

Research Paper

Accelerated charging and discharging of phase change materials in vertical double-pipe heat exchangers with void-embedded twisted inner tubes

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ABSTRACT

Latent heat thermal energy storage (LHTES) is widely employed to buffer intermittency in solar, building, and industrial heat systems; however, achievable power density is constrained by the low thermal conductivity of phase-change materials (PCMs). This study evaluates whether a void-embedded twisted inner tube in a double-pipe exchanger can accelerate charging and discharging without adding solid mass. Three-dimensional simulations of melting and solidification were performed using the Boussinesq enthalpy-porosity method with an isothermal inner wall boundary; a parametric sweep varied void number, void height, and helical revolutions. Increasing the void count from 3 to 5 raised the heat-storage rate from 84.6 to 119.4 W and shortened melting time from 2269 to 1570 s; optimizing void height to 20 mm further increased storage to 153.8 W and reduced melting time to 1185 s. During discharge, recovery rates up to 94.6 W were obtained, decreasing solidification time to 2037 s. Relative to a smooth baseline (49.2 W melting rate, solidification >3600 s), the optimized configuration delivered +212 % in melting-rate and + 191 % in solidification-rate improvements. The observed gains are attributed to synergy between void-induced flow steering and helical-induced secondary motions, which thin thermal boundary layers and shorten conduction paths. Collectively, the results demonstrate that void-embedded geometries provide a practical, scalable route to higher LHTES power density for medium-temperature applications in solar thermal systems, building HVAC, and industrial waste-heat recovery.

1. Introduction

Latent heat thermal energy storage (LHTES) systems utilize phase change materials (PCMs) to store and release thermal energy during phase transitions, typically between solid and liquid states, offering high energy density and near-isothermal behavior [1]. These characteristics make LHTES critical for applications such as solar thermal energy systems, building thermal management, and industrial waste heat recovery, where efficient energy storage mitigates the mismatch between energy supply and demand [2,3]. Recent reviews emphasize LHTES as a

practical bridge between intermittent supply and continuous heat demand across solar thermal, industrial waste-heat recovery, and building applications, while documenting rapid progress in materials, integration, and system design [4,5]. Beyond materials and configurations, recent work has applied artificial neural networks (ANN) and multi-objective optimization to accelerate LHTES design, demonstrating systematic improvement of charging/discharging behavior under practical constraints. Examples include ANN-assisted design optimization of latent heat storage systems and Pareto-based optimization of finned-tube storage tanks that jointly evaluate charging/discharging performance [6,7].

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Nomenclature

Symbol Meaning, Unit

A	Cross-sectional flow area, m^2
c_p	Specific heat capacity of PCM, $\text{J kg}^{-1} \text{K}^{-1}$
D_h	Hydraulic diameter, m
E	Total PCM energy (sensible + latent), J
E_e, E_i	PCM energy at end / beginning of process, J
E_T	Overall energy (storage/release) rate metric, J s^{-1}
g	Gravitational acceleration, m s^{-2}
h	Specific enthalpy, J kg^{-1}
k	Thermal conductivity of PCM, $\text{W m}^{-1} \text{K}^{-1}$
L_f	Latent heat of fusion of PCM, J kg^{-1}
P	Wetted perimeter, m
p	Static pressure, Pa
t_m, t_s	Melting / solidification time, s
t_p	Process duration, s

T	Temperature, K (or $^{\circ}\text{C}$ where shown)
T_{wall}	Inner wall (HTF-side) temperature, $^{\circ}\text{C}$
$T_{\text{solidus}}, T_{\text{liquidus}}$	Solidus / liquidus temperatures of PCM, $^{\circ}\text{C}$
T_{ref}	Reference temperature (Boussinesq), $^{\circ}\text{C}$
V	Velocity vector, m s^{-1}
U_m	Bulk mean velocity (for Reynolds number), m s^{-1}
β	Thermal expansion coefficient (Boussinesq), K^{-1}
ρ	Density of PCM, kg m^{-3}
ρ_0	Reference density (Boussinesq), kg m^{-3}
μ	Dynamic viscosity, Pa s
λ	Liquid fraction (0: solid, 1: liquid), —
ΔT	Imposed temperature span, K
ε	Small numerical constant to avoid singularity, —
A_m	Mushy-zone (Darcy) constant, —
S	Momentum source term (mushy-zone resistance), N m^{-3}
S_L	Latent-heat source term, W m^{-3}
Re	Reynolds number, —

A comprehensive review by Jayathunga et al. [8] examined PCM candidates for LHTES in concentrated solar power (CSP) applications, emphasizing the global potential of solar energy to displace fossil fuels and reduce greenhouse gas emissions in alignment with the United Nations Sustainable Development Goals. Specifically, they highlight the use of organic PCMs for CSP applications, which exhibit higher latent heat capacities, but lower thermal conductivity compared to conventional inorganic materials. Despite these benefits, they noted the challenge of improving the thermal conductivity of organic PCMs, which remains a key area of research. Furthermore, a systematic assessment by Liu et al. [9] addressed progress in direct contact heat storage utilizing phase change mechanisms, emphasizing its efficacy in mitigating intermittency issues in renewable energy and waste heat recovery. They noted the challenge of inefficient heat transfer in traditional direct-contact systems due to high thermal resistance at the PCM interface.

This manuscript investigates a novel double-pipe heat exchanger design to enhance LHTES performance, addressing the growing need for sustainable energy solutions. Unlike traditional designs that rely on fins to enhance heat transfer, this study explores void-based modifications in the heat exchanger to disrupt flow dynamics and improve thermal conductivity. By optimizing the void geometry and pipe configuration, this study aims to overcome the limitations of conventional designs. This work presents an alternative approach to improving LHTES systems in medium-temperature applications, contributing to the advancement of thermal energy storage technologies for a sustainable energy future.

LHTES systems offer advantages over conventional storage like batteries and sensible TES, including lower cost, longer lifespan, and direct utility for heating/cooling [10,11]. While sensible TES systems like rock beds are mature and scalable, they suffer from low energy density and slow heat transfer, requiring larger volumes [12]. In contrast, LHTES provides higher energy density and near-isothermal operation, enabling compact storage. The practical viability of PCM-based LHTES across various operational conditions has been demonstrated for applications like building thermal management [13].

Taken together, these advantages position LHTES as a cost-effective and reliable alternative to conventional storage technologies, particularly for solar thermal systems, building thermal management, and waste heat recovery, where mismatches between energy supply and demand must be mitigated [14]. Building upon this foundation, the present study advances LHTES technology by introducing an innovative double-pipe heat exchanger with strategically embedded voids. This design directly addresses the heat-transfer limitations identified in sensible TES systems [12,15] and complements the system-level robustness observed in PCM-based storage [13], thereby enhancing both energy density and operational efficiency in medium-temperature

applications.

The dominant limitation in PCM systems remains low thermal conductivity, which slows charging/discharging and caps power density [16]. Consequently, research has focused on geometric and material interventions, fins (including perforated/constructal forms), high-conductivity matrices (e.g., metal foams), encapsulation strategies, and flow-field optimization, to accelerate conduction and strengthen natural/forced convection within the PCM domain. Recent high-quality studies show that geometry and flow control can shorten melt/solidification times substantially without sacrificing capacity [17,18]. The primary challenge in PCM-based LHTES systems is the low thermal conductivity of PCMs, which slows charging and discharging processes and limits power density. Yang et al. [19] reinforced that low thermal conductivity is the principal challenge for PCM-based LHTES, showing that it directly limits power density and charging/discharging rates, even when advanced PCM materials are used. The study highlights the development of organic and inorganic PCMs, shape stabilization, encapsulation, and heat transfer enhancement techniques (e.g., fins, nanoparticles, external fields) to address issues like supercooling, phase segregation, and slow energy transfer, applicable to thermal energy storage systems for renewable energy and waste heat recovery [20,21]. Another challenge lies in balancing enhancement techniques with material and volume constraints. Wang et al. [22] showed that while conductive structures such as fins can improve conduction, they also increase system mass and may obstruct convective pathways, leading to diminishing returns in overall performance.

These trade-offs highlight the need for optimized geometries that enhance heat transfer without significantly increasing material usage or compromising long-term stability [23,24]. Building on these insights, the present study explores void-based geometric modifications as a lightweight alternative to fins. Unlike solid conductive structures, voids reduce material mass while simultaneously promoting fluid mixing and turbulence within the PCM region, thereby addressing the conductivity bottleneck without sacrificing system compactness or efficiency. Traditional heat exchanger configurations for PCM-based TES, such as finned, shell-and-tube, and triplex-tube designs, enhance conduction but often face limitations. Extended surfaces like fins add mass and can impede buoyant flow paths, particularly during late melting stages, reducing convective efficiency [25]. Emerging configurations, including non-uniform or perforated fins and tree-shaped constructs, balance conduction and convection by relieving flow clogging and improving kinetics. Similarly, pin-fin optimization enhances local heat transfer without overly restricting natural convection [26]. These advancements inspire novel designs, such as void-enhanced systems, that prioritize lightweight structures and convective mixing to address PCM thermal

conductivity challenges.

Venkateswarlu and Ramakrishna [27] provided a comprehensive review of recent advances in PCMs for TES, with implications for heat exchanger configurations in applications such as solar thermal systems, building thermal management, and industrial waste heat recovery. The review discusses traditional configurations, including capsule-based shell-and-tube designs for PCM encapsulation and multitube heat transfer arrays, which enhance conduction but face limitations in thermal conductivity, leading to slow charge/discharge rates. Emerging enhancements, such as PCM slurries and composite materials (e.g., with metal foams or graphite nanoplatelets), are highlighted for improving heat transfer in these configurations, offering trade-offs in stability and scalability while inspiring novel geometries to balance conduction and convection.

Moreover, A numerical investigation by [28] examines hybrid microchannel heat sinks with PCM insets for electronic thermal management, focusing on six novel configurations integrating convective heat transfer with latent heat storage. Traditional MCHSs without PCM are limited by fluctuating heat loads, but the hybrid models demonstrate superior hydrothermal performance, achieving up to 12 % improvement in thermal performance factor and 7.3 % reduction in thermal resistance. The study analyzes the negligible impact of PCM channel shape and aspect ratio on overall efficiency, underscoring the potential of PCM-integrated microchannel designs as compact, high-performance alternatives to conventional finned or triplex-tube heat exchangers for PCM-based TES.

Together, these studies indicate that the future of PCM-based TES lies in configurations that minimize structural mass while maximizing conductive pathways and convective mixing. Building on this insight, the present study introduces void-based heat exchanger designs as a novel alternative to fins. By embedding voids within the twisted inner tube of a double-pipe heat exchanger, the system reduces material use while enhancing fluid circulation and turbulence within the PCM region. This approach directly addresses the conduction–convection trade-offs identified in finned and shell-tube designs, offering a pathway to more efficient and lightweight LHTES systems.

Voids in PCM-based LHTES systems offer a lightweight alternative to fins, enhancing heat transfer by promoting turbulence and convective mixing while reducing material use. Unlike fins, which add mass and may impede buoyant flow, voids disrupt thermal boundary layers, accelerating melting and solidification (e.g., by up to 37 % in optimized configurations) [29]. Shaik et al. [30] conducted a numerical study on advanced fin designs in PCM-based latent heat storage systems, using paraffin wax enhanced with CuO nanoparticles in a cylindrical heat exchanger with innovative fin geometries. While focusing on fins, the review implies void-enhanced systems through perforated or bionic structures that reduce material mass and thermal resistance, serving as lightweight alternatives to conventional solid fins by promoting buoyancy-driven convection and reducing melting time by up to 88.6 %. The synergy of nanoparticle dispersion (6 % CuO) further cuts melting time by 21 % and improves temperature uniformity by 47.23 %, suggesting void-like perforations in fins can optimize heat transfer efficiency without excessive structural weight, aligning with emerging designs for sustainable TES applications. In addition, Chen et al. [31] investigated thermal performance of a PCM-based thermal energy storage system enhanced with metal foam, using paraffin as PCM and water as heat transfer fluid (HTF) in a slab configuration. The study highlighted metal foam's porous structure, characterized by voids (pores), as a lightweight alternative to traditional fins, reducing material use while significantly improving heat transfer through increased effective thermal conductivity and interfacial area. By inserting foam in both HTF and PCM sides, the total charging/discharging time is reduced by 84.9 % compared to pure PCM and 40 % compared to foam only in the PCM side, with porosity effects showing lower porosities enhancing conduction but potentially restricting convection, underscoring voids' role in balancing heat transfer mechanisms for medium-temperature

applications like solar thermal systems.

These studies demonstrate that voids, whether through perforations, bionic fins, or porous foams, provide an effective route to reducing material use while accelerating PCM phase transitions. However, they also highlight trade-offs, such as increased pressure drop and manufacturing complexity, which motivate the search for simpler yet effective geometries. Building on these insights, the present study introduces a void-enhanced double-pipe heat exchanger with a twisted inner tube, where strategically embedded voids are used not as passive perforations or porous foams but as active design elements to simultaneously promote convective mixing and reduce structural mass. Using a controlled vertical setup with constant PCM volume across cases, we systematically vary: (i) void count and dimensions, and (ii) helical revolutions of the inner tube, to map their influence on melting/solidification times, heat-storage/release rates, and temperature/flow fields. The working hypothesis, supported by recent advances in helical PCM exchangers and optimized perforated constructs, is that void-assisted flow steering synergizes with helical-induced secondary flows, achieving faster, more uniform phase change at modest structural mass. The results provide design guidance for next-generation, lightweight PCM exchangers that avoid excessive fin mass yet deliver enhanced power density, thereby advancing practical, scalable LHTES for renewable and industrial thermal applications. This novel approach directly addresses the conductivity and convection balance identified in prior studies, positioning voids as a viable and sustainable enhancement strategy for medium-temperature LHTES applications.

2. Problem definition and physical model

2.1. Problem description

This work investigates an innovative double-pipe heat exchanger design incorporating a twisted inner tube with strategically embedded voids (Fig. 1). The HTF flows inside a copper inner tube and the PCM fills the annular region outside the inner tube and inside the outer shell. Thus, the PCM is separated from the HTF by the copper tube wall and no direct PCM–HTF contact occurs. Unlike traditional finned configurations, the proposed concept utilizes voids (i.e., empty spaces) to influence flow behavior and promote enhanced thermal performance. These voids are expected to reduce material usage while simultaneously altering heat transfer dynamics through improved fluid mixing and disruption of thermal boundary layers.

The twisted inner tube can be fabricated from annealed copper by mandrel-assisted twist forming; the axial voids are created prior to twisting by precision laser drilling or CNC milling and then deburred to mitigate stress concentrations. The tube is subsequently vacuum-brazed into the outer shell and pressure-tested. Potential challenges include geometric tolerances along the helical pitch, local wall thinning near void edges, and increased pressure drop that may raise pumping power. From an economic perspective, the concept introduces extra forming and joining steps relative to a plain tube; however, shorter charge/discharge times and higher energy-throughput per unit volume can compensate these costs in applications where cycling frequency is high.

The study focuses on a vertical arrangement of the heat exchanger and examines eight distinct cases. Case 1 represents the baseline with a smooth inner pipe and no voids, while Cases 2–8 introduce variations in void number (3–5), void dimensions (e.g., 18×8 mm, 14×6.15 mm, 20×4.4 mm), and helical revolutions (2 or 4 turns). To isolate the effect of void design, critical geometric parameters, including the pipe length (304 mm), the inner diameter (ID) and outer diameter (OD) of the PCM region (24 mm and 80 mm, respectively, except for Case 1 with ID = 39 mm), and material properties, are kept constant. Thus, the PCM volume is maintained equal across all cases by proportionally adjusting void dimensions. Fig. 2 illustrates schematic representations of the eight cases, showing both cross-sectional views and 3D geometrical models of the twisted inner tube with different void arrangements. The variations

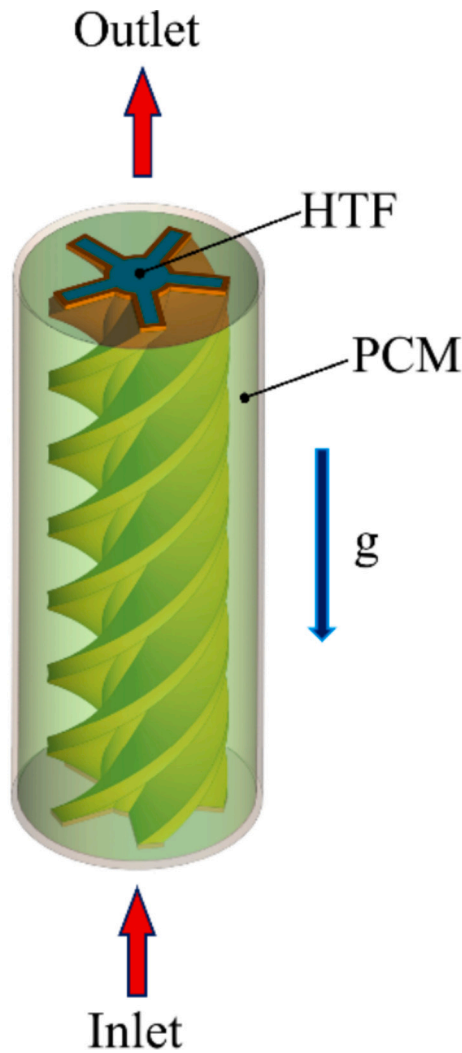


Fig. 1. Schematic of the vertical double-pipe exchanger with a copper inner tube wall separating the HTF (core) from the PCM (annulus).

clearly demonstrate how void geometry and helical twisting interact to create distinct flow pathways and heat transfer surfaces.

Table 1 summarizes the geometric characteristics of all studied cases, including void dimensions, number of voids, pipe thickness, and helical revolutions. The pipe thickness in Case 1 was increased to maintain a constant total pipe and PCM volume across all cases. This systematic parameter variation provides a controlled framework to evaluate how void configuration affects the heat transfer performance of the system. These include density, specific heat, thermal conductivity, latent heat, thermal expansion coefficient, dynamic viscosity, and melting temperature range (32–38 °C). Such properties directly govern energy storage and release behavior during melting and solidification cycles, ensuring the reliability of simulation results. To maintain clarity and reproducibility, the cases are denoted simply as Case 1–8, thereby avoiding excessive technical details in the text while retaining explicit correspondence to geometric variations. Note that Case 8 is a conventional double-pipe heat exchanger with smooth fins connected to the inner pipe which is used for comparison. For this case, the inner pipe diameter is considered similar to the pipe diameter for cases 2–7.

Prior enhancement approaches for PCM-based LHTES mitigate, but do not remove, the conductivity bottleneck. Extended surfaces (fins, pins) raise area yet add structural mass, obstruct buoyancy-driven mixing, and can create channel clogging and non-uniform melting; metal foams and packed inserts improve conduction but reduce PCM

volume and increase hydraulic losses; shell-and-tube arrays distribute heat but suffer from tree-shaped conduction paths and persistent stratification at late stages. Unlike fins, which serve as conductive extensions, voids reduce structural mass while simultaneously generating beneficial convective flow disturbances. Overall, this study introduces voids as a lightweight and efficient alternative to fins, aiming to optimize heat transfer efficiency while preserving structural stability. What has been less attended in the literature and industrial applications is a lightweight geometry that shortens solid/mushy conduction distances and intensifies convective mixing without additional solid mass. The present work addresses this gap by introducing a void-embedded twisted inner tube in a vertical double-pipe exchanger and by systematically varying (a) void number, (b) void height, and (c) helical revolutions to balance conduction-convection pathways.

2.2. PCM context and applications

PCMs used in LHTES are commonly grouped into (i) organic (e.g., paraffins, fatty acids), (ii) inorganic (salt hydrates, molten salts), and (iii) eutectic formulations. Organics offer congruent melting, chemical stability, and corrosion-free operation, but typically have low thermal conductivity. Inorganics provide higher volumetric storage density and k , but may suffer from supercooling, phase segregation, and corrosion issues. Eutectics can tune melting points, yet their availability and long-term stability are case-dependent. LHTES with melting ranges between 30 and 45 °C is widely applied to solar domestic hot-water buffering, space-heating preheating, building HVAC load shifting, PV/PVT temperature smoothing, and low-grade waste-heat recovery, where compactness and passive operation are valued [32]. These application domains motivate geometries that intensify heat transfer to compensate the inherently low k of many organics.

The present study employs paraffin RT35 (organic), whose melting interval (32–38 °C in this work) aligns with the aforementioned application band. RT35 is selected owing to (i) high latent heat with good cycling stability and limited subcooling, (ii) favorable materials compatibility and safety (non-corrosive, non-toxic, low vapour pressure), and (iii) suitability for enthalpy–porosity and Boussinesq-based modelling [33]. Its principal limitation, low thermal conductivity, motivates the proposed void-embedded twisted inner-tube configuration to increase interfacial area and promote secondary flows during charging and discharging. Key thermophysical properties are reported in Table 2.

Paraffin RT35 was selected because its melting range (30–40 °C) aligns with the intended medium-temperature use cases (building HVAC storage, low-grade solar/waste-heat recovery). RT35 provides high latent heat, chemical stability over many cycles with negligible subcooling, non-corrosiveness and broad material compatibility at low cost. Although paraffins have low thermal conductivity, the proposed helical/void geometry increases HTF–PCM interfacial area and convective mixing, mitigating this limitation (as seen in Section 5) [33,35].

3. Mathematical modelling

3.1. Governing equations

The numerical investigation of phase change phenomena within the proposed multi-lobed tube geometries is conducted using a rigorous mathematical framework designed to capture the coupled mechanisms of heat transfer, fluid motion, and phase transformation. The formulation is based on the well-established enthalpy–porosity method proposed by Brent et al. [36], which provides a reliable approach for modelling solid–liquid transitions in PCM systems. In this method, the phase change zone is treated as a porous medium, where the liquid fraction evolves continuously from 0 (completely solid) to 1 (completely liquid), allowing accurate characterization of the mushy region that naturally develops during melting and solidification [37].

To reflect the physical nature of the system, several assumptions are

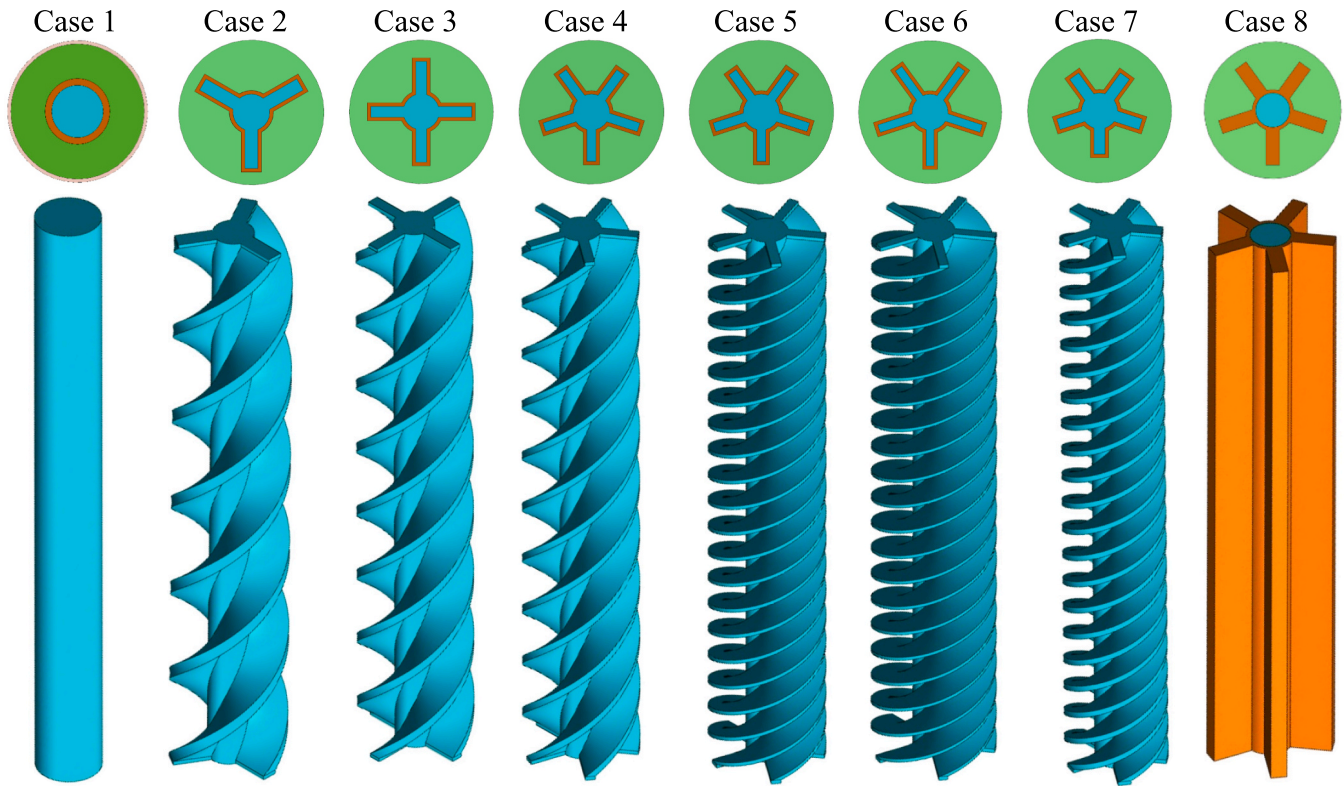


Fig. 2. Eight investigated cases, including cross-sectional and three-dimensional views of the copper twisted inner tube with and without embedded voids.

Table 1
Geometrical specifications of the heat exchanger configurations.

case	Number of voids	Dimensions of void (mm)	Length of pipe (mm)	Thickness of pipe (mm)	ID of PCM (mm)	OD of PCM (mm)	Revolutions of helix
1	0	–	304	4	39	80	–
2	3	18 × 8	304	2	24	80	2
3	4	18 × 6	304	2	24	80	2
4	5	18 × 4.8	304	2	24	80	2
5	5	18 × 4.8	304	2	24	80	4
6	5	20 × 4.4	304	2	24	80	4
7	5	14 × 6.15	304	2	24	80	4
8	0	–	304	2	24	80	–

Table 2
Thermophysical properties of RT-35 (GmbH) used as the PCM and copper used as the pipe material in the simulations [34].

Property	Copper	PCM	Water	Unit
Mean density (ρ)	8978	815	998.2	kg m ⁻³
Specific heat (c_p)	381	2000	4182	J kg ⁻¹ K ⁻¹
Thermal conductivity (k)	387.6	0.2	0.6	W m ⁻¹ K ⁻¹
Latent heat of fusion (L_f)	–	160,000	–	J kg ⁻¹
Thermal expansion coefficient	–	0.0006	–	K ⁻¹
Dynamic viscosity (μ)	–	0.023	0.001003	Pa s
Melting temperature range	–	32–38	–	°C

incorporated into the model. First, density variation of the PCM during phase change is explicitly considered by employing appropriate thermophysical properties. Natural convection within the PCM is accounted for using the Boussinesq approximation, ensuring buoyancy-driven flow is properly represented. The HTF is assumed to be laminar, incompressible, and Newtonian, while temporal periodicity is imposed to achieve steady-state convergence. System boundaries are modelled as adiabatic, isolating the influence of the geometric modifications on heat transfer. Additionally, viscous dissipation is neglected due to the

relatively low Reynolds numbers (Re) typically observed in TES applications.

Under these assumptions, the governing equations for mass, momentum, and energy conservation are expressed in differential form and solved numerically to evaluate the phase change dynamics. This framework enables accurate prediction of melting and solidification behavior in the modified heat exchanger designs while capturing the interplay of conduction, convection, and latent heat effects [17,38]:

Continuity Equation:

$$\left[\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{V}) \right] = 0 \quad (1)$$

Momentum Conservation:

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

Energy Conservation:

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

The momentum conservation equation includes a specific source

term (S) that models the influence of phase change on fluid flow within the computational field. This resistance effect is formulated using the Darcy law analogy for porous media flow [39]:

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

The mushy zone constant (A_m) is assigned a value of (10^5) following recommendations from previous studies [40–42]. This parameter ensures sufficient momentum damping in partially solid regions, thereby preserving numerical stability. The liquid fraction variable (λ) serves as the key indicator of the local phase state, varying continuously from zero in the solid phase to one in the liquid phase. Its definition is established through an enthalpy-based formulation [40]:

$$\lambda = \frac{\Delta H}{L_f} = \begin{cases} 0, & T < T_{\text{Solidus}} \\ \frac{T - T_{\text{Solidus}}}{T_{\text{Liquidus}} - T_{\text{Solidus}}}, & T_{\text{Solidus}} \leq T \leq T_{\text{Liquidus}} \\ 1, & T > T_{\text{Liquidus}} \end{cases} \quad (5)$$

In the energy conservation equation, the latent heat source term (S_L) represents the temporal and spatial variations of thermal energy that arise during phase change processes:

$$S_L = \left[\nabla \cdot (\rho \vec{V} L_f) \right] + \left[\frac{\partial(\rho \lambda L_f)}{\partial t} \right] \quad (6)$$

This latent heat formulation captures two fundamental mechanisms. The temporal term $\partial(\rho \lambda L_f) / \partial t$ quantifies the local rate of energy absorption or release due to phase state transitions, while the convective term $(\rho \vec{V} L_f)$ represents the transport of latent heat carried by fluid motion. During melting, an increasing (λ) leads to a negative source term, signifying energy consumption as solid material transforms into liquid. Conversely, during solidification, a decreasing λ produces a positive source term, corresponding to the release of latent heat as liquid crystallizes into solid. The overall energy storage and release rates for different multi-lobed tube geometries, during both charging and discharging operations, are mathematically expressed as:

$$\dot{E}_T = \left[\frac{E_e - E_i}{t_p} \right] \quad (7)$$

here (t_p) denotes the characteristic duration required for full melting or solidification, while (E_e) and (E_i) represent the total PCM energy content at the end and beginning of the respective process. The total stored energy (E) includes contributions from sensible heating ($MC_p dT$) and latent heat (ML_f), providing a complete measure of the PCM's thermal capacity. During charging, the parameter (E_T) reflects the overall energy accumulation rate, combining sensible heat gain and latent heat absorption during melting. In discharging, it quantifies the rate of energy release, incorporating sensible cooling together with latent heat liberation during solidification.

Under the assumptions stated above, laminar, incompressible flow; constant thermophysical properties; Boussinesq treatment of buoyancy with Eq. (8) acting only in the body-force term; no-slip walls; and adiabatic outer wall; the governing relations in Eqs. (2)–(6) are obtained. Specifically, Eq. (2) is the momentum balance augmented by the Boussinesq buoyancy force and the mushy-zone Darcy resistance S of Eq. (4); Eq. (3) is the energy balance solved with the enthalpy–porosity formulation, in which the liquid fraction λ defined in Eq. (5) and the latent-heat source term S_L in Eq. (6) account for phase change. These modelling choices ensure that each assumption is explicitly reflected in the mathematical model used throughout the study.

3.1.1. Boussinesq approximation

To capture buoyancy-driven natural convection during phase change while keeping the flow incompressible, we adopt the Boussinesq

approximation. Density variations are neglected everywhere except in the gravitational body-force term of the momentum equation where.

$$\rho = \rho_0 [1 - \beta(T - T_{\text{ref}})] \quad (8)$$

with reference density $\rho_0 = 815 \text{ kg m}^{-3}$ (RT35), thermal expansion coefficient $\beta = 6.0 \times 10^{-4} \text{ K}^{-1}$, and $T_{\text{ref}} = 35^\circ \text{C}$ (approximately the melting temperature of RT35). The temperature span imposed in both charging and discharging cycles is ($\Delta T = 30 \text{ K}$ ($60 \rightarrow 30^\circ \text{C}$ and $40 \rightarrow 10^\circ \text{C}$, respectively), yielding $\beta \Delta T = 1.8\%$). Because density variations remain $\ll 5\%$, and the fluid is essentially incompressible, the Boussinesq assumption is valid for this configuration. The momentum equation (Eq. (2)) thus uses ρ_0 everywhere except in the buoyancy source term, which becomes $\rho_0 \beta (T - T_{\text{ref}}) g$.

3.1.2. Enthalpy–porosity method

Phase change in the PCM is modelled via the enthalpy–porosity formulation, which treats the mushy zone as a porous medium whose permeability depends on the local liquid fraction (λ) (0: solid, 1: liquid). The total enthalpy is

$$h(T, \lambda) = h_{\text{ref}} + \int_{T_{\text{ref}}}^T c_p(\theta) d\theta + \lambda L \quad (9)$$

and the energy equation (Eq. (3)) includes the latent heat source term (Eq. (6)) that accounts for both temporal and convective transport of latent energy. The liquid fraction is obtained from a piecewise-linear relation (Eq. 5). Momentum damping inside partially solid regions is introduced through the Darcy-like source term (Eq. (4)). In this equation ($A_m = 10^5$) and ($\varepsilon = 10^{-3}$) to prevent division by zero. This penalization suppresses velocities as ($\lambda \rightarrow 0$), ensuring physical no-flow in the solid while permitting smooth transition across the mushy region. The approach is numerically stable and widely used for paraffin PCMs due to its robustness and modest computational cost. Classical developments of the enthalpy–porosity method and recent applications to LHTES systems, including optimization and ANN-assisted studies, provide further background and modelling practice [6,7].

3.1.3. HTF modelling

The conservation equations for the HTF flow through the inner pipe is described as:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho \vec{V}) = 0 \quad (10)$$

$$\rho \frac{\partial \vec{V}}{\partial t} + \rho (\vec{V} \cdot \nabla) \vec{V} = -\nabla P + \mu (\nabla^2 \vec{V}) \quad (11)$$

$$\rho C_p \frac{\partial T}{\partial t} + \nabla \cdot (\rho C_p \vec{V} T) = \nabla \cdot (k \nabla T) \quad (12)$$

The flow is assumed laminar based on the velocity considered for the HTF flow. Nominal Reynolds number (HTF reference) can be evaluated as:

$$Re = \frac{\rho_{\text{HTF}} \bullet U_m \bullet D_h}{\mu_{\text{HTF}}} \quad (13)$$

for the circular inner tube $D_h = D_i$ (otherwise $D_h = \frac{4A}{P}$). Here U_m is the bulk mean velocity. For all configurations, the Reynolds number remains below 2000, i.e. within the laminar /onset-of-transition range for internal flows (the classical critical value for smooth circular pipes being around $Re \approx 2300$). Note that a constant inlet velocity and identical inlet area (except case 1 with no voids) were used in all cases, so the volumetric flow rate is the same; only the hydraulic diameter changes slightly with geometry, leading to modest variations in Reynolds number.

It should be noted that the twisted / helical inner tube promotes

secondary flows and enhances mixing, but at the relatively low Reynolds numbers considered here the flow field in our simulations remains steady and does not exhibit unsteady or fluctuating behavior that would typically require a turbulence model [43]. Similar studies on twisted-tube heat exchangers at comparable Reynolds numbers have also successfully employed laminar flow models for predicting global quantities such as melting/solidification time and average heat transfer coefficients [43,44].

The table summarizes the operating conditions for the eight simulated cases, where a constant inlet velocity of 0.0322 m s^{-1} was applied to the heat transfer fluid (HTF), resulting in a uniform volumetric flow rate (Q) of $24 \text{ cm}^3 \text{ s}^{-1}$ across all configurations except Case 8. The Reynolds number (Re) varies due to differences in hydraulic diameter caused by the geometric modifications, specifically, the number and dimensions of the embedded voids and the helical revolutions of the inner tube. For instance, Case 1 (smooth tube, no voids) has the highest Re of 993, while Case 6 (5 voids, 4 revolutions, 20 mm height) has the lowest Re of 368. Case 8, included as a reference with no voids and a reduced flow rate ($Q = 10 \text{ cm}^3 \text{ s}^{-1}$), has a Re of 641, illustrating how both geometry and flow conditions influence the flow regime and

thermal performance.

3.2. Modelling assumptions

A1: PCM phase change model. Phase change is treated with the enthalpy–porosity formulation; the mushy zone is a porous medium with a Darcy damping source (Eqs. (3)–(6)).

A2: Boussinesq buoyancy. Natural convection in the PCM is captured using the Boussinesq approximation: density is constant everywhere except in the buoyancy term, where $\rho = \rho_0 [1 - \beta (T - T_{ref})]$ with $\rho_0 = 815 \text{ kg m}^{-3}$ (RT35 reference density), $\beta = 6 \times 10^{-4} \text{ K}^{-1}$, and $T_{ref} = 35 \text{ }^\circ\text{C}$ (midpoint of the RT35 melting interval used here, $32\text{--}38 \text{ }^\circ\text{C}$). The imposed $\Delta T = 30 \text{ K}$ gives $\beta \Delta T = 1.8 \%$, validating Boussinesq.

A3: Flow regime. The PCM liquid and any auxiliary fluid regions are modelled as laminar, incompressible, Newtonian; viscous dissipation is neglected. The Reynolds number refers to the HTF inside the inner tube. Interfacial boundary conditions enforce continuity of temperature and heat flux across HTF–copper and copper–PCM interfaces, with thermal contact resistance assumed zero. For the PCM, the solved flow is the natural convection, which is modelled as laminar, consistent with

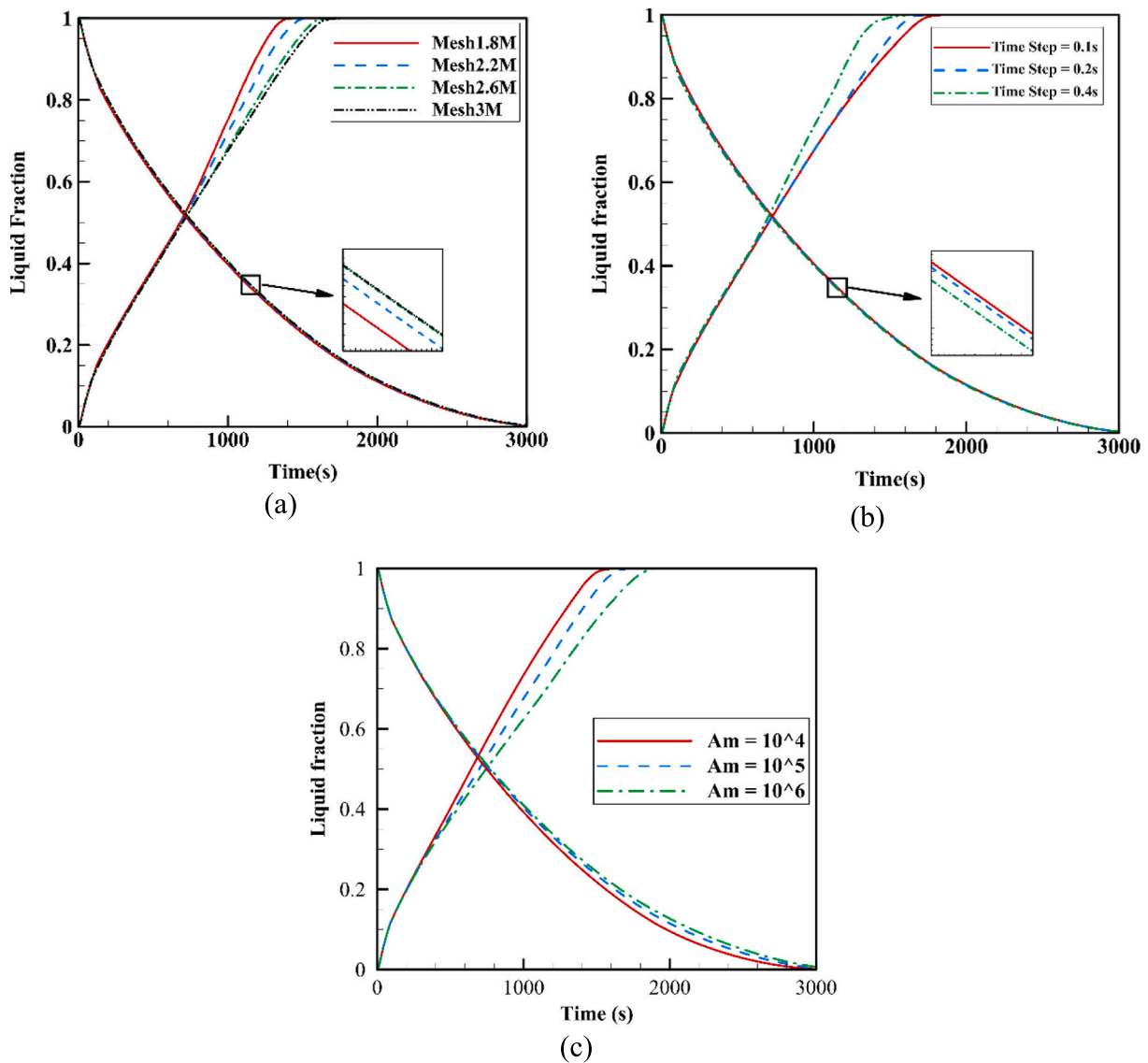


Fig. 3. Effect of numerical resolution on the predicted melting performance of the PCM: (a) influence of mesh density on the temporal evolution of the liquid fraction, (b) influence of time-step size on the accuracy and stability of the solution, and (c) influence of mushy zone constant on temporal evolution of liquid fraction.

standard enthalpy–porosity practice in this temperature range.

A4: Thermophysical properties. Material properties of RT-35 are taken from Table 2; k , c_p , and L are treated as temperature-independent within each phase; latent heat is included via $\lambda(T)$ between $T_{\text{solidus}} = 32^\circ\text{C}$ and $T_{\text{liquidus}} = 38^\circ\text{C}$.

A5: Geometry and orientation. The exchanger is vertical; gravity acts in $-z$. The twisted inner tube has 0–5 embedded voids and 2–4 helical revolutions per the eight cases (Table 1).

A6: Walls and interfaces. All solid walls are no-slip. Conduction in the metal walls is neglected; the inner wall is prescribed isothermal, representing a controlled HTF surface.

A7: Thermal losses. Radiation and external convection are neglected; outer boundaries are adiabatic.

A8: Temporal setting. Fully transient simulations are performed for two operating modes (charge/discharge); convergence within each time step is set by residuals $< 10^{-6}$ for energy and $< 10^{-4}$ for continuity/momentum.

A9: Mushy constant. The momentum damping coefficient is $A_m = 10^5$; a small $\epsilon = 10^{-3}$ avoids singularity as $\lambda \rightarrow 0$.

A10: Mesh independence and time-step sensitivity. Grid and Δt studies are provided in Fig. 3; the selected mesh/time step (Mesh -2 M, $\Delta t = 0.2$ s) ensures $< 0.5\%$ variation in liquid fraction and stable solutions.

3.3. Boundary and initial conditions

Boussinesq is applied exclusively in the buoyancy term with $\rho_0 = 815 \text{ kg m}^{-3}$ and $T_{\text{ref}} = 35^\circ\text{C}$, elsewhere the flow is incompressible with constant ρ_0 . The liquid-fraction law $\lambda(T)$ follows the piecewise relation between 32°C and 38°C . The accuracy of the mathematical model relies on the precise definition of boundary and initial conditions that represent the actual operation of the proposed latent heat thermal energy storage unit. The system undergoes cyclic operation involving two distinct modes: charging (melting) and discharging (solidification), each governed by specific thermal conditions.

During the charging process, the entire PCM domain is initially set to a uniform temperature of 30°C , corresponding to the solid state. The HTF in the inner multi-lobed tube, serving as the primary heat transfer source, is circulated at a constant temperature of 60°C and velocity of 0.0322 m s^{-1} to provide the thermal gradient required for melting. In this stage, the outer tube and any auxiliary heat transfer surfaces are assumed to be adiabatic, thereby eliminating unintended heat losses and ensuring that energy transfer occurs only through the designated boundary.

Following completion of the charging process, the system proceeds to the discharging phase. At this point, the PCM is initialized at 40°C in the liquid state. The thermal boundary conditions are then inverted: the HTF is passed through the inner tube at the inlet temperature of 10°C and velocity of 0.0322 m s^{-1} to promote solidification, while the remaining boundaries are treated as thermally insulated. The imposed temperature differences of 30°C in both charging and discharging cycles are chosen to reflect practical operating ranges for medium-temperature thermal storage applications, such as building energy management and solar thermal systems.

It should be noted that a constant inlet velocity boundary condition was applied in all simulations and as mentioned the PCM and pipe volumes were kept constant among cases. Since the inlet flow area is the same in all configurations (except case 1 with no voids), this also results in a constant volumetric flow rate for all studied cases. However, the wetted perimeter, and thus the hydraulic diameter, differ between geometries, so the corresponding Reynolds numbers are not identical. This choice is consistent with practical applications, where a constant flow rate is often imposed, while the internal pipe geometry may vary.

3.3.1. Charging (melting) stage

- Initial condition (PCM): uniform $T = 30^\circ\text{C}$ (solid, $\lambda = 0$).
- Inlet HTF side: $T_{\text{inlet}} = 60^\circ\text{C}$ and $V_{\text{inlet}} = 0.0322 \text{ m s}^{-1}$.
- Outer wall and end caps: adiabatic ($n \cdot k \nabla T = 0$).
- Velocity: no-slip at all walls; initial $v = 0$ in PCM.
- Gravity: $g = 9.81 \text{ m s}^{-2}$ along $-z$ to drive buoyancy in the PCM.
- Stopping criterion: simulations continue until the domain-averaged liquid fraction reaches 1 (complete melting) and the melting time t_m is recorded.

3.3.2. Discharging (solidification) stage

- Initial condition (PCM): uniform $T = 40^\circ\text{C}$ (liquid, $\lambda = 1$).
- Inlet HTF side: $T_{\text{inlet}} = 10^\circ\text{C}$ and $V_{\text{inlet}} = 0.0322 \text{ m s}^{-1}$.
- Outer wall and end caps: adiabatic.
- Velocity: no-slip at all walls; initial $V = 0 \text{ m s}^{-1}$ in PCM.
- Gravity: $g = 9.81 \text{ m s}^{-2}$ along $-z$ to drive buoyancy in the PCM.
- Stopping criterion: simulations continue until the domain-averaged liquid fraction falls to 0 (complete solidification) and the solidification time t_s is recorded.

3.3.3. Wall, symmetry, and gravity settings

- No symmetry planes are imposed; the full 3D geometry is solved to capture helical-induced secondary flows.
- Wall motion: all walls are stationary; no slip and impermeable.
- Buoyancy model: Boussinesq, $\rho_0 = 815 \text{ kg m}^{-3}$, $\beta = 6 \times 10^{-4} \text{ K}^{-1}$, and $T_{\text{ref}} = 35^\circ\text{C}$.

4. Numerical modelling and validation

The numerical simulations were performed in ANSYS Fluent using the finite volume method to discretize the governing equations. The SIMPLE algorithm was employed for pressure–velocity coupling, while the QUICK scheme was applied to discretize the momentum and energy equations, providing third-order accuracy in capturing heat transfer and fluid flow behavior during the PCM melting and solidification processes. To ensure reliable pressure correction, particularly in the presence of complex multi-lobed geometry, the PRESTO scheme was adopted. The under-relaxation factors were fixed at 0.3 for both pressure and velocity, 0.5 for the liquid fraction, and 1.0 for the energy equation. Convergence thresholds were defined as 10^{-4} for continuity and momentum equations, and 10^{-6} for the energy equation.

The solver was validated using 2D canonical benchmarks that exercise the enthalpy–porosity phase-change formulation and Boussinesq coupling (the Stefan problem; natural-convection melting in a differentially heated square cavity). All production simulations were conducted in 3D owing to the helical geometry; mesh and time-step independence were verified, and global energy-balance errors were kept below 1 %.

A mesh independence study was carried out for the reference configuration (Case 6) using structured quadrilateral grids within both the PCM and HTF domains. Fig. 3(a) illustrates the effect of mesh refinement on the predicted liquid fraction during the melting phase. It can be observed that successive grid refinements result in negligible differences, with variations of less than 0.5 %, indicating that the solution is grid independent.

In addition to spatial resolution, a time-step sensitivity analysis was performed by testing intervals of 0.1 s, 0.2 s, and 0.4 s. As shown in Fig. 3 (b), the results confirm that a time step of 0.2 s offers a balance between accuracy and computational efficiency. While the smallest time step (0.1 s) produces nearly identical results, the computational cost is higher, whereas the largest time step (0.4 s) introduces noticeable deviations. Therefore, 0.2 s was selected as the optimal value. The use of implicit time discretization guarantees unconditional stability for

varying thermal diffusion scales during phase change processes.

The selected computational mesh used for the main simulations is depicted in Fig. 4. The mesh distribution across the PCM and HTF regions demonstrates a structured arrangement, with local refinements near the multi-lobed inner tube and void surfaces to accurately capture steep gradients in velocity and temperature. The enlarged view of the helical section further highlights the smooth and uniform grid distribution, ensuring accurate resolution of complex flow and thermal fields within the domain.

The computational domain was meshed using a structured hexa/quad topology with local cell refinement concentrated near the multi-lobed inner tube and the void edges to capture steep temperature and velocity gradients (see Fig. 4). To accurately resolve wall-normal (boundary-layer) gradients adjacent to the inner tube and outer shell, prism (inflation) layers were generated on all solid walls, five layers with a growth ratio of 1.2 and a first-layer thickness selected to provide fine near-wall spacing consistent with accurate resolution of the natural-convection boundary layer. The production mesh (denoted Mesh -2 M in the mesh-sensitivity study) contained approximately 2.0×10^6 cells. Mesh quality was verified in ANSYS: minimum orthogonal quality >0.10 , maximum cell skewness <0.85 , and average skewness ≈ 0.25 . Global energy-balance errors were maintained below 1 % for all simulations, confirming overall numerical consistency.

Although the PCM region may appear relatively coarsely meshed in the visualization, the mesh was intentionally graded to concentrate cell density in zones of steep gradients (particularly near the inner wall, void boundaries, and phase-change fronts) while employing larger cells in the bulk PCM where temperature and velocity fields vary slowly. This graded strategy balances computational efficiency with numerical accuracy. The mesh-independence analysis (Section 4, Fig. 3) confirmed that the adopted grid yields less than 0.5 % deviation in predicted liquid fraction and melting time compared with a finer reference mesh ($\approx 3 \times 10^6$ cells). Therefore, the chosen mesh resolution, boundary-layer refinement, and time-step size ($\Delta t = 0.2$ s) collectively ensure that the simulation results are both grid- and time-convergent while maintaining efficient computational performance.

To address potential concerns regarding mesh sensitivity, it is clarified that the mesh in Fig. 4 is the final production mesh used for all simulations. The grid was strategically graded, with high refinement near the inner tube, void boundaries, and phase-change fronts to resolve steep gradients, while employing larger cells in the PCM bulk where variations are gradual. This approach, validated by the mesh independence study showing less than 0.5 % deviation in key outputs, ensures accuracy while maintaining computational efficiency. A structured hex-

dominant mesh was preferred over polyhedral for its superior control over boundary layer resolution and proven robustness with the enthalpy-porosity model in ANSYS Fluent for transient phase-change simulations.

To verify that the chosen mushy-zone constant ($Am = 10^5$) is correct, a sensitivity test was performed using the representative configuration (Case 6), with the results are displayed in Fig. 3(c). Three values ($Am = 10^4$, 10^5 , and 10^6) were examined, corresponding to the range typically adopted in the literature for enthalpy-porosity formulations of PCM melting and solidification [40–42]. During the conduction-dominated period at the beginning of both melting and solidification, natural convection is still weak, so the flow field remains almost quiescent, and the thermal gradients are primarily governed by pure conduction. In this regime, the influence of the mushy-zone constant on the momentum sink term is minimal, and the differences between the cases are therefore negligible. Quantitatively, in the first half of the phase-change processes, the evolution of the liquid fraction differs by less than 1 % across the tested Am values, indicating that the global melting/solidification rate is essentially insensitive to the choice of Am in this early stage. However, toward the end of melting/solidification, when convection becomes dominant and stronger recirculating flows develop in the liquid region, the damping effect of the mushy-zone term becomes more pronounced. As a result, larger deviations in the liquid-fraction histories are observed between the different Am values, particularly near the completion of the phase change, which is consistent with trends reported in previous numerical studies [45]. Following the approach recommended by Karami and Kamkari [46], the mushy-zone constant is selected based on agreement with experimental data; for the present configuration, this corresponds to 10^5 . It is acknowledged that prior works employing $Am = 10^5$ used different geometrical configurations, such as horizontally oriented PCM units, spherical capsules, or rectangular enclosures. However, the present results confirm that, for the current vertical double-pipe geometry, $Am = 10^5$ provides an appropriate balance between physical realism and numerical stability.

The numerical results were validated using the experimental data reported by Al-Abidi et al. [47] for a finned triplex-tube system employing PCM with similar thermophysical properties and subjected to comparable thermal boundary conditions. In their experiments, calibrated thermocouples with an accuracy of ± 0.5 % were placed at selected positions within the PCM to record the temporal evolution of temperature and to capture the advancement of the phase change interface.

Fig. 5 illustrates the comparison between the present numerical predictions and the experimental measurements for both the charging

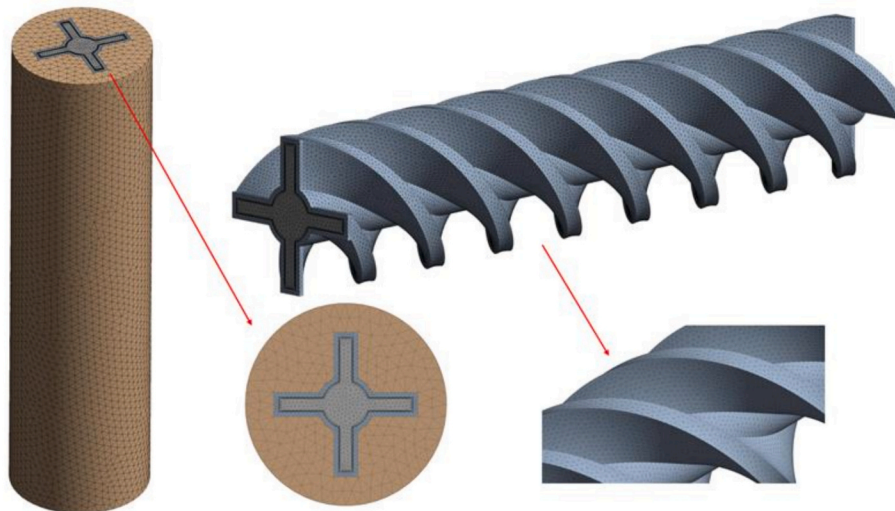


Fig. 4. Computational mesh adopted for the selected configuration.

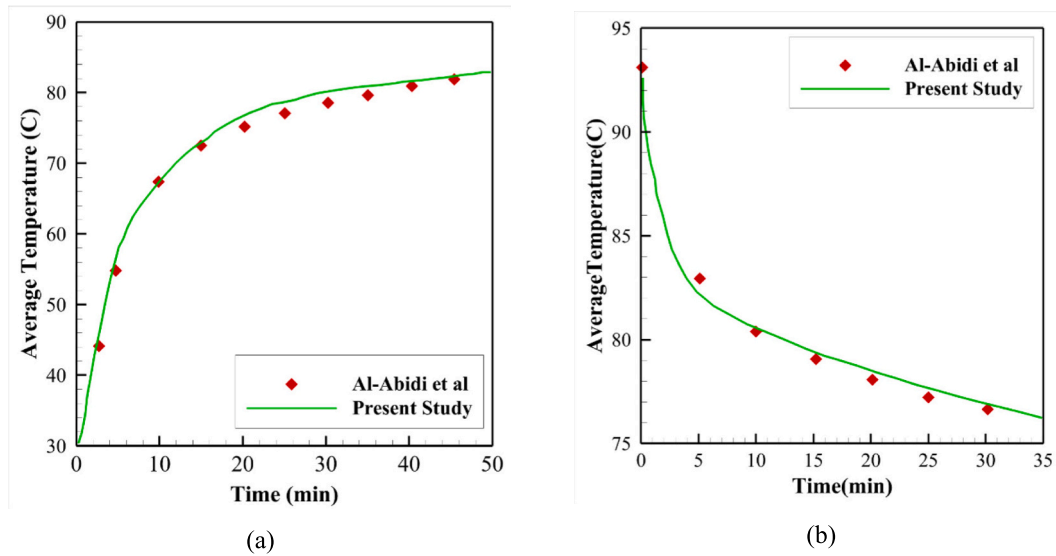


Fig. 5. Validation of the numerical model against the experimental data of Al-Abidi et al. [47] for a finned triplex-tube system with comparable PCM properties and boundary conditions: (a) charging phase, showing close agreement during sensible heating, isothermal melting, and superheating stages; (b) discharging phase.

and discharging processes. As shown in Fig. 5(a), the charging phase demonstrates excellent agreement, with deviations remaining below 3 % throughout the sensible heating, isothermal melting, and superheating stages. Similarly, Fig. 5(b) presents the discharging phase, where the model accurately reproduces the cooling behavior and solidification kinetics of the PCM. Minor discrepancies are observed during the intermediate solidification period; however, the overall trends are consistently well captured.

This validation confirms the reliability of the developed numerical model in simulating both melting and solidification phenomena. The close agreement with experimental benchmarks establishes a solid foundation for extending the analysis to multi-lobed geometries, thereby supporting the credibility of conclusions drawn regarding enhanced heat transfer mechanisms and optimal geometric configurations.

5. Results and discussion

The observed reductions (or increases) in melting and solidification times can be interpreted by considering the dominant heat-transfer mechanisms. Additional helical revolutions and height strengthen secondary flows along the tube and thin the thermal boundary layer at the wall, which elevates the local wall-normal temperature gradient and accelerates heat intake/release. Introducing axial voids shortens conduction paths through the PCM and helps organize natural-convection cells in the liquid phase; this expedites front propagation up to a point, beyond which the reduction in active PCM volume leads to diminishing returns. These effects jointly explain the trends reported in the figures without altering the underlying model.

This section presents a comprehensive analysis of the thermal performance of a novel void-enhanced double-pipe heat exchanger compared to a conventional design. Through systematic evaluation of key parameters (including void number, height, helical revolutions, and Reynolds number), the impact of geometric modifications on melting (charging) and solidification (discharging) processes have been quantified. The results demonstrate significant enhancements in heat storage/recovery rates and phase-change efficiency, supported by liquid fraction and temperature contour visualizations. Discussion focuses on the underlying heat transfer mechanisms, performance trade-offs, and practical implications for thermal energy storage applications. These findings provide critical insights for optimizing heat exchanger designs to achieve faster thermal cycling and improved energy efficiency.

5.1. Charging phase analysis

5.1.1. Effect of void number on melting dynamics during the charging phase

Fig. 6 presents liquid fraction contours for Cases 2, 3, and 4, which correspond to twisted inner pipes with 3, 4, and 5 voids, respectively. The contours are visualized at four different time intervals (up to 1500 s) to illustrate the phase change dynamics of the PCM within the heat exchanger. The height (H) of the voids is fixed at 18 mm, and the pipe completes 2 helical revolutions, ensuring consistent geometric constraints across cases. By comparing the liquid fraction distributions, the figure highlights how the number of voids influences melting rates and heat transfer uniformity. For instance, Case 4 (5 voids) exhibit faster melting due to increased surface disruption, while Case 2 (3 voids) could show more localized heat absorption. The time-resolved snapshots (e.g., 300 s, 600 s) reveal transient behaviors, such as asymmetric melting patterns or void-induced convection effects, which are critical for evaluating thermal performance.

In this geometry, each axial void shortens the effective conduction path across the PCM and seeds buoyancy cells adjacent to the inner wall. As the void count increases, these recirculations become more numerous and the near-wall thermal boundary layer is thinned, advancing the melting front and improving early/mid-stage heat absorption.

The liquid fraction contours in Fig. 6 provide a direct visual comparison of how void quantity impacts PCM melting. The progression over time suggests that higher void numbers (e.g., Case 4) enhance heat transfer by creating more turbulence and increasing the effective contact area between the pipe and PCM. However, trade-offs exist, such as uneven melting or reduced structural stability with excessive voids. The figure's focus on early to mid-stage melting (up to 1500 s) is particularly useful for identifying design optimizations, for example, whether fewer, larger voids (Case 2) or more, smaller voids (Case 4) achieve faster phase change.

The gains from 3 $\rightarrow$ 4 voids are larger than from 4 $\rightarrow$ 5 voids because the beneficial effects of shorter conduction distance and better-organized convection rolls begin to saturate once the liquid layer is well established. Beyond a threshold, additional voids also reduce contiguous PCM volume and can create isolated hot spots that couple less effectively to the bulk, yielding diminishing returns. The late stage of melting is increasingly limited by mushy-zone resistance, so differences between cases narrow as completion is approached. This cause-effect picture explains the ordering observed in Fig. 6 and the quantitative trend later summarized in Table 3.

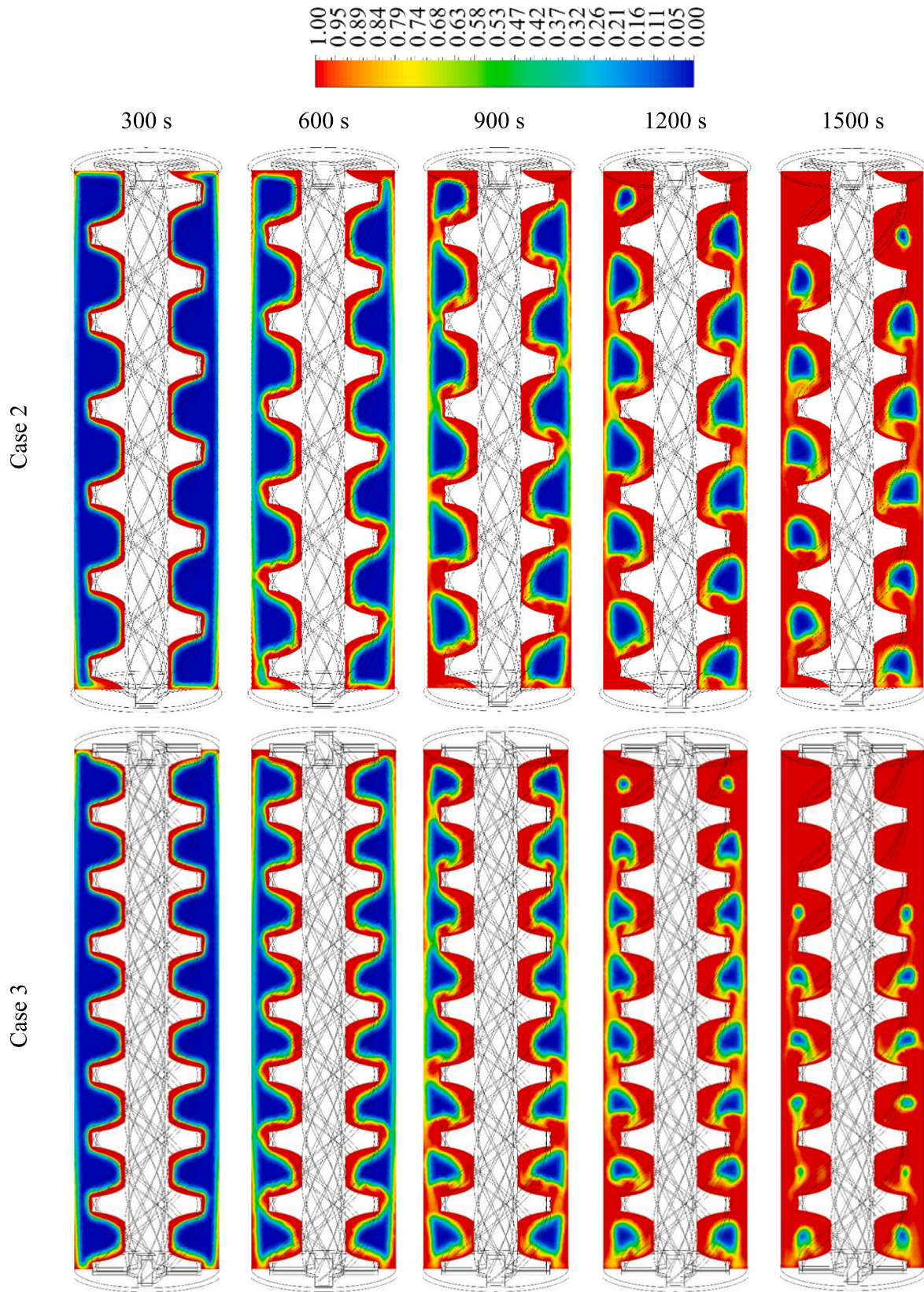


Fig. 6. Liquid fraction contours for cases 2, 3, and 4 for four various times up to 1500 s.

Fig. 7 displays temperature contours for Cases 2, 3, and 4 at four distinct time intervals (300 s, 600 s, 800 s, and 1200 s, up to 1500 s), illustrating the thermal evolution within the heat exchanger. The cases

represent varying void configurations: Case 2, 3, and 4. The temperature gradients visualized in the contours reveal how heat propagates through the PCM, with warmer regions indicating active melting and cooler

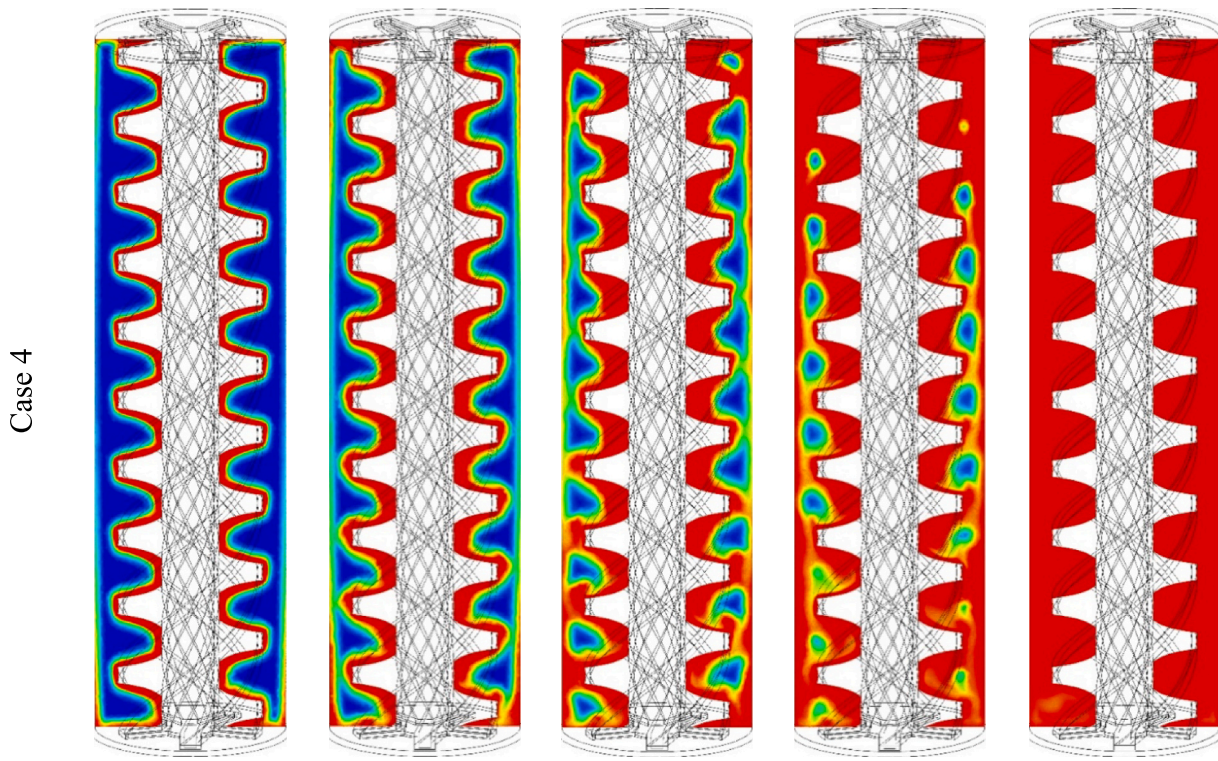


Fig. 6. (continued).

Table 3

Operating conditions for the simulated cases of the void-embedded twisted-tube heat exchanger.

case	Number of voids	Inlet Velocity (m s^{-1})	Re	Q ($\text{cm}^3 \text{s}^{-1}$)
1	0	0.0322	993	24
2	3	0.0322	560	24
3	4	0.0322	463	24
4	5	0.0322	394	24
5	5	0.0322	641	24
6	5	0.0322	368	24
7	5	0.0322	471	24
8	0	0.0322	641	10

zones reflecting slower heat absorption. The figure highlights differences in thermal distribution attributable to the number of voids, such as more uniform heating in Case 4 (5 voids) compared to localized hot spots in Case 2 (3 voids). By describing snapshots at progressive time steps, the figure effectively demonstrates the transient thermal behavior and the impact of void quantity on heat diffusion efficiency. The progression from 300 s to 1500 s suggests that increasing the number of voids (e.g., Case 4) promotes faster and more even heat distribution, due to improved fluid mixing and larger interfacial contact areas. However, the figure also reveals potential drawbacks, such as thermal stratification or regions of delayed melting, which could inform design refinements.

Fig. 8a presents velocity contours at $t = 1200$ s for Cases 2, 3, and 4, depicting the flow dynamics within the PCM region of the twisted-pipe heat exchanger. The velocity magnitude is capped at 1 mm s^{-1} to emphasize the distribution patterns rather than extreme values, revealing how the number of voids (3, 4, and 5, respectively) influences fluid motion. The contours show enhanced circulation and turbulence in Case 4 (5 voids) due to greater flow disruption from the voids, while Case 2 (3 voids) exhibit more localized or weaker vortices. This visualization underscores the role of void quantity in promoting convective heat transfer, as higher void numbers appear to correlate with more

uniform and vigorous fluid mixing. The capped scale ensures clarity in comparing subtle differences between cases, providing insights into design trade-offs between void-induced flow enhancement and potential energy losses.

Fig. 8b presents the velocity streamlines within the PCM at $t = 1200$ s for Cases 2, 3, and 4, which feature 3, 4, and 5 voids respectively, with the velocity magnitude capped at 1 mm s^{-1} to elucidate the distribution patterns. The visualization reveals that increasing the number of voids significantly enhances fluid motion and circulation within the PCM domain. Case 2 (3 voids) exhibits more localized and weaker vortices, whereas Case 4 (5 voids) demonstrates more vigorous and uniformly distributed flow, characterized by stronger secondary flows and recirculation zones. This enhanced fluid dynamics, induced by the greater flow disruption from additional voids, directly promotes convective heat transfer by thinning thermal boundary layers and improving the mixing of warmer and cooler PCM regions. The figure thus provides clear visual evidence that a higher void count intensifies the internal flow field, which is a key mechanism behind the accelerated melting rates observed in these configurations.

Fig. 9 provides a comparative analysis of Cases 2, 3, and 4 by examining two critical parameters during the melting process: liquid fraction (Fig. 9(a)), mean temperature (Fig. 9(b)) and inlet-outlet temperature difference of the HTF. The liquid fraction curves illustrate the progression of PCM melting over time, with Case 4 (5 voids) reaching higher liquid fractions earlier compared to Cases 3 (4 voids) and 2 (3 voids). For instance, at $t = 900$ s, Case 4 might exhibit a liquid fraction of 0.7, while Cases 3 and 2 could lag at 0.5 and 0.4, respectively. This trend suggests that increasing the number of voids accelerates melting due to enhanced heat transfer from greater surface disruption and convective mixing. However, the curves also reveal diminishing returns, for example, the performance gap between Cases 3 and 4 might narrow after $t = 1200$ s, indicating an optimal void count beyond which gains plateau.

The temperature (Fig. 9(b)) complements these observations by showing the thermal response of the PCM. Case 4 demonstrates a steeper temperature rise, reaching 54°C by $t = 1550$ s, whereas Cases 3 and 2

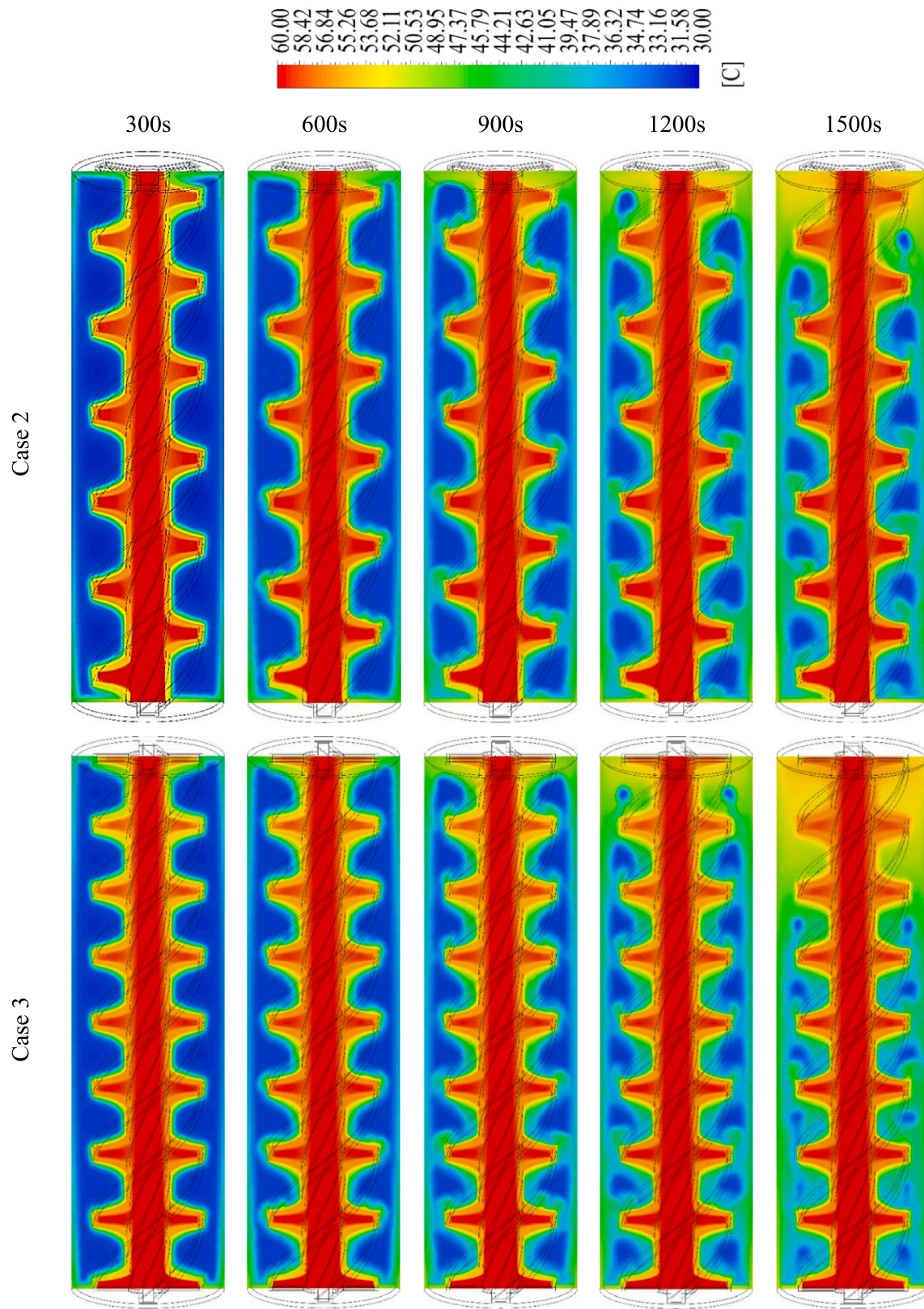


Fig. 7. Temperature contours for cases 2, 3, and 4 for four various times up to 1500 s.

might achieve only 45 °C and 42 °C at the same time. This aligns with the liquid fraction data, confirming that more voids promote faster heat absorption. However, the temperature curves also expose transient inefficiencies, such as slower initial warming in Case 2 due to fewer void-

induced flow disturbances. The convergence of temperatures at later stages ($t > 1500$ s) could imply that all cases eventually approach similar steady-state conditions, emphasizing the importance of early-stage performance for practical applications. While Case 4 offers superior

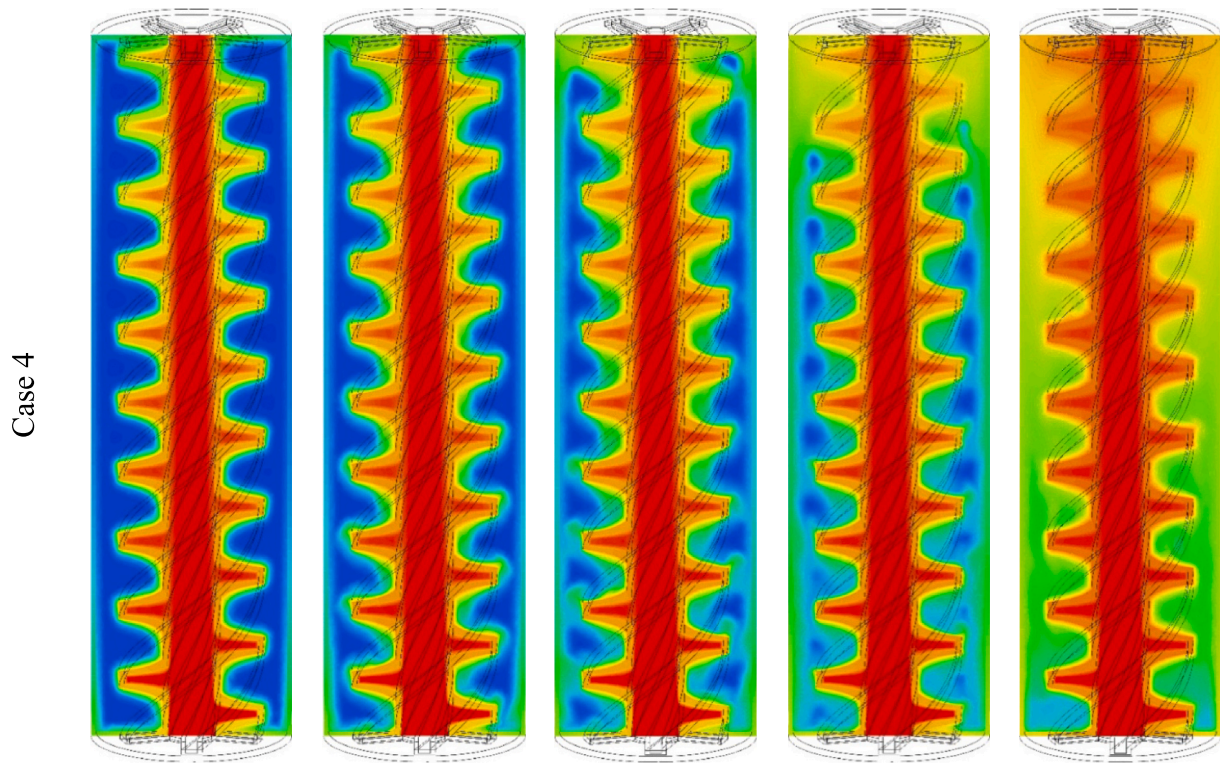


Fig. 7. (continued).

initial melting rates, its complexity (5 voids) introduces manufacturing challenges or material stresses compared to Case 3. Engineers might prioritize Case 3 if it delivers a balance between performance (e.g., 90 % liquid fraction by $t = 1500$ s) and practicality.

The temperature difference between the HTF inlet and outlet, shown in Fig. 9c exhibits a sharp initial peak followed by a gradual decay toward nearly zero for all void configurations. At the beginning of the charging, the PCM is entirely solid and relatively cold, so the HTF experiences a large heat sink, leading to a high ΔT , especially for the case with five voids, which provides the largest heat transfer area and thus the greatest initial cooling of the fluid. As melting progresses and the PCM temperature increases, the driving temperature difference between the HTF and PCM decreases and the melt fraction grows, so the outlet temperature approaches the inlet temperature and ΔT steadily diminishes. The cases with more voids show slightly higher ΔT values in the early stages, reflecting their enhanced heat transfer capability, but the difference among the 3-, 4-, and 5-void designs becomes small at longer times when most of the PCM has melted, and the system approaches a quasi-steady condition. Overall, the relatively modest ΔT after the initial transient indicates that the HTF temperature variation along the flow direction is limited, supporting the assumption of an approximately constant wall temperature during the melting process. It should be noted that, as mentioned, in this study, the HTF flow is numerically modelled.

The data presented in Table 4 demonstrate a clear relationship between the number of voids in the heat exchanger design and its thermohydraulic performance. As the void count increases from 3 (Case 2) to 5 (Case 4), significant improvements are observed in both heat storage rate and melting time, with only a marginal reduction in pressure drop and required working power. Case 2, with 3 voids, exhibits the lowest heat storage rate of 84.60 W and the longest melting time of 2269 s, alongside a pressure drop of 460.88 Pa and a pumping power of 0.011071 W. In contrast, Case 4, with 5 voids, achieves a substantially higher heat storage rate of 119.40 W, reduces the melting time to 1570 s, and also slightly lowers the flow resistance, evidenced by a pressure

drop of 459.78 Pa and a pumping power of 0.011044 W. Case 3, with 4 voids, represents an intermediate performance level. These results indicate that additional voids enhance heat transfer by increasing the surface area and promoting turbulence within the PCM, leading to more efficient energy storage and faster phase change, without incurring a hydraulic penalty. The improvements in thermal performance, however, follow a pattern of diminishing returns, as the transition from 3 to 4 voids yields a 24.3 % increase in heat storage rate and a 21.3 % reduction in melting time, while the move from 4 to 5 voids results in smaller gains of 13.5 % and 12.0 %, respectively. This suggests that while adding voids initially provides substantial benefits, the incremental advantages decrease as the void count rises. For practical applications, this implies that an optimal void count must balance performance enhancements with design feasibility.

This pattern of diminishing thermal returns occurs despite a minimal associated hydraulic penalty. As shown in Table 4, the pressure drops and pumping power remain nearly constant, decreasing slightly from 460.88 Pa to 459.78 Pa and from 0.011071 W to 0.011044 W as the void count increases from 3 to 5. This indicates that the saturation of thermal performance is not driven by a significant increase in flow resistance but is instead a result of the physical limits of the enhancement mechanism, namely, the reduced contiguous PCM volume and the eventual ineffectiveness of additional voids in further disrupting the thermal boundary layer once the flow is sufficiently mixed.

5.1.2. Effect of helical revolutions on PCM melting performance during charging

Fig. 10 compares the liquid fraction contours of Case 4 (2 helical revolutions) and Case 5 (4 helical revolutions) at 600 s and 1200 s, maintaining a constant void height ($H = 18$ mm) and count (5 voids). The visualization reveals that increasing the number of revolutions from 2 to 4 significantly enhances PCM melting uniformity and speed. At 600 s, Case 5 (4 revs) exhibits more advanced melting fronts and fewer stagnant PCM regions compared to Case 4, indicating stronger convective mixing induced by the tighter helical twist. By 1200 s, this

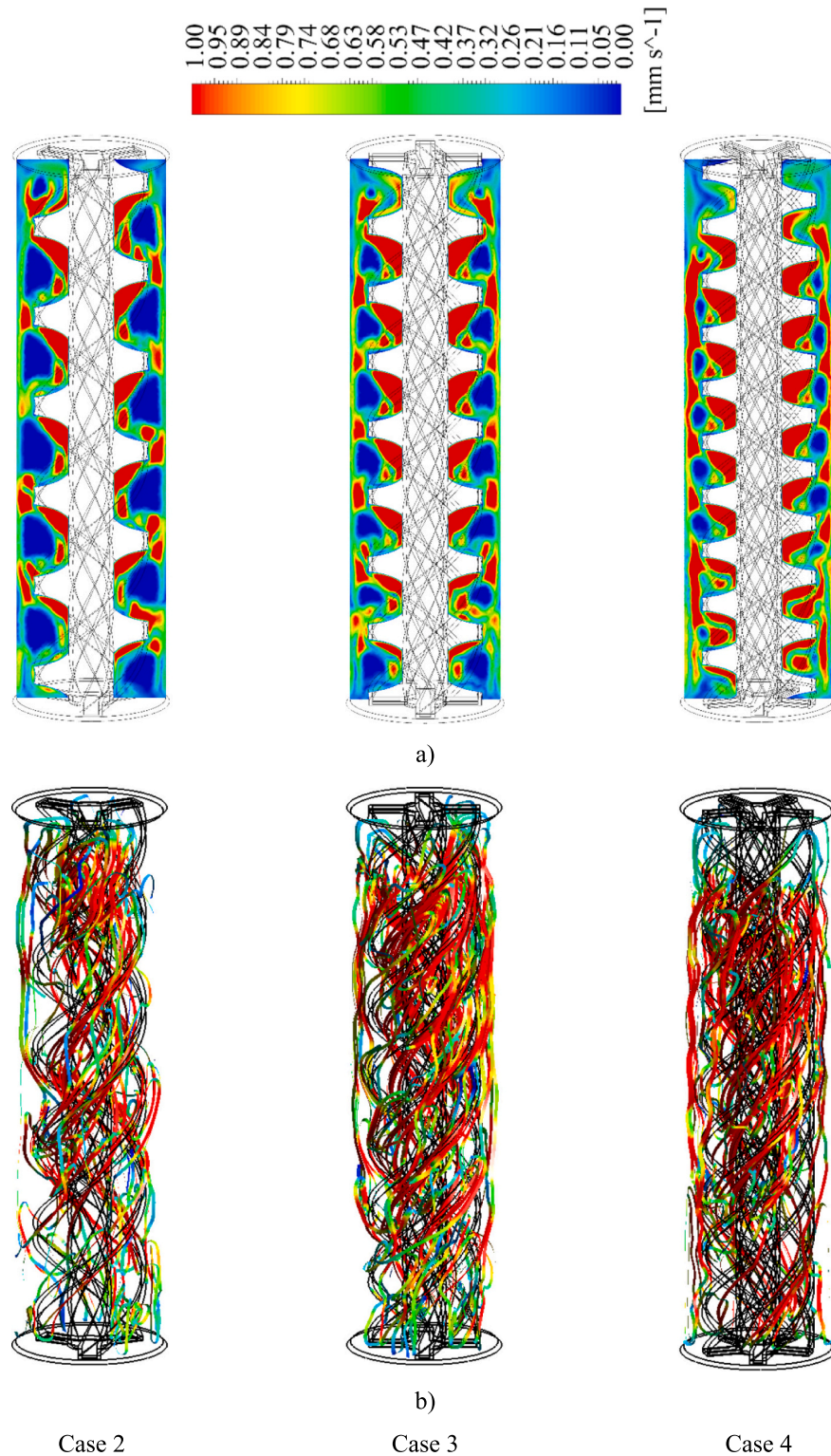


Fig. 8. a) Velocity contours and b) streamlines at $t = 1200$ s for the cases 2, 3, and 4, the maximum value is capped at 1 mm s^{-1} to better show the distribution.

difference becomes more pronounced, with Case 5 achieving near-complete melting in broader zones, while Case 4 still shows localized solid pockets. The results demonstrate that higher helical revolutions improve heat transfer by elongating the flow path and increasing turbulence, which accelerates phase change. However, the trade-off between enhanced performance and potential pressure drops or manufacturing complexity warrants further investigation for practical applications.

Increasing the number of helical turns strengthens secondary flows (Dean-type vortices) and thins the thermal boundary layer along the inner wall. The longer, more tortuous path also increases local residence time of the liquid PCM near the wall. Together these effects raise the near-wall temperature gradient and sustain a higher effective transfer, which explains the earlier and more uniform melting fronts observed for the 4-revolution case.

Fig. 11 compares the temperature distributions of Case 4 (2 helical

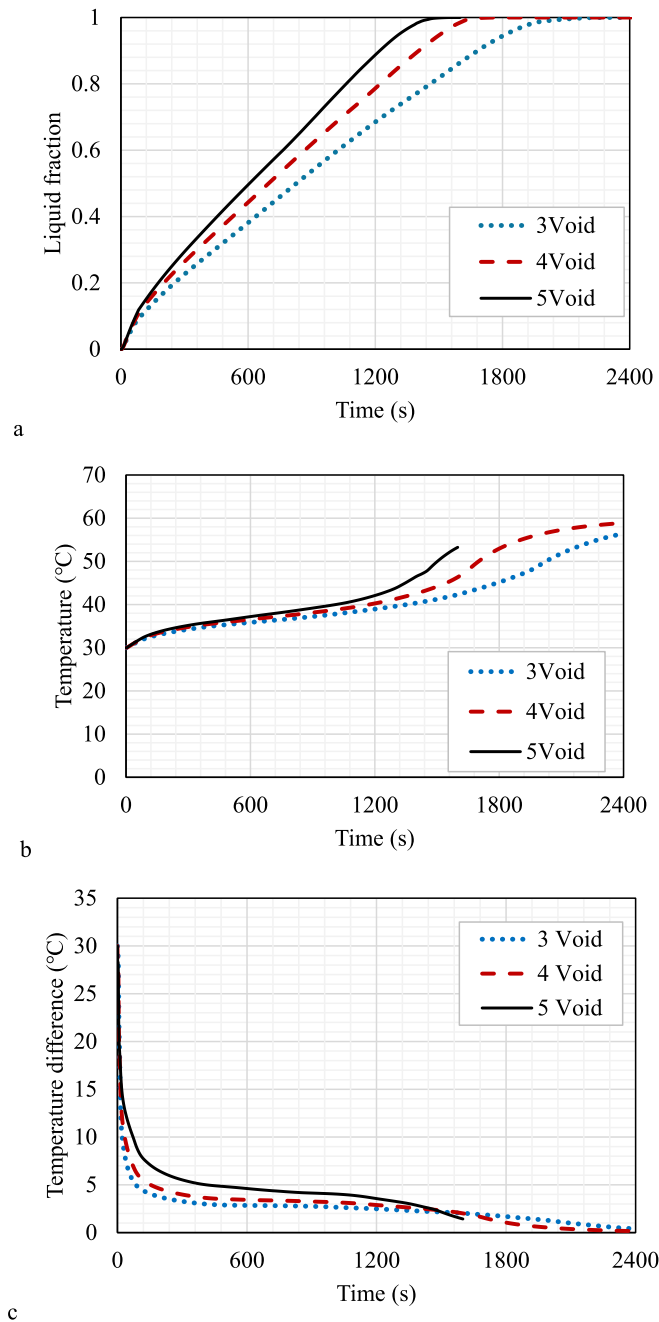


Fig. 9. Comparison of the cases 2, 3, and 4 regarding a) liquid fraction, and b) temperature during the melting process.

Table 4
The charging period and the thermal storage rate for the cases 2, 3 and 4.

	Heat storage rate (W)	Melting time (s)	Pressure drop (Pa)	Pumping power (W)
Case 2	84.60	2269	460.88	0.011071
Case 3	105.16	1785	460.69	0.011066
Case 4	119.40	1570	459.78	0.011044

revolutions) and Case 5 (4 revolutions) at 600 s and 1200 s, revealing that increased helical turns enhance heat transfer efficiency. At both timesteps, Case 5 exhibits more uniform and higher temperature

distributions compared to Case 4, with fewer cold spots and faster thermal propagation. This improvement occurs because additional revolutions create a longer flow path, increasing fluid residence time and promoting turbulent mixing. The tighter twist in Case 5 also generates stronger secondary flows (Dean vortices), which disrupt thermal boundary layers and improve convection. While Case 4 shows gradual heating with distinct thermal gradients, Case 5 demonstrates rapid, homogeneous warming, confirming that helical intensity directly governs thermal performance. The trade-off, however, is potential energy losses from higher flow resistance, suggesting an optimal revolution count exists between these cases.

Fig. 12 provides a dual-axis comparison of Cases 4 (2 revolutions) and 5 (4 revolutions), tracking both liquid fraction (left y-axis) and mean temperature (right y-axis) during the melting process. The data demonstrate that Case 5 consistently achieves higher liquid fractions and temperatures at earlier timesteps than Case 4, confirming that increased helical revolutions accelerate PCM melting and improve heat transfer efficiency. For instance, at 900 s, Case 5 reaches 70 % liquid fraction and 41 °C, while Case 4 lags at 65 % and 39 °C. This performance gap stems from enhanced convective mixing in Case 5, where additional revolutions create stronger secondary flows (Dean vortices) that disrupt thermal stratification and distribute heat more uniformly. However, the curves converge at later stages, suggesting diminishing returns for extended melting durations. As melting nears completion, the remaining solid persists as thin films/pockets within the mushy region, where permeability is low and heat removal becomes conduction-limited; the benefit of stronger secondary flows therefore saturates and the liquid-fraction curves merge. These results highlight the critical role of helical geometry in optimizing thermal storage systems, though practical implementation must balance performance gains against pressure drop penalties and manufacturing constraints.

The higher liquid fraction and mean temperature for 4 revolutions are consistent with suppressed stagnant pockets and more uniform near-wall heating induced by the tighter twist. The data in Table 5 reveal important differences in the thermo-hydraulic performance between the two helical configurations. Case 5 (4 revolutions) demonstrates a 5.0 % higher heat storage rate (125.4 W vs. 119.4 W) and a 5.6 % faster melting time (1482 s vs. 1570 s) compared to Case 4 (2 revolutions). While these improvements confirm that additional helical turns enhance heat transfer, the relatively modest gains are accompanied by an increase in flow resistance; the pressure drop rises from 459.78 Pa to 462.79 Pa, and the pumping power increases from 0.011044 W to 0.011116 W. This pattern occurs because the additional turns create more turbulent flow and longer fluid residence time, improving heat exchange, but the greater flow path tortuosity also introduces greater hydraulic losses that partially offset these benefits. The results indicate that while more revolutions do boost performance, the relationship between revolution count and system efficiency is not linear, reflecting a trade-off.

From a design perspective, the choice between these configurations depends on application requirements. The 4-revolution design (Case 5) offers better thermal performance and would be preferable in systems where maximum heat transfer efficiency is critical. However, the 2-revolution design (Case 4) is more practical for many applications, as it provides nearly comparable performance (within 5 %) while being easier to manufacture and operating with marginally lower pressure drops and working power. The small performance difference, when weighed against the increased hydraulic penalty, suggests that factors such as cost and system integration requirements may outweigh the modest thermal advantages of the 4-revolution design in real-world implementations.

5.1.3. Effect of void height on melting characteristics during charging

Fig. 13 compares the liquid fraction contours of Case 6 (void dimensions: 20 × 4.4 mm) and Case 7 (14 × 6.15 mm) under identical conditions (5 voids, 4 helical revolutions) during the melting process up

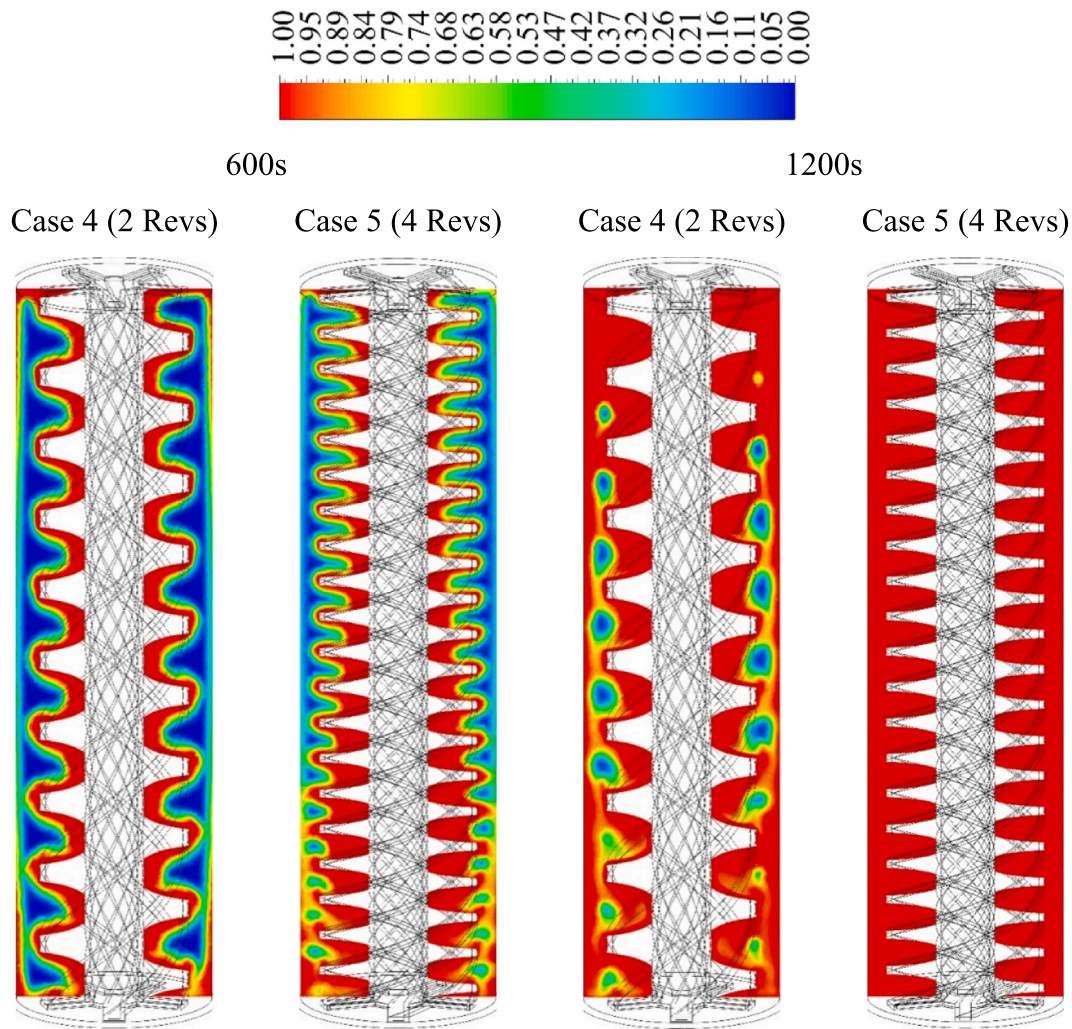


Fig. 10. Liquid fraction contours for cases 4 (2 Revolutions of helix) and 5 (4 Revolutions of helix) for the time steps 600 and 1200 s.

to 1200 s. The results reveal that Case 6 with taller voids (20 mm height) achieves significantly more advanced and uniform melting compared to Case 7 with shorter voids (14 mm). This performance difference occurs because taller voids create larger flow passages that enhance several heat transfer mechanisms. First, the increased height allows for greater fluid penetration and more vigorous convective mixing within the PCM, disrupting thermal boundary layers more effectively. Second, the expanded void volume in Case 6 provides a larger surface area for thermal energy exchange between the heat transfer fluid and PCM. Third, the aspect ratio difference (Case 6's 20×4.4 mm vs Case 7's 14×6.15 mm) alters the flow patterns, with taller voids generating stronger secondary flows that improve heat distribution throughout the PCM domain. The faster melting in Case 6 is particularly evident in the reduced solid PCM regions and more homogeneous liquid fraction distribution by 1200 s. Increasing the void height enlarges the open flow passages and wetted perimeter adjacent to the inner wall. The taller voids therefore intensify buoyancy-driven recirculation and thin the thermal boundary layer, while also shortening the cross-wall conduction path from the wall into the PCM. These effects raise the local wall heat flux and promote earlier front advance, consistent with the more uniform liquid-fraction fields of Case 6. These findings demonstrate that void height is a critical geometric parameter that directly influences thermal performance, suggesting that careful optimization of this dimension could yield substantial improvements in heat exchanger efficiency. However, designers must balance these benefits against potential trade-offs such as increased pressure drop or structural

weakening that accompany larger void dimensions. The results underscore the importance of considering three-dimensional void geometry in heat exchanger optimization, beyond just the number or arrangement of voids.

Fig. 14 presents mean temperature contours for Cases 6 (20×4.4 mm voids) and 7 (14×6.15 mm voids) during PCM melting up to 1200 s, maintaining identical void counts (5) and helical revolutions (4). The contours reveal that Case 6 achieves both higher temperatures and more uniform thermal distribution compared to Case 7, particularly in later stages (e.g., 900–1200 s). This thermal superiority stems from the taller voids in Case 6, which enhance heat transfer through three mechanisms: (1) increased fluid-PCM contact area improves conductive heat transfer, (2) larger flow channels promote stronger convective mixing and reduce dead zones, and (3) the altered void aspect ratio (4.55 for Case 6 vs. 2.28 for Case 7) optimizes vortex generation, disrupting thermal stratification more effectively. For instance, at 1200 s, Case 6 exhibits temperatures exceeding 50°C in most regions, while Case 7 shows cooler pockets (45°C) near void boundaries, indicating less efficient heat penetration. These results demonstrate that void height critically governs thermal performance, with taller voids outperforming shorter ones, though designers must evaluate trade-offs against structural integrity and pressure drop constraints when optimizing void geometry.

The data in Table 6 demonstrate a clear relationship between void height and the thermo-hydraulic performance of the heat exchanger system. Case 6, with the tallest voids (20 mm), achieves the highest heat storage rate of 153.80 W and the shortest melting time of 1185 s,

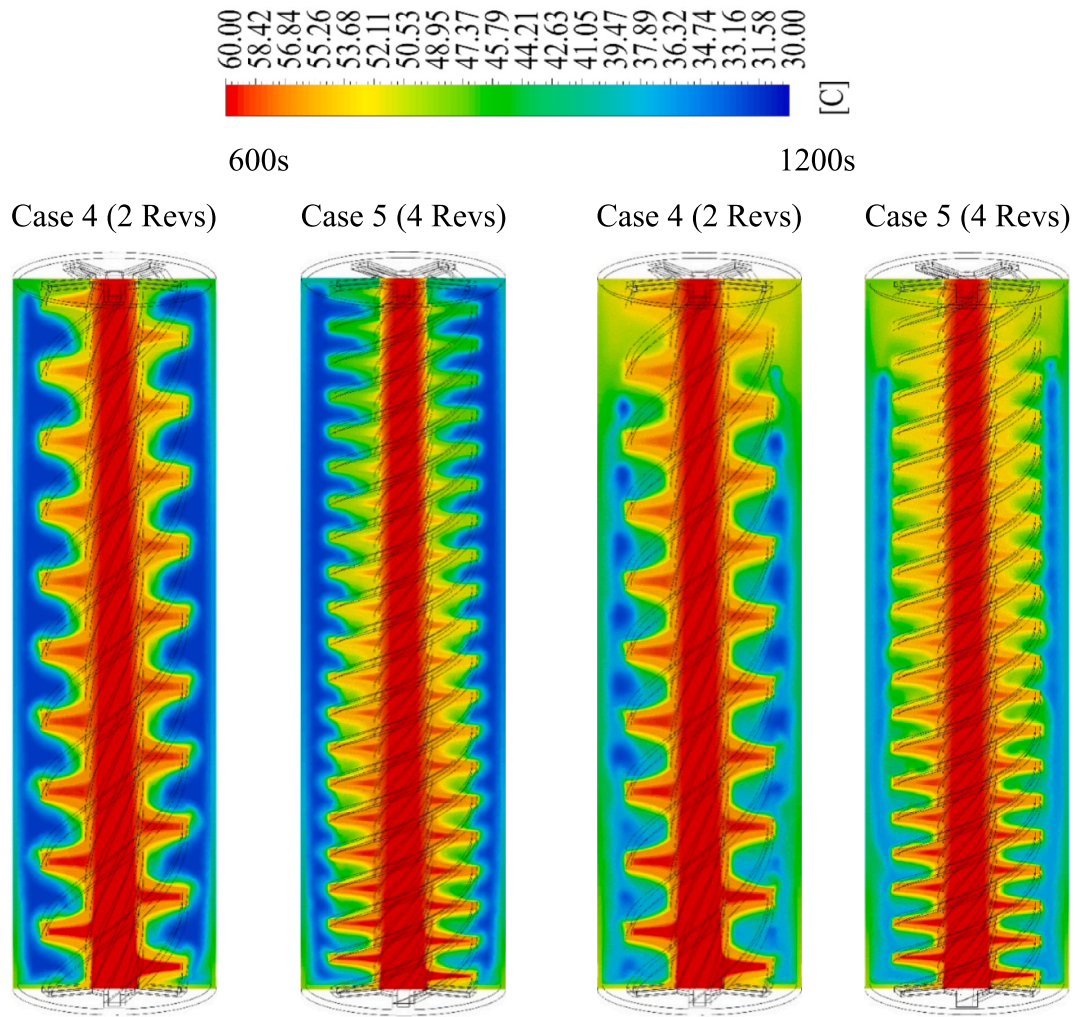


Fig. 11. Temperature contours for cases 4 (2 Revolutions of helix) and 5 (4 Revolutions of helix) for the time steps 600 and 1200 s.

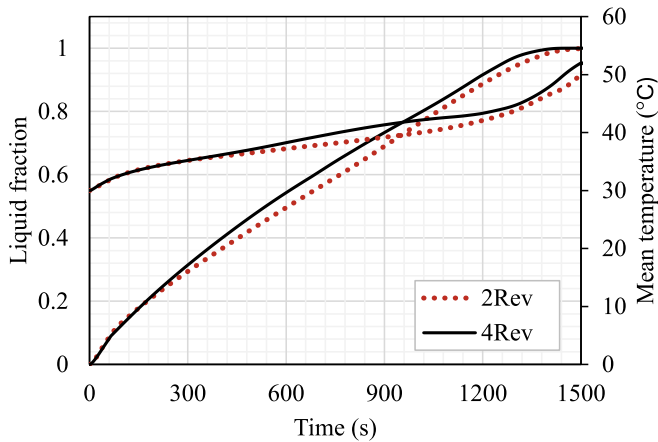


Fig. 12. Comparison of the cases 4 and 5 regarding liquid fraction (y-axis at the left), and mean temperature (y-axis at the right) during the melting process.

outperforming both Case 5 (18 mm voids) and Case 7 (14 mm voids). This significant thermal improvement is coupled with a favorable hydraulic profile; Case 6 exhibits a slightly lower pressure drop (462.72 Pa) than Case 7 and a nearly identical drop to Case 5, despite the enhanced performance. The marginal increase in pumping power to 0.011234 W is a minor trade-off for the substantial gains in charging

Table 5

The charging period and the thermal storage rate for the cases 4 and 5.

	Heat storage rate (W)	Melting time (s)	Pressure drop (Pa)	Pumping power (W)
Case 4 (2 Revolutions of helix)	119.4	1570	459.78	0.011044
Case 5 (4 Revolutions of helix)	125.4	1482	462.79	0.011116

speed and efficiency. The improvement can be attributed to the larger void height in Case 6, which enhances convective heat transfer by increasing the flow passage area and promoting more vigorous fluid mixing without proportionally increasing flow resistance. In contrast, Case 7's shorter voids (14 mm) exhibit the lowest thermal performance, as the restricted flow area limits convective mixing and results in slower heat penetration, while also incurring the highest pressure drop (463.2 Pa).

The performance trends highlight the critical role of void height in optimizing system efficiency. Case 6's 22.7 % higher heat storage rate and 20.1 % faster melting compared to Case 5 suggest that a modest increase in void height can yield significant thermal gains with minimal hydraulic penalty. The severely compromised performance of Case 7 underscores that reducing void height is detrimental to overall efficiency. These findings indicate that an optimal void height, such as 20

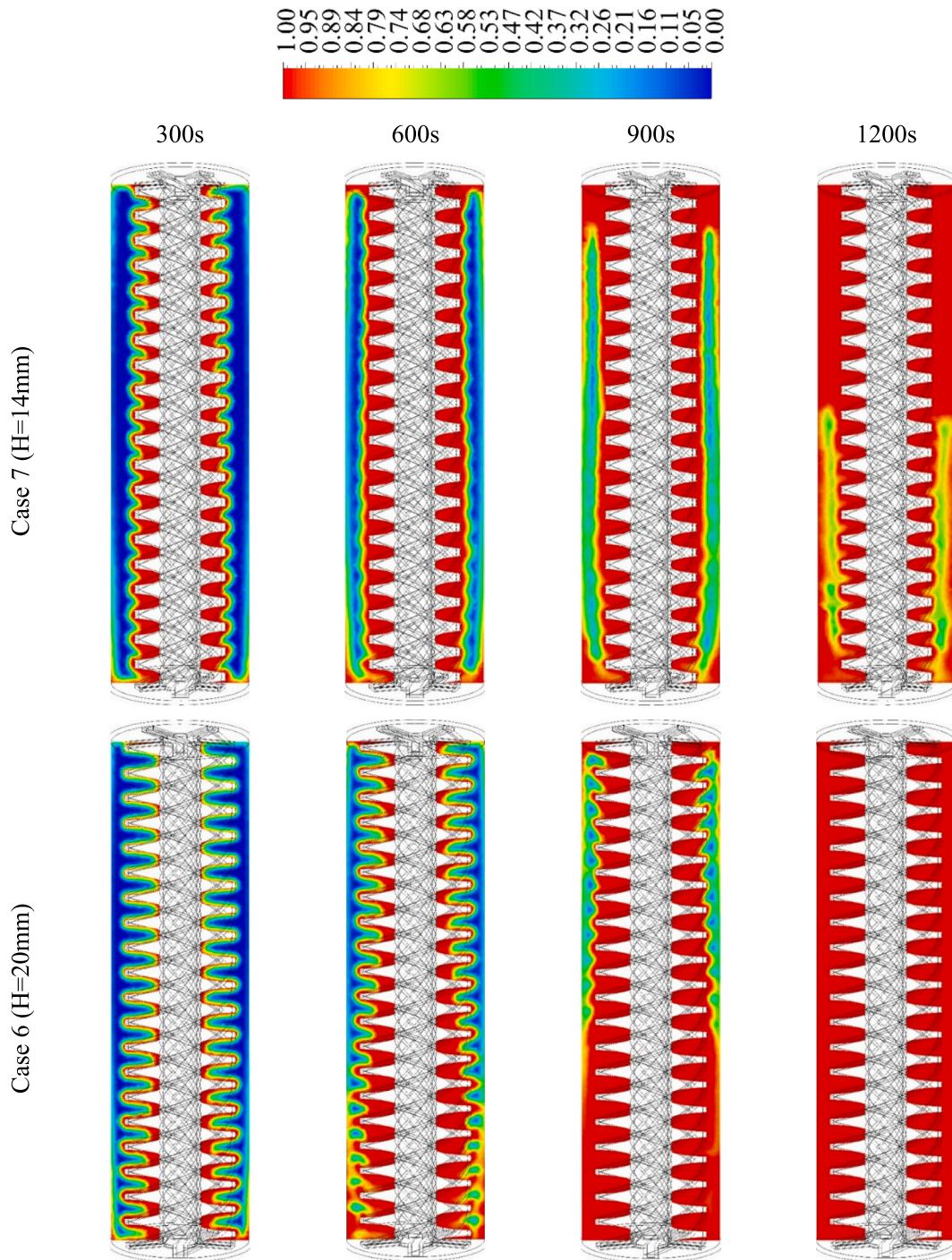


Fig. 13. Liquid fraction contours for cases 6 (20×4.4) and 7 (14×6.15) for the time steps up to 1200 s, number of voids is 5, number of revolutions is 4.

mm, successfully balances enhanced heat transfer with manageable flow resistance. For practical applications, Case 6's design offers superior energy storage capabilities, and engineers must assess whether the associated performance benefits justify any potential manufacturing complexities.

The superiority of $H = 20\text{ mm}$ over $H = 18\text{ mm}$ reflects stronger secondary flows and reduced conduction distance; however, the gain saturates as melting proceeds because the rate becomes constrained by mushy-zone resistance and by impingement of the front on remaining solid pockets. Further reducing H to 14 mm restricts channel width, weakening convection and lengthening the heat path, which explains the cooler pockets and slower propagation observed. In practice, larger

H may also increase hydraulic losses and impose structural-tolerance demands; thus, the optimal height balances faster cycling against pressure-drop and manufacturability constraints.

Since the Re of all previous cases was fixed at 2000, Table 7 investigates the influence of varying Re on the heat storage rate and melting time for Case 6, which exhibited the best performance among all configurations. The data reveal a strong positive correlation between Re and thermal performance. At $Re = 1000$, the heat storage rate is 98.55 W with a melting time of 1814 s , while increasing Re to 1500 improves these values to 122.96 W and 1480 s , respectively. The highest performance is achieved at $Re = 2000$, with a heat storage rate of 153.80 W and a melting time of 1185 s . This trend occurs because higher Re values

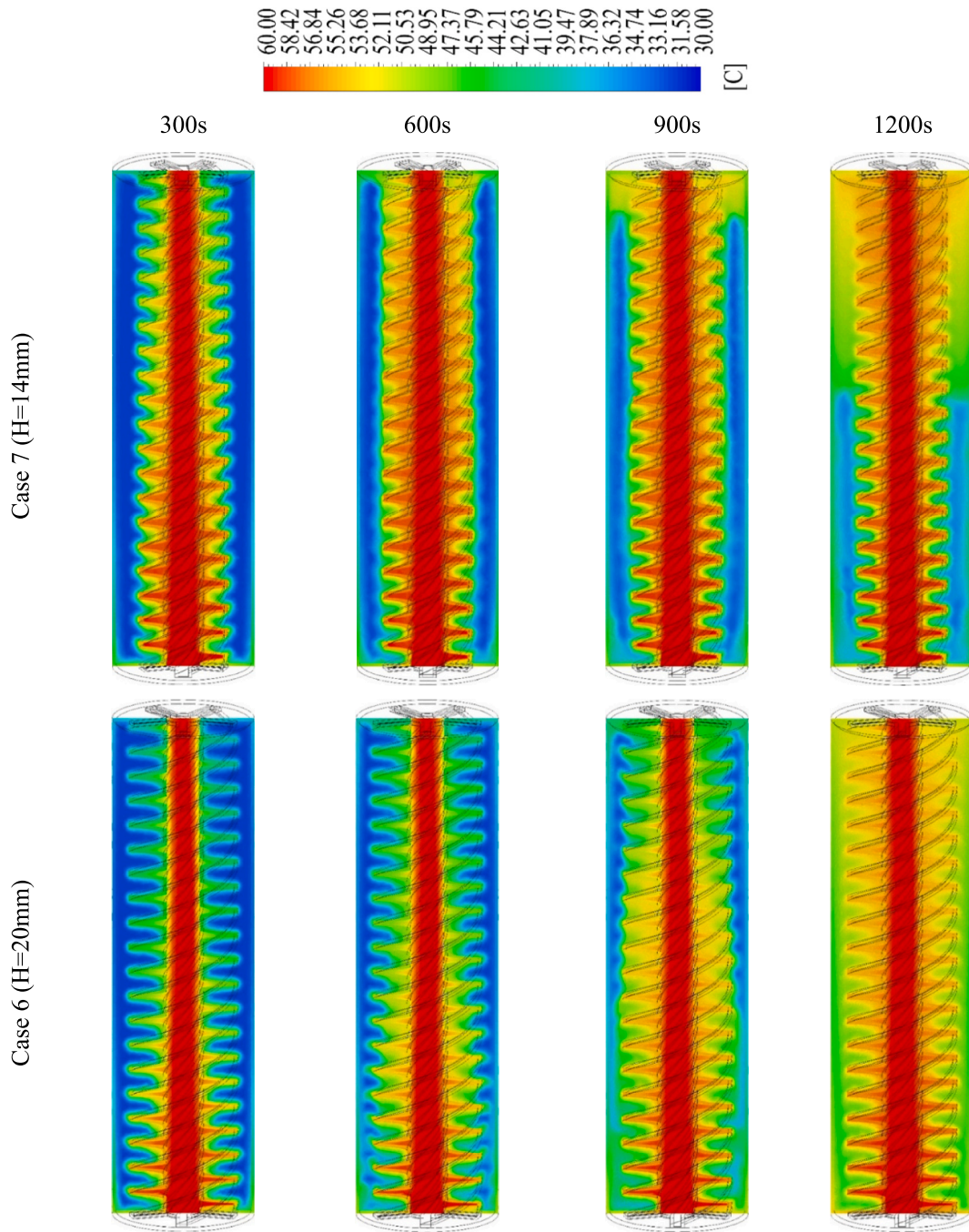


Fig. 14. Mean temperature contours for cases 6 (20 * 4.4) and 7 (14 * 6.15) for the time steps up to 1200 s, number of voids is 5, number of revolutions is 4.

Table 6 The charging period and the thermal storage rate for the cases 5, 6 and 7.				
	Heat storage rate (W)	Melting time (s)	Pressure drop (Pa)	Pumping power (W)
Case 5 ($H = 18$ mm)	125.37	1482	462.79	0.011116
Case 6 ($H = 20$ mm)	153.80	1185	462.72	0.011234
Case 7 ($H = 14$ mm)	120.90	1579	463.2	0.011104

enhance turbulent flow and convective heat transfer, leading to more efficient energy distribution and faster phase change in the PCM.

Table 7 The charging period and the thermal storage rate for the case 6 at different Re .		
	Heat storage rate (W)	Melting time (s)
$Re = 1000$	98.55	1814
$Re = 1500$	122.96	1480
$Re = 2000$	153.80	1185

The improvement in thermal performance with increasing Re can be attributed to the transition from laminar to turbulent flow regimes, which disrupt thermal boundary layers and promote better mixing. At $Re = 1000$, the flow is laminar or weakly turbulent, resulting in slower heat transfer and prolonged melting. As Re increases to 1500 and 2000, the flow becomes more turbulent, significantly boosting the heat storage

rate and reducing melting time. However, while higher Re values improve performance, they also entail greater pumping power requirements due to increased pressure drops. For practical applications, this trade-off must be carefully considered. The data suggest that operating at $Re = 2000$ offers the best thermal performance, but systems with energy constraints might benefit from a compromise at $Re = 1500$, which still provides substantial improvements over $Re = 1000$.

5.2. Solidification process analysis

5.2.1. Effect of void number on solidification performance

Fig. 15 illustrates the liquid fraction contours during the solidification process for Cases 2, 3, and 4, which feature 3, 4, and 5 voids, respectively, observed at intervals up to 2400 s. The results demonstrate that increasing the number of voids significantly accelerates the

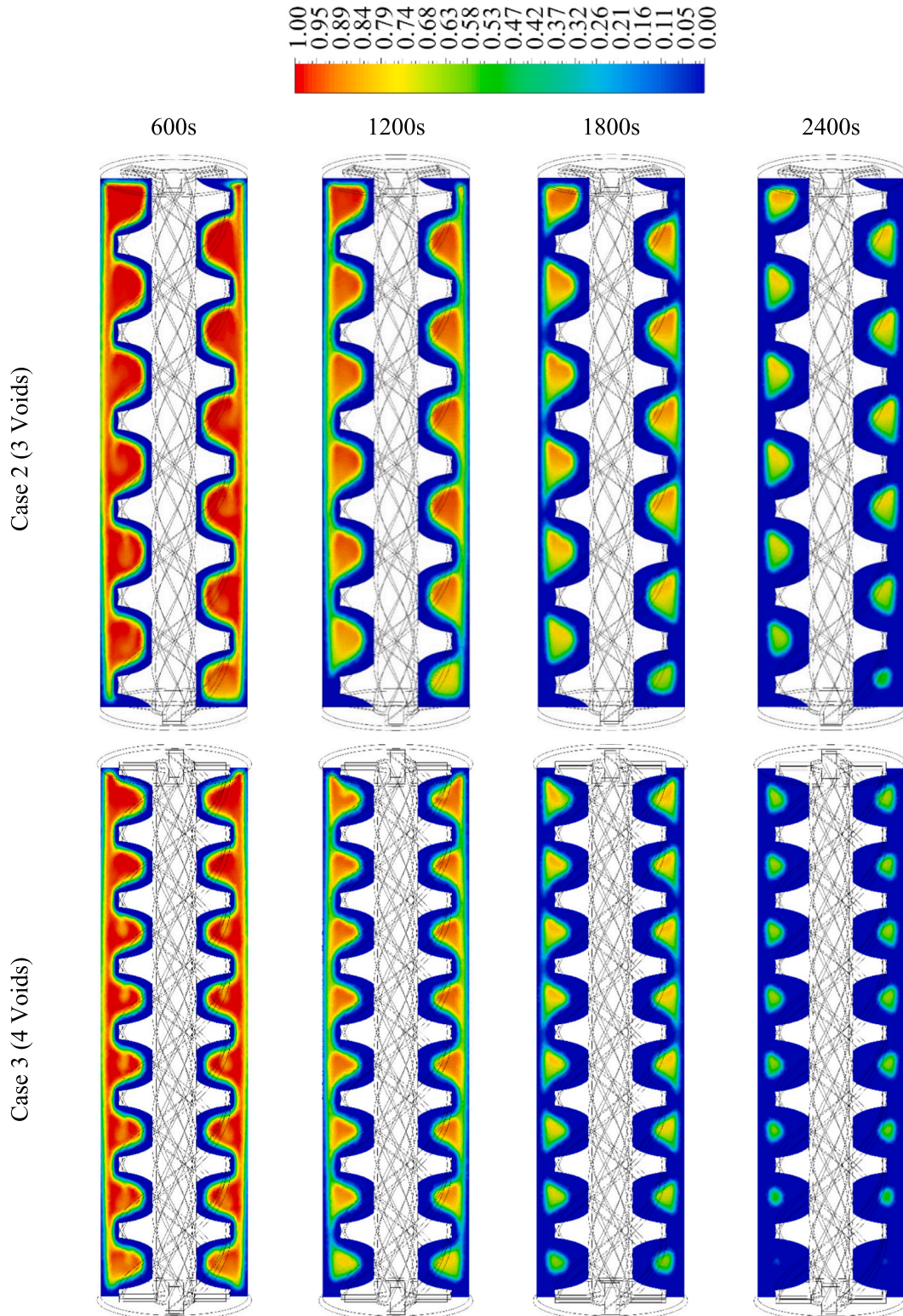


Fig. 15. Liquid fraction contours for cases 2, 3, and 4 for four various times up to 2400 s during the solidification process.

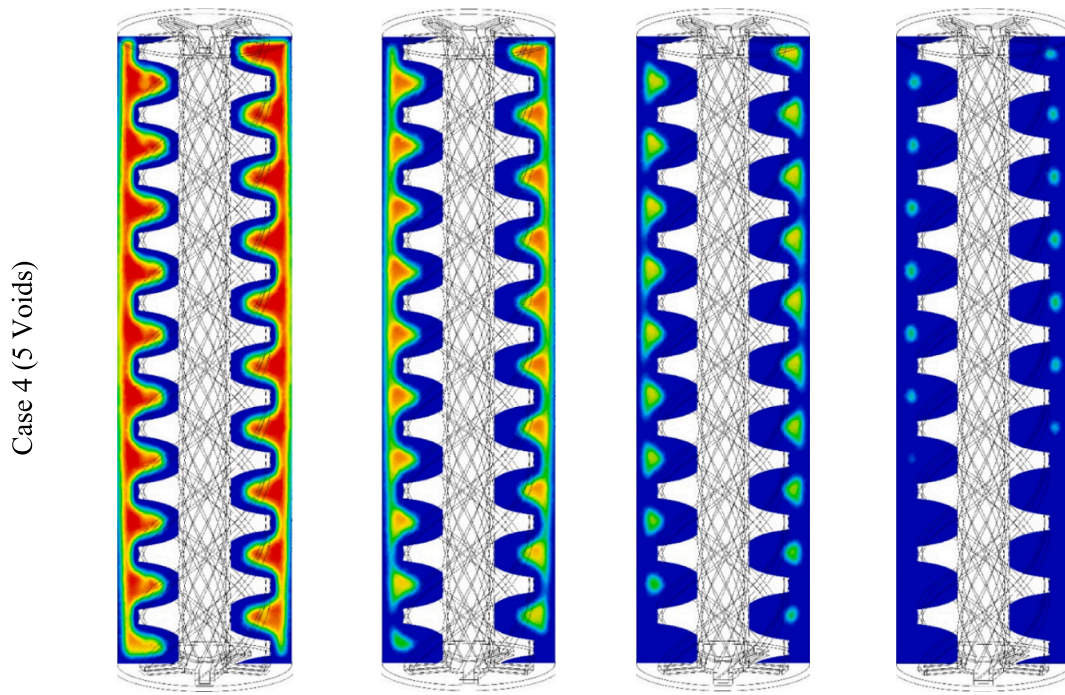


Fig. 15. (continued).

solidification process. Case 4 (5 voids) exhibits the most rapid reduction in liquid fraction, with larger solid regions appearing earlier compared to Cases 3 (4 voids) and 2 (3 voids). This enhancement occurs because additional voids improve heat extraction by increasing the contact area between the PCM and the cooling surface, while also promoting more uniform temperature distribution through enhanced convective cooling. The voids act as channels for faster heat dissipation, preventing localized heat accumulation and ensuring a more homogeneous phase transition. By 2400 s, Case 4 shows near-complete solidification, whereas Cases 2 and 3 still retain isolated liquid pockets, underscoring the critical role of void quantity in optimizing solidification efficiency. During discharge, the dominant pathway is heat extraction from the liquid PCM to the cooled wall. Additional voids (3 $\rightarrow$ 4 $\rightarrow$ 5) (i) shorten the conduction path from the interior to the wall, (ii) increase interfacial area available for heat removal, and (iii) guide convective currents within the liquid PCM so that thermal stratification is disrupted and cold fluid reaches previously stagnant pockets. These effects explain the earlier appearance and faster growth of solid regions with higher void counts.

The solidification behavior across the three cases reveals a nonlinear relationship between void count and performance gains. While Case 4 (5 voids) outperforms the others, the marginal improvement over Case 3 (4 voids) is less pronounced than the difference between Cases 2 and 3. This suggests diminishing returns beyond a certain void density, where further additions complicate manufacturing without proportional thermal benefits. The faster solidification in void-rich designs is attributed to disrupted thermal gradients and intensified heat transfer, but practical trade-offs (such as mechanical stability or material costs) must be considered. For applications prioritizing rapid energy discharge (e.g., thermal storage systems), higher void numbers (Case 4) are advantageous, but designs like Case 3 offer a balanced compromise.

The improvement from 3 $\rightarrow$ 4 voids is larger than from 4 $\rightarrow$ 5 because the benefits of reduced heat-path length and better-organized extraction channels begin to saturate as the liquid layer thins; the late stage becomes conduction-limited through the solid/mushy framework, so case differences narrow. At very high void density, the active PCM volume decreases and local cold spots may not couple efficiently to the bulk, yielding diminishing returns. In practice, higher void counts can also increase flow resistance on the coolant side in a fully conjugate system;

thus, designs like 4 voids offer a balanced compromise between discharge speed and practicality.

The number of voids in the heat exchanger significantly influences the temperature distribution of the PCM during the solidification process. As shown in Fig. 16, configurations with a higher number of voids, such as Case 4 (5 voids), exhibit more uniform and faster cooling compared to designs with fewer voids, like Case 2 (3 voids). This occurs because additional voids increase the surface area available for heat transfer, allowing for more efficient dissipation of thermal energy from the PCM to the cooling medium. The voids act as conduits for heat extraction, disrupting thermal stratification and promoting convective cooling, which leads to a more homogeneous temperature distribution. In Case 4, the temperature gradients are minimized, and the PCM solidifies more evenly, whereas Case 2 displays localized regions of slower cooling due to less effective heat removal. The enhanced thermal uniformity in void-rich designs underscores their potential for applications requiring consistent and rapid energy discharge.

The relationship between void count and cooling efficiency is nonlinear, with diminishing returns observed as the number of voids increases. While Case 4 (5 voids) demonstrates superior performance, the improvement over Case 3 (4 voids) is less pronounced than the difference between Cases 2 and 3. This suggests that beyond a certain threshold, additional voids not proportionally enhance solidification rates, possibly due to limitations in heat transfer dynamics or increased flow resistance. For practical applications, this implies that an optimal void count exists, balancing performance gains with manufacturing and operational constraints. Designs like Case 3 offer a viable compromise, achieving near-optimal cooling without excessive complexity.

Fig. 17 compares Cases 2 (3 voids), 3 (4 voids), and 4 (5 voids) during PCM solidification, tracking liquid fraction (Fig. 17a), mean PCM temperature (Fig. 17b) and inlet-outlet temperature difference of the HTF (Fig. 17c). The data reveal a clear trend: increasing void count accelerates solidification and improves thermal uniformity. For instance, Case 4 (5 voids) achieves 90 % solidification by 1800 s, while Cases 3 and 2 require 2100 s and 2700 s, respectively, for the same progress. Similarly, the temperature Fig. 17(b) shows Case 4 cooling to 17 °C by 2400 s, whereas Cases 3 and 2 remain at 22 °C and 25 °C at the same time. This enhancement occurs because additional voids expand the

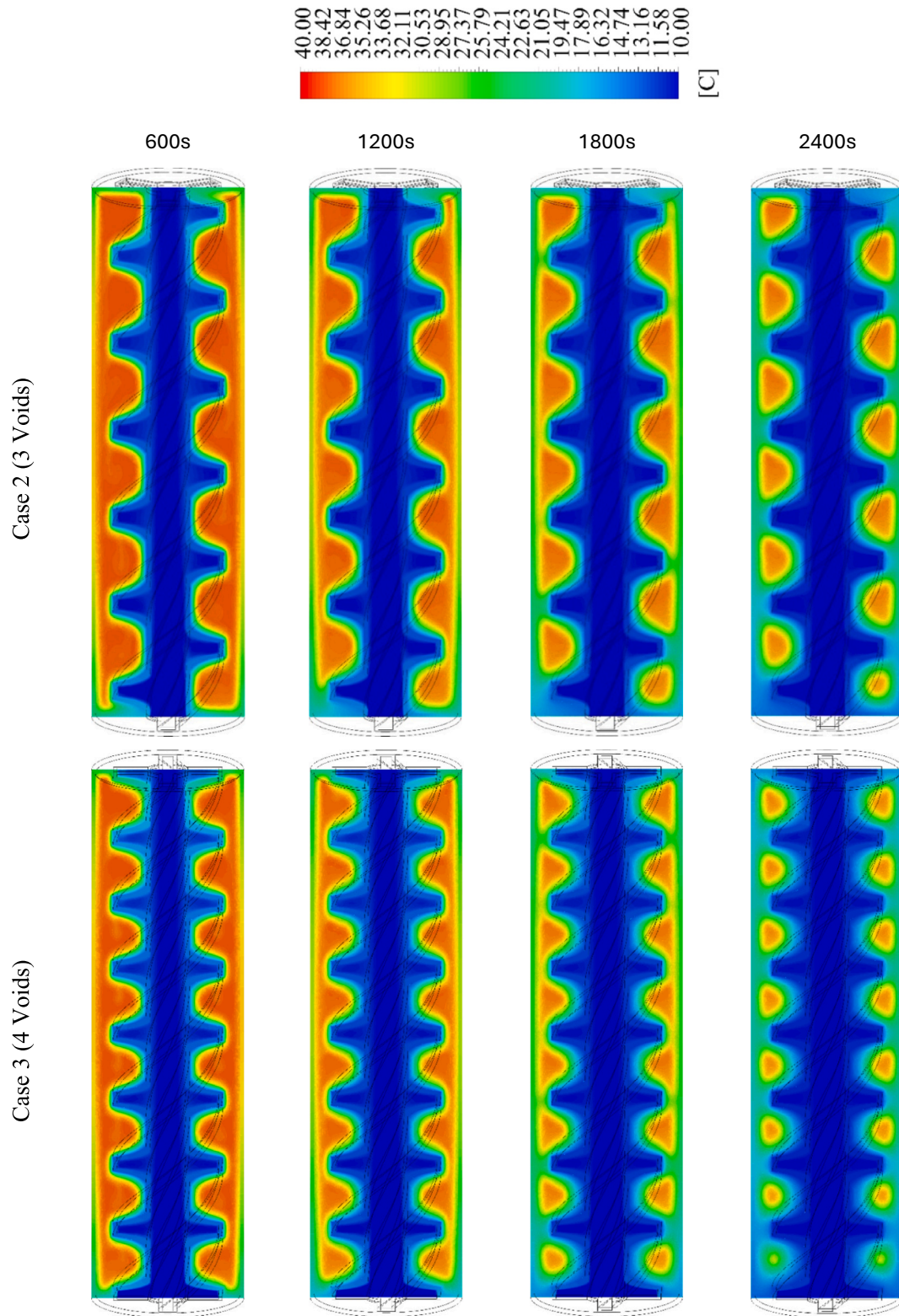


Fig. 16. The temperature contours for cases 2, 3, and 4 for four various times up to 2400 s during the solidification process.

heat transfer surface area and intensify convective cooling through two mechanisms: (1) disrupted thermal stratification from void-induced flow turbulence, and (2) reduced thermal resistance via shorter heat conduction paths through the PCM. The taller/larger voids in higher-count cases (e.g., Case 4's 5 voids) particularly amplify these effects by creating more pronounced flow channels for coolant interaction.

While Case 4's performance is superior, the marginal gains over Case 3 diminish beyond 4 voids, suggesting an optimal void density. For example, the 5-void design (Case 4) solidifies only 14 % faster than the 4-void (Case 3), compared to the ~30 % improvement from Case 2 (3 voids) to Case 3. This nonlinearity stems from flow resistance limits and overlapping thermal influence zones between densely packed voids.

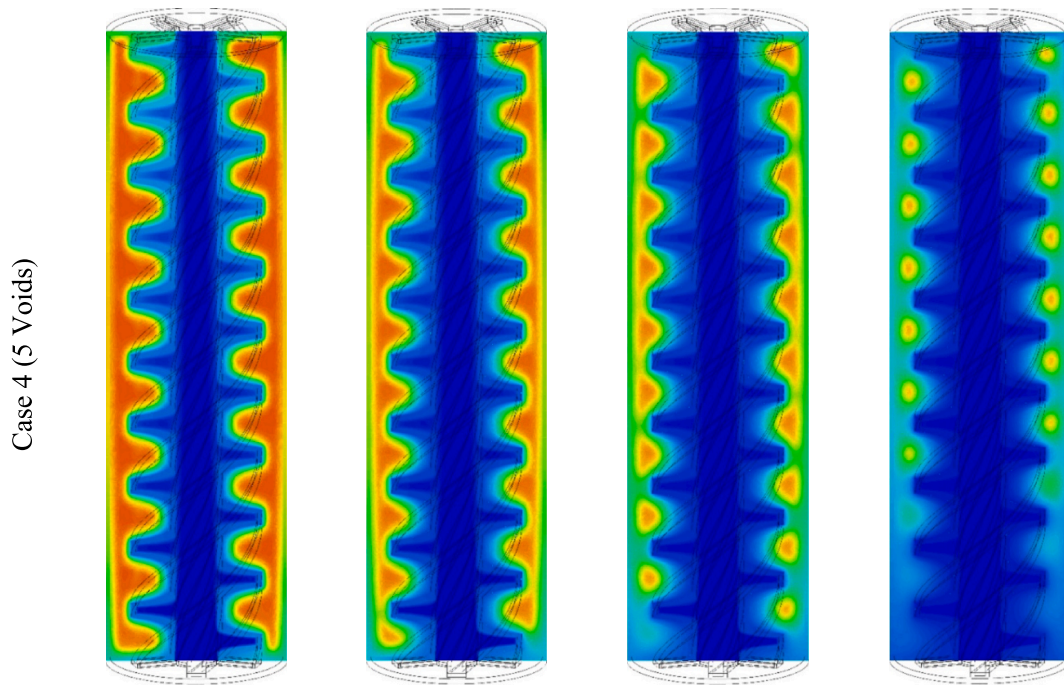


Fig. 16. (continued).

Practically, Case 3 offer the best balance, as its near-optimal performance (e.g., ~85 % solidification by 1800 s) could justify simpler manufacturing versus Case 4.

The inlet–outlet temperature difference of the HTF, shown in Fig. 17c, shows a sharp initial peak, followed by a gradual decrease as time progresses similar to this diagram during melting. The 3-void case maintains the lowest ΔT throughout the process, indicating weaker heat transfer between the HTF and PCM, while the 4- and 5-void configurations exhibit higher ΔT values due to their enhanced heat transfer area and stronger conduction paths through the voids. At later times, the ΔT curves converge as the PCM approaches full solidification, with the 5-void design reaching a low ΔT (and thus near-thermal equilibrium) earlier than the other cases, consistent with its shorter solidification time.

The data in Table 8 demonstrate a significant improvement in thermal recovery rates and solidification times as the number of voids increases from 3 to 5. Case 2 (3 voids) exhibits the lowest heat recovery rate (46.17 W) and longest solidification time (4218 s), while Case 4 (5 voids) achieves the highest recovery rate (72.45 W) and shortest solidification time (2656 s). Case 3 (4 voids) represents an intermediate performance level, with a recovery rate of 59.94 W and solidification time of 3229 s. These trends can be attributed to the enhanced heat transfer mechanisms enabled by additional voids. More voids increase the effective surface area for heat dissipation and promote stronger convective cooling through improved fluid flow dynamics. The voids disrupt thermal stratification and create additional pathways for heat extraction, leading to faster and more uniform solidification. The 5-void configuration (Case 4) outperforms the others by leveraging these effects most effectively, resulting in a 56.9 % higher recovery rate and 37.0 % faster solidification compared to the 3-void design (Case 2).

The relationship between void count and solidification performance follows a nonlinear pattern, with diminishing returns observed as the number of voids increases. While the transition from 3 to 4 voids yields a 29.8 % improvement in recovery rate and a 23.4 % reduction in solidification time, the step from 4 to 5 voids offers smaller gains of 20.9 % and 17.8 %, respectively. This suggests that beyond a certain point, additional voids provide limited thermal benefits, possibly due to flow resistance or overlapping thermal influence zones. From a practical

standpoint, the 4-void design (Case 3) represent an optimal balance, offering substantial performance improvements over the 3-void configuration without the added complexity of the 5-void design. Engineers must weigh these thermal advantages against potential trade-offs, such as increased manufacturing costs or pressure drop, when selecting a void configuration for real-world applications.

The observed nonlinear relationship and diminishing returns are primarily due to the saturation of heat extraction pathways rather than a prohibitive increase in hydraulic losses. Although not tabulated here for the solidification phase, the pressure drop trend is consistent with the charging analysis, showing only minor variations with void count. Therefore, the optimal void number is determined by the point at which the incremental reduction in conduction path length and improvement in convective organization no longer yield significant thermal gains, without being limited by flow resistance.

5.2.2. Effect of helical revolutions on solidification performance

Fig. 18 compares the solidification performance of Case 4 (2 helical revolutions) and Case 5 (4 revolutions) through liquid fraction (Fig. 18(a)) and temperature contours (Fig. 18(b)) at 1200 s and 2400 s. The results demonstrate that increasing the number of revolutions from 2 to 4 significantly enhances the solidification process. Case 5 exhibits faster and more uniform phase change, with larger solidified regions visible at 1200 s compared to Case 4. By 2400 s, Case 5 achieves near-complete solidification, while Case 4 still shows residual liquid PCM pockets. This improvement is attributed to the intensified convective cooling in Case 5, where additional revolutions create longer flow paths and stronger secondary flows (Dean vortices). These vortices disrupt thermal stratification and improve heat extraction efficiency, leading to more homogeneous temperature distribution (Fig. 18(b)) and accelerated solidification. The temperature contours further reveal that Case 5 maintains lower and more uniform temperatures, confirming its superior thermal performance. Additional helical turns strengthen curvature-induced secondary flows (Dean-type vortices) and increase the tortuosity/residence time of the coolant-cooled liquid PCM adjacent to the wall. These effects suppress thermal stratification, thin the thermal boundary layer, and maintain a larger wall-normal temperature gradient, which explains the earlier appearance and faster growth of

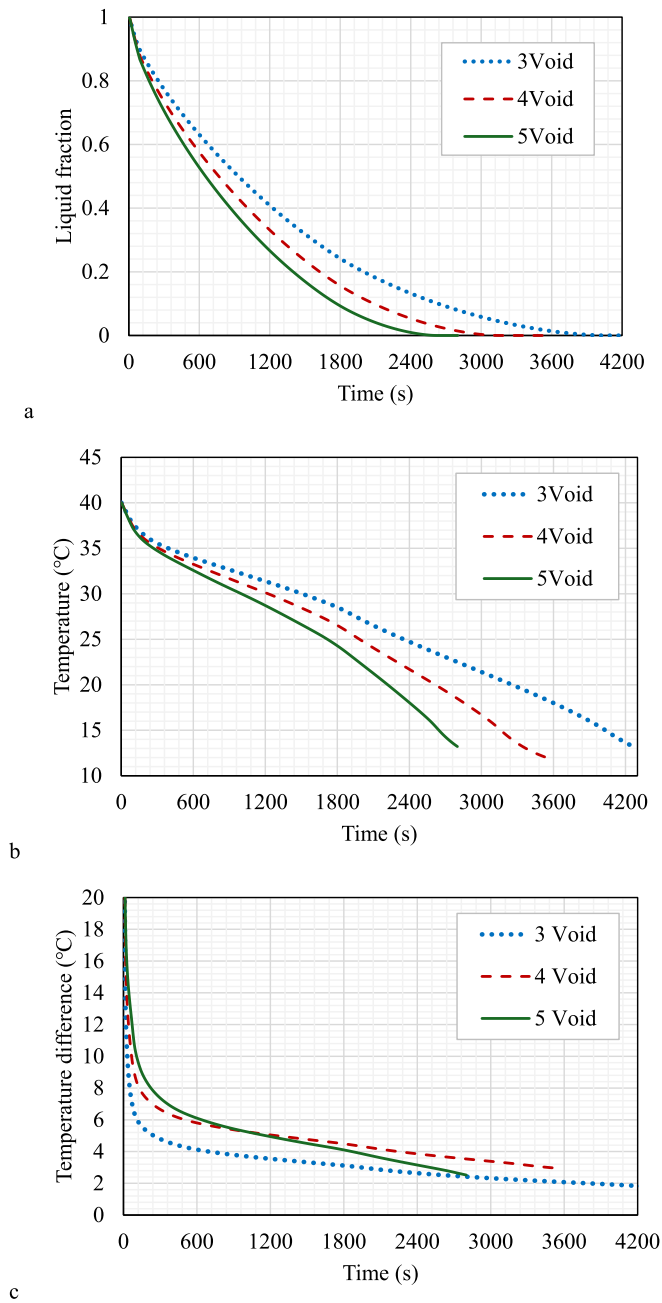


Fig. 17. Comparison of the cases 2, 3, and 4 regarding a) liquid fraction, and b) temperature during the solidification process.

Table 8

The discharging period and the thermal recovery rate for the case 2, 3, and 4 during the solidification process.

	Heat recovery rate (W)	Solidification time (s)
3Void	46.17	4218
4Void	59.94	3229
5Void	72.45	2656

solid regions for the 4-revolution case.

The enhanced solidification in Case 5 underscores the critical role of helical geometry in optimizing heat exchanger designs for thermal energy storage. The 4-revolution configuration's superior performance stems from its ability to maximize fluid-PCM interaction time and turbulence, which are essential for efficient heat dissipation. However, the

trade-off between thermal performance and practical constraints must be considered. While Case 5 offers faster solidification, its higher flow resistance and potential manufacturing complexity limit its applicability in certain systems. For scenarios requiring rapid energy discharge, such as high-demand thermal storage, the 4-revolution design is advantageous. Conversely, the 2-revolution design (Case 4) suffice for applications where moderate performance and simpler implementation are prioritized.

Fig. 19 compares the solidification performance of Case 4 (2 helical revolutions) and Case 5 (4 revolutions) by tracking liquid fraction (left axis) and mean temperature (right axis) over time. The data reveal that Case 5 achieves significantly faster solidification, reaching ~80 % liquid fraction reduction by 1300 s, while Case 4 requires 2400 s to attain the same level. This acceleration correlates with a steeper decline in mean temperature for Case 5, which cools to 22 °C by 1800 s compared to Case 4's 24 °C at the same time. The performance gap stems from enhanced convective heat transfer in Case 5, where additional revolutions amplify secondary flows (Dean vortices), improving thermal uniformity and heat extraction rates. However, the diminishing temperature difference observed beyond 2000 s suggests that the 4-revolution design's advantage is most pronounced during early to mid-stage solidification. For applications prioritizing rapid energy discharge, Case 5 is superior, but its higher flow resistance warrant trade-off evaluations against Case 4's simpler geometry in practical implementations.

The faster decay of liquid fraction for 4 turns is consistent with stronger secondary flows and more uniform near-wall cooling which reduce residual liquid pockets late in the cycle. The data in Table 8 demonstrate that increasing the number of helical revolutions from 2 (Case 4) to 4 (Case 5) enhances the thermal recovery rate by 11.9 % (from 72.45 W to 81.05 W) while reducing the solidification time by 11.6 % (from 2656 s to 2348 s). This improvement occurs because additional revolutions create a longer and more tortuous flow path for the cooling fluid, which intensifies convective heat transfer through two primary mechanisms: (1) extended residence time of the coolant within the heat exchanger, allowing for more complete heat extraction from the PCM, and (2) stronger secondary flows (Dean vortices) that disrupt thermal boundary layers and promote uniform cooling. The higher recovery rate and faster solidification in Case 5 confirm that helical geometry plays a critical role in optimizing thermal energy discharge, particularly in systems requiring rapid heat rejection.

The data in Table 9 demonstrate that increasing the number of helical revolutions from 2 (Case 4) to 4 (Case 5) enhances the thermal recovery rate by 11.9 %, confirming that this modification improves performance when all other geometric parameters, specifically the void height ($H = 18$ mm), are held constant. It is important to note that the subsequent analysis in Section 5.2.3 reveals that void height (H) can have an even more significant influence on solidification performance, as seen when comparing cases with different H values in Table 9.

While Case 5's performance is superior, the marginal gains (12 % improvement in both metrics) must be weighed against potential drawbacks. The 4-revolution design increases flow resistance, which could elevate pumping power requirements and operational costs. For applications where energy efficiency is paramount, such as industrial thermal storage systems, the enhanced recovery rate of Case 5 justifies these trade-offs. Conversely, in cost-sensitive or space-constrained applications, the simpler 2-revolution design (Case 4) might offer a more balanced solution, as it achieves ~90 % of Case 5's performance with potentially lower manufacturing and operational complexity.

The benefit of four turns is most pronounced during early-mid solidification when convective extraction governs the rate. As solid layers thicken, heat removal becomes increasingly conduction/mushy-zone limited, so the curves converge and incremental gains shrink. In practical systems, extra turns also imply higher pressure drop and pumping power on the HTF side; therefore, the optimal revolution count balances cycle-time reduction against hydraulic penalties and manufacturability.

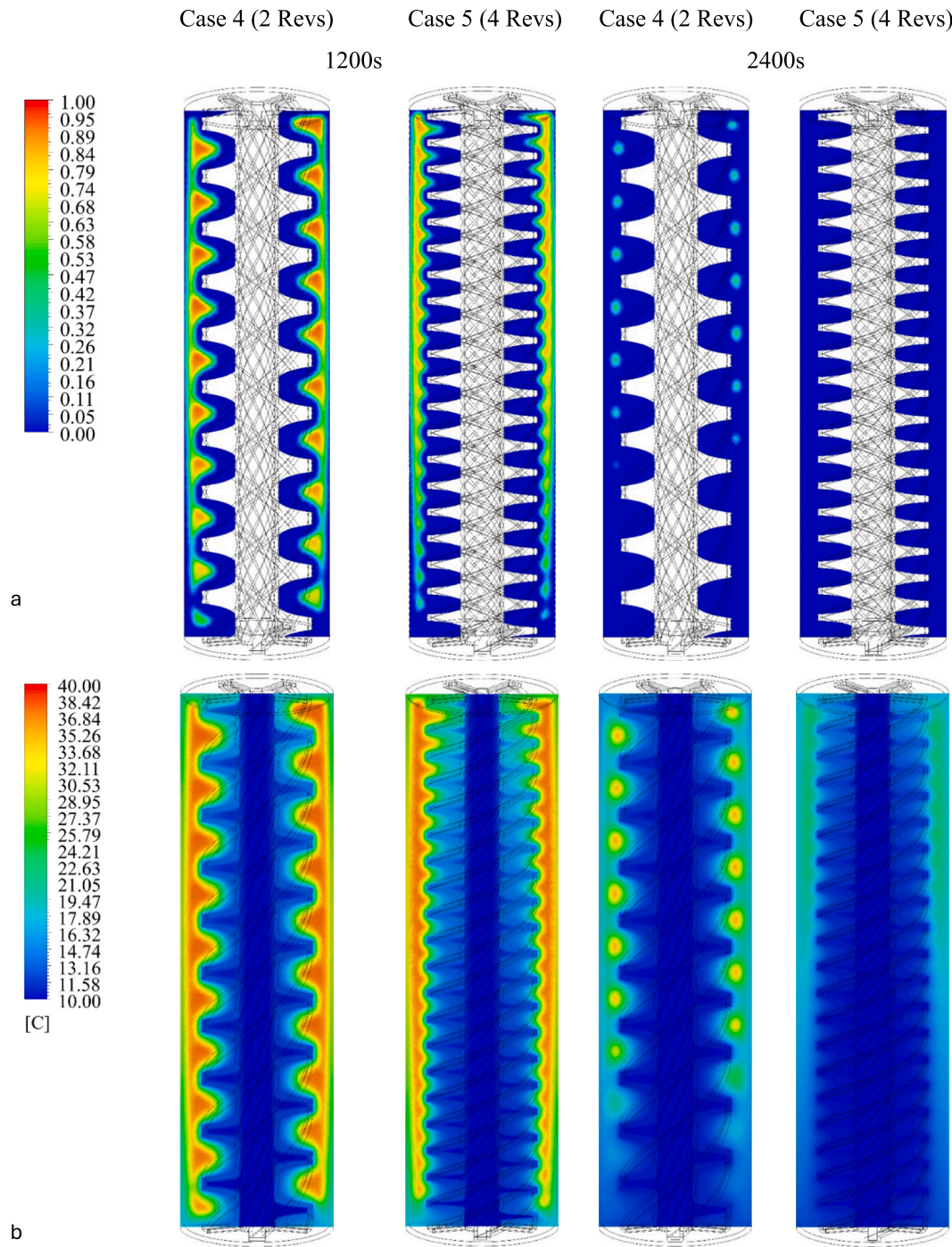


Fig. 18. a) Liquid fraction and b) temperature contours for cases 4 (2 Revolutions of helix) and 5 (4 Revolutions of helix) for the time steps 1200 and 2400 s during the solidification process.

5.2.3. Influence of void height on solidification performance

Fig. 20 compares the solidification process of Cases 6 (20 mm void height) and 7 (14 mm void height) through liquid fraction contours at intervals up to 2400 s, while maintaining identical void counts (5) and helical revolutions (4). The results demonstrate that taller voids (Case 6) significantly accelerate PCM solidification compared to shorter voids (Case 7). By 1200 s, Case 6 exhibits substantially larger solidified regions, while Case 7 retains more liquid PCM. This performance gap widens at 1800 s and 2400 s, with Case 6 achieving near-complete solidification earlier. The enhanced performance of Case 6 stems from three key mechanisms: (1) increased void height expands the heat transfer surface area between the PCM and cooling fluid, (2) taller voids

create larger flow channels that promote stronger convective currents and better coolant penetration, and (3) the greater void volume reduces thermal resistance by shortening heat conduction paths through the PCM. These factors collectively enable more efficient heat extraction from the PCM in Case 6.

The superior solidification performance of Case 6 highlights void height as a critical geometric parameter for thermal energy storage systems. The 42.9 % increase in void height (from 14 mm to 20 mm) leads to markedly faster phase change, suggesting that void height influences solidification rates more significantly than either void count or helical revolutions in this configuration. However, practical implementation requires balancing these thermal advantages against

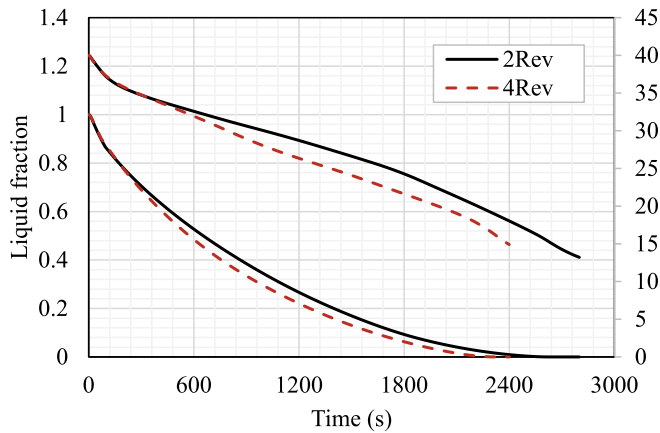


Fig. 19. Comparison of the cases 4 and 5 regarding liