since the perforations are located in the lower region of the fin, the contact surface is reduced to 228.8 mm—which is still approximately six times greater than in Case 2. Similarly, in Case 5, semicircular perforations are centered along the midline of the solid fin. Case 6 features fully circular perforations, with 12 holes per fin and a radius of 11 mm for each hole.

Despite differences in shape, size, and location of the holes, the total material removal ratio was maintained at 20 % for Cases 2–6. Alternative removal ratios for selected best fins configurations are discussed in Section 3.

With the recent developments in next-generation CSP systems, operation temperature can reach up to 1000 °C [37]. Under such high-temperature conditions, the use of conventional fluids and structural materials becomes impractical due to issues such as phase separation, increased corrosiveness, higher costs, and reduced durability. Consequently, the design of shell-and-tube latent heat storage (LHS) units must account for these challenging conditions. Regarding the selection of HTFs, the study of Niedermeier [38] provides a comprehensive comparison of liquid metals that can be proposed priorly due to high thermal diffusivity and durability skills at high temperatures. Among all, liquid sodium is selected due to comparatively better thermal conductivity, low density, and cost-effectiveness.

On the other hand, commonly used molten salts (NaNO_3 , KNO_3 , or solar salt) significantly corrode Ni-based alloys, similar to the corrosion caused by liquid metals, and their decomposition rate increases at temperatures of 565 °C and beyond [39,40]. Nevertheless, few salt mixtures exist that are able to work at temperatures exceeding 700 °C. One such candidate is FLiNaK (LiF-NaF-KF, 46.5–11.5–42 mol%), which was selected as an alternative HTF to liquid metals due to its specific

heat capacity, which is approximately 1.5 times higher than that of metals. In addition, Inconel alloy 617 is employed as the shell and pipe material due to its excellent resistance to corrosion and oxidation, attributed to its composition of chromium, molybdenum, nickel, and cobalt [41]. The thermophysical properties of both HTF and structural material are summarized in Table 3. Similar to the selection of HTFs from a thermal performance perspective, MgCl_2 is chosen since it is a promising candidate due to its high latent heat of fusion and has been extensively studied both numerically and experimentally [42–44]. The thermophysical properties of the PCM is presented in Table 4.

2.2. Assumptions and governing equations

Transient heat transfer from HTF to PCM cavity through the solid domain, fluid flow inside the tube and buoyancy-driven convection circulations of the HTF, and melting/solidification of the PCM are simulated via ANSYS 2024R2 in a 3D computational domain. The software employs the enthalpy-porosity technique which implicitly tracks the solid–liquid front considering buoyancy force activeness and enthalpy equilibrium during the phase change process [46]. For the aim of simplifying the melting/solidification computation steps, the following assumptions are applied for setting up the numerical model:

- Both HTF flow inside the tube and circulation flow of PCM in liquid phase are unsteady and in laminar regime,
- Buoyancy effect due to density variation in melted PCM via temperature gradient is modeled by governing Boussinesq approximation,
- Isothermal phase change conditions exist during melting/solidification processes,
- Radiation, viscous dissipation and volumetric expansion of PCM throughout the charging and discharging processes are neglected.

Radiative heat transfer makes its presence felt more and more while

Table 3

Thermophysical properties of Liquid Sodium (liquid metal) and FLiNaK (molten salt) as HTF, and Inconel alloy 617 used as the structural material for tube, shell, and fins.

HTFs	Density (kg/m ³)	Thermal Conductivity (W/mK)	Specific heat (J/kgK)	Dynamic viscosity (Pas)	Source
Liquid Sodium	966.74–0.1274874 T- 5.38*10 ⁻⁵ T ²	98.91–0.046 T-3.714x10 ⁻⁶ T ²	1380	0.103 T ^{0.9103}	[45]
FLiNaK	2018.9	0.921	1890	0.0029	[43]
Structure Material	Density (kg/m ³)	Thermal Conductivity (W/mK)	Specific heat (J/kgK)	T _{melting} (°C)	Source
Inconel alloy 617	8360	24.2	586	1332–1380	[43]

Table 4

Thermophysical properties of MgCl₂ as PCM.

PCM	Melting point (°C)	Latent heat of fusion (kJ/kg)	Thermal Conductivity (W/mK)	Density (kg/m ³)	Specific heat (kJ/kgK)	Dynamic viscosity (Pas)	Source
MgCl ₂	714	452	0.46 (at 714 °C)	2320 (at 25 °C), 1680 (at 714 °C)	0.7602 + 1.6567 10 ⁻⁴ *T (absolute temp.)	0.0032	[44]

elevating the operating temperature to high levels, and PCM can emit/absorb radiation. However, the study of Shen et al. [47] uncovers that uniform distribution of fins inside the PCM domain makes conduction dominant over radiation and its influence becomes weak in a high-conductive environment.

Following these simplifications and activation of ‘Solidification/Melting Module’, conservation equations of mass, momentum, and energy are updated for the 3D domain as below Eqs. (1–4):

$$(Mass) \frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \mathbf{u}) = 0 \quad (1)$$

$$(X \text{ Momentum}) \frac{\partial (\rho u)}{\partial t} + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) = \nabla \cdot (\mu \nabla u) \quad \frac{\partial P}{\partial x} + A u \quad (2)$$

$$(Y \text{ Momentum}) \frac{\partial (\rho v)}{\partial t} + \nabla \cdot (\rho \mathbf{u} \mathbf{v}) = \nabla \cdot (\mu \nabla v) \quad \frac{\partial P}{\partial y} + A v + S_{buoyancy} \quad (3)$$

$$(Z \text{ Momentum}) \frac{\partial (\rho w)}{\partial t} + \nabla \cdot (\rho \mathbf{u} \mathbf{w}) = \nabla \cdot (\mu \nabla w) \quad \frac{\partial P}{\partial z} + A w \quad (4)$$

Here, the velocity vector, $\mathbf{u}$, has u , v , and w velocity components in x , y , and z directions, respectively. ρ and μ are the density and dynamic viscosity of material; P and T terms represent pressure and temperature, respectively.

A temperature change in melted PCM creates a density gradient and buoyancy forces, causing the melted PCM to move upward or downward, together with gravitational force in Eq. (5).

$$S_{buoyancy} = \rho_{ref} g \beta (T - T_{ref}) \quad (5)$$

The terms g and β terms represent gravitational acceleration and coefficient of volumetric thermal expansion, respectively.

The distinction between the present governing equations and the general form of the Navier-Stokes equations lies in the momentum dissipation terms. The porosity function denoted by ‘ A ’ is added to the momentum equations as a multiplication factor to diminish the velocity while melted PCM solidifies. As formulated in Eq. (6), the liquid fraction term lf corresponds to the porosity term in the Carman-Kozeny relation. When full solidification occurs, this term equals zero, resulting in momentum dissipation. The default small number b is used to prevent division by zero. The “mushy zone constant”, with a default value of 10⁵ kg/(m³.s) in Fluent, governs the evolution of interface morphology and the extent of the phase change process in Eq. (6).

$$A = C \frac{(1 - lf)^2}{lf^3 + b} \quad (6)$$

The energy equation employs the enthalpy instead of temperature term since it accounts for the changes in both sensible and latent heat storage

in a single equation. One consequence of heat transfer is the temperature rise of the PCM either in the liquid or solid phase. The second consequence is the solid–liquid phase change, during which no dramatic temperature changes are shown even if heat absorption/release continues. This is because, as shown in Eq. (7), the effects of temperature rise and phase change processes are represented in the energy equation by sensible and latent enthalpies, respectively.

$$\frac{\partial (\rho h_{sens})}{\partial t} + \nabla \cdot (\rho \mathbf{u} h_{sens}) = \nabla \cdot (k \nabla T) + S_{ph} \quad (7)$$

Here, the energy source term (Eq. (8)), S_{ph} is included for establishing the dynamics of melting/solidification when PCM reaches the phase change temperature.

$$S_{ph} = \frac{\partial (\rho h_{lat})}{\partial t} + \nabla \cdot (\rho \mathbf{u} h_{lat}) \quad (8)$$

The summation of sensible heat and latent heat enthalpies gives the total enthalpy as in Eq. (9).

$$h_{tot} = h_{sens} + h_{lat} = \int_{T_{ref}}^T c_p dT + Hlf \quad (9)$$

The stored heat is calculated as stated in Eq. (10) by subtracting the initial enthalpy from the final enthalpy of the PCM after the same melting process duration. Initially, all cases have the same initial enthalpy values due to the uniform temperature distribution across all domains until the melting process starts. Similarly in Eq. (10), for the released heat enthalpy, the initial enthalpy values are the same before the solidification process begins, as the temperature distribution is uniform. The released heat is then determined by subtracting the final enthalpy from the initial enthalpy of the PCM for all fin configurations.

$$h_{stored} = h_{final} - h_{initial} \text{ and } h_{released} = h_{initial} - h_{final} \quad (10)$$

The heat absorption until solidus temperature results in sensible heating with temperature increase. Further charging in the solidus to liquidus temperature range, heat transfer initiates the melting process while simultaneously contributing to sensible heat storage. Once the PCM is fully melted, additional heat transfer raises the temperature and increases the sensible heat content of the storage domain. Eq. (11) provides a method for calculating enthalpy based on the relationship between instantaneous PCM temperature and phase change temperatures.

$$h = \begin{cases} \int_{T_{ref}}^{T_s} c_p dT & \text{if } T < T_s \\ \int_{T_s}^{T_l} c_p dT + L & \text{if } T_s < T < T_l \\ \int_{T_l}^T c_p dT & \text{if } T_l < T \end{cases} \quad (11)$$

The simulations were conducted at the High Performance Computer (HPC) center, bWUnicluster, of Karlsruhe Institute of Technology (Germany), using 4 nodes x 40 CPUs.

2.3. Boundary conditions and numerical scheme

As demonstrated in Fig. 2, the physical domain of shell-and-tube LHS unit consists of fluid, tube, PCM, and shell subdomains. This study investigates the melting and solidification processes of $MgCl_2$ with melting temperatures of 714 °C. During both processes, all subdomains are assumed to have a uniform temperature distribution, and the temperature difference between PCM melting temperature and uniform initial temperature, $|T_{melting} - T_{initial}|$, is maintained at 24 K. Hot or cold HTF flows inside the tube, which initiates the heat transfer between fluid and PCM through the tube and fins for melting and solidification processes, respectively. The temperature difference between HTF inlet temperature and PCM melting temperature, $|T_{inlet} - T_{melting}|$, is set at 36 K for each process. Detailed initial and boundary conditions are presented in Table 5.

Fins are attached to the tube surface, facilitating conjugate heat transfer between the HTF, finned tube, and PCM domains. The shell is assumed to be well-insulated, with an adiabatic boundary condition at its outermost surface. However, the shell touches the HTF tube at the inlet and outlet sides, therefore, it is heated, stores thermal energy in sensible form, and subsequently transfers heat to the PCM.

SIMPLE algorithm is used to couple pressure and velocity for HTF flow and PCM convection circulations and PRESTO is selected as pressure correction scheme which is recommended for buoyancy driven [48]. The first-order upwind scheme is utilized for momentum and energy equations and convergence criteria are set to 10^{-5} for the conservation of mass and momentum equations, and 10^{-10} for the conservation of energy equations, ensuring the attainment of accurate solutions.

2.4. Dependency tests and validation

Fig. 4 illustrates the representative computational grid of the finned shell-and-tube LHS unit generated via the ANSYS meshing tool. While PCM, shell, tube, and fluid subdomains are meshed with hexahedral mesh type, extended fins are predominantly formed via tetrahedral mesh type due to the complexities in fin geometry and joint surfaces on HTF tube.

Mesh and time independency studies are tested for unfinned base case. Three different grid sizes with 1,148,881, 856,910, and 610,018 elements are compared based on the melting rate over 8-hour charging process. As shown in Fig. 5a, the distinction starts to appear after about 300 min. The maximum difference between fine and base meshes occurs at the end of melting process and is only 1 %. Additionally, the

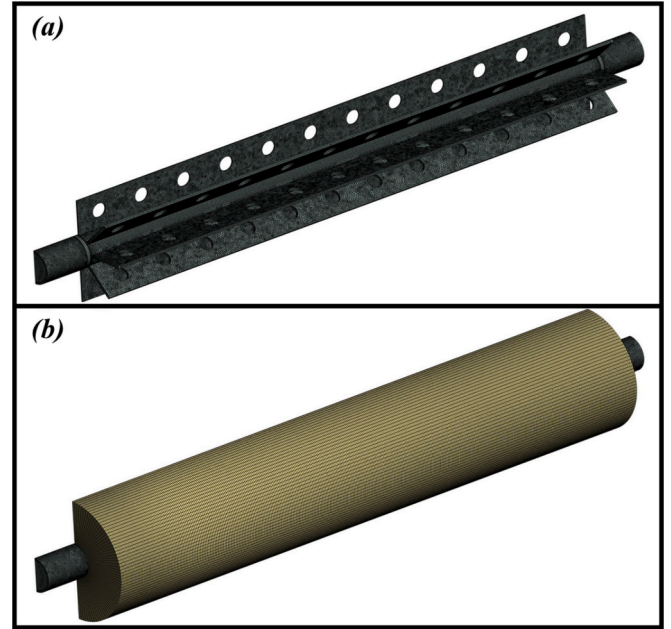


Fig. 4. Mesh structure of (a) finned tube and (b) shell parts of HEX.

numerical model demonstrated independence from time step size. Fig. 5b shows consistent outputs of the CFD model using time steps of 0.05 s, 0.1 s, and 0.15 s. Therefore, mesh density with 856,190 elements and time step size of 0.1 s is chosen as the reference model setup to the finned configurations. The numerical model, with the current solution setup, requires approximately 8–10 iterations per 0.1 s time step to satisfy the established convergence criteria after the first seconds of the melting process. As for solidification, the model reaches convergence within 4 iterations per unit time step. This difference in the required iteration between the processes stems from natural convection dominating for a shorter period during solidification.

The validation of the current numerical model is carried out by remodeling the well-documented study of Parry et al. [49] that reports melting and solidification processes for 8 h both experimentally and numerically. Identical geometric parameters and operation conditions are implemented for repeating the charging process. The authors use “Frozen Flux Formulation” in the 3D numerical domain, which takes the mass flux of previous time steps to suppress the nonlinear terms in momentum equations, thereby reducing the number of iterations per time step and accelerating convergence.

The model is recreated using identical geometric parameters (shell tube diameter of 152.4 mm, HTF tube diameter of 54 mm, HTF tube length of 1 m, and insulation thickness of 100 mm) and operating conditions (HTF inlet temperature of 140 °C, initial temperature of 25 °C, HTF mass flow rate of 30 kg/min, and a heat transfer coefficient of 10 $W/m^2 \cdot K$ at the external surface of the insulation) for validation. During the charging process, industrial heat transfer oil (HTF) flows through the inner tube. As the process progresses, the phase change material (PCM) in the annular region between the HTF tube and the PCM container wall begins to melt. Fig. 6 demonstrates the agreement between the results obtained from both approaches concerning PCM temperature values. However, even if the output of the Frozen Flux Formulation matches well with average temperature data, significant differences in temperature distributions were observed, as highlighted in the contour plots from previous comparisons [50].

Table 5

Initial and boundary conditions for the melting and solidification processes of $MgCl_2$.

$MgCl_2$			
Initial and Boundary Conditions for Melting (°C)		Initial and Boundary Conditions for Solidification (°C)	
$T_{initial}$	690	$T_{initial}$	738
$T_{melting}$	714	$T_{melting}$	714
T_{inlet}	750	T_{inlet}	678

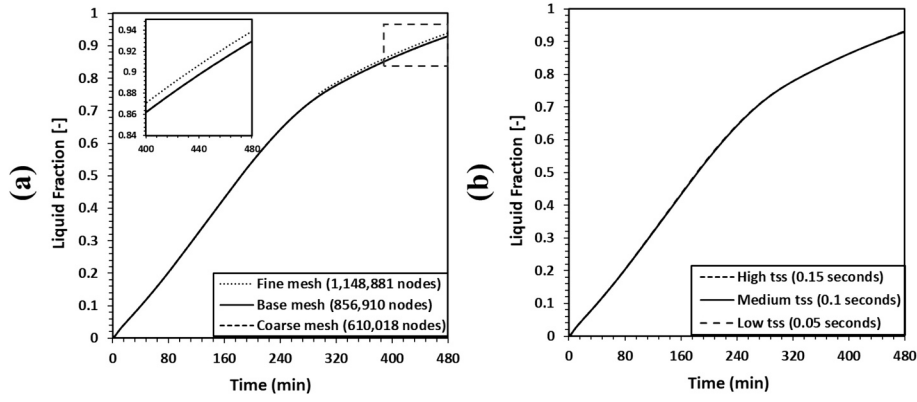


Fig. 5. (a) Mesh independency and (b) Time independency tests over variation of liquid fraction in time.

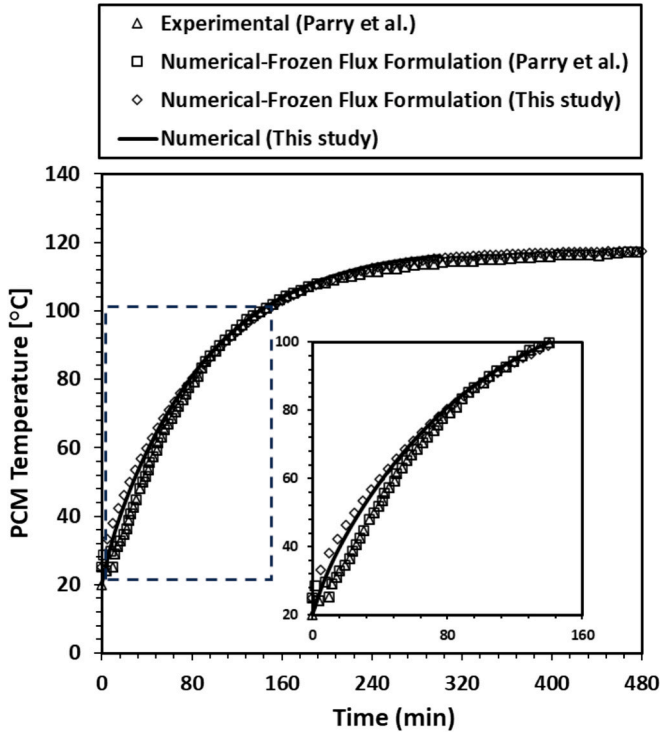


Fig. 6. Validation of present numerical model over comparing numerical/experimental data of Parry et al. [49].

3. Results and Discussions

3.1. Effects of mass flow Rate: Molten salt vs. Liquid metal

Firstly, the influence of different HTFs on the melting process was

analysed. Fig. 7 demonstrates liquid fraction contour graphs at the midsection of the three-dimensionally modeled shell-and-tube LHS unit using liquid sodium as HTF and MgCl_2 as PCM. The melting front progress of MgCl_2 resembles that of low-temperature PCMs. Initially, all subdomains are subcooled to 690°C , which is 24°K below melting temperature of MgCl_2 , and the melting process begins with the pumping of hot liquid sodium at 750°C . The charging process is simulated firstly employing liquid sodium as HTF for 8 h. Non-uniformity in the melting front becomes noticeable after 30 min due to the effects of buoyancy forces at the upper part of the tube. The strength of buoyant flow generates powerful natural convection in the upward direction, causing the top region to melt significantly faster. In contrast, the lower part of the tube has a slow-paced progress due to the absence of convection. Although the primary objective of heat transfer is to achieve uniform and rapid melting of the PCM domain, conduction-dominated melting in the bottom region leads to an asymmetric phase change process. The increasing thickness of the melted PCM gradually acts as an insulating layer between the tube and the solid-liquid interface.

As shown in Fig. 8, the liquid fraction increases from only 82 % to 93 % during the final 120 min, which corresponds to one-quarter of the total charging time. This observation highlights the diminishing efficiency of heat transfer as the process near completion. The 8-hour charging process for the base case is repeated for different HTFs and Re numbers to investigate the effects of HTF velocity and thermofluid types. Fig. 8 demonstrates the liquid fraction outputs of distinct flow conditions. If the Reynolds (Re) number is changed from 1000 to 1500, the liquid fraction after 360 min considerably rises from 93 % to 97 %, but a further increase of the Re number to 1800 causes only 1 % improvement at the end of melting process. Moreover, the comparison of liquid sodium (a liquid metal as HTF) and the FLiNaK (a molten salt as HTF) show no difference in liquid fraction performance. The outputs indicate that neither liquid sodium with various Re numbers nor the selection of molten salt instead of liquid sodium result in substantial changes in the liquid fraction of the PCM.

Transient heat transfer performance with changing conditions is more clearly illustrated by variations in heat flux trends. According to

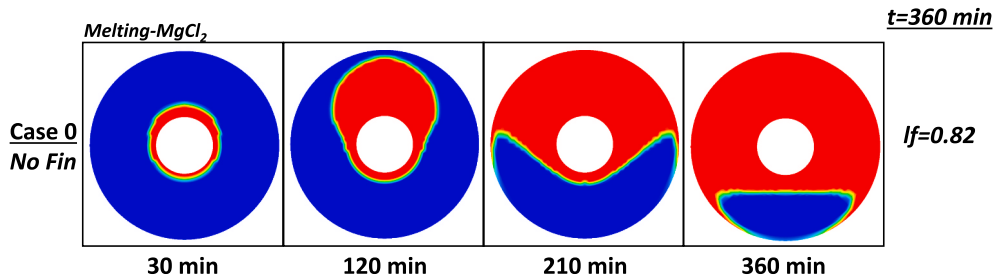


Fig. 7. Contours of liquid fraction during melting process using sodium as HTF and MgCl_2 as PCM without fins. If liquid fraction after 360 min.

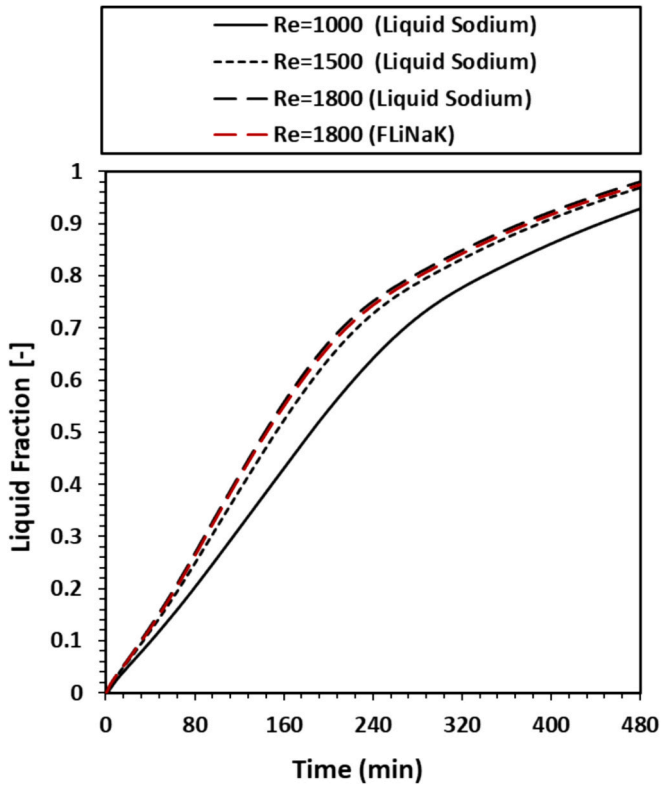


Fig. 8. Liquid fraction of MgCl_2 as PCM during the melting process using different HTFs and Re numbers.

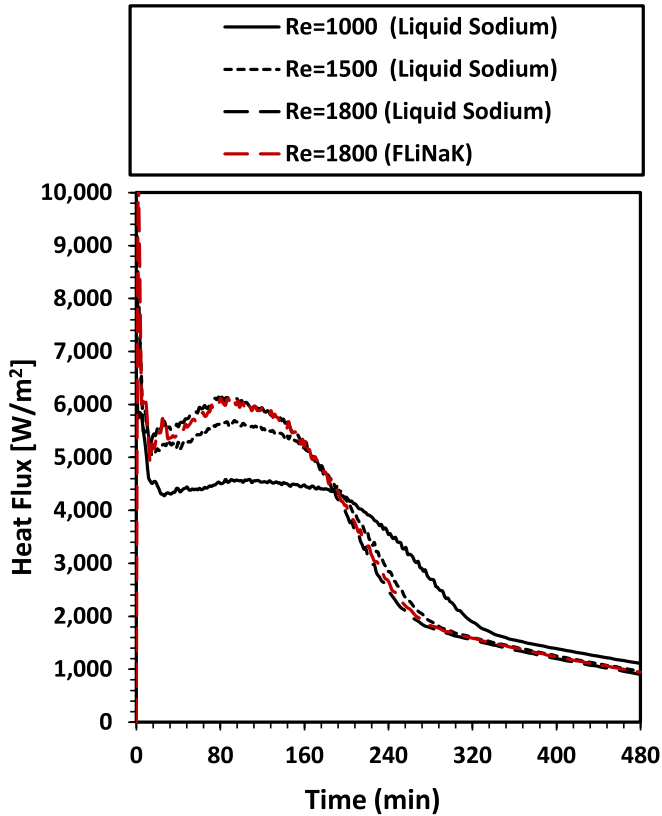


Fig. 9. Heat flux during the melting process of MgCl_2 as PCM using different HTFs and Re numbers.

Fig. 9, heat flux values are exceptionally high at the beginning of the melting process due to the greatest difference between PCM initial temperature and HTF inlet temperature. During this early stage, the PCM absorbs a large amount of thermal energy carried by the HTF, but heat flux dramatically decreases as the thickness of the melted PCM increases. Natural convection compensates for the reduction occurring in conduction heat transfer due to thermal resistance, and it stays almost constant in the condition with $\text{Re} = 1000$ during about half of the melting process. In contrast, heat flux values continue to rise in conditions with greater Re number until no solid PCM remained above the HTF tube. Beyond this point, heat transfer performance declines since the low thermal conductivity of MgCl_2 inhibits melting at the bottom region.

The objective of developing a high-temperature shell-and-tube LHS unit in this work is not only to enhance heat transfer and maximize total stored thermal energy but also to reduce the HTF outlet temperature to meet the requirements of the hybridized sensible and latent heat storage system discussed in previous sections. Fig. 10 presents the main discrepancy between the two different HTFs selected. When molten salt flows inside the tube, the HTF outlet temperature rises suddenly in the first minutes. After that, it remains nearly constant and very close to the HTF inlet temperature (750°C) throughout the charging process. On the other side, for liquid sodium under all Re number cases, the outlet temperature remains stable between 725°C and 735°C until approximately 200 min. After this time point, as heat flux values decrease, outlet temperature of liquid sodium oppositely rises. This behavior can be attributed to the specific heat capacity ($\rho \cdot c_p$) of FLiNaK, which is 3.3 times higher than that of liquid sodium. Even when both fluids absorb the same amount of thermal energy, the temperature of molten salt remains relatively unchanged, whereas the outlet temperature of liquid sodium is significantly affected after it releases its thermal content.

Furthermore, Fig. 9 reveals that the HTF with the lowest Re number has the lowest outlet temperatures since it allows for more time for heat transfer interaction with the storage medium. After 200 min, a rising

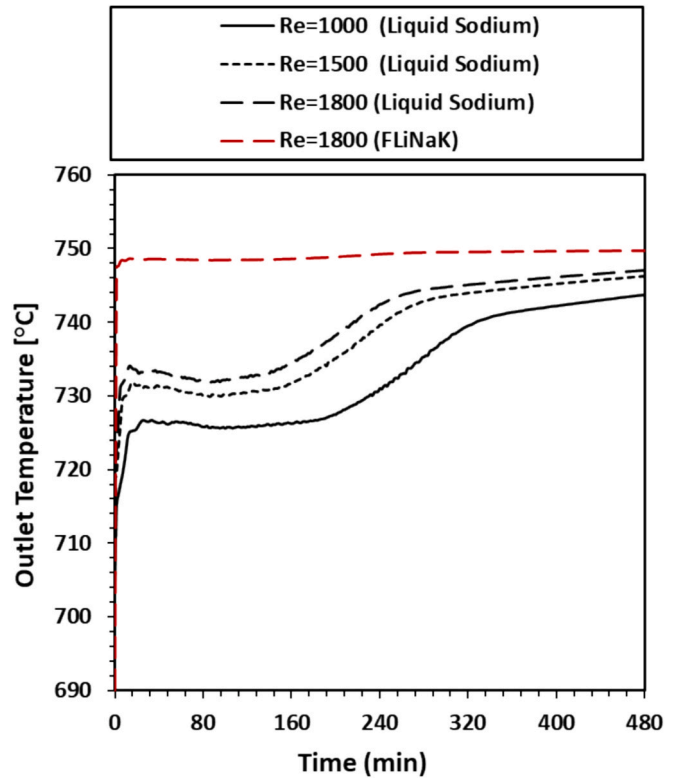


Fig. 10. Outlet temperature of the HTFs during the melting process of MgCl_2 as PCM using different HTFs and Re numbers.

trend in outlet temperatures is observed due to decreasing heat fluxes caused by the increasing thermal resistance from the growing thickness of the melted PCM layer. Ultimately, the results suggest that liquid sodium with a low Re number is preferable when considering thermal storage performance, and heat transfer effectiveness together.

3.2. Influence of perforated fins

3.2.1. Melting process

Firstly, the effect of using perforated fins on the melting process is investigated. For that, liquid sodium is being used as HTF and MgCl_2 as PCM. Fig. 11 and Fig. 12, in which fin removal rates are 20 % and 10 %, respectively, demonstrate liquid fraction contour graphs at the midsection of the three-dimensional modeled shell-and-tube LHS unit. Since the midsection views coincide with perforated and hollowed regions of the fins, a discontinuity exists depending on hole geometries except in Case 1. In these regions, PCM fills these holes, whereas other section views have solid straight fin structures. The solid fin configuration reaches fully-melted status in 6 h, while an 8-hour charging period remains insufficient for the *No-fin* case (see Fig. 7). Therefore, melting results are compared based on the 6-hour melting performance. The addition of fins accelerates the melting process by over 2 h due to faster conductive heat transfer compared with bare tube configuration. However, the 8.6 % volumetric occupation rate of fins reduces the available PCM volume, hinders natural convection circulation, and increases material costs.

In the solid fin case, melting begins around the HTF tube and bottom points of the fins, progressing along the fin penetration length over time. Although the existence of metal fins raises the influence of heat

conduction inside the PCM domain, the effect of circulations makes the melting front asymmetric, with the upper region the melting completely before 210 min. According to Fig. 13, the best melting performance belongs to Case 1. Despite having the same penetration depth as other fin configurations, Case 2 performs poorly, almost resembling the scenario without fins. This behavior is linked to the hollowed region at the base of the fin, which limits conductive heat transfer. In the first stages of the melting process, the liquid fraction in Case 2 matches that of other cases. However, once the PCM in the hollowed gap melts, it stops acting as a heat sink. This creates a thermal barrier between the hot tube surface and fins. In Case 3, relocating the hollow region from the bottom to the middle of the fins improves performance. The liquid fraction increases from 86 % to 93 % after 360 min. Liquid fraction contours at 210 min and 360 min reveal noticeable differences in the remaining solid PCM and it highlights the importance of hole location under identical removal rate conditions. Alternatively, perforated fins with semicircular holes are presented in Case 4. Despite the perforations at the bottom, the fins maintain contact with the HTF tube due to the interval arrangement of the semicircular holes. The comparison of Case 2 and 4, with corresponding liquid fraction values of 86 % and 96 % respectively, proves that hole shape has a significant influence on melting performance. In Case 5, relocating the perforations yields improvements similar to those observed with hollow fins. Further enhancement occurs in Case 6, where circular perforations with smaller diameters are used instead of semicircular ones at the middle location. This improvement is due to less interruption in the fin structure, which enhances radial conduction.

When the present findings are compared with the existing literature summarized in Table 1, most researchers (except for Karami and Kamkari[21]) report either negligible gains or no improvement at all when

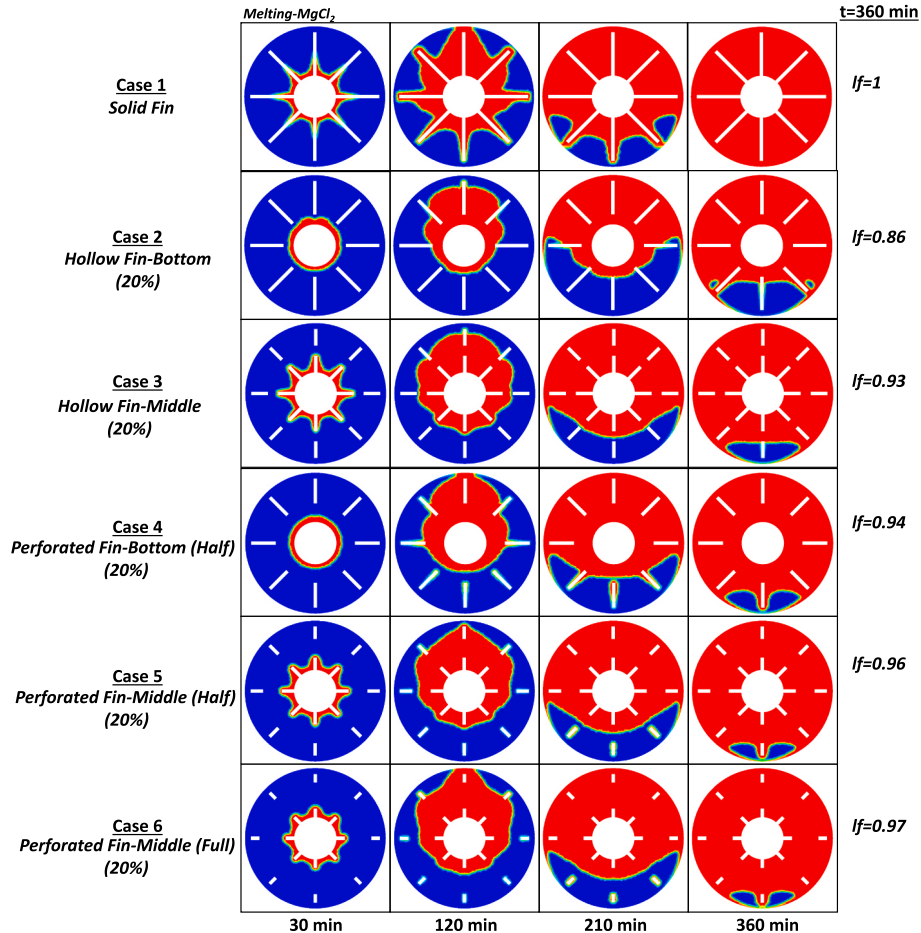


Fig. 11. Contours of liquid fraction during the melting process with hollowed and perforated fins (removal rate 20 %).

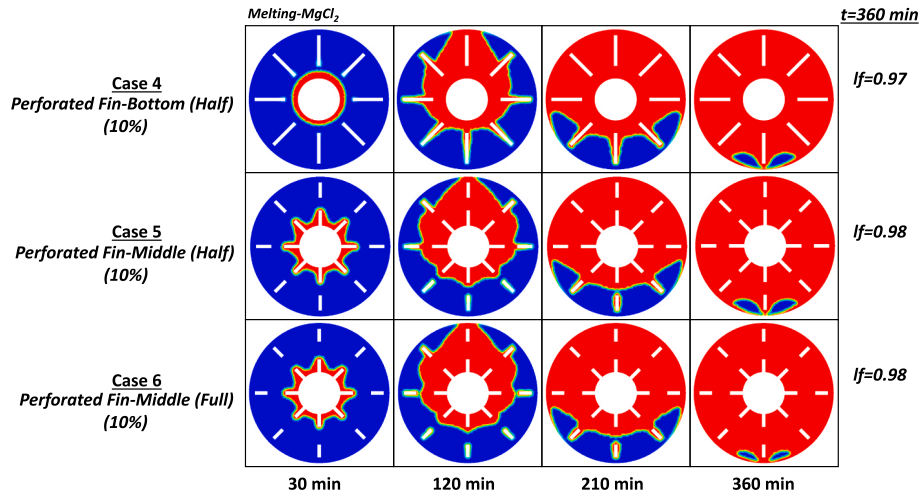


Fig. 12. Contours of liquid fraction of selected cases during the melting process with hollowed and perforated fins (removal rate 10 %).

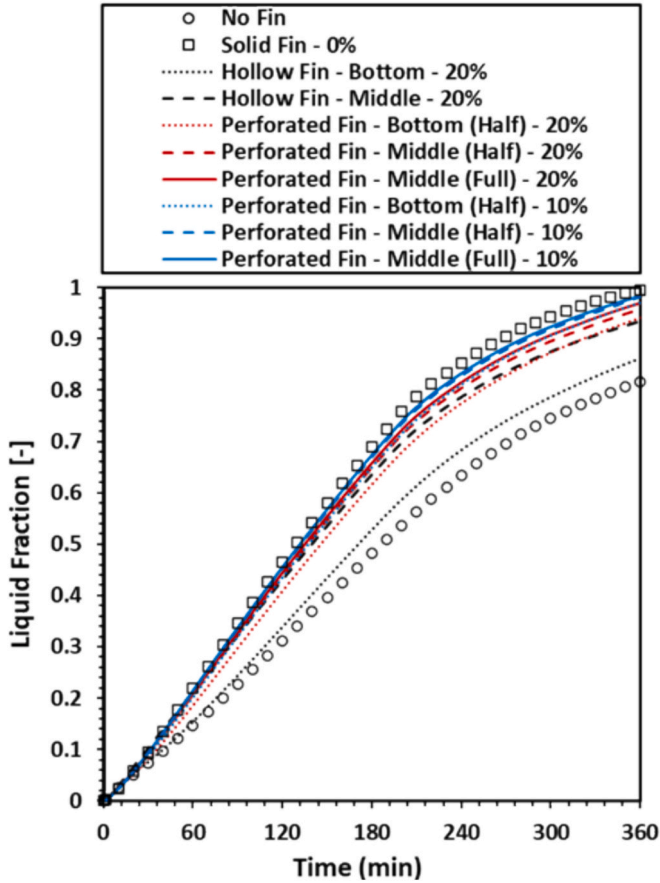


Fig. 13. Liquid fraction variation during the melting process with hollowed and perforated fins.

using low perforation rates of 2–3.5 %. Such small removal rates do not meaningfully reduce fin material use or increase PCM volume. Further, they may also result in high fabrication costs. Therefore, the present study endeavors to analyze the 10 % fin removal rate and compares it with a 20 % removal rate for the best fin configurations, namely Cases 4 to 6. As shown in Fig. 12, comparative examination of the liquid fraction uncovers that the melting front progresses slightly faster at both the upper and lower parts when the fin removal rate is reduced to 10 %. However, the performance remains only marginally below that of the

solid fin case.

Fig. 13 shows that melting performances of perforated fins with different removal rates are almost the same with solid fin case, yet using less fins. From a financial perspective, the reduction in material costs is significant. Another critical aspect is the net stored thermal energy during the charging process.

Fig. 14 presents the net charged thermal energy in a 6-hour melting time duration in kJ unit. To enable a more meaningful comparison, initial stored heat in sensible form is subtracted from the total stored heat at the end of the process for each case as stated in Eq. (10). In the *No-fin* case, 5833 kJ thermal energy is stored, but it is important to note that only 82 % of PCM is melted in 6 h. Interestingly, as seen from Fig. 14, the *Hollow fin-Bottom* (Case 2) configuration exhibits lower storage performance, with a melting rate of 86 %. This is caused by impaired conduction from the hollow base and reduced PCM volume due to fin insertion. Subsequently, the *Hollow Fin-Middle* (Case 3) performs better than Case 2 but remains significantly below the performance of the solid fin configuration. Among perforated fin configurations, Case 6 shows better results with regard to the stored heat, but reducing the removal rate from 20 % to 10 % does not lead to a significant increase in stored energy. Another important output is that the location of perforation holes becomes less important if the removal rate is reduced to 10 %.

The best performance is achieved by *Solid Fin* configuration (Case 1) with 6488 kJ of thermal energy. This is followed by the *Perforated-Middle (Full)* Case 6 with charged enthalpy values of 6406 kJ and 6453 kJ for 20 % and 10 % removal rates, respectively. These configurations exhibit only 1.25 % and 0.53 % performance losses compared to the solid fin case, while achieving significant fin material reductions of 20 % and 10 %, respectively.

Initially, all subdomains have a temperature of 690 °C, and the melting process starts with the flow of liquid sodium into the tube at 750 °C. A greater temperature difference between PCM's initial temperature and HTF's inlet temperature causes a lower outlet temperature in the first half of the melting process. As shown in Fig. 15, the HTF exits the shell-and-tube LHS unit at a lower temperature when fins are included. However, it is noteworthy that variations in fin configurations do not lead to significant differences in outlet temperatures among themselves. Moreover, in the second half of the process, the finned cases are unable to maintain constant outlet temperatures and perform worse than the *No-fin* case. Fig. 15 clearly demonstrates that the addition of fins is insufficient to reduce the outlet temperature to the extent required for HTF that will subsequently be stored in a sensible heat storage system. Based on the results in Fig. 15, it is recommended to use a cascaded LHS system. By selecting PCMs with decreasing phase change

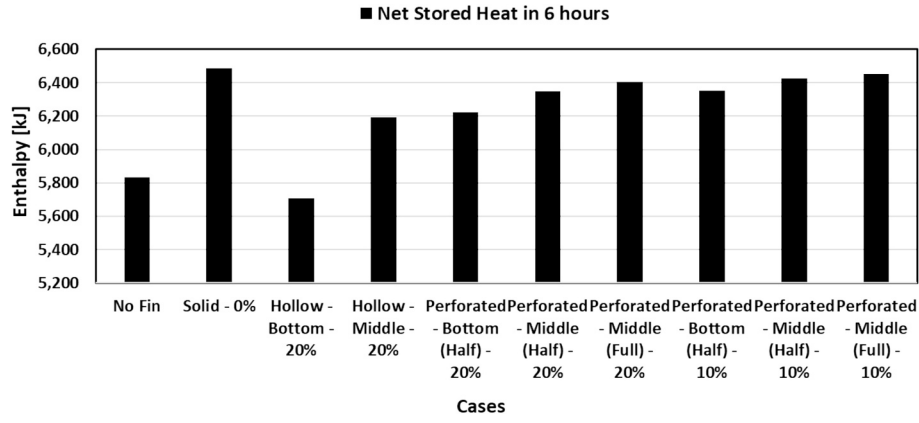


Fig. 14. Net stored energy values for each case at the end of the 6-hours melting process.

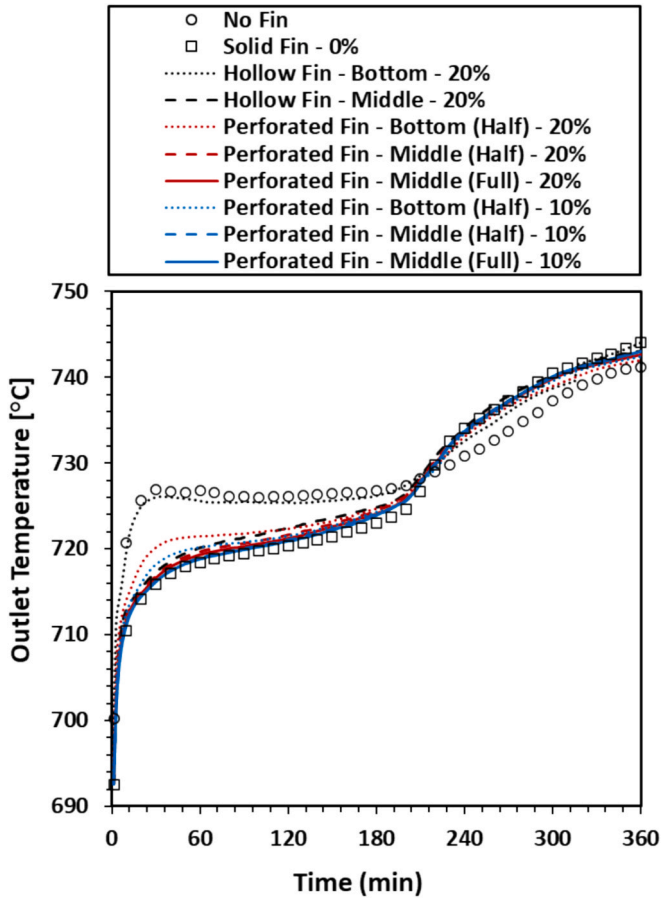


Fig. 15. Outlet temperature of the HTF during the melting process with hollowed and perforated fins.

temperatures, the outlet temperature can be brought down to the desired level more effectively. However, it is essential to ensure compatibility between the melting temperatures of the selected PCMs and the outlet temperature of the HTF exiting the preceding LHS unit.

3.2.2. Solidification process

Figs. 16 and 17 show the evolution of liquid fraction during the solidification process at the midsection of the LHS unit with fin removal rates of 20 % and 10 %, respectively. Initially, all subdomains have a uniform temperature distribution at 738 °C. Cold liquid sodium flows into the tube at 638 °C, creating a 60 K temperature difference between

the PCM and HTF subdomains. A 7-hour discharging time is applied, and the comparisons are made within this period for all solidification processes, as the best performance case reaches fully-solidified condition in 420 min.

In the *No-fin* case, solidification begins around the cold tube surface in a non-uniform pattern. The liquid PCM, whose density increases as its temperature decreases, accumulates below the tube surface and solidifies first due to natural convection. Then, the unsymmetrical solidification front diminishes in the following discharging process because the strength of convection decreases in the remaining time. By the end of the solidification process, only 47 % of the liquid PCM has solidified, whereas 87 % of the PCM melts during the 7-hour charging process (section 3.2.1). This discrepancy arises because natural convection has a different influence during the melting and solidification processes. In the best performing case, Case 1 with solid fins, PCM totally solidifies. Due to the dominant influence of high-conductivity fins, convection effects are not observed during the early stages of the discharging process. However, when comparing at the liquid fraction at 360 min, the share of unsolidified PCM at the bottom region is slightly less than that of the top region. This effect occurs because high-conductivity fins are surrounded by solidified, low-conductivity PCM. In this setup, natural convection helps transfer heat from the remaining liquid PCM to the colder surfaces at the bottom. This behavior is evident across all fin configurations.

Similarly, Case 2 exhibits the worst performance during discharging, achieving only 62 % solidification. Due to the minor influence of natural convection, it is expected that the conduction-dominant solidification process will suffer from removing high conductive fins. However, a long continuous gap between the fin and the cold tube surface along the axial direction is the least desired design option. This observation is supported by the comparative performance difference between Case 1 and Case 2, which is 14 % during the melting process and 38 % during the solidification process. A similar comparison between Case 2 and Case 3 reveals the impact of changing the hollowed gap location, as highlighted in Fig. 18. While the difference is approximately 8 % during the melting process, it increases to approximately 38 % in solidification process. This finding underscores the critical importance of maintaining contact with hot and cold surfaces. Perforated fin designs also improve solidification performance compared to hollowed fins. Middle-perforated fins (Case 5) outperform bottom-perforated fins (Case 4). Nevertheless, the solid fin configuration (Case 1) demonstrates the best performance. As shown in Figs. 16 and 18 at 120 min, solidification trends diverge, with solidification around the fins occurring exclusively in Case 1. This superior performance is attributed to the uninterrupted conduction heat transfer along the fins, which is not impeded by perforations.

To further investigate the performance of solid fin configuration, the solidification of perforated fin cases is tested with a 10 % fin removal rate. The greatest improvement was observed in Case 4. In this configuration, reducing the gap thickness allowed the fins to behave similarly

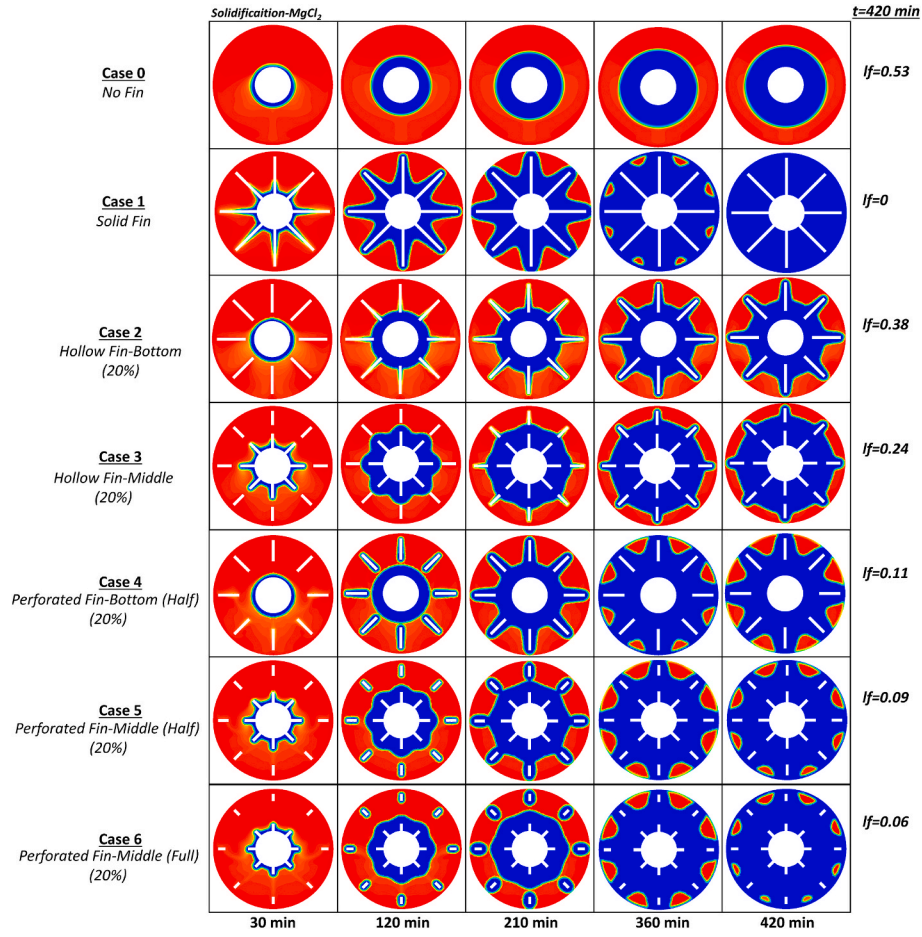


Fig. 16. Contours of liquid fraction during the solidification process with hollowed and perforated fins and without fins. (Removal rate 20 %).

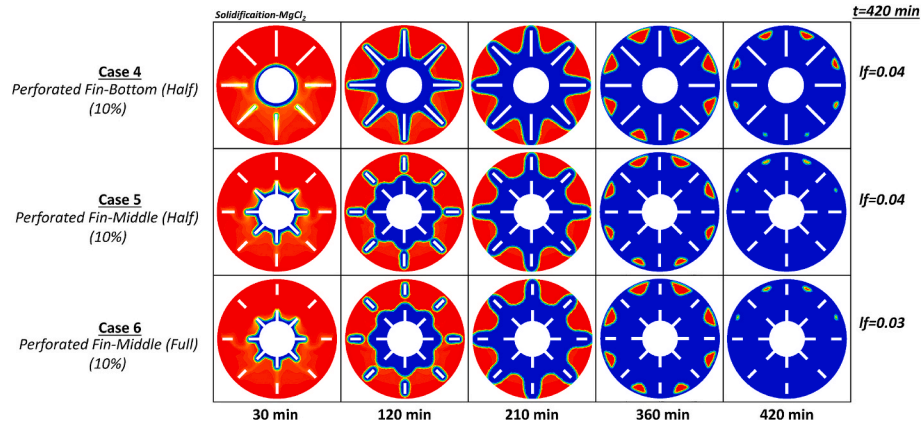


Fig. 17. Contours of liquid fraction of selected cases during solidification process with hollowed and perforated fins (removal rate 10 %).

to solid fins, improving solidification. Among the perforated fin cases, Case 6 performed best, achieving nearly full solidification by the end of 420 min. Natural convection clearly contributed to this outcome. In particular, the bottom region solidified completely due to convection, although 3 % of the PCM at the top remained unsolidified.

To provide a comprehensive comparison of both melting and solidification processes, Fig. 19 presents the charged and discharged enthalpy values for 6 h and 7 h, corresponding to the completion times of both processes for the solid fin configuration (Case 1). The final heat content was subtracted from the initial enthalpy as stated in Eq. (10) to ensure accurate comparisons. A similar trend was observed between charged

and discharged enthalpy values. As seen in Fig. 19, the *No-fin* and hollow fin cases performed poorly compared to other solid and perforated fin configurations, for reasons previously discussed. One important point is the large variation in charged and discharged enthalpy between the cases. This highlights the extent to which natural convection is suppressed or enabled, depending on the design. The percentage values with dark blue color in Fig. 19 represent the ratio of this difference to the discharged enthalpy, so the enthalpy difference between net stored enthalpy in 6 h during the melting process and net released enthalpy in 7 h of solidification processes. In the *No-fin* case, if the weak convection influence at the initial state of discharging is neglected, the solidification

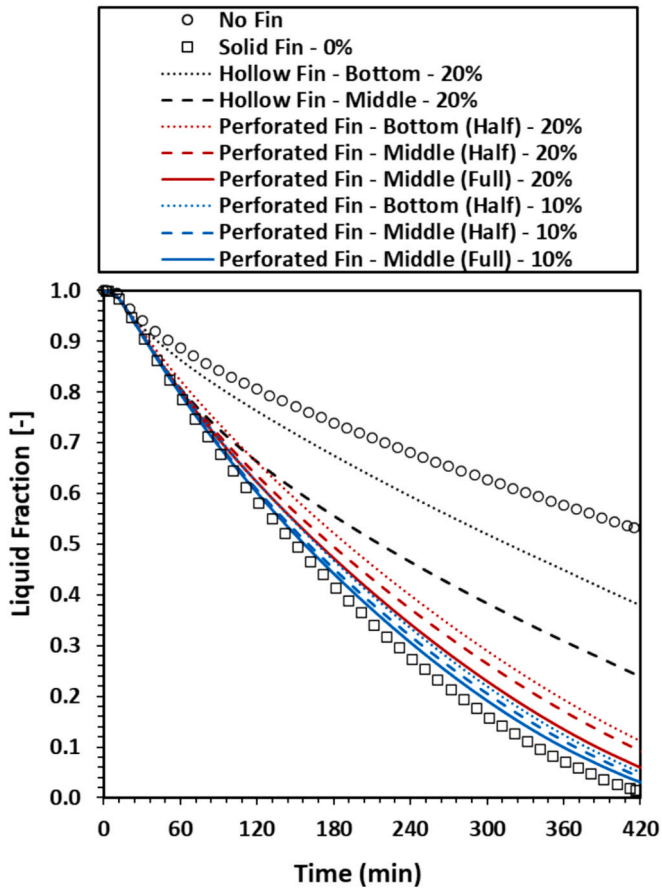


Fig. 18. Liquid fraction variation during the solidification process with hollowed and perforated fins.

process can almost be considered predominantly conduction-drive. On the other hand, natural convection has a determinant role in the charging process, particularly during the melting of the upper parts of the PCM domain. This 41 % difference between melting and solidification processes represents a measure of natural convection effectiveness. The absence of fins results in slow heat release due to stagnant regions by limited thermal contact.

In the solid fin configuration, there was no notable difference between charged and discharged enthalpy, indicating that fins significantly suppressed convection. Continuous solid fins that do not leave any gaps for buoyancy-driven flow act as obstacles to the movement of the melted PCM. Moreover, uniformly distributed longitudinal fins

create hot spots at various locations within the PCM domain, which disrupt the necessary temperature gradient for the upward and downward motion of the liquid PCM. Briefly, the presence of such fins hinders the formation of natural convective flow patterns and shifts the melting behavior toward a conduction-dominated process, similar to solidification. The observation of a 0 % difference between the charged enthalpy after the 6-hour melting process and the discharged enthalpy following the subsequent 7-hour solidification process may look like convection is almost entirely suppressed, as shown in Fig. 18. However, the one-hour time difference between the processes still points to partially maintaining the presence of convection on the effectiveness of the melting process. At the same time, it is important to note that solid fins primarily enhance heat transfer through conduction, and they still yield the highest stored and released enthalpy values despite their limitations on flow. Another notable result is seen in the Hollow fin–Bottom configuration (Case 2). Here, a 28 % performance drop is observed. The continuous axial gap and direct PCM contact with the hot tube facilitates a convenient environment for circulations. Yet, this configuration is significantly disadvantageous due to the absence of natural convection in the solidification process. Shifting the hollow gap location partially mitigated the shortcomings of the hollow fin configurations. Perforated fin designs further closed the enthalpy difference between melting and solidification processes. Even if the expectation from the perforation of fins worsens conduction, it allows suitable gaps in fins for PCM movement. The results in Fig. 19 show that holes with smaller sizes and larger numbers create harmony from the perspectives of both natural convection and conduction. In addition, the influence of natural convection even during solidification is obviously seen in perforated fin configurations with a lower removal rate of 10 %, especially in *Perforated fin-Middle (Full)*, Case 6. Notably, *Solid fin* configuration (Case 1) and perforated fin configurations (Case 4–5–6) balance heat storage and release, optimizing both conduction and natural convection. Considering material costs, the *Perforated Fin-Middle (full)* configuration (Case 6) with a 20 % fin removal rate was identified as the superior option.

Fig. 20 shows outlet temperature variation of HTF during solidification process. Cold liquid sodium enters the HTF tube at 678 °C and leaves the LHS unit increasing its temperature almost to the initial temperature of PCM at the first steps of discharging process. As expected, outlet temperature values for each case reduces since the distance between hot tube surface and solid–liquid interface increases. The smooth decline in temperature during solidification, in contrast to the melting process, provides further evidence of the absence of natural convection.

In the *No-fin* case, the outlet temperatures exhibit a sudden drop and stabilize at approximately 685 °C, which is not much greater than the PCM initial temperature. It implies that heat flux pursues at low levels due to lack of high conductive structures. The behavior of the *Hollow fin-Bottom* configuration (Case 2) closely resembles that of the *No-fin* case.

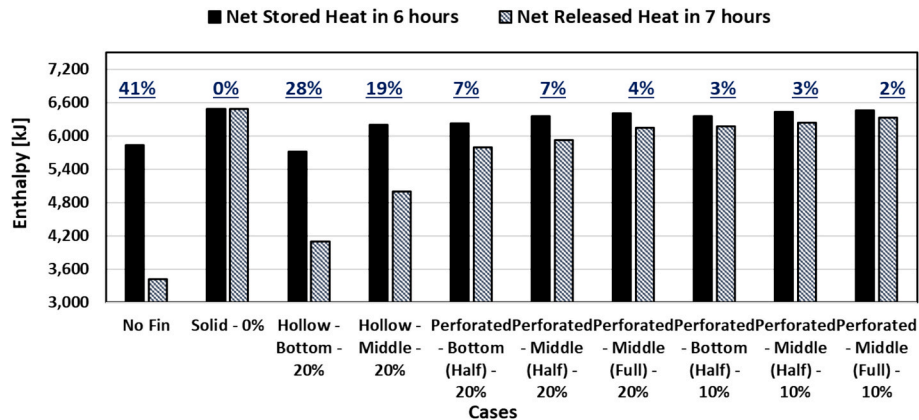


Fig. 19. Comparison of net charged and discharged enthalpy values for each case for 6-hour melting and 7-hour solidification processes, respectively.

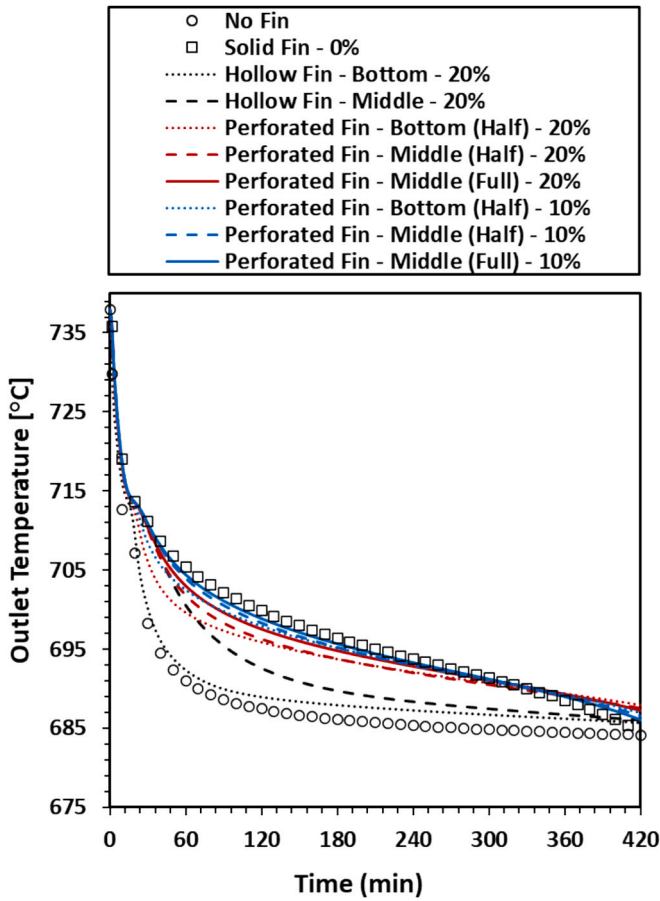


Fig. 20. Outlet temperature of the HTF during the solidification process with hollowed and perforated fins.

However, the outlet temperature trend of the *Hollow fin-Middle* configuration (Case 3) deviates considerably. The remaining cases exhibit nearly identical performance, indicating no significant distinction based on the design variations of the perforated fins.

3.2.3. Fin effectiveness

To evaluate fin effectiveness, a common approach is to compare the heat fluxes or the stored/released enthalpies between finned and unfinned configurations. In the present study, a correction factor accounting for the material usage is introduced to ensure a fair comparison, since perforation alters the amount of fin material. For the *No-fin* case, only HTF tube material is considered. In contrast, for finned configurations, the material amount includes both the HTF tube and the fins. The material factor (M) is defined as the ratio of the material quantity in a finned case to that in the *No-fin* case, with the material amount represented by the lateral surface area of the tube and attached fins. After calculating the ratio of stored and released enthalpies during charging and discharging processes, the fin effectiveness values ($\epsilon_{\text{charging}}$, $\epsilon_{\text{discharging}}$) are obtained by normalizing with the material factor, as shown in Eq. (12).

$$\epsilon_{\text{charging}} = \frac{1}{M} \frac{h_{\text{stored,finned}}}{h_{\text{stored,unfinned}}} \text{ and } \epsilon_{\text{discharging}} = \frac{1}{M} \frac{h_{\text{released,finned}}}{h_{\text{released,unfinned}}} \quad (12)$$

where

$$M = \frac{(\text{material amount})_{\text{finned case}}}{(\text{material amount})_{\text{unfinned case}}}$$

Table 6 presents the material usage for each configuration along with the corresponding material factors, which indicate how many times more material is used in finned cases compared to the *No-fin* case. Fin

Table 6

Material amount, material factor and fin effectiveness values of each fin configuration for melting and solidification processes.

Case Number	Case Name	Material Amount (mm ²)	Material Factor (M)	$\epsilon_{\text{charging}}$	$\epsilon_{\text{discharging}}$
Case 0	No Fin	311	1.0		
Case 1	Solid 0 %	1,199	3.9	2.9	23.4
Case 2	Hollow Bottom 20 %	1,021	3.3	0.6	6.1
Case 3	Hollow Middle 20 %	1,021	3.3	1.9	14.1
Case 4	Perforated Bottom (Half) 20 %	1,021	3.3	2.0	21.2
Case 5	Perforated Middle (Half) 20 %	1,021	3.3	2.7	22.4
Case 6	Perforated Middle (Full) 20 %	1,021	3.3	3.0	24.4
Case 4	Perforated Bottom (Half) 10 %	1,110	3.6	2.5	22.6
Case 5	Perforated Middle (Half) 10 %	1,110	3.6	2.9	23.2
Case 6	Perforated Middle (Full) 10 %	1,110	3.6	3.0	23.9

effectiveness evaluates how efficiently a unit amount of highly conductive material contributes to thermal energy storage and release, depending on the fin design strategy. Consistent with trends observed in liquid fraction and enthalpy values, the fin effectiveness of hollowed configurations (Cases 2 and 3) remains lower during both melting and solidification. It proves again that hollow fins are not advantageous in fin removal strategy. Among the perforated designs, bottom-perforated fins yield lower effectiveness, while middle-perforated fins perform comparably to the solid fin case. Notably, Case 6 (*Perforated-Middle, (Full)*) with a 20 % material removal rate exhibits higher effectiveness than the solid fin configuration (Case 1). On the other side, when the fin removal rate is changed to 10 %, fin effectiveness values are increasing for Cases 4 and 5, while it stays constant for Case 6. When fin effectiveness is investigated for the solidification process, it is seen that those values are greater than melting process since the released enthalpy in the *No-fin* case is significantly lower than the finned cases. There is a resemblance for the fin effectiveness of the configurations between melting and solidification processes, with one exception: *Perforated-Middle (Full)* (Case 6) with 20 % fin removal rate has a greater fin effectiveness value than its 10 % fin removal rate option. It indicates that full circular perforation offers a superior performance with less amount of fin material compared to the semicircular perforation option, which makes *Perforated-Middle (Full)* (Case 6) with 10 % fin removal rate the best configuration.

4. Conclusion

This study investigated for the first time the thermal performance of hollowed and perforated fins in a shell-and-tube LHS unit designed for HiHyTES applications using liquid sodium as heat transfer fluid. The 6-hour charging and 7-hour discharging durations were selected based on typical solar energy availability and night-time power demand in CSP plants, ensuring that the TES system effectively supports continuous power generation during non-sunlight hours.

The key insights from the findings are:

1. Perforated fins offer a trade-off between conduction and convection: Solid fins maximize conduction but hinder natural convection, while perforated fins allow for improved convection circulations without significant losses in both charging and discharging performances.
2. Fin removal rate significantly affects performance: A 20 % perforation rate (Case 6, *Perforated Fin-Middle (Full)*) provided the best balance, maintaining melting/solidification efficiency while reducing material use and increasing PCM storage capacity.
3. Placement of perforations is crucial: Middle-perforated fins performed better than bottom-perforated designs due to enhanced convection pathways, enabling more uniform heat distribution.
4. HTF selection impacts thermal performance: Liquid sodium demonstrated superior heat transfer effectiveness compared to molten salts, allowing for lower HTF outlet temperatures, which is beneficial for hybrid TES integration.
5. Solidification is more conduction-dominated: Unlike the melting process, which benefits from natural convection, solidification relies primarily on conduction, making fin continuity and contact with the cold surface essential for efficient heat extraction.
6. Outlet temperature trends indicate multi-stage storage optimization: The inability of a single LHS unit to drastically lower the HTF outlet temperature suggests the need for cascaded LHS configurations, where multiple PCMs with varying melting points can be used sequentially.
7. Material and cost considerations: The *Perforated fin-Middle (full)* design (20 % removal rate) proved to be a cost-effective alternative to solid fins, achieving similar thermal performance with reduced material usage, making it more viable for large-scale deployment.

Overall, this study highlights the importance of finned geometry optimization, HTF selection, and hybrid storage design for improving the efficiency of high-temperature thermal energy storage systems. Future work should focus on experimental validation, exploring alternative fin configurations, and integrating cascaded PCM designs for further efficiency gains.

CRediT authorship contribution statement

İsmail Gürkan Demirkıran: Writing – review & editing, Writing – original draft, Visualization, Validation, Software, Methodology, Investigation, Formal analysis, Conceptualization. **Klarissa Niedermeier:** Writing – review & editing, Supervision, Methodology, Conceptualization. **Erdal Çetkin:** Writing – review & editing, Supervision, Methodology, Conceptualization.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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Data availability

Data will be made available on request.

References

- [1] IEA, Electricity 2024: Analysis and forecast to 2026, IEA Publ. (2024) 1–170. <https://www.iea.org/news/global-coal-demand-expected-to-decline-in-coming-years>.
- [2] Rodat S. Status of Concentrated Solar Power Plants Installed Worldwide : Past and Present Data 2024:365–78.
- [3] Frate GF, Ferrari L, Sdringola P, Desideri U, Sciacovelli A. Thermally integrated pumped thermal energy storage for multi-energy districts: Integrated modelling, assessment and comparison with batteries. *J Energy Storage* 2023;61:106734. <https://doi.org/10.1016/J.EST.2023.106734>.
- [4] Laube T, Marocco L, Niedermeier K, Pacio J, Wetzel T. Thermodynamic Analysis of High-Temperature Energy Storage Concepts Based on Liquid Metal Technology. *Energy Technol* 2020;8. <https://doi.org/10.1002/ente.201900908>.
- [5] Ding Z, Wu W, Leung M. Advanced/hybrid thermal energy storage technology: material, cycle, system and perspective. *Renew Sustain Energy Rev* 2021;145:111088. <https://doi.org/10.1016/J.RSER.2021.111088>.
- [6] Pacio J, Wetzel T. Assessment of liquid metal technology status and research paths for their use as efficient heat transfer fluids in solar central receiver systems. *Sol Energy* 2013;93:11–22. <https://doi.org/10.1016/J.SOLENER.2013.03.025>.
- [7] Stahl D, Boese FK, Kostrzewa S. System and components design of a sodium heat transfer circuit for solar power plants. *Electr Power Syst Res* 1980;3:151–61. [https://doi.org/10.1016/0378-7796\(80\)90002-4](https://doi.org/10.1016/0378-7796(80)90002-4).
- [8] Coventry J, Andracka C, Pye J, Blanco M, Fisher J. A review of sodium receiver technologies for central receiver solar power plants. *Sol Energy* 2015;122:749–62. <https://doi.org/10.1016/J.SOLENER.2015.09.023>.
- [9] Niedermeier K, Marocco L, Flesch J, Mohan G, Coventry J, Wetzel T. Performance of molten sodium vs. molten salts in a packed bed thermal energy storage. *Appl Therm Eng* 2018;141:368–77. <https://doi.org/10.1016/J.APPLTHERMALENG.2018.05.080>.
- [10] Ahmed N, Elfeky KE, Lu L, Wang QW. Thermal and economic evaluation of thermocline combined sensible-latent heat thermal energy storage system for medium temperature applications. *Energy Convers Manag* 2019;189:14–23. <https://doi.org/10.1016/j.enconman.2019.03.040>.
- [11] Tehrani SSM, Taylor RA, Saberi P, Dierce G. Design and feasibility of high temperature shell and tube latent heat thermal energy storage system for solar thermal power plants. *Renew Energy* 2016;96:120–36. <https://doi.org/10.1016/j.renene.2016.04.036>.
- [12] Michels H, Pitz-Paal R. Cascaded latent heat storage for parabolic trough solar power plants. *Sol Energy* 2007;81:829–37. <https://doi.org/10.1016/j.solener.2006.09.008>.
- [13] Liu M, Riahi S, Jacob R, Belusko M, Bruno F. Design of sensible and latent heat thermal energy storage systems for concentrated solar power plants: Thermal performance analysis. *Renew Energy* 2020;151:1286–97. <https://doi.org/10.1016/j.renene.2019.11.115>.
- [14] Mostafavi Tehrani SS, Shoraka Y, Nithyanandam K, Taylor RA. Shell-and-tube or packed bed thermal energy storage systems integrated with a concentrated solar power: A techno-economic comparison of sensible and latent heat systems. *Appl Energy* 238 2019:887–910. <https://doi.org/10.1016/j.apenergy.2019.01.119>.
- [15] Riahi S, Jovet Y, Saman WY, Belusko M, Bruno F. Sensible and latent heat energy storage systems for concentrated solar power plants, exergy efficiency comparison. *Sol Energy* 2019;180:104–15. <https://doi.org/10.1016/j.solener.2018.12.072>.
- [16] Zauner C, Hengstberger F, Mörzinger B, Hofmann R, Walter H. Experimental characterization and simulation of a hybrid sensible-latent heat storage. *Appl Energy* 2017;189:506–19. <https://doi.org/10.1016/J.APENERGY.2016.12.079>.
- [17] He F, Hu C, Gao W, Li S, Meng X. Effect of inclination angles on heat transfer characteristics of solid and perforated spiral finned heat exchangers. *Int Commun Heat Mass Transf* 2025;164:108920. <https://doi.org/10.1016/J.ICHEATMASTRANSFER.2025.108920>.
- [18] Ben Khedher N, Mahdi JM, Majidi HS, Khosravi K, Al-Azzawi WK, Al-Qrimli FA, et al. CFD analysis of phase-change material-based heat storage with dimple-shaped fins: evaluation of fin configuration and distribution pattern. *J Comput Des Eng* 2022;9:2055–72. <https://doi.org/10.1093/JCDE/QWAC105>.
- [19] Gürsoy E. Melting and energy storage performance enhancement of rectangular cavity with metal foam by nano-PCM and recessed/protruding dimpled fin wall. *J Energy Storage* 2025;119:116327. <https://doi.org/10.1016/J.EST.2025.116327>.
- [20] Li C, Li Q, Ge R. Enhancement of melting performance in a shell and tube thermal energy storage device under different structures and materials. *Appl Therm Eng* 2022;214:118701. <https://doi.org/10.1016/J.APPLTHERMALENG.2022.118701>.
- [21] Karami R, Kamkari B. Experimental investigation of the effect of perforated fins on thermal performance enhancement of vertical shell and tube latent heat energy storage systems. *Energy Convers Manag* 2020;210:112679. <https://doi.org/10.1016/J.ENCONMAN.2020.112679>.
- [22] Li H, Hu C, He Y, Tang D, Wang K, Huang W. Effect of perforated fins on the heat-transfer performance of vertical shell-and-tube latent heat energy storage unit. *J Energy Storage* 2021;39:102647. <https://doi.org/10.1016/J.EST.2021.102647>.
- [23] Modi N, Wang X, Negnevitsky M. Experimental investigation of the effects of inclination, fin height, and perforation on the thermal performance of a longitudinal finned latent heat thermal energy storage. *Energy* 2023;274:127327. <https://doi.org/10.1016/J.ENERGY.2023.127327>.
- [24] Wang Z, Wang Y, Yang L, Cui Y, Song L, Yue G. Multi-objective optimization of heat charging performance of phase change materials in tree-shaped perforated fin heat exchangers. *Energy* 2024;294:130839. <https://doi.org/10.1016/J.ENERGY.2024.130839>.

- [25] He F, Yan B, Zou J, Hu C, Meng X, Gao W. Experimental evaluation of the effect of perforated spiral fins on the thermal performance of latent heat storage units. *J Energy Storage* 2023;58:106359. <https://doi.org/10.1016/J.EST.2022.106359>.
- [26] Dhaidan NS, Hassan AF, Rasheed Al-Gaheeshi AM, Al-Mousawi FN, Homod RZ. Experimental investigation of thermal characteristics of phase change material in finned heat exchangers. *J Energy Storage* 2023;71:108162. <https://doi.org/10.1016/J.EST.2023.108162>.
- [27] Khobragade S, Devanuri JK. Experimental investigation to assess the influence of inclination on thermal performance of latent heat storage system with solid and perforated fins under simultaneous charging and discharging process. *Therm Sci Eng Prog* 2023;41:101809. <https://doi.org/10.1016/J.TSEP.2023.101809>.
- [28] Kalapala L, Krishna Devanuri J. Comparative analysis of solid and perforated fins for thermal enhancement of a latent heat storage unit positioned at various inclinations. *Energy Sources, Part A Recover Util Environ Eff* 2021;00:1–20. <https://doi.org/10.1080/15567036.2021.1886202>.
- [29] Raj LS, Yashwanth K, Madan C, Anusha A, Sridarsi G, Sreenivasulu S, et al. Optimization study of radial finned latent heat storage system employing novel circumferential perforations. *J Energy Storage* 2024;79:110068. <https://doi.org/10.1016/j.est.2023.110068>.
- [30] Wang Z, Wang Y, Yang L, Song L, Jia H, Ren Y, et al. Study on solidification characteristics of bionic finned phase change heat exchanger and multi-objective optimization design. *J Energy Storage* 2024;86:111105. <https://doi.org/10.1016/J.EST.2024.111105>.
- [31] Modi N, Wang X, Negnevitsky M. Numerical investigation into selecting the most suitable shell-to-tube diameter ratio for horizontal latent heat thermal energy storage. *Energy. Sustain Dev* 2023;73:188–204. <https://doi.org/10.1016/j.esd.2023.02.004>.
- [32] Panisilvam J, Tay NHS, Wang PC. Experimental investigation during the melting process of a vertical and horizontal tube-in-shell Latent Heat Energy Storage System. *J Energy Storage* 2022;55:105401. <https://doi.org/10.1016/J.EST.2022.105401>.
- [33] Khobragade S, Devanuri JK. Impact of inclination on the thermal performance of shell and tube latent heat storage system under simultaneous charging and discharging: Numerical investigation. *Appl Therm Eng* 2022;214:118811. <https://doi.org/10.1016/J.APPLTHERMALENG.2022.118811>.
- [34] Seddegh S, Wang X, Henderson AD. A comparative study of thermal behaviour of a horizontal and vertical shell-and-tube energy storage using phase change materials. *Appl Therm Eng* 2016;93:348–58. <https://doi.org/10.1016/J.APPLTHERMALENG.2015.09.107>.
- [35] Kalapala L, Devanuri JK. Energy and exergy analyses of latent heat storage unit positioned at different orientations – An experimental study. *Energy* 2020;194:116924. <https://doi.org/10.1016/J.ENERGY.2020.116924>.
- [36] Riahi S, Evans M, Belusko M, Liu M, Bruno F. Orientation impact on structural integrity of a shell and tube latent heat thermal energy storage system. *J Energy Storage* 2022;52:104829. <https://doi.org/10.1016/j.est.2022.104829>.
- [37] Opolot M, Zhao C, Liu M, Mancini S, Bruno F, Hooman K. A review of high temperature ($\geq 500^\circ\text{C}$) latent heat thermal energy storage. *Renew Sustain Energy Rev* 2022;160. <https://doi.org/10.1016/j.rser.2022.112293>.
- [38] Niedermeier K. A perspective on high-temperature heat storage using liquid metal as heat transfer fluid. *Energy Storage* 2023;5:1–12. <https://doi.org/10.1002/est2.530>.
- [39] McConohy G, Kruienza A. Molten nitrate salts at 600 and 680 °C: Thermophysical property changes and corrosion of high-temperature nickel alloys. *Sol Energy* 2014;103:242–52. <https://doi.org/10.1016/J.SOLENER.2014.01.028>.
- [40] Kunkel S, Klasing F, Hanke A, Bauer T, Bonk A. Concentrating solar power at higher limits: First studies on molten nitrate salts at 600 °C in a 100 kg-scale hot tank. *Sol Energy Mater Sol Cells* 2023;258:112412. <https://doi.org/10.1016/J.SOLMAT.2023.112412>.
- [41] Range M. Inconel Alloy 617. *Alloy Dig* 1972;21:1–12. <https://doi.org/10.31399/asm.ad.ni0179>.
- [42] Kim T, France DM, Yu W, Zhao W, Singh D. Heat transfer analysis of a latent heat thermal energy storage system using graphite foam for concentrated solar power. *Sol Energy* 2014;103:438–47. <https://doi.org/10.1016/j.solener.2014.02.038>.
- [43] Zhao W, France DM, Yu W, Kim T, Singh D. Phase change material with graphite foam for applications in high-temperature latent heat storage systems of concentrated solar power plants. *Renew Energy* 2014;69:134–46. <https://doi.org/10.1016/j.renene.2014.03.031>.
- [44] Singh D, Yu W, Zhao W, Kim T, France DM, Smith RK. Development and prototype testing of MgCl₂/graphite foam latent heat thermal energy storage system. *Sol Energy* 2018;159:270–82. <https://doi.org/10.1016/j.solener.2017.10.084>.
- [45] Khuda MA, Khalesi J, Tu Z, Long Y, Piccioni Koch D, Sarunac N. Numerical analysis of a developing turbulent flow and conjugate heat transfer for molten salt and liquid sodium in a solar receiver. *Appl Therm Eng* 2022;217. <https://doi.org/10.1016/j.applthermaleng.2022.119156>.
- [46] V.V.R. & P. C., A Fixed grid numerical modelling methodology for convection diffusion mushy region phase change problems, *Int. Journal Heat Mass Transf.* 30 (1978) 1709–1719.
- [47] Shen ZG, Chen S, Chen B. Heat transfer performance of a finned shell-and-tube latent heat thermal energy storage unit in the presence of thermal radiation. *J Energy Storage* 2022;45:103724. <https://doi.org/10.1016/J.EST.2021.103724>.
- [48] ANSYS FLUENT 12.0 User's Guide - 13.2.4 Natural Convection and Buoyancy-Driven Flows, (n.d.). <https://www.afs.enea.it/project/neptunius/docs/fluent/html/ug/node470.htm> (accessed February 26, 2025).
- [49] Parry AJ, Eames PC, Agyenim FB. Modeling of thermal energy storage shell-and-tube heat exchanger. *Heat Transf Eng* 2014;35:1–14. <https://doi.org/10.1080/01457632.2013.810057>.
- [50] Demirkiran İG, Cetkin E. Computation time reduction of PCM melting process by changing modeling parameters. *Numer Heat Transf Part A Appl* 2022;83:50–67. <https://doi.org/10.1080/10407782.2022.2149229>.