



Multi-shell PCM heat exchangers with wavy walls and fins: Parametric analysis of melting dynamics and storage efficiency

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ABSTRACT

This study presents a comprehensive numerical investigation of multi-shell phase change material (PCM) heat exchangers incorporating wavy walls and fins, aimed at enhancing melting dynamics and latent heat storage efficiency. An eighteen-case parametric framework was developed to assess the combined effects of wavy-wall wavelength and amplitude, fin arrangement and length, number of PCM shells, Reynolds number, and inlet temperature under consistent thermal and material conditions. The model, validated against experimental melting data for RT82 PCM in finned triplex-tube exchangers, accurately captured the coupled conduction–convection processes governing phase transition. Results reveal that geometric optimization significantly accelerates melting and improves storage efficiency: reducing the wavy-wall wavelength from 31.4 mm to 21 mm shortened melting time by 21.9% and increased heat storage by 26.4%, while increasing amplitude from 5 mm to 10 mm halved melting time and nearly doubled storage capacity (+97.1%). Optimal fin arrangement enhanced performance by 50% compared with less effective configurations, and increasing the number of PCM shells from one to three reduced melting time by 60% and more than doubled storage power. Operational parameters produced similarly strong effects: raising Reynolds number from 500 to 2000 increased storage rate by 88.5%, while elevating inlet temperature from 50 °C to 70 °C halved melting time and raised power output to 2.68 kW. The results provide validated, quantitative trends linking geometry, flow, and thermal driving force, offering practical guidance for the design of compact latent heat storage systems.

1. Introduction

Latent Heat Thermal Energy Storage (LHTES) is a cornerstone technology for sustainable and efficient energy systems, offering compact, high-energy-density storage by utilizing the latent heat of PCMs [1,2]. Its near-isothermal operation and sealed, mechanically simple design enable scalable deployment with minimal maintenance requirements [3,4], making it suitable for applications such as solar heating, building energy management, and industrial waste heat recovery. However, the

inherently low thermal conductivity of most PCMs limits heat transfer and overall system performance [5,6]. To address this, various enhancement strategies have been proposed, including fin structures (pin, rectangular, and biomimetic designs) [7–10], wavy-wall geometries, and nanoparticle-doped PCMs [11], all of which substantially improve melting and solidification rates. More recently, hybrid approaches that combine passive geometry optimization with active forced convection have shown synergistic gains in charging and discharging efficiency [12,13]. Overall, ongoing advances in material engineering and system integration are steering LHTES toward scalable, high-

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Nomenclature		t	Time (s)
<i>Greek symbols</i>		t_m	Total melting time (s)
β	Thermal expansion coefficient (K^{-1})	$\vec{V}$	Velocity vector ($m \cdot s^{-1}$)
λ	Liquid fraction (1)	<i>Subscripts</i>	
μ	Dynamic viscosity (Pa·s)	eff	Effective value
ρ	Density ($kg \cdot m^{-3}$)	e	Equilibrium
φ	Branched-fins' angle (deg)	i	Initial
<i>Latin symbols</i>		L	Latent heat
A_m	Mushy zone Darcy coefficient (–)	m	Momentum
C_p	Specific heat capacity ($J \cdot kg^{-1} \cdot K^{-1}$)	ref.	Reference
E	internal energy (J)	T	Total
$\dot{E}$	Energy input rate (W)	in	Inlet
g	Gravitational acceleration ($m \cdot s^{-2}$)	out	Outlet
H	Total enthalpy (J)	<i>Acronyms</i>	
k	Thermal conductivity ($W \cdot m^{-1} \cdot K^{-1}$)	LHTES	Latent Heat Thermal Energy Storage
L_f	Latent heat of fusion ($J \cdot kg^{-1}$)	TES	Thermal energy storage
P	Pressure (Pa)	HTF	Heat Transfer Fluid
S	source term ($W \cdot m^{-3}$)	PCM	Phase Change Material
T	Temperature (K)		

efficiency solutions for renewable and waste heat applications [14].

Geometrical configuration of the inner tube and enclosure plays a decisive role in enhancing heat transfer performance in LHTES systems by governing natural convection patterns, melting and solidification rates, and overall thermal efficiency. Various studies have demonstrated that altering the enclosure shape, introducing wavy or finned surfaces, and adjusting tube eccentricity or rotation can markedly accelerate heat transfer and improve temperature uniformity. Ahmed et al. [15] reported that wavy boundaries and rotating cylinders intensify convective motion, while Shojaeinasab et al. [16] showed that optimized fin spacing in double-tube systems reduces melting time and enhances heat recovery. Sokakini et al. [17] confirmed that diamond-shaped multi-tube cores increase surface area and boost melting–solidification kinetics. In ice storage applications, Izadi et al. [18] found that coil curvature and orientation strongly influence freezing rate and capacity, and Shahsavari et al. [19] demonstrated that tube rotation and eccentric placement improve fluid mixing and suppress thermal stratification.

Beyond internal tube configurations, the geometry of the enclosure body plays a decisive role in governing convective strength and overall thermal efficiency in LHTES systems. Abderrahmane et al. [20] demonstrated that introducing optimally shaped internal baffles in nanofluid-filled enclosures enhanced mixed convection and increased the average Nusselt number by nearly 30%, significantly improving heat transfer uniformity and reducing entropy generation. Alomari et al. [21] further showed that wavy cylindrical cavities promote stronger double-diffusive convection, resulting in up to 35% higher heat transfer rates compared to smooth walls. Aly and Alsedais [22] reported that wavy star-shaped cavities with nano-enhanced PCMs improved the overall melting rate by approximately 42% due to the expanded surface area and intensified circulation zones. Similarly, Parsa et al. [23] found experimentally that optimized shell geometries in shell-and-tube LHTES units reduced complete melting time by nearly 55% while improving thermal uniformity by 20%. Complementary studies by Shahsavari et al. [24] confirmed that rotating or eccentric inner tubes accelerated phase change by 40–50%, highlighting the critical influence of geometric asymmetry on enhancing convective motion. Collectively, these findings confirm that wavy, trapezoidal, or star-shaped enclosures and eccentric or rotating cores substantially intensify natural convection and heat transfer, leading to faster melting–solidification cycles and more compact, energy-efficient thermal storage systems adaptable to renewable and industrial applications.

The geometry of the inner tube and the configuration of fins are critical factors influencing phase change behavior and heat transfer performance in LHTES systems [25]. Numerous experimental and numerical studies have shown that optimizing these geometrical parameters enhances melting and solidification rates, promotes thermal uniformity, and minimizes resistance within the PCM domain. Parsa et al. [26] demonstrated that introducing tube eccentricity accelerates melting by intensifying buoyancy-driven flow, while elliptic tubes with optimized aspect ratios and inclination angles reduce total melting time by up to 25% [27]. Fin configuration further amplifies heat transfer efficiency: spiral and discontinuous fins enhance conduction pathways and improve uniformity [18,28], and Y-shaped or T-shaped fins yield faster melting and smaller unmelted regions [29,30]. Additionally, studies incorporating nanoparticles within finned structures show that adjusting fin orientation can improve melting uniformity and boost heat storage capacity by up to 200% compared to sensible systems [31–33]. Bouzidi et al. [34], Park et al. [35] and Hekmat et al. [36] reported that multi-tube arrangements and optimized enclosure contours can significantly reduce thermal resistance and enhance convective heat transport. Collectively, these findings highlight that synergistic optimization of inner tube geometry and fin design provides an effective, low-cost pathway to accelerate PCM phase transition, increase energy efficiency, and advance the performance of next-generation LHTES technologies.

Non-circular and biomimetic tube geometries have shown strong potential for enhancing heat transfer in LHTES systems. Zigzag-shaped inner tubes increased melting rates by about 30% through secondary flow enhancement [37], while vertical multi-tube designs improved melting uniformity via stronger buoyant circulation [38]. Quasi-petal tubes further boosted nano-enhanced PCM performance, achieving up to 40% faster melting by eliminating stagnant zones [39,40]. Optimized petal-shaped fins with an offset ratio of 0.458 reduced solidification time by nearly 24% compared with standard annular fins [41]. Overall, complex wavy and petal-like geometries effectively couple conduction and convection, enabling faster, more uniform phase change in compact, high-efficiency thermal storage systems.

While corrugated or wavy-wall PCM enclosures have been explored as a means of enhancing heat transfer, previous studies have predominantly considered such features within single-shell geometries or under fixed operating conditions, thereby providing limited insight into their interaction with distributed heat-transfer interfaces and forced

convection. The present work advances this line of research by integrating wavy walls into a multi-shell LHTES configuration and systematically examining the combined effects of wall geometry, fin arrangement, shell count, and operating parameters within a unified parametric framework. This approach reveals strong synergies between surface modulation, extended conduction pathways, and convective transport, leading to substantially higher melting rates and storage power than achievable through isolated geometric modifications.

In contrast to conventional studies that vary individual parameters sequentially, the present analysis treats geometry and flow as a coupled design space, enabling consistent interpretation of how multiple structural and operational factors jointly govern phase-change dynamics and storage efficiency. This integrated methodology yields physically interpretable design guidelines for selecting corrugation wavelength and amplitude, fin topology, and shell distribution in concert with Reynolds number and inlet temperature. Such co-design principles are essential for translating PCM-based latent thermal energy storage from case-specific prototypes to scalable, high-performance systems for applications such as HVAC load leveling, electronics cooling, and renewable energy integration [42,43].

2. Methodology

2.1. Model configuration

In the present work, a concentric shell-and-tube TES module is

configured to systematically decouple and co-analyze geometric and flow/thermal effects on PCM melting through an 18-case matrix illustrated in Fig. 1 and characterized in Table 1. The module comprises an inner pipe carrying a forced HTF and one to three surrounding PCM annuli, enabling controlled modification of the heat-exchange interface while preserving consistent materials, outer envelope, and boundary idealizations for fair comparison. Geometry is varied along three axes: wall morphology (smooth versus sinusoidally wavy with prescribed wavelength λ and amplitude A), extended surfaces (fins attached on the PCM-side wall using arrangements A–C with specified length and thickness), and shell count (single, double, or triple PCM annuli) to tune local heat spreading, interfacial area, and melt-front topology. Operating conditions are varied via the HTF inlet Reynolds number Re and inlet temperature T_{in} , which together set the convective driving and thermal power delivered to the PCM domain.

Fig. 1 depicts all 18 configurations grouped by design intent. Cases 1–5 sweep wall undulation (λ , A) without fins to isolate surface-shape effects on near-wall transport. Cases 6–10 introduce fins and vary both arrangement (A–C) and fin length relative to the local gap, capturing conduction spreading and convection coupling introduced by extended surfaces. Cases 11–13 alter the number of PCM shells at otherwise fixed geometric features to reveal how distributed annuli reconfigure thermal pathways. Cases 14–15 reduce Re at fixed geometry to examine buoyancy-forced flow interplay, while Cases 16–17 vary inlet temperature (T_{in}) to quantify sensitivity to thermal driving. Case 18 serves as a baseline with smooth walls and no fins at the nominal Re and T_{in} ,

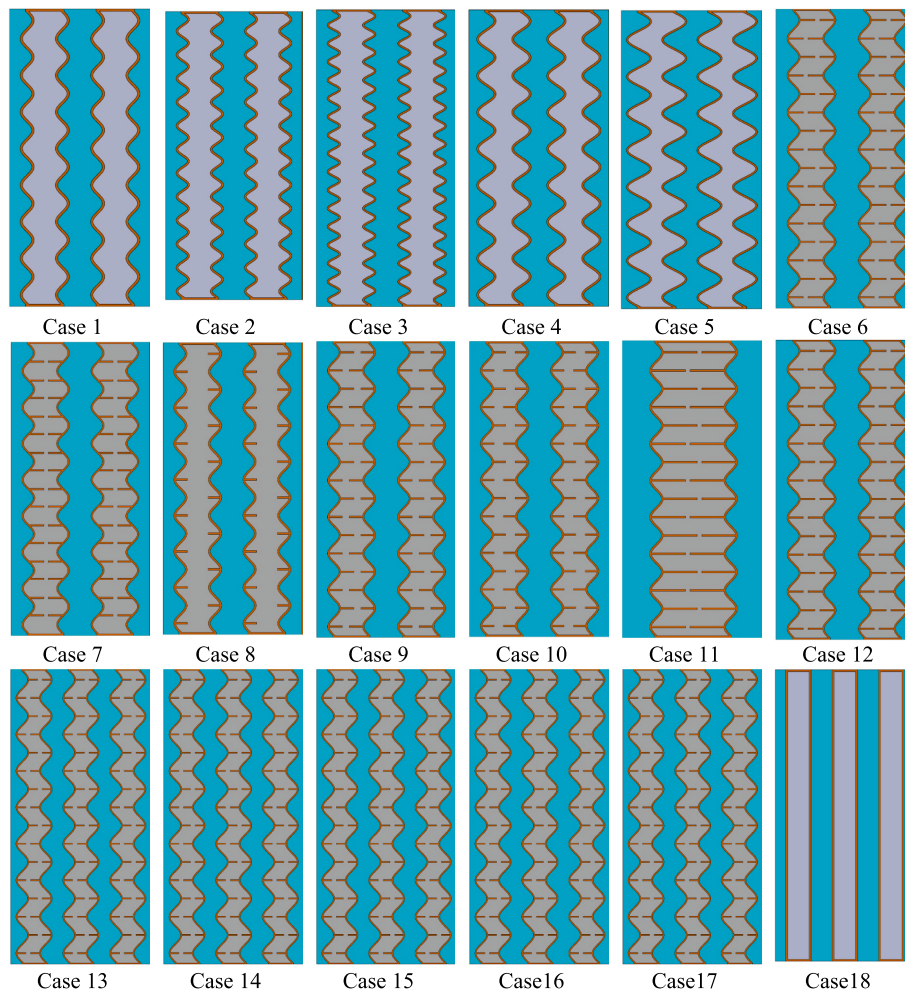


Fig. 1. Schematic of all cases. Geometric and operating variants used in the parametric study, illustrating wall settings (wavelength, amplitude), fin layouts and lengths (Arrangements A–C), PCM shell counts (1–3), and the baseline geometry.

Table 1

Case matrix and model inputs. Summary of configurations evaluated, listing PCM shell count, wavy-wall wavelength and amplitude, wall thickness, fin arrangement and length, inlet Reynolds number, and inlet temperature; the baseline (Case 18) has no waviness and no fins for controlled comparison.

Case Number	number of PCM shells	Wave length (mm)	Wave amplitude (mm)	Pipe thickness (mm)	Fin arrangement	Fin length* (mm)	Inlet Reynolds	Inlet temp. (°C)
1	2	31.4	5	2	No fin	–	2000	60
2	2	21	5	1.7	No fin	–	2000	60
3	2	15.7	5	1.44	No fin	–	2000	60
4	2	31.4	7.5	1.7	No fin	–	2000	60
5	2	31.4	10	1.44	No fin	–	2000	60
6	2	31.4	5	1.3	A	12.35	2000	60
7	2	31.4	5	1.4	B	12.01	2000	60
8	2	31.4	5	1.7	C	10.15	2000	60
9	2	31.4	5	1.4	A	11.3	2000	60
10	2	31.4	5	1.45	A	10.15	2000	60
11	1	31.4	5	1.6	A	12.35	2000	60
12	2	31.4	5	1.3	A	12.35	2000	60
13	3	31.4	5	1.1	A	12.35	2000	60
14	3	31.4	5	1.1	A	12.35	1000	60
15	3	31.4	5	1.1	A	12.35	500	60
16	3	31.4	5	1.1	A	12.35	2000	50
17	3	31.4	5	1.1	A	12.35	2000	70
18	3	No Wave	No fin				2000	60

* Length is chosen based on the distance from the wall.

anchoring all comparisons.

Table 1 enumerates the exact inputs for each case: PCM shell count (1–3), wavy-wall wavelength and amplitude (where applicable), wall/pipe thickness, fin arrangement and fin length specification, HTF inlet Re (e.g., 500, 1000, 2000), and T_{in} (e.g., 50, 60, 70 °C). Across all cases, the HTF is imposed at a fixed inlet condition; external boundaries are adiabatic; and the PCM exchanges heat only through the shaped wall and any attached fins. Phase change is treated using a fixed-grid enthalpy–porosity formulation under laminar, incompressible, Newtonian assumptions with Boussinesq buoyancy in the molten region; viscous dissipation and environmental heat losses are neglected, and the governing continuity, momentum, and energy equations retain their standard forms. This configuration establishes a clean, orthogonal design space in which geometry and flow/thermal drivers are varied both individually and jointly, providing a rigorous basis for deriving model-supported design rules for compact, fast-charging latent TES modules.

The overarching goal is to mitigate the inherent limits of PCMs, low thermal conductivity and slow phase transition, by combining geometric modifications (wavy walls, fins) with controlled flow and thermal drivers (Re , inlet temperature) to accelerate melting and better utilize storage capacity. The case matrix is structured for clear comparisons: Cases 1–5 probe wall undulation, 6–10 examine fin layout/length, 11–13 assess shell count, 14–15 vary Reynolds number, and 16–17 test inlet temperature, with Case 18 as a no-wave, no-fin baseline. This design enables identification of effective geometry–operation combinations that improve melting dynamics and overall latent TES performance.

2.2. Mathematical modelling

In the present study, the enthalpy–porosity method proposed by Brent et al. [44,45] was employed to simulate the melting and freezing behavior of the PCM. This method conceptualizes the mushy region, where solid and liquid phases coexist, as a porous structure whose permeability varies with the local liquid fraction. Each computational element is assigned a porosity value between 0 (solid state) and 1 (liquid state), enabling a continuous representation of the evolving phase interface without explicitly tracking its motion.

The buoyancy-driven flow induced by density variations during melting was incorporated using the Boussinesq approximation [46]. To simplify the numerical model while preserving essential physics, the flow was considered laminar, Newtonian, and incompressible, and all outer walls were assumed to be adiabatic, neglecting both viscous

dissipation and heat exchange with the environment [24,47]. With these assumptions, the continuity, momentum, and energy equations maintain their classical forms, expressed in Eqs. (1)–(3). These governing relations are consistent with standard formulations reported in previous numerical investigations of phase-change processes in latent heat TES systems [48].

$$\nabla \cdot \rho \vec{V} = 0 \quad (1)$$

$$\rho \frac{\partial \vec{V}}{\partial t} + \rho (\vec{V} \cdot \nabla) \vec{V} = -\nabla P + \mu (\nabla^2 \vec{V}) - \rho \beta (T - T_{ref}) \vec{g} - \vec{S} \quad (2)$$

$$\rho C_p \frac{\partial T}{\partial t} + \nabla \cdot (\rho C_p \vec{V} T) = \nabla \cdot (k \nabla T) - S_L \quad (3)$$

where $\vec{V}$ is the velocity vector, P is the pressure, T is the temperature, ρ is the density, μ is the dynamic viscosity, k is the thermal conductivity, C_p is the specific heat capacity, $\vec{g}$ is the gravitational acceleration, and β is the thermal expansion coefficient.

In the enthalpy–porosity formulation, the effects of the solid–liquid phase transition are incorporated by introducing additional source terms into the governing equations. In the momentum equation, a damping term $\vec{S}$ is included in Eq. (4) to account for the Darcy-type resistance that develops within the mushy region. As the material transitions from liquid to solid, the permeability of this region declines, gradually restricting the fluid motion and thereby simulating the momentum reduction caused by the formation of the solid phase.

Similarly, in the energy equation, a latent heat source term S_L is incorporated in Eq. (5) to represent the energy absorption or release associated with melting and solidification. This treatment ensures that the enthalpy variation during phase transition is accurately captured in the temperature field.

Overall, these modifications allow the phase change process to be seamlessly integrated into the standard conservation equations without explicit interface tracking, while preserving the thermodynamic consistency of the enthalpy–porosity framework [49].

$$\vec{S} = A_m \frac{(1 - \lambda)^2}{\lambda^3 + 0.001} \vec{V} \quad (4)$$

$$S_L = \rho L_f \frac{\partial \lambda}{\partial t} + \rho L_f \nabla \cdot (\vec{V} \lambda) \quad (5)$$

In these equations, A_m is the mushy-zone constant, and λ denotes the

local liquid fraction. The term S_L represents the latent heat source term accounting for the energy absorbed or released during phase change, while L_f is the latent heat of fusion of the PCM. This formulation introduces a Darcy-type damping force in the momentum equation to suppress velocity in solid regions and incorporates latent heat effects into the energy equation.

In line with previous numerical studies, the mushy-zone constant (A_m) was assigned a value of 10^5 in this work [50–52]. This coefficient determines the magnitude of the artificial damping force within partially melted regions, ensuring a gradual yet sharp transition between liquid and solid phases. The resistance term is formulated as a function of the local liquid fraction (λ), as defined in Eq. (6) [53]. Through this dependency, the permeability of the mushy region decreases with solidification, effectively suppressing fluid motion in solid zones while maintaining flow in melted areas.

After computing the liquid fraction, the associated latent heat exchange during phase transition is evaluated using Eq. (7), which quantifies the rate of energy storage or release within the PCM. This approach enables the model to consistently capture both momentum damping and thermal energy accumulation during melting and solidification, providing a unified and physically coherent framework for simulating the complex dynamics of phase-change processes.

$$\lambda = \frac{\Delta H}{L_f} = \begin{cases} 0 & \text{if } T < T_{\text{Solidus}} \\ 1 & \text{if } T > T_{\text{Liquidus}} \\ \frac{T - T_{\text{Solidus}}}{T_{\text{Liquidus}} - T_{\text{Solidus}}} & \text{if } T_{\text{Solidus}} < T < T_{\text{Liquidus}} \end{cases} \quad (6)$$

$$\dot{E}_T = \frac{E_e - E_i}{t_m} \quad (7)$$

In Eq. (6), the liquid fraction λ varies between 0 (solid phase) and 1 (liquid phase) and determines the permeability of the mushy zone during phase transition. In Eq. (7), t_m denotes the total charging (melting) duration of the thermal storage system, while E_i and E_e represent the initial and final total energies of the PCM, respectively, as obtained from the numerical simulations. The total energy input rate, expressed as $\dot{E}_T$, consists of two contributions: the sensible heat term $MC_p dT$ associated with the temperature rise of the PCM and the latent heat term ML_f corresponding to the energy absorbed during melting. The total enthalpy of the PCM is denoted by H and is evaluated as the sum of sensible and latent heat contributions, ensuring consistent quantification of energy storage during the phase-change process.

By integrating both contributions, the formulation accurately describes the overall energy storage capability of the PCM during the charging process. This combined treatment enables the model to reflect the full extent of thermal energy accumulation, encompassing both temperature-dependent heating and latent heat absorption throughout the phase-change cycle.

3. Numerical modelling, grid independence and validation

3.1. Numerical approach

The computational modelling of heat transfer and phase change was performed using ANSYS Fluent, a CFD solver widely employed for PCM analysis. The model simultaneously resolved thermal conduction in the solid phase and natural convection within the molten PCM to capture the coupled transport phenomena during melting. To link the velocity and pressure fields in the incompressible flow regime, the SIMPLE algorithm was utilized, offering stable convergence and mass conservation. The Green–Gauss cell-based approach was adopted to calculate spatial gradients, ensuring accurate evaluation of field variables across the unstructured computational mesh.

For the discretization of governing equations, the QUICK scheme was applied to both momentum and energy terms to minimize artificial

diffusion and to resolve steep temperature gradients near the phase front with higher precision. The PRESTO scheme was chosen for pressure interpolation due to its robustness in handling buoyancy-dominated flows, which are significant in melting-driven convection.

Stringent convergence criteria were imposed to maintain numerical accuracy: 10^{-4} for the continuity and momentum equations and 10^{-6} for the energy equation. These tolerances ensured that residual errors were sufficiently minimized, providing confidence in the stability and physical fidelity of the final computed solutions.

A comprehensive grid independence analysis was conducted for the reference case featuring a uniformly finned, inline arrangement. As illustrated in Fig. 2, simulations were performed with progressively refined meshes containing approximately 100 K, 200 K, 300 K, 400 K, and 500 K elements. To assess convergence, both global metrics including total charging duration and mean energy storage rate and local flow characteristics, such as the evolution of the liquid fraction field, were carefully examined.

The comparison revealed that results obtained using the 400 K and 500 K meshes were nearly indistinguishable, indicating that further refinement yielded negligible improvement in accuracy. However, coarser grids (≤ 300 K) introduced noticeable deviations, particularly in the thermal gradients near the inner tube wall and fin bases, where heat transfer is most intense during the initial melting phase.

Considering the trade-off between computational efficiency and solution precision, the mesh with 400 K elements was adopted for all subsequent simulations. Contour maps of the liquid fraction verified that beyond this resolution, the morphology of the melting front and the thermal boundary layer distribution remained virtually unchanged, confirming that the selected grid was sufficiently fine to capture all essential physical features without excessive computational cost.

To assess the influence of time step resolution, simulations were conducted with $\Delta t = 0.4, 0.2, 0.1$, and 0.05 s on the 400 K mesh. Fig. 3 demonstrates that the liquid-fraction profiles for $\Delta t = 0.1$ and 0.05 s overlap closely, indicating that time-step errors are negligible at 0.1 s. In contrast, the 0.2 s step shows minor discrepancies during the melting period. Based on these findings, $\Delta t = 0.1$ s was adopted for all subsequent simulations, offering a reliable compromise between accuracy and computational efficiency.

3.2. Validation

The numerical model developed in this study was validated by replicating the melting process of RT82 PCM in a triplex-tube heat exchanger with internal–external fins, as investigated by Mat et al. [50].

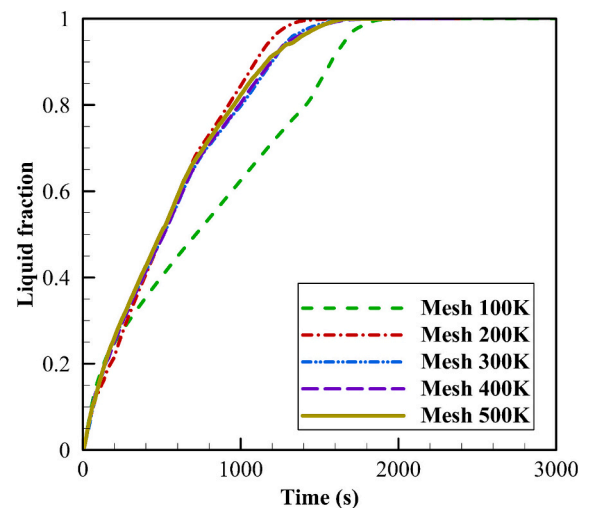


Fig. 2. Mesh independence study showing the effect of grid resolution on the results.

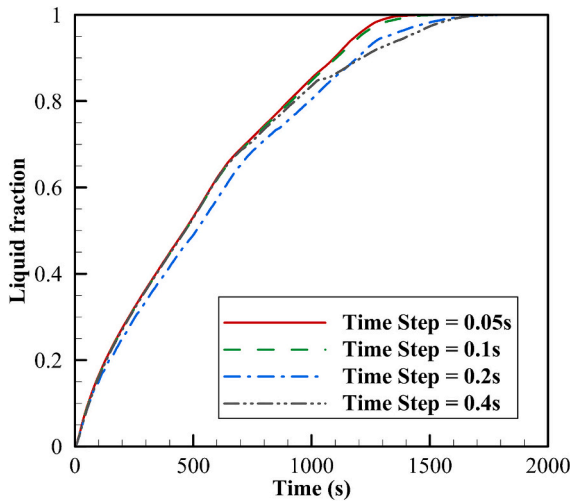


Fig. 3. Time-step independence study for the 400 K-element mesh, showing that results for $\Delta t = 0.05$ s and 0.1 s are nearly identical.

Their study provides detailed experimental data on the melting behavior of RT82 PCM under various operating conditions, making it an ideal benchmark for validating computational models.

In the replication study, we employed identical geometric configurations, boundary conditions, and thermophysical properties as those used by Mat et al. [50]. The HTF was modeled with the same inlet temperature and flow rate, and the PCM was initialized at its solid state. The numerical simulations were conducted using the same mesh density and time-stepping criteria to ensure consistency in the comparison.

The PCM used in both the model validation and the subsequent parametric investigations is RT82 paraffin wax. Employing the same PCM in all simulations ensures consistency between the validation results and the performance trends reported in the main study [16,54]. The properties of the materials utilized have been listed in Table 2.

Fig. 4 presents the comparison between the present simulation results and the experimental data reported by Mat et al. The figure illustrates the temporal evolution of the liquid fraction within the PCM, highlighting the accuracy of the numerical model in capturing the melting process. A rigorous comparison between the present numerical model and the numerical results of Mat et al. [50] confirms an excellent quantitative agreement in both the mean temperature and liquid fraction evolution of the PCM. As illustrated in Fig. 4, the predicted temperature profile from the present study precisely follows the numerical data of Mat et al., with a maximum deviation below approximately 1–2 °C throughout the melting period. At 10 min, both studies report an average PCM temperature of nearly 55 °C; at 30 min, the temperature

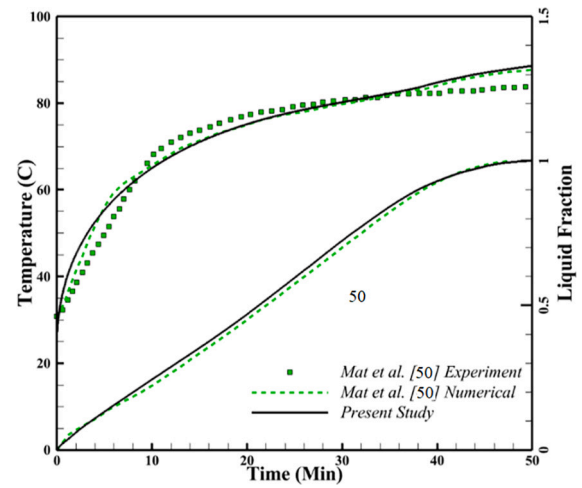


Fig. 4. Comparison of the present simulation results with experimental data from Mat et al. [50] for the melting process of RT82 PCM in a triplex-tube heat exchanger with internal–external fins.

reaches about 78 °C in Mat et al.'s simulation and 79 °C in the present model; and at the end of the melting process (45 min), both attain approximately 90 °C. A similar level of consistency is observed for the liquid fraction evolution. The current model predicts a liquid fraction of about 0.34 at 15 min, 0.70 at 30 min, and 1.0 at 45 min, which are nearly identical to the values of 0.33, 0.68, and 1.0 reported in Mat et al. [50].

The strong agreement observed in the temperature profiles, including the shape during both solid and liquid phases, the intermediate transition times, and the final values, demonstrates that the numerical framework is reliable for evaluating the effects of fin number, size, and orientation under the consistent boundary conditions applied in this study.

4. Results and discussion

The thermal performance of the storage unit is evaluated using melting time, heat storage rate, liquid fraction evolution, and spatially averaged temperature. These indicators are closely linked to the dominant heat transfer mechanisms governing the phase change process. In particular, variations in melting time reflect changes in the effective thermal resistance between the heat transfer surface and the bulk PCM, while the heat storage rate represents the combined contribution of sensible heating and latent heat absorption. Moreover, the evolution of liquid fraction and temperature fields provides insight into the relative roles of conductive heat transfer and buoyancy-driven convection. Accordingly, the following parametric results are interpreted not only in terms of quantitative performance metrics but also with respect to their influence on convective flow structure, conductive heat transfer pathways, and phase-front propagation.

4.1. Wave length

Figs. 5 and 6 illustrate the effect of wavy wall wavelength on PCM melting for Cases 1 (31.4 mm), 2 (21 mm), and 3 (15.7 mm) at 300, 600, and 1200 s. At 300 s, partial melting occurs near the heated surface for all cases, with Case 2 showing a larger melted region than Cases 1 and 3. By 600 s, Case 2 maintains the fastest melting, while Case 1 lags with a lower liquid fraction. At 1200 s, Case 2 nearly reaches complete melting, followed by Case 3, whereas Case 1 remains slowest. These results indicate that reducing wavelength enhances heat transfer by increasing surface contact with the fluid; however, extremely short wavelengths (Case 3) offer limited benefit due to flow resistance. An intermediate wavelength of 21 mm (Case 2) provides the best balance, maximizing

Table 2
Thermophysical properties of materials used in the present study.

Material	Property	Value
RT82 PCM	Melting temperature range (°C)	77–82
	Latent heat (J·kg ⁻¹)	~170,000
	Density (solid/liquid) (kg·m ⁻³)	~880 / ~770
	Thermal conductivity (W·m ⁻¹ ·K ⁻¹)	~0.2
	Specific heat (J·kg ⁻¹ ·K ⁻¹)	~2000
Fin material (Aluminium)	Density (kg·m ⁻³)	2700
	Thermal conductivity (W·m ⁻¹ ·K ⁻¹)	237
	Specific heat (J·kg ⁻¹ ·K ⁻¹)	900
	Density (kg·m ⁻³)	8960
Heat exchanger wall (Copper)	Density (kg·m ⁻³)	8960
	Thermal conductivity (W·m ⁻¹ ·K ⁻¹)	401
	Specific heat (J·kg ⁻¹ ·K ⁻¹)	385

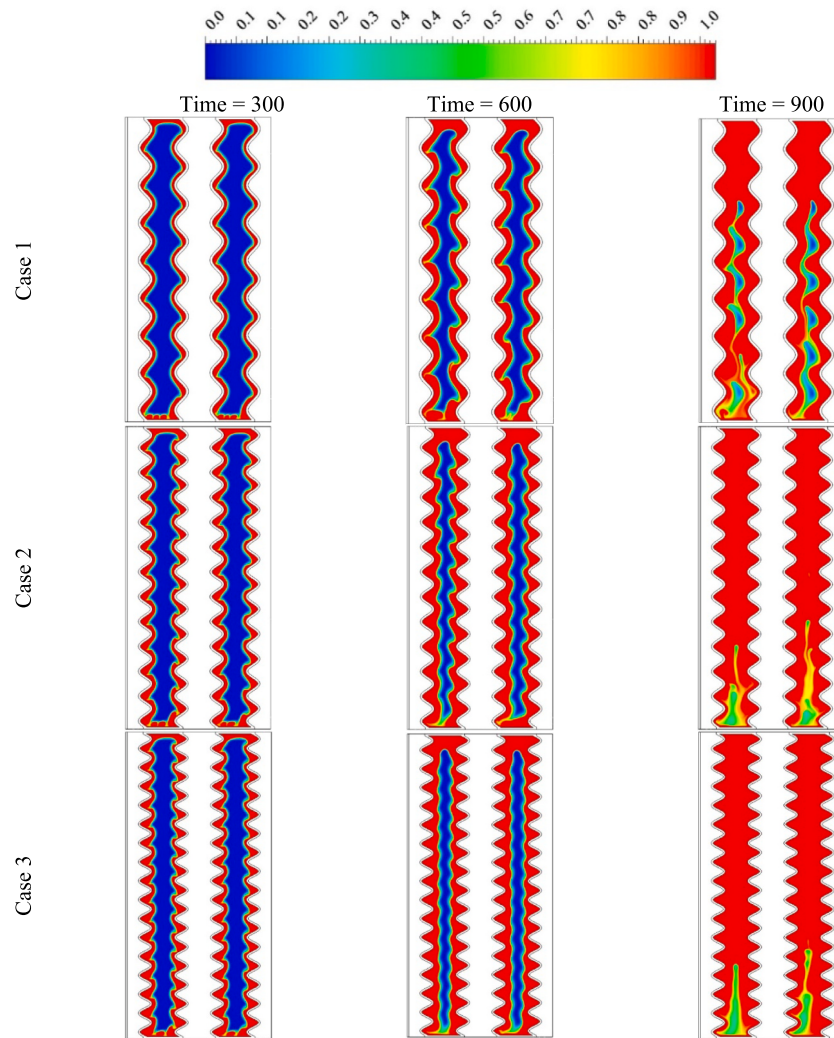


Fig. 5. Liquid fraction contours of PCM melting at different time intervals (300 s, 600 s, 1200s) for Cases 1, 2, and 3, illustrating the impact of varying wavy wall wavelengths on melting performance.

melting efficiency while avoiding the diminishing returns observed in Case 3. Very long wavelengths reduce surface area exposure and slow heat absorption, while excessively short wavelengths hinder fluid circulation.

Temperature contours show similar trends: at 300 s, Cases 2 and 3 have larger high-temperature regions than Case 1. By 600 s, Case 2 achieves more uniform thermal distribution, indicating enhanced mixing, whereas Case 1 retains strong gradients. At 1200 s, Case 2 exhibits near-uniform temperatures, Case 3 still has localized gradients, and Case 1 remains least efficient. Temperature distribution closely correlates with melting performance, reinforcing that the intermediate wavelength optimizes thermal penetration and phase change rates.

Overall, these results confirm that geometry modification through wavelength adjustment is critical for accelerating PCM melting. Case 2, with an intermediate wavelength, demonstrates the most balanced and efficient thermal performance, highlighting the importance of optimizing wall geometry to overcome PCM's low thermal conductivity and enhance energy storage efficiency.

Fig. 7 compares PCM melting times and temperatures for Cases 1 (31.4 mm), 2 (21 mm), and 3 (15.7 mm), showing the impact of wavy wall wavelength on thermal performance. Case 1 exhibits the longest melting time (1820 s) and lowest heat storage rate, while Case 2 achieves the shortest melting time (1421 s) and highest energy storage, indicating superior thermal enhancement. Case 3 performs slightly

worse than Case 2 (1463 s) but remains significantly better than Case 1. These results highlight that intermediate wavelength optimize the balance between surface area and fluid circulation, whereas excessively short or long wavelengths reduce efficiency due to flow resistance or limited heat exchange. Case 2 demonstrates nearly a 22% reduction in melting time and a 26% increase in heat storage rate relative to Case 1. Overall, the study confirms that careful geometric optimization of wavy walls is essential to enhance both conductive and convective heat transfer, accelerating PCM melting and improving TES performance.

Table 3 compares PCM melting time and heat storage rate for Cases 1 (31.4 mm), 2 (21 mm), and 3 (15.7 mm). Case 1 shows the slowest melting (1820 s) and lowest heat storage (502.27 W). Case 2 achieves the best performance, reducing melting time to 1421 s (−21.9%) and increasing heat storage to 634.92 W (+26.4%). Case 3 also improves over Case 1 (1463 s, 624.23 W) but lags slightly behind Case 2 (−2.9% melting time, +1.7% heat storage). These results indicate that intermediate wavelengths optimize the balance between surface area and fluid circulation, while excessively small or large wavelengths limit performance. Case 2 represents the optimal configuration for efficient PCM thermal storage.

Although the melting behavior is primarily illustrated using temperature and liquid fraction contours, the underlying heat transfer mechanism is governed by buoyancy-driven convection within the molten PCM. As melting progresses, density gradients induced by

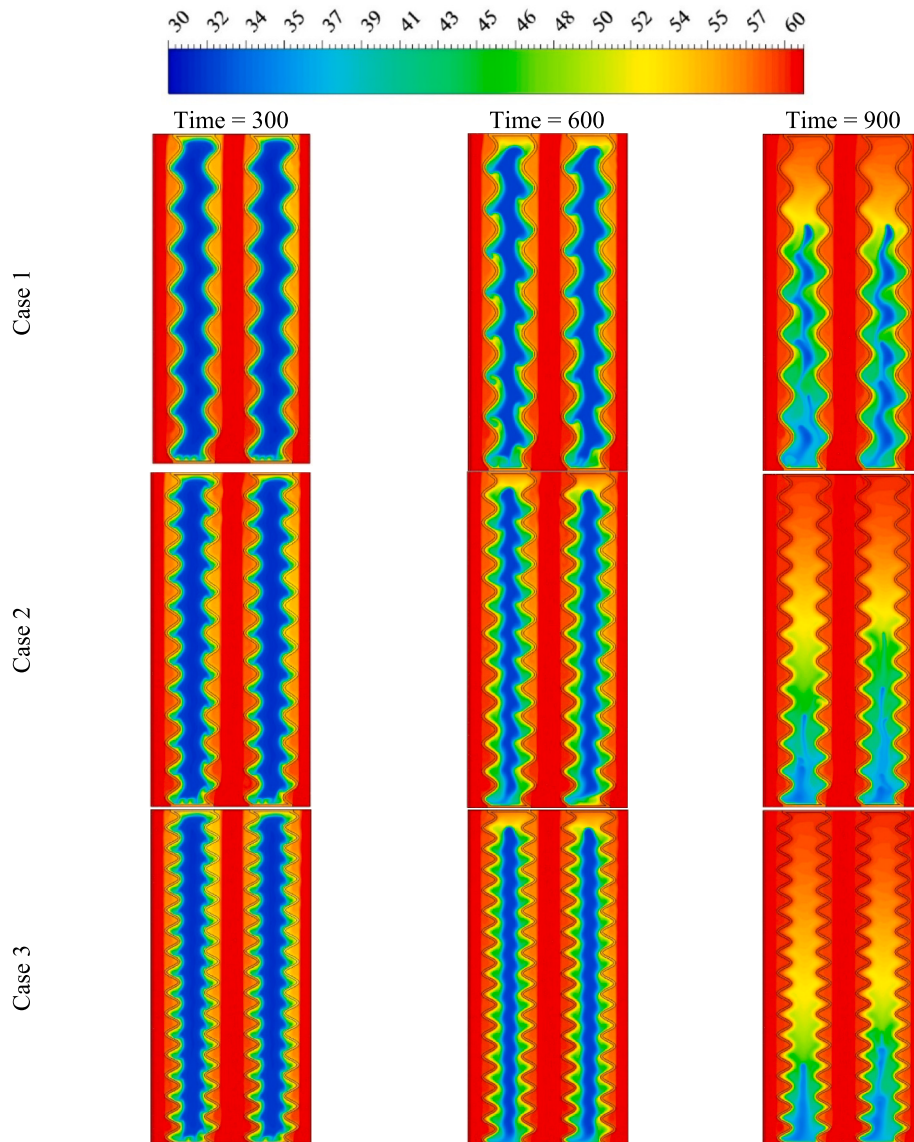


Fig. 6. Temperature contours of PCM melting at different time intervals (300 s, 600 s, 1200s) for Cases 1, 2, and 3, illustrating the impact of varying wavy wall wavelengths on melting performance.

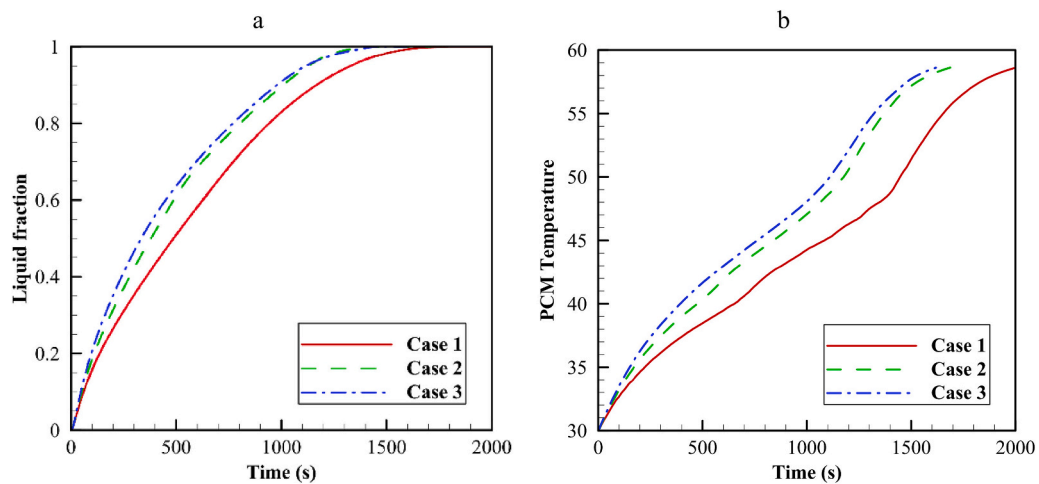


Fig. 7. Comparison of (a) melting times (b) PCM temperatures across Cases 1, 2, and 3, showing the impact of wavelength of the wavy wall geometry on phase change completion rates.

Table 3

Heat storage rate performance (W) and the melting time for Cases 1–3, quantifying the TES capacity enhancement through geometric optimization.

Cases	Melting time (s)	Heat storage rate (W)
Case 1 (Wavelength 31.4)	1820	502.27
Case 2 (Wavelength 21)	1421	634.92
Case 3 (Wavelength 15.7)	1463	624.23

temperature differences generate ascending hot plumes along the heated corrugated walls and descending cooler streams near solid–liquid interfaces. The introduction of wall corrugation disrupts the formation of a single dominant circulation cell and promotes multiple localized vortices, which enhances fluid mixing and sustains higher convective heat transfer coefficients.

Furthermore, the presence of fins and multi-shell interfaces divides the PCM domain into smaller sub-regions, effectively shortening convection loop lengths and increasing the frequency of flow impingement on solid–liquid fronts. This mechanism explains the accelerated melting observed in optimized cases compared with smooth-wall configurations, where convection becomes stratified at later stages and heat transfer

reverts toward conduction-dominated behavior. Although velocity fields are not explicitly shown, the consistent acceleration of melting and elevated heat storage rates across cases with increased corrugation amplitude and shell number provide indirect but robust evidence of intensified convective motion. Future work will incorporate detailed velocity and streamline visualizations to quantitatively characterize these flow structures and further refine the mechanistic understanding of geometry-driven convection enhancement in LHTES systems.

4.2. Wave amplitude

Figs. 8 and 9 illustrate the effect of wavy wall amplitude on PCM melting for Cases 1 (5 mm), 4 (7.5 mm), and 5 (10 mm) at 300, 600, and 900 s. At 300 s, Case 5 already exhibits a substantially larger liquid fraction than Cases 1 and 4, indicating faster initial melting. By 600 s, Case 5 maintains nearly uniform melting, Case 4 shows moderate progress, and Case 1 lags with significant solid PCM remaining. At 900 s, Case 5 is almost fully melted, Case 4 demonstrates advanced melting with limited solid zones, and Case 1 still has incomplete phase change. This progression confirms that increasing wall amplitude accelerates melting by enhancing natural convection and increasing surface area

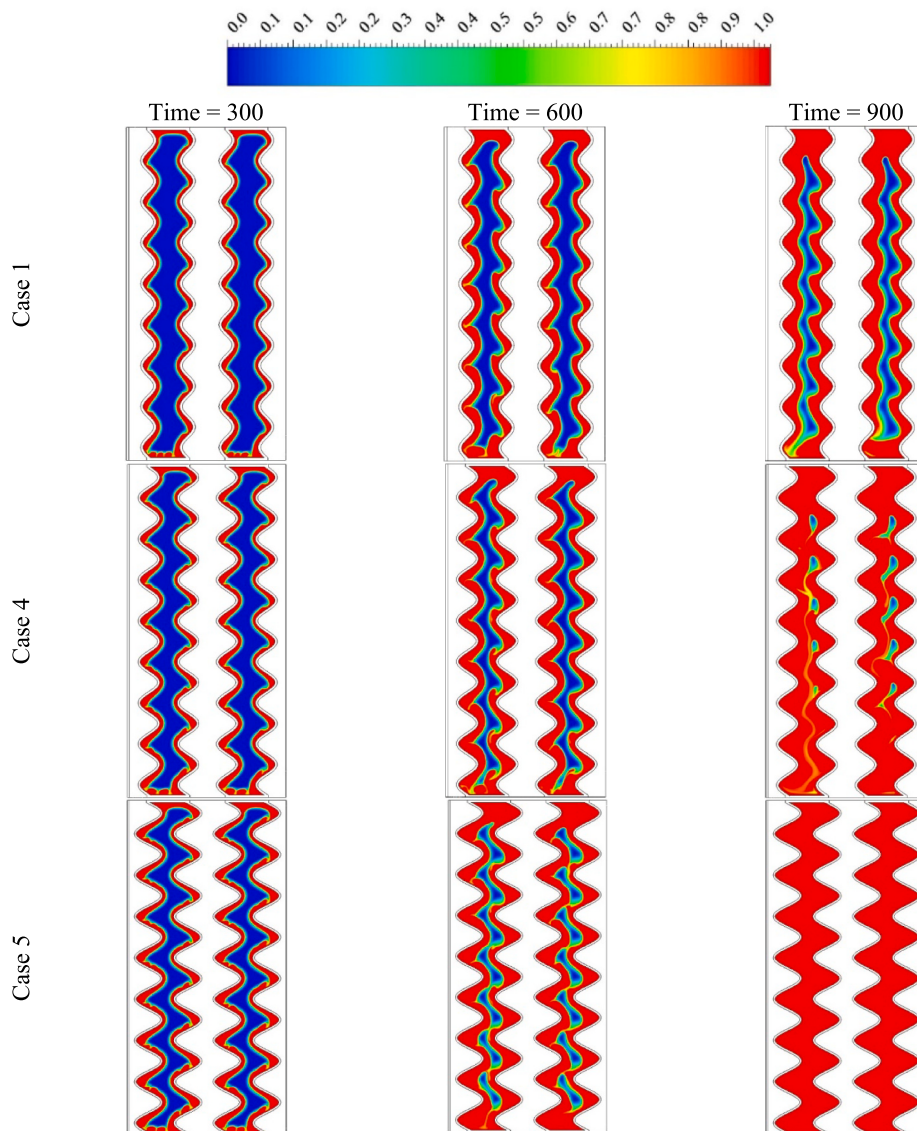


Fig. 8. Liquid fraction contours of PCM melting at different time intervals (300 s, 600 s, 900 s) for Cases 1, 4, and 5, illustrating the impact of varying wavy wall wave amplitude on melting performance.

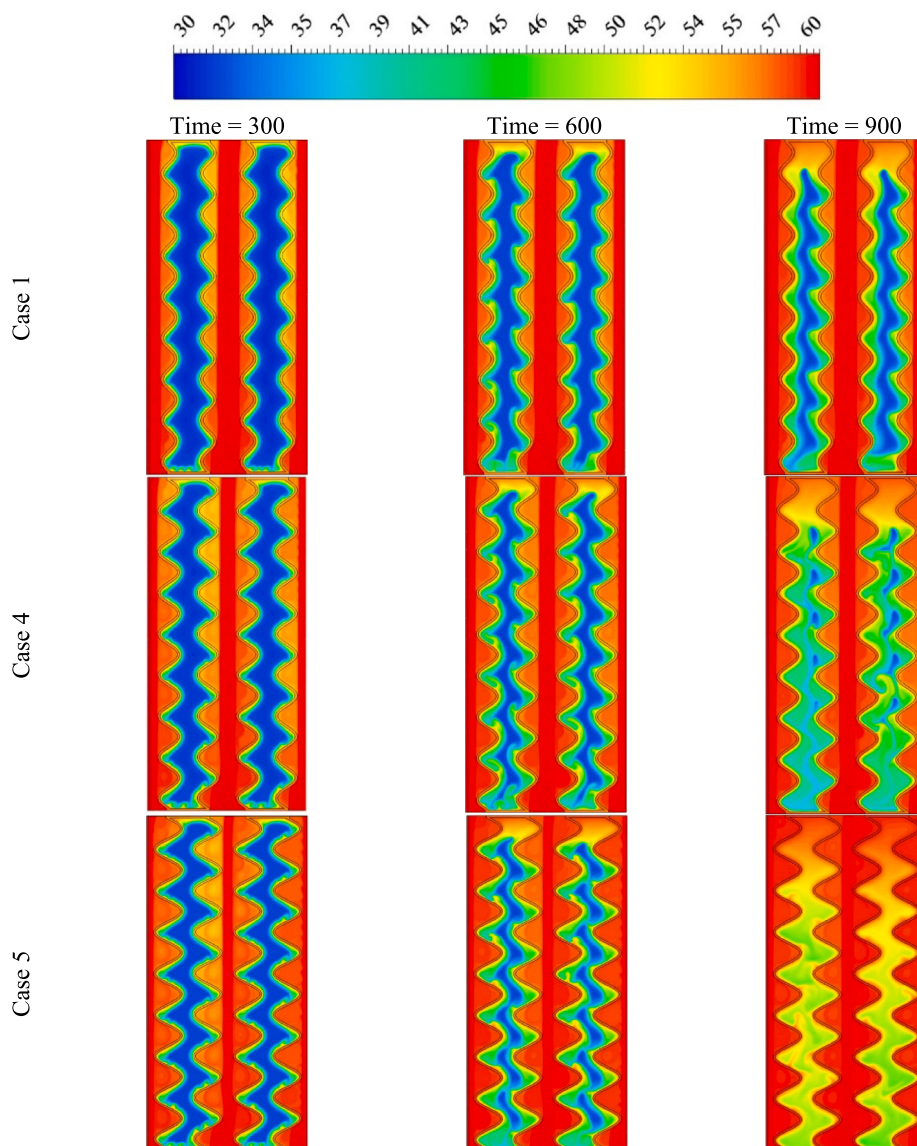


Fig. 9. Temperature contours of PCM melting at different time intervals (300 s, 600 s, 900 s) for Cases 1, 4, and 5, illustrating the impact of varying wavy wall wave amplitude on melting performance.

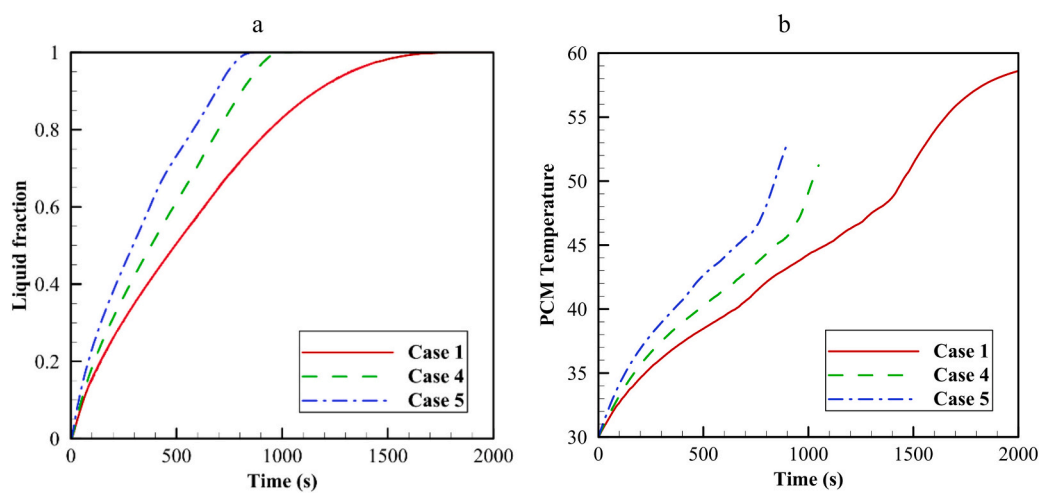


Fig. 10. Comparison of a) melting times b) PCM temperatures across Cases 1, 4, and 5, showing the impact of wavy amplitude of the PCM wall on phase change completion rates.

exposure, which promotes fluid circulation and heat transfer.

Temperature contours corroborate these observations. At 300 s, Case 5 shows deeper heat penetration compared to Cases 1 and 4, which remain near the heated wall. By 600 s, Case 5 exhibits uniform temperature distribution with minimal gradients, Case 4 shows moderate improvement, and Case 1 retains strong stratification. At 900 s, Case 5 reaches nearly uniform high temperatures, Case 4 shows incomplete spread, and Case 1 continues to suffer from slow heat penetration. These results highlight that larger amplitudes improve both melting rate and thermal uniformity through enhanced convective transport.

Overall, Cases 5 and 4 outperform the baseline Case 1, with Case 5 demonstrating the fastest melting and most uniform temperature distribution. However, higher amplitudes increase geometric complexity, potentially affecting manufacturability and structural stability. Therefore, while amplitude increase is highly effective for PCM melting enhancement, practical design requires identifying an optimal amplitude range that balances thermal performance with real-world feasibility.

Fig. 10 compares PCM melting times and temperatures for Cases 1 (5 mm), 4 (7.5 mm), and 5 (10 mm), highlighting the effect of wavy wall amplitude. Increasing amplitude significantly reduces melting time and improves thermal response. Case 1 exhibits the slowest melting, Case 4 shows marked improvement, and Case 5 achieves the fastest melting with nearly uniform temperature rise, indicating enhanced heat transfer and convective mixing. Quantitatively, Case 5 melts almost twice as fast as Case 1 and achieves the highest heat storage rate, while Case 4 provides intermediate performance. The smaller difference between Cases 4 and 5 compared to the jump from Case 1 to Case 4 suggests diminishing returns for very large amplitudes. These results confirm that amplitude adjustment is effective for accelerating PCM melting, with higher amplitudes maximizing surface contact and strengthening natural convection. Case 5 delivers the best thermal performance, while Case 4 offers a practical compromise, balancing efficiency with potential design and manufacturing constraints.

Table 4 compares PCM melting time and heat storage rate for Cases 1 (5 mm), 4 (7.5 mm), and 5 (10 mm), showing the impact of wavy wall amplitude. Case 1 has the longest melting time (1820 s) and lowest heat storage (502.27 W). Case 4 reduces melting time by 45.2% to 998 s and increases heat storage by 68.1% to 844.74 W. Case 5 achieves the shortest melting time (876 s, -51.9%) and highest heat storage (990.02 W, +97.1%). Improvements are largest from small to medium amplitude, with diminishing gains at higher amplitudes. Case 5 maximizes thermal performance, while Case 4 offers a practical balance between efficiency and design feasibility.

Although the effects of corrugated-wall wavelength and amplitude were examined separately to isolate their individual influences, their impact on melting behavior is inherently coupled because both parameters simultaneously modify the effective heat transfer area and the strength of buoyancy-driven convection within the PCM. Increasing the amplitude intensifies wall undulation, enlarges the contact area, and enhances convective circulation, whereas decreasing the wavelength increases the number of corrugation cycles, further augmenting surface area but also introducing greater geometric complexity and flow resistance.

The present results indicate the presence of a synergistic regime at intermediate wavelength and moderate-to-high amplitude, where

surface area enhancement and convective mixing act cooperatively. For instance, the intermediate wavelength (21 mm) promotes effective circulation without excessive flow obstruction, and its thermal benefit is amplified when combined with larger amplitudes that strengthen convective cells. In contrast, a trade-off emerges when short wavelengths are combined with high amplitudes: although surface area continues to increase, excessive wall curvature can impede fluid motion and weaken convective transport, leading to diminishing thermal returns. At long wavelengths, the effectiveness of increasing amplitude is similarly limited by insufficient corrugation density, which restricts heat penetration into the PCM core.

These findings demonstrate that corrugated-wall design should not treat wavelength and amplitude as independent tuning parameters but as a coupled geometric pair. Within the investigated parameter space, an intermediate wavelength combined with a relatively high amplitude provides the most favorable balance between surface area growth and flow accessibility, yielding improved melting rate and heat storage efficiency. This interaction-based interpretation enhances the design relevance of the present study and provides practical guidance for selecting corrugation geometries in fin-assisted LHTEs systems.

4.3. Fin arrangement

Figs. 11 and 12 illustrate the influence of fin arrangement on PCM melting performance for Cases 6, 7, and 8 at 200, 400, and 600 s. At 200 s, Case 6 exhibits the largest melted region and widespread high-temperature zones, followed by Case 7, while Case 8 lags with smaller liquid fractions and localized heating near the wall. By 400 s, melting in Case 6 progresses uniformly across the PCM, Case 7 shows moderate improvement, and Case 8 remains delayed. At 600 s, Case 6 achieves near-complete melting and almost uniform temperature distribution, Case 7 reaches an advanced but uneven melting stage, and Case 8 retains substantial unmelted PCM with strong thermal gradients.

These results highlight that fin arrangement critically affects heat transfer enhancement. Case 6's optimized fin placement maximizes conduction pathways and strengthens natural convection, ensuring rapid and balanced temperature rise, which accelerates melting. Case 7 benefits from fins but shows uneven heat distribution due to suboptimal arrangement. Case 8, despite having fins, performs poorly, indicating that inappropriate positioning can create stagnant zones or hinder convection, undermining thermal effectiveness.

Figs. 11 and 12 demonstrate that the mere presence of fins is insufficient for performance gains. Optimal fin arrangement is essential to fully exploit their capacity, enhancing both conduction and convective heat transfer, avoiding thermal stagnation, and ensuring uniform melting. Case 6 emerges as the most effective configuration, providing the fastest and most uniform PCM melting, while Cases 7 and 8 illustrate that suboptimal designs can significantly reduce thermal performance despite fin inclusion.

Fig. 13 compares melting times and PCM temperature evolution for Cases 6, 7, and 8, highlighting the impact of fin arrangement on thermal performance. Case 6 (arrangement A) shows the shortest melting time and fastest temperature rise, outperforming the others by promoting both conduction through the fins and convection within the PCM. Case 7 (arrangement B) provides moderate improvement, with longer melting time and less uniform heat distribution, while Case 8 (arrangement C), despite having fins, performs poorly due to inefficient placement that creates stagnant thermal zones. These results demonstrate that fin presence alone does not guarantee enhanced melting; strategic arrangement is critical. Case 6 ensures rapid temperature equalization, reduced thermal gradients, and superior heat storage, whereas Case 7 shows partial uniformity and Case 8 exhibits significant temperature differences. Overall, the figure confirms that optimal fin configuration not only accelerates melting but also improves thermal consistency and energy storage, providing clear design guidance for practical PCM applications.

Table 4

Heat storage rate performance (W) and the melting time (s) for Cases 1, 4, and 5 quantifying the TES capacity enhancement through changing the wave amplitude of the wavy wall.

Cases	Melting time (s)	Heat storage rate (W)
Case 1 (Wave amp. 5)	1820	502.27
Case 4 (Wave amp. 7.5)	998	844.74
Case 5 (Wave amp. 10)	876	990.02

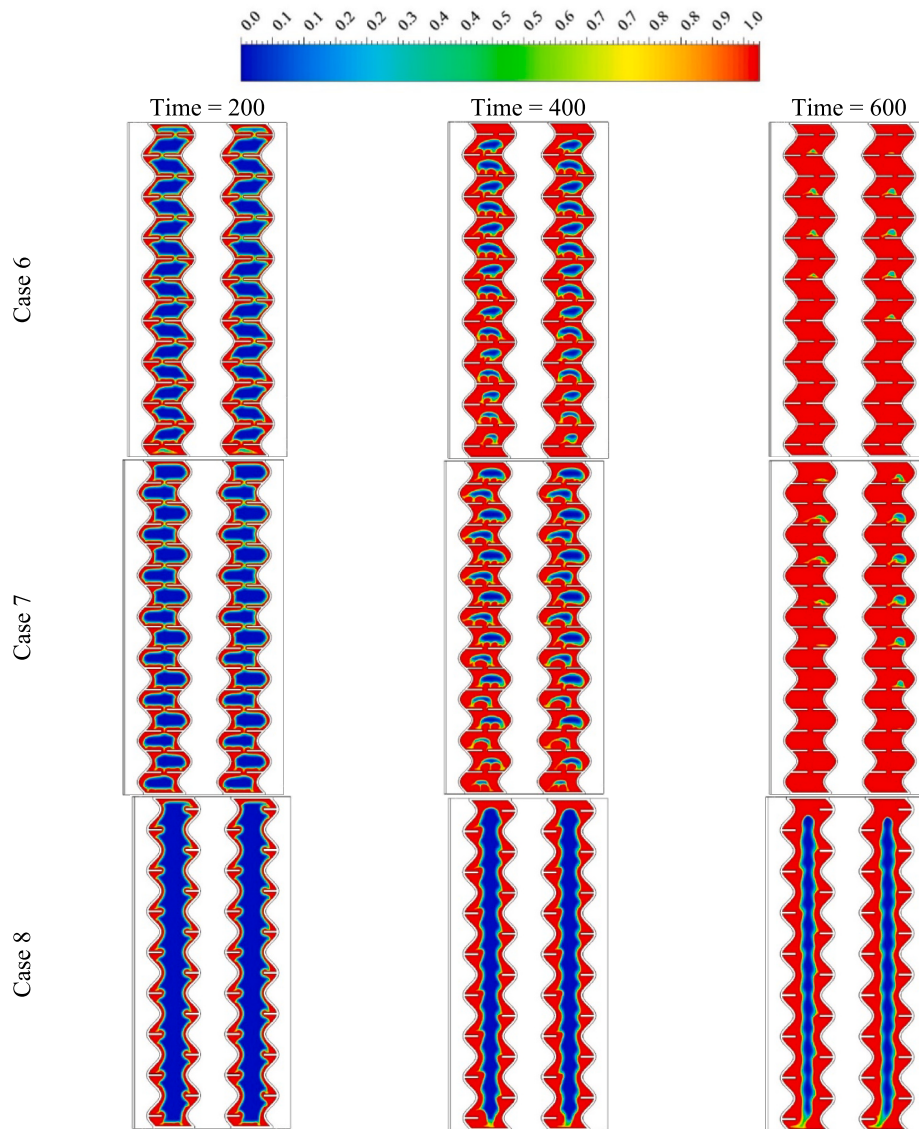


Fig. 11. Liquid fraction contours of PCM melting at different time intervals (200 s, 400 s, 600 s) for Cases 6, 7, and 8, illustrating the impact of varying fins arrangement on melting performance.

Table 5 highlights the critical role of fin arrangement on PCM thermal performance. Case 6 (arrangement A) delivers the highest heat storage rate (1355.81 W) and shortest melting time (657 s). Case 7 (arrangement B) shows moderate reduction, with a 6.3% lower heat storage rate and 7.9% longer melting time. Case 8 (arrangement C) performs worst, with a 50.1% decrease in heat storage and a 103.2% longer melting time, due to disrupted convection and isolated PCM zones. These results emphasize that fin placement, not just presence, dictates efficiency. Optimal arrangements, like Case 6, maximize conduction and natural convection, ensuring rapid melting and superior energy storage.

From a heat transfer perspective, the reduction in melting time and the increase in heat storage rate observed for the improved configurations arise from a simultaneous shortening of conductive heat transfer paths and intensification of convective circulation within the molten PCM. The geometric modifications promote more uniform temperature fields and suppress thermally stagnant regions, thereby lowering the overall thermal resistance of the storage unit.

4.4. Fin length

Figs. 14 and 15 illustrate the effect of fin length on PCM melting, showing that while fin length influences performance, its impact is less pronounced than fin arrangement. Fig. 14 presents liquid fraction contours for Cases 6 (12.3 mm), 9 (11.3 mm), and 10 (10.3 mm). Cases 6 and 9 demonstrate nearly identical melting progress, achieving near-complete phase change by 600 s, whereas Case 10 exhibits a slightly smaller melted region, indicating a minor delay. This suggests that small reductions in fin length have negligible effect, but further shortening begins to hinder conduction.

Temperature contours in Fig. 15 reinforce these observations. Heat penetration is most uniform and rapid in Case 6, almost identical in Case 9, while Case 10 shows localized heating and stronger thermal gradients at 400 s. The minimal difference between Cases 6 and 9 indicates a threshold beyond which longer fins provide diminishing returns, as the primary role of fins is to overcome PCM's low thermal conductivity.

Quantitatively, the performance gap between the longest (Case 6) and shortest fins (Case 10) is modest, with a 3.3% longer melting time and 5.1% lower heat storage rate for Case 10. These results demonstrate that fin length is a secondary design parameter compared to

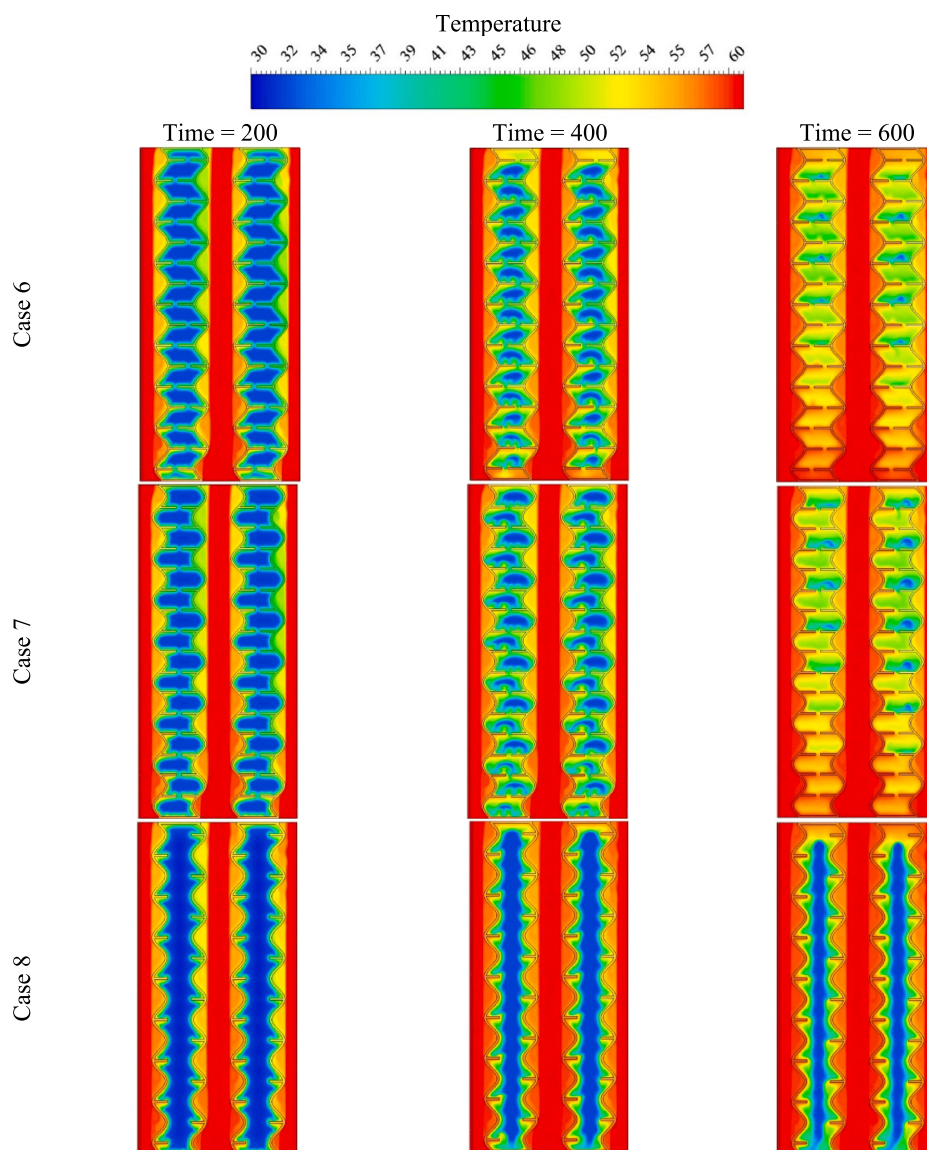


Fig. 12. Temperature contours of PCM melting at different time intervals (200 s, 400 s, 600 s) for Cases 6, 7, and 8, illustrating the impact of varying fins arrangement on melting performance.

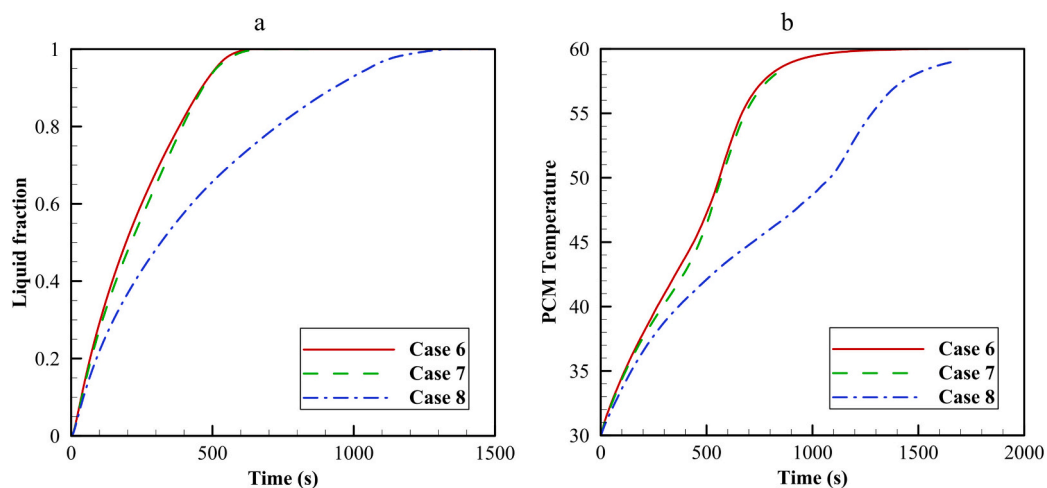


Fig. 13. Comparison of a) melting times b) PCM temperatures across Cases 6, 7, and 8, showing the impact of fins arrangements on the PCM wall on phase change completion rates.

Table 5

Heat storage rate performance (W) and the melting time for Cases 6, 7, and 8, quantifying the TES capacity enhancement through changing the fins arrangement.

Cases	Melting time (s)	Heat storage rate (W)
Case 6	657	1355.81
Case 7	709	1270.06
Case 8	1335	676.64

arrangement. Engineers can slightly reduce fin length to save material and weight with minimal performance penalty, provided the fin placement is optimized, ensuring efficient heat distribution and effective PCM melting.

Fig. 16 quantitatively illustrates the effect of fin length on PCM melting time and temperature, supporting observations from Figs. 14 and 15. Case 9 (11.3 mm) achieves the shortest melting time at 654 s, marginally faster than Case 6 (12.3 mm, 657 s), while Case 10 (10.3 mm) requires 679 s, showing a measurable delay. This indicates a point of diminishing returns, where increasing fin length beyond ~ 11.3 mm offers negligible benefit, but shortening below this optimal range begins to reduce performance.

Temperature evolution confirms this trend: Cases 6 and 9 display nearly identical heating trajectories, whereas Case 10 lags consistently, reflecting slower melting. Reducing fin length from 12.3 mm to 10.3 mm ($\sim 18\%$) increases melting time by only 3.4%, suggesting material and weight savings are possible with minimal performance loss. Overall, fin length is a secondary factor compared to arrangement, and an optimal range exists where thermal performance is maximized, balancing efficiency, material use, and cost-effectiveness in PCM thermal storage systems.

Table 6 quantifies the effect of fin length on thermal performance, showing measurable yet modest differences compared to the stronger influence of fin arrangement. Case 9 (11.3 mm) achieves the best balance, with the shortest melting time of 654 s and a heat storage rate of 1342.28 W. Case 6 (12.3 mm) performs nearly identically, melting 0.5% slower but exhibiting only 1.0% higher heat storage. Case 10 (10.3 mm) shows a noticeable decline, with a 3.8% longer melting time and 4.1% lower heat storage rate, confirming that excessively short fins hinder conduction and convection. The negligible gap between Cases 6 and 9 suggests diminishing returns beyond ~ 11.3 mm, where added fin material provides no significant advantage and may even restrict flow. Thus, fin length plays a secondary yet tunable role: designers can adopt

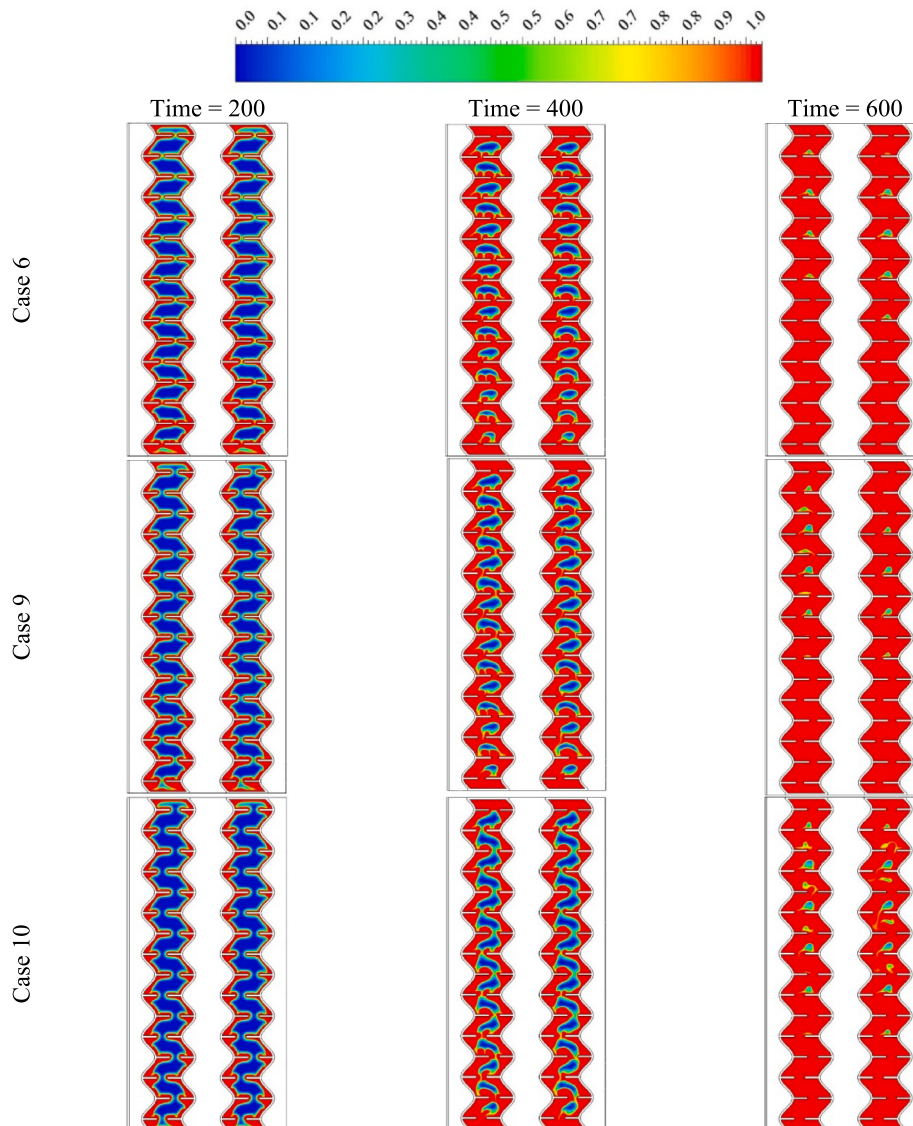


Fig. 14. Liquid fraction contours of PCM melting at different time intervals (200 s, 400 s, 600 s) for Cases 6, 9, and 10, illustrating the impact of varying fins length on melting performance.

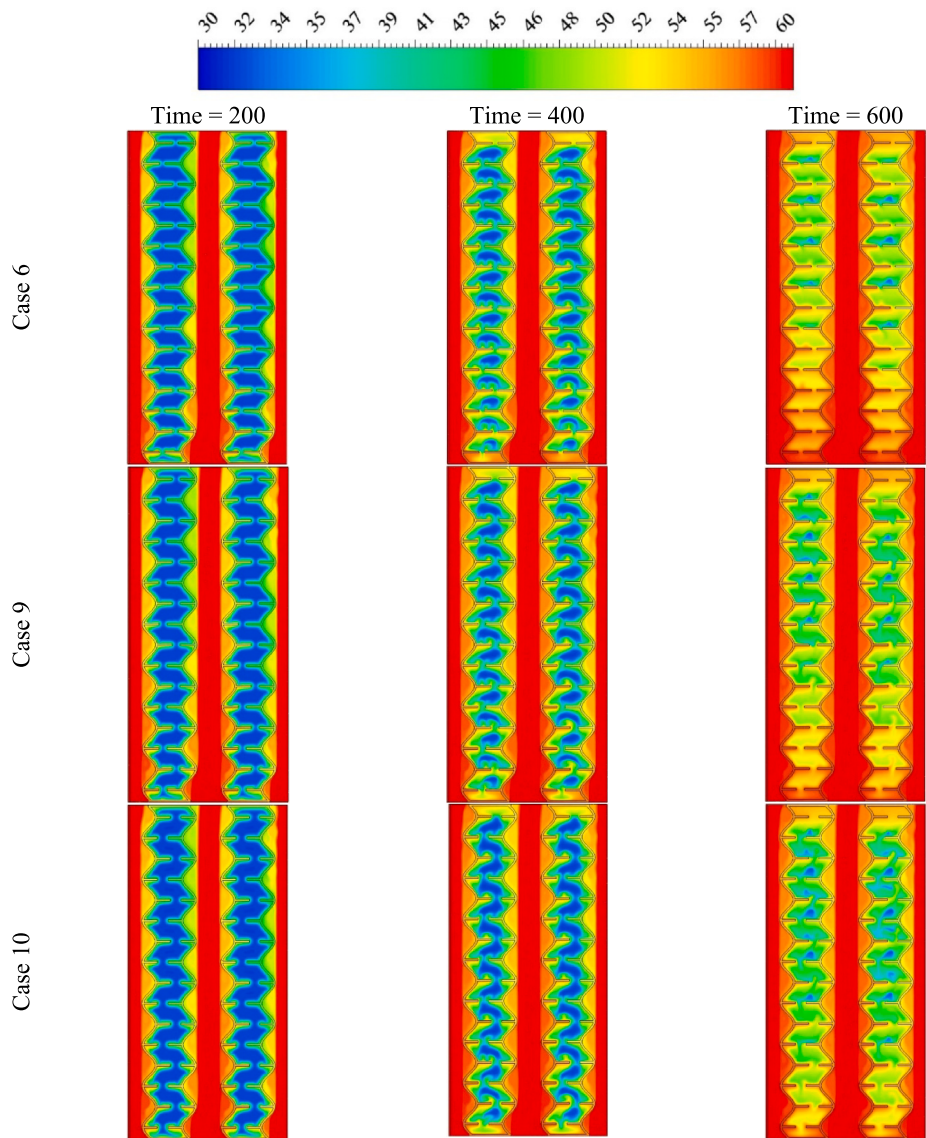


Fig. 15. Temperature contours of PCM melting at different time intervals (200 s, 400 s, 600 s) for Cases 6, 9, and 10, illustrating the impact of varying fins length on melting performance.

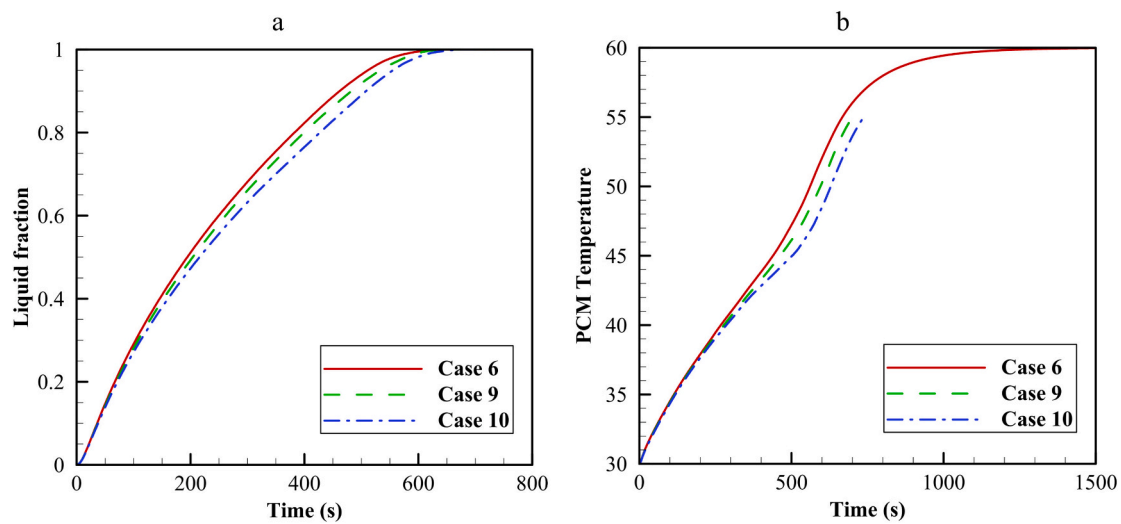


Fig. 16. Comparison of a) melting times b) PCM temperatures across Cases 6, 9, and 10, showing the impact of fins lengths on the PCM's thermal performance.

Table 6
Heat storage rate performance (W) and the melting time (s) for Cases 6, 9, and 10, quantifying the TES capacity enhancement through changing the fins length.

Cases	Melting time (s)	Heat storage rate (W)
Case 6 (Fin length = 12.3 mm)	657	1355.81
Case 9 (Fin length = 11.3 mm)	654	1342.28
Case 10 (Fin length = 10.3 mm)	679	1287.07

the 11.3 mm configuration to minimize material and weight while maintaining near-optimal performance. Overall, fin arrangement remains the dominant parameter governing PCM melting efficiency.

4.5. Number of PCM

Figs. 17 and 18 vividly demonstrate the significant influence of increasing the number of PCM shells on the melting process, fully consistent with the quantitative results in Table 7. Fig. 17 (liquid fraction contours) illustrates a pronounced acceleration in melting as the number of shells increases from one to three. At 200 s, Case 11 (single shell) shows melting limited to the region adjacent to the heat transfer wall, whereas Case 12 (two shells) and Case 13 (three shells) display

extensive melting across their additional annular regions, with Case 13 clearly most advanced. This trend continues at 400 and 600 s, where Case 13 reaches near-complete melting significantly earlier than the others. The results indicate that each added shell acts as an additional heat transfer interface, multiplying the effective surface area and enabling simultaneous melting fronts that absorb energy from multiple directions rather than relying on a single heating path.

Fig. 18 (temperature contours) further elucidates the mechanism behind this enhancement. In Case 11, steep temperature gradients remain, with high temperatures concentrated near the inner wall and much cooler regions persisting outward, a signature of conduction-dominated behavior. In contrast, Cases 12 and 13 exhibit far more uniform temperature distributions, showing rapid and balanced heat propagation throughout the PCM domain. The multi-shell structure drastically shortens conductive distances and promotes multidirectional heat diffusion, converting the system from a slow, conduction-limited configuration into a high-efficiency thermal storage unit. This structural modification explains the sharp reduction in melting time and the substantial gain in heat storage capacity observed experimentally.

Fig. 19 quantitatively confirms the strong performance enhancement achieved by increasing the number of PCM shells, complementing the

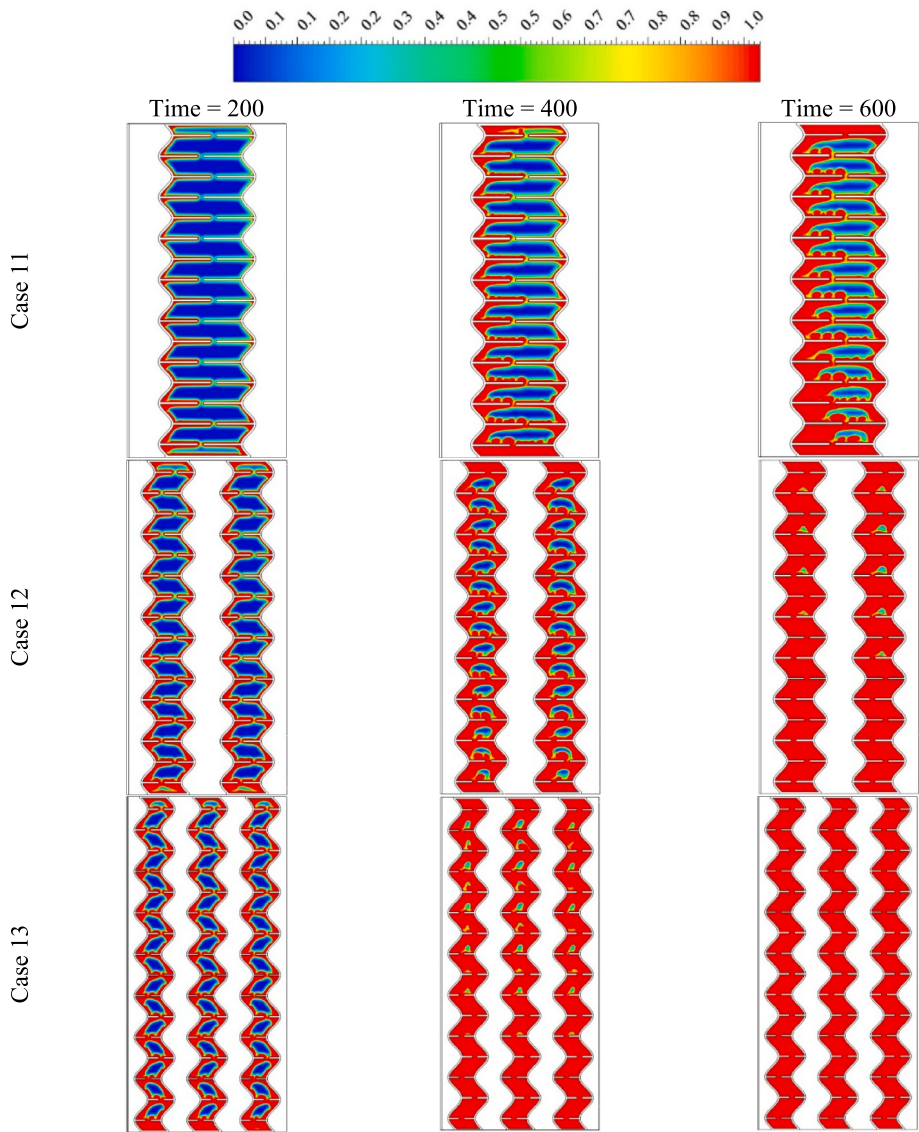


Fig. 17. Liquid fraction contours of PCM melting at different time intervals (200 s, 400 s, 600 s) for Cases 11, 12, and 13, illustrating the impact of varying number of the PCM shell on melting performance.

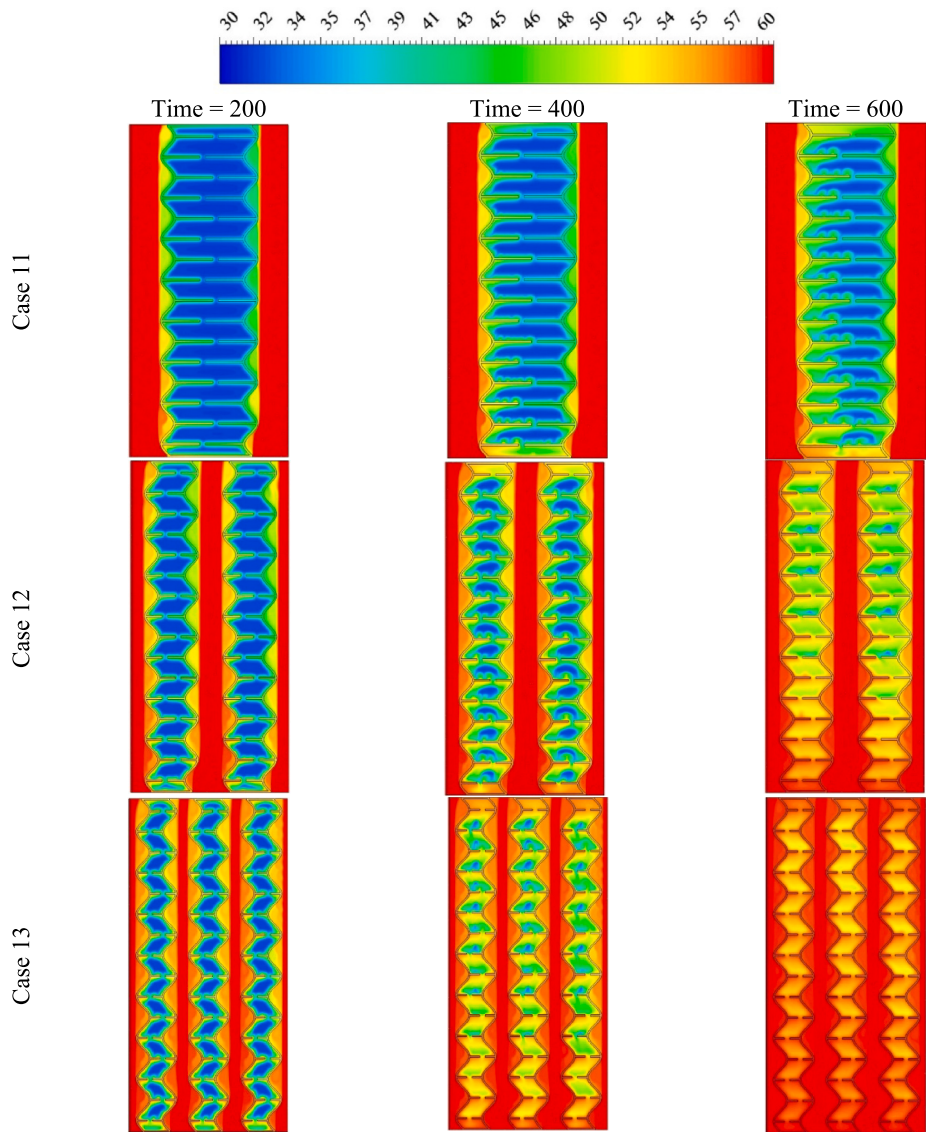


Fig. 18. Temperature contours of PCM melting at different time intervals (200 s, 400 s, 600 s) for Cases 11, 12, and 13, illustrating the impact of varying number of the PCM shell on melting performance.

Table 7
Heat storage rate performance (W) and the melting time (s) for Cases 11, 12, and 13, quantifying the TES capacity enhancement through changing the number of the PCM shell.

Cases	Melting time (s)	Heat storage rate (W)
Case 11 (Number of PCM =1)	1145	770.7185
Case 12 (Number of PCM = 2)	662	1347.931
Case 13 (Number of PCM = 3)	458	1948.083

data trends in Table 7. The melting time curve exhibits a steep decline, demonstrating a dramatic reduction from 1145 s for a single shell to just 458 s for three shells, an improvement of nearly 60%. This clearly illustrates the superior efficiency gained through multiple heat transfer surfaces. The PCM temperature evolution in Fig. 19b further reinforces this behavior, showing a markedly faster temperature rise for the three-shell configuration (Case 13) compared to the single-shell case (Case 11). The sharper slope of Case 13's temperature curve indicates an accelerated thermal response and higher energy absorption rate. These results establish a direct correlation between shell number and melting performance, emphasizing that expanding the heat transfer area is the

most effective method to mitigate the PCM's low thermal conductivity. Overall, the multi-shell configuration represents a transformative design advancement, enabling simultaneous heat penetration from multiple directions and significantly enhancing system power density and charging efficiency. Table 7 demonstrates that increasing the number of PCM shells produces a non-linear, transformative improvement in thermal performance. Adding a second shell (Case 12) reduces the melting time by 42.2% (from 1145 s to 662 s) and increases the heat storage rate by 74.9% (from 770.7 W to 1347.9 W) relative to the single-shell baseline (Case 11). Introducing a third shell (Case 13) further enhances performance, cutting melting time by 30.8% (to 458 s) and boosting the heat storage rate by 44.5% (to 1948.1 W). Overall, moving from one to three shells shortens melting time by 60% and more than doubles the heat storage rate. These gains arise because each additional shell provides a new heat transfer surface, generating multiple simultaneous melting fronts and greatly expanding the effective area for energy exchange. This geometric optimization mitigates the PCM's low conductivity, enabling faster, more uniform melting and transforming the system into a high-power, rapid-response thermal storage unit.

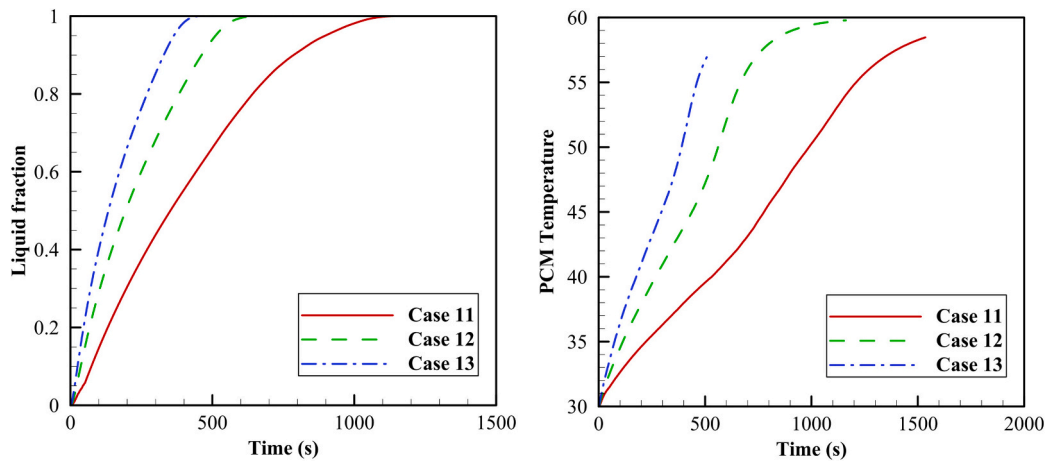


Fig. 19. Comparison of a) melting times b) PCM temperatures across Cases 11, 12, and 13, showing the impact of number of PCM shells on the PCM's thermal performance.

4.6. Effect of Reynolds number

Figs. 20 and 21 vividly illustrate the strong influence of the Reynolds number (Re) on PCM melting behavior, emphasizing that flow dynamics are as crucial as geometry in determining thermal performance. In Fig. 20, the liquid fraction contours at 300 s reveal that Case 13 ($Re = 2000$) achieves the most extensive melting, with the phase front advancing deeply into the PCM. Case 14 ($Re = 1000$) shows moderate progress, while Case 15 ($Re = 500$) exhibits limited melting confined near the heat transfer surface. This progression clearly demonstrates that higher Reynolds numbers, corresponding to stronger convection and mixing, dramatically accelerate melting by enhancing the convective heat transfer coefficient.

Fig. 21 reinforces these observations through temperature contours: Case 13 displays a widespread, uniform temperature field, while Case 15 retains steep gradients and large cold regions. The high Re case sustains vigorous fluid motion that continuously renews the boundary layer, maintaining strong thermal gradients and high heat flux into the PCM. In contrast, the low Re flow forms a stagnant layer that insulates the PCM, restricting heat penetration and slowing phase change. Collectively, these results confirm that increasing the Reynolds number substantially boosts energy transfer efficiency, transforming the system from conduction-dominated to convection-driven heat transport.

Fig. 22 comprehensively illustrates the effect of Reynolds number on the system's thermal behavior through three distinct aspects: melting

time (a), PCM temperature (b), and outlet temperature (c).

In part (a), the melting time shows a sharp exponential rise as the Reynolds number decreases. Case 15 ($Re = 500$) takes 1038 s to complete melting, more than twice the duration of Case 13 ($Re = 2000$) at 458 s—a 126% increase for a 75% reduction in Re . The corresponding heat storage rate drops from 1948 W to 870 W, confirming that sluggish, laminar flow at low Re weakens convective heat transfer, leading to thicker thermal boundary layers and slower energy transfer into the PCM.

Part (b) further reveals that a low Reynolds number impedes the heating process itself. The PCM temperature curve for Case 13 ($Re = 2000$) rises steeply, indicating rapid energy absorption, while that of Case 15 ($Re = 500$) increases slowly due to a diminished temperature gradient between the HTF and PCM. Consequently, melting occurs primarily through inefficient conduction rather than convection.

Finally, part (c) shows that the outlet temperature is lowest for the high Re case, reflecting superior heat transfer efficiency. In contrast, the fluid exits hotter at low Re , signifying insufficient thermal energy exchange and reduced system effectiveness.

Table 8 clearly demonstrates that the Reynolds number (Re) of the HTF plays a decisive role in governing the PCM system's performance, with higher flow rates yielding exponential gains in efficiency. When Re decreases from 2000 (Case 13) to 1000 (Case 14), the melting time rises by 19.4% (458 s to 547 s) and the heat storage rate drops by 17.1% (1948 W to 1616 W). A further reduction to $Re = 500$ (Case 15) leads to a

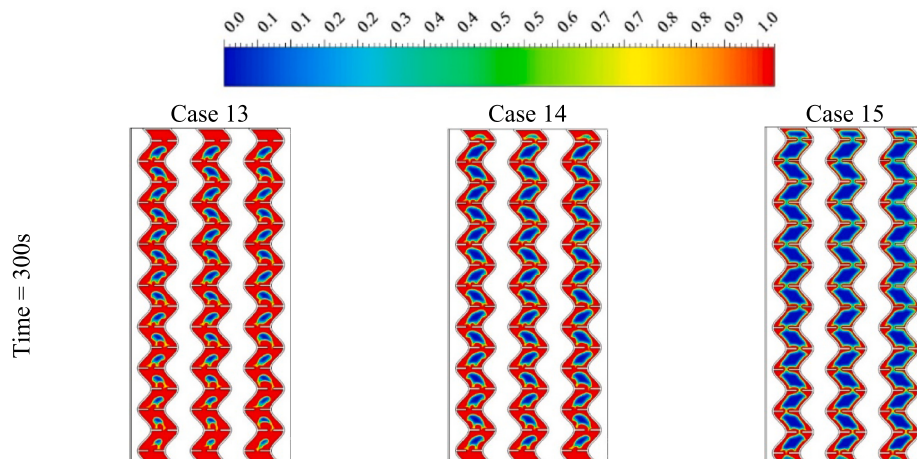


Fig. 20. Liquid fraction contours of PCM melting at time interval 300 s for Cases 13, 14, and 15, illustrating the impact of varying Reynolds numbers on melting performance.

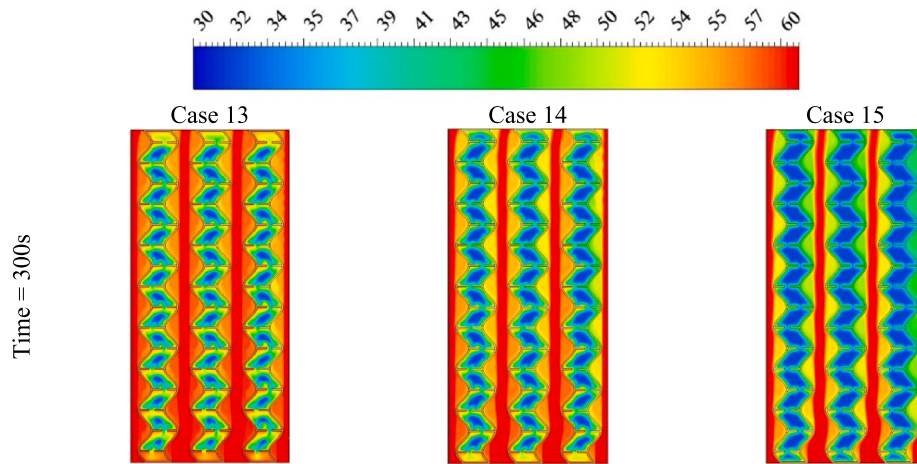


Fig. 21. Temperature contours of PCM melting at time interval 300 s for Cases 13, 14, and 15, illustrating the impact of varying Reynolds numbers on melting performance.

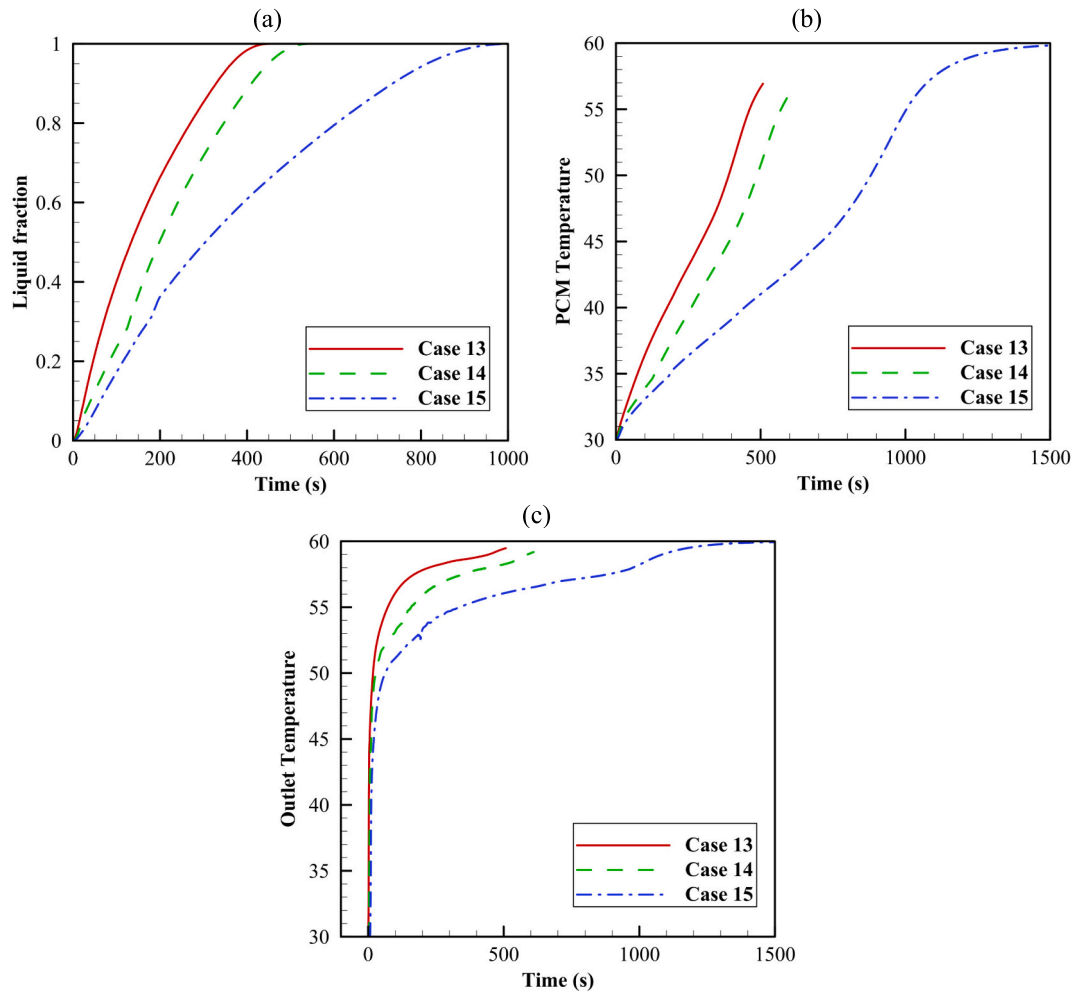


Fig. 22. Comparison of (a) melting times (b) PCM temperatures and (c) Outlet temperature across Cases 13, 14, and 15, showing the impact of Reynolds numbers on the PCM's thermal performance.

dramatic 126.6% increase in melting time and a 55.3% decline in heat storage rate, confirming the system's high sensitivity to flow regime.

This nonlinear degradation occurs because a lower Re weakens convection. At Re (2000), mixing enhances the convective heat transfer coefficient and sustains a strong temperature gradient, accelerating

energy absorption. Conversely, at Re (500), laminar flow thickens the thermal boundary layer, limiting heat flux into the PCM and drastically slowing melting.

Table 8

Heat storage rate performance (W) and the melting time (s) for Cases 13, 14, and 15, quantifying the TES capacity enhancement through changing Reynolds number.

Cases	Melting time (s)	Heat storage rate (W)
Case 13 ($Re = 2000$)	458	1948.08
Case 14 ($Re = 1000$)	547	1616.01
Case 15 ($Re = 500$)	1038	870.37

4.7. Inlet temperature

Based on Figs. 23 and 24, the profound impact of the HTF inlet temperature on the melting process is immediately and visually apparent, demonstrating that thermal driving force is a primary dictator of system performance. Fig. 23, the liquid fraction contours at the 200-s mark, reveals a staggering difference in melting progression. Case 17 (70 °C) already exhibits a vast, well-developed melted region, while Case 13 (60 °C) shows moderate melting, and Case 16 (50 °C) has only a small fraction of liquid PCM near the wall. This visual evidence directly correlates a higher temperature difference between the HTF and the PCM's melting point with a dramatically accelerated phase change initiation. The greater thermal potential in Case 17 drives a much higher heat flux into the PCM, causing rapid and widespread melting from the very beginning of the process.

Fig. 24, the temperature contours at the same time interval, provides the thermal rationale behind the observed melting behavior. The image for Case 17 shows a large volume of PCM at significantly elevated temperatures, indicating deep and powerful heat penetration. In contrast, Case 16 displays a very steep thermal gradient, with high temperatures confined to a thin layer adjacent to the wall and the vast majority of the PCM remaining much closer to its solid-state temperature. This stark contrast underscores that the inlet temperature defines the intensity of the thermal driving force. A higher inlet temperature not only increases the rate of heat transfer but also ensures that the heat can penetrate deeper into the PCM domain before the HTF cools down, effectively increasing the “thermal reach” within the system. The evaluation of these figures confirms that while geometry and flow optimize the heat transfer mechanism, the inlet temperature controls the fundamental magnitude of the energy transfer. It is the most direct lever for controlling the power output of the storage unit, as a 20 °C increase in inlet temperature (from 50 °C to 70 °C) is shown to transform the system's response from slow and limited to rapid and comprehensive.

Fig. 25 quantitatively demonstrates the dominant effect of inlet temperature on the PCM system's thermal behavior and energy efficiency. The melting time plot (a) shows a clear inverse relationship

between inlet temperature and phase change duration. Case 17 (70 °C) completes melting in only 354 s, which is 43% faster than Case 13 (60 °C) and 86% faster than Case 16 (50 °C). This acceleration occurs because a higher temperature difference (ΔT) between the HTF and PCM enhances convective heat transfer, as expressed by $q = hA\Delta T$, driving a much greater thermal flux into the PCM and enabling rapid phase transition.

The PCM temperature plot (b) further clarifies this effect. Case 17 displays a steep temperature rise, signifying fast energy absorption and early melting onset, while Case 16 shows a gradual increase, indicating limited heat penetration and delayed melting.

The outlet temperature plot (c) completes the performance picture. Case 16 retains the highest outlet temperature, meaning less energy is transferred to the PCM, whereas Case 17 exhibits the lowest outlet temperature, confirming superior heat utilization and system efficiency. Overall, Fig. 25 highlights inlet temperature as a critical operational parameter, capable of delivering faster charging and higher power density when properly optimized.

Table 9 clearly demonstrates the dominant influence of inlet temperature on the PCM system's thermal behavior and efficiency. Increasing the HTF inlet temperature from 50 °C (Case 16) to 60 °C (Case 13) reduces melting time by 30.7% (661 s to 458 s) and increases heat storage rate by 56% (1248.8 W to 1948.1 W). A further rise to 70 °C (Case 17) enhances performance even more, cutting melting time by 22.7% (to 354 s) and raising the heat storage rate by 37.4% (to 2676.1 W). Overall, a 20 °C temperature increase reduces melting time by nearly half and more than doubles the system's storage power.

These dramatic gains stem from the fundamental heat transfer law $q = hA\Delta T$. A higher inlet temperature increases the temperature difference (ΔT) between the HTF and PCM, producing a stronger driving force for convection. This accelerates melting, amplifies heat flux, and enables all geometric enhancements, such as fins and multi-shell designs, to operate at maximum effectiveness.

A comprehensive evaluation of the data allows ranking the cases based on overall thermal performance, primarily indicated by Heat Storage Rate (W), which reflects how rapidly the system can store energy. Higher values correspond to better performance, while Melting Time serves as a complementary metric, with shorter times being favorable. The “best” configuration is context dependent: Case 17 achieves the highest power output but requires a 70 °C inlet temperature, which may not always be feasible. Case 13 provides an excellent compromise, delivering high performance under more practical operating conditions. Note that Table 10 also includes Case 18, which has no fins and no waves, for benchmarking and performance comparison.

To assess the reliability and contribution of the present results, a

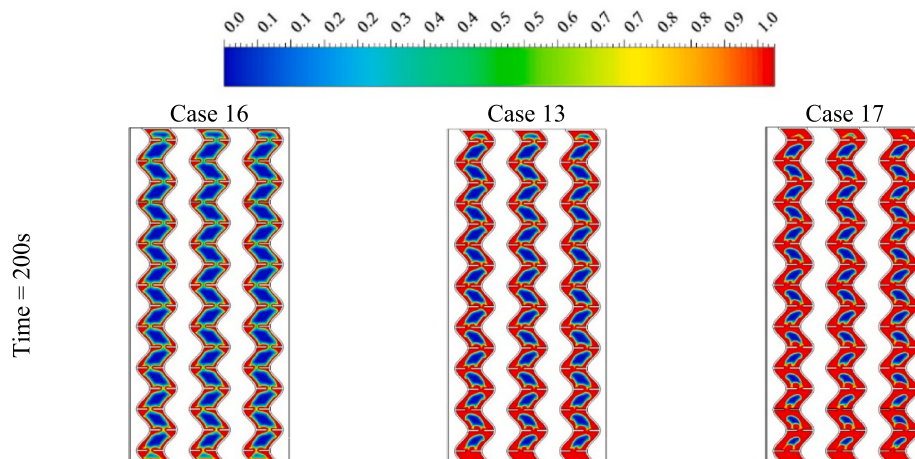


Fig. 23. Liquid fraction contours of PCM melting at time interval 200 s for Cases 16, 13, and 17, illustrating the impact of varying inlet temperature on melting performance.

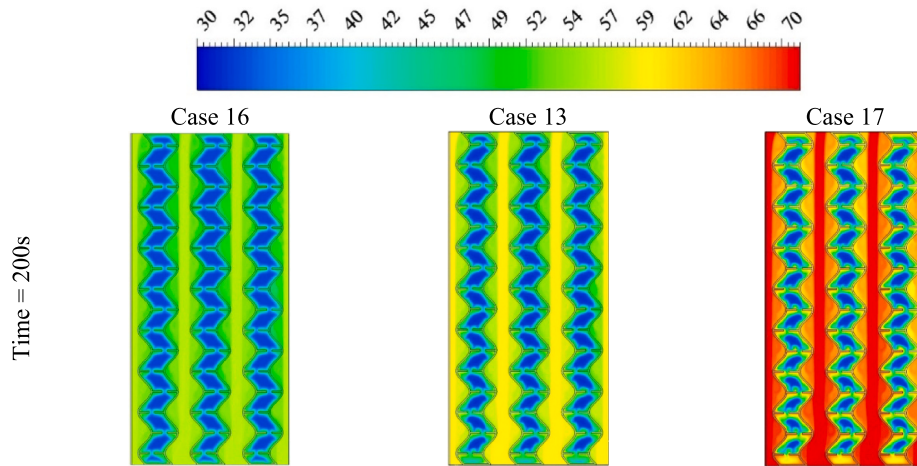


Fig. 24. Temperature contours of PCM melting at time interval 200 s for Cases 16, 13, and 17, illustrating the impact of varying inlet temperature on melting performance.

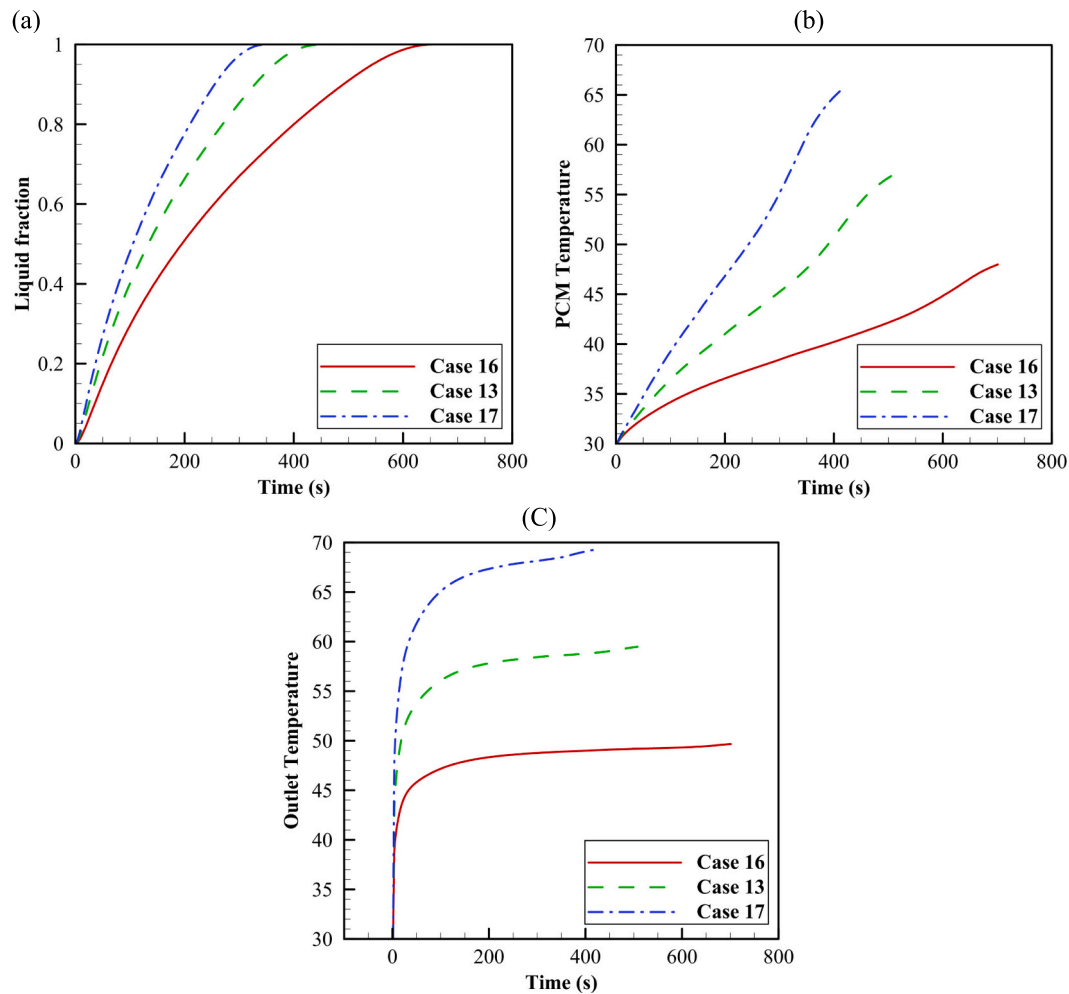


Fig. 25. Comparison of (a) melting times (b) PCM temperatures and (c) Outlet temperature across Cases 16, 13, and 17, showing the impact of inlet temperature on the PCM's thermal performance.

qualitative and quantitative comparison with representative literature is conducted. Previous investigations of corrugated or wavy-wall PCM enclosures typically report melting-time reductions of approximately 20–35% relative to smooth-wall baselines, primarily due to increased heat transfer area and enhanced natural convection. Likewise, studies on

finned-tube and shell-and-tube PCM systems commonly achieve melting-time reductions of about 30–45%, depending on fin geometry, spacing, and orientation. These ranges are consistent with the trends observed in the present study when individual geometric parameters are varied independently.

Table 9

Heat storage rate performance (W) and the melting time (s) for Cases 16, 13, and 17, quantifying the TES capacity enhancement through changing the inlet temperature.

Cases	Melting time (s)	Heat storage rate (W)
Case 16 ($T = 50$)	661	1248.81
Case 13 ($T = 60$)	458	1948.1
Case 17 ($T = 70$)	354	2676.1

When multiple enhancement mechanisms are combined within a unified structural framework, substantially larger performance gains are obtained. In particular, the three-shell configuration incorporating optimized fin arrangement and wavy walls yields an approximately ~60% reduction relative to the single-shell configuration (Case 11), which is higher than most values reported for single-modification designs in the existing literature. Compared with the smooth-wall, no-fin baseline (Case 18), the best case achieves an 82.2% shorter melting time, corresponding to an $\sim 5.6\times$ higher effective melting rate. In addition, the predicted heat storage rates (up to approximately 2.7 kW under elevated inlet temperature) exceed those commonly reported for conventional finned or corrugated PCM heat exchangers of comparable scale. These comparisons indicate that the present model reproduces established heat transfer behavior while extending prior studies by demonstrating the synergistic impact of coupled geometric and operational enhancements. Accordingly, the proposed framework provides both physically consistent and comparatively improved performance relative to existing PCM heat exchanger designs.

Table 10 consolidates the findings, revealing that peak performance arises from the combined effect of multiple design and operational strategies rather than a single factor. Top-performing cases, such as Cases 17 and 13, excel by maximizing the heat transfer driving force through higher inlet temperatures, enhancing surface area via multiple PCM shells and fin arrangements, and maintaining strong convective flow with high Reynolds numbers. In contrast, low-performing cases, like Cases 15 and 8, demonstrate that suboptimal operational conditions (low flow rates) or poorly configured fins that impede convection can severely limit performance. This analysis highlights a key principle in TES design: the interdependence of geometry and operational parameters, where only a carefully balanced combination achieves optimal system efficiency.

Although the present investigation concentrates on the charging (melting) process, the discharge (solidification) behavior of the optimized configurations also warrants discussion in light of the identified design principles. From a physical standpoint, the dominant resistances during PCM solidification arise from low thermal conductivity and increasing solid-layer thickness near the heat extraction surface. The structural features identified as optimal in this study (namely multi-shell PCM layouts, wavy walls with moderate wavelength and higher amplitude, and optimally arranged fins) directly address these limitations by expanding the effective heat transfer area and reducing conduction distances during both melting and solidification. Multi-shell configurations enable simultaneous heat extraction from multiple interfaces, which is expected to significantly accelerate solidification compared with single-shell designs. Similarly, fins and wavy walls promote more uniform temperature gradients and suppress the formation of thick, insulating solid layers that typically slow discharge. While extremely aggressive geometries could, in principle, influence flow patterns differently during discharge, prior solidification studies on finned and corrugated PCM systems report either neutral or enhanced heat release rates when similar design principles are applied. Therefore, the optimized cases identified herein are expected to maintain, or even improve, discharge performance relative to conventional smooth-wall configurations. A full transient solidification analysis will be the subject of future work to quantitatively confirm these expectations and to further optimize the designs for complete thermal cycling.

Table 10

Ranking of TES cases from highest to lowest performance based on heat storage rate, with key parameters and primary advantages listed.

Rank	Case	Key Parameters	Heat Storage Rate (W)	Melting Time (s)	Primary Advantage & Reason for Performance
1	Case 17	3 Shells, $T = 70^\circ\text{C}$	2676.10	354	Highest Thermal Driving Force. The 70°C inlet temperature creates the largest temperature difference, maximizing the heat flux into the system. Optimal Combination. Excellent geometry (max surface area from 3 shells & fins) combined with optimal flow (high Re for strong convection). The benchmark for high power under standard conditions. Highly Efficient Design. The addition of a second shell dramatically increases heat transfer surface area compared to a single shell, offering a massive performance jump. Optimal Fin Design. Demonstrates the critical importance of fin arrangement. Arrangement A optimally enhances conduction without hindering convection. Enhanced Convection. Large wave amplitude increases surface area and disrupts flow, strengthening natural convection within the PCM. Optimized Geometry. The intermediate wavelength provides the best balance between increased surface area and maintained fluid circulation. Good Performance Boost. A moderate increase in amplitude provides a significant improvement over the baseline case. Geometric Baseline. The reference case against which all geometric modifications are compared. Performance is limited by low surface area. Hampered Flow. Shows the
2	Case 13	3 Shells, $Re = 2000$	1948.08	458	
3	Case 12	2 Shells, $Re = 2000$	1347.93	662	
4	Case 6	Fin Arrangement A	1355.81	657	
5	Case 5	Wave Amp. = 10 mm	990.02	876	
6	Case 2	Wave Length = 21 mm	634.92	1421	
7	Case 4	Wave Amp. = 7.5 mm	844.74	998	
8	Case 1	Baseline (Long Wave)	502.27	1820	
9	Case 14	$Re = 1000$	1616.01	547	

(continued on next page)

Table 10 (continued)

Rank	Case	Key Parameters	Heat Storage Rate (W)	Melting Time (s)	Primary Advantage & Reason for Performance
10	Case 16	T = 50 °C	1248.81	661	performance penalty of reducing flow rate. Lower <i>Re</i> weakens convection, dropping performance significantly from Case 13. Low Thermal Potential. A lower inlet temperature reduces the driving force for heat transfer, limiting the system's power output despite good geometry. Poor Fin Design. Worst Geometric Modifier.
11	Case 8	Fin Arrangement C	676.64	1335	Arrangement C is counterproductive, blocking natural convection currents and creating thermal shadows, making it worse than having no fins at all. Worst Overall Performer. Crippled by Laminar Flow.
12	Case 15	<i>Re</i> = 500	870.37	1038	Even an excellent geometry (3 shells, fins) cannot overcome the incredibly weak heat transfer caused by low, laminar flow. Proves operation is as important as design. The absolute baseline.
13	Case 18	No Wave, No Fin	360.09	2577	Expected to have the lowest performance of all, worse than Case 1.

The optimized composite PCM heat exchanger structures investigated in this study are relevant to several engineering applications that require compact, high-power TES, such as solar thermal collectors, waste-heat recovery units, building HVAC load leveling, and battery thermal management systems. In these applications, the combination of multi-shell PCM layouts, fin-assisted conduction, and corrugated heat transfer surfaces can significantly enhance charging and discharging rates while maintaining a relatively small system footprint.

Across all investigated cases, a consistent relationship is observed between the selected performance indicators and the dominant heat transfer mechanisms. Configurations that enhance natural convection, increase the effective heat transfer area, and reduce the characteristic conduction length simultaneously yield shorter phase change times and higher heat storage rates. In contrast, geometries that restrict fluid motion or generate poorly mixed thermal zones exhibit slower melting despite comparable surface area. These trends indicate that the chosen evaluation metrics capture the combined influence of geometric design and operating conditions on the governing thermal resistances, enabling a physically interpretable comparison of different configurations rather than a purely empirical ranking.

5. Conclusion

A central contribution of this study is the combined use of multi-shell PCM architecture with wavy walls and fins within a unified parametric framework, covering wall wavelength and amplitude, fin arrangement

and length, number of PCM shells, Reynolds number, and inlet temperature. Unlike conventional single-surface enhancement approaches, this configuration redistributes heat transfer interfaces throughout the PCM domain, generating multiple simultaneous melting fronts and modifying the dominant thermal resistance pathways. These findings indicate that optimal LHTES performance is governed primarily by the reduction of dominant thermal resistances through distributed heat-transfer interfaces rather than by isolated geometric modifications.

Quantitatively, reducing the corrugation wavelength from 31.4 mm to 21 mm shortened the melting time by 21.9% and increased the heat storage rate by 26.4%, while increasing the wave amplitude from 5 mm to 10 mm reduced the melting time by 51.9% and raised the storage rate by 97.1%. Increasing the number of PCM shells from one to three produced the strongest structural effect, reducing melting time by about 60% and more than doubling the storage rate, demonstrating that interface distribution is more influential than local surface refinement. Fin configuration further controlled performance, with optimized arrangements yielding nearly 50% higher storage rates than unfavorable layouts, confirming that fins act as flow-structuring elements rather than simple conductive extensions. Operating conditions amplified these geometric gains: increasing the Reynolds number from 500 to 2000 enhanced the storage rate by 88.5%, and raising the inlet temperature from 50 °C to 70 °C nearly halved the melting time and increased the power output to 2.68 kW.

By linking these improvements to changes in transport topology rather than isolated geometric trends, the present study provides transferable design guidance for compact, high-power latent heat storage systems.

From a feasibility perspective, the proposed geometries are compatible with existing manufacturing techniques, including metal extrusion, precision machining, and advanced additive manufacturing for finned and corrugated components. However, several challenges remain for practical implementation. These include increased fabrication complexity and cost compared with smooth-wall designs, potential PCM leakage or void formation in complex geometries, and the need to ensure long-term mechanical and thermal reliability under repeated phase-change cycling. In addition, optimizing the balance between enhanced heat transfer and pressure drop on the heat transfer fluid side remains a key design constraint. Addressing these bottlenecks through experimental validation, manufacturability-oriented optimization, and lifecycle analysis represents an important direction for future work and will be essential for transitioning the proposed composite structures from numerical design to industrial application.

CRedit authorship contribution statement

Fathi Alimi: Writing – review & editing, Writing – original draft, Methodology, Formal analysis, Data curation, Conceptualization. **Nashmi H. Alrasheedi:** Writing – review & editing, Writing – original draft, Resources, Project administration, Methodology, Funding acquisition, Formal analysis. **Mohammad Hossein Heidarshenas:** Writing – review & editing, Writing – original draft, Visualization, Software, Investigation, Formal analysis. **Hayder I. Mohammed:** Writing – review & editing, Writing – original draft, Methodology, Investigation, Formal analysis, Data curation, Conceptualization. **Mohamed Bouzidi:** Writing – review & editing, Writing – original draft, Visualization, Methodology, Investigation, Formal analysis, Data curation. **Mansour Mohamed:** Writing – review & editing, Writing – original draft, Visualization, Methodology, Data curation, Conceptualization. **Mohammad Ghalebaz:** Writing – review & editing, Writing – original draft, Visualization, Validation, Supervision, Resources, Methodology, Formal analysis, Conceptualization. **Khalil Hajlaoui:** Writing – review & editing, Writing – original draft, Validation, Supervision, Resources, Methodology, Formal analysis, Conceptualization.

Declaration of competing interest

The authors clarify that there is no conflict of interest for report.

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Data availability

Data will be made available on request.

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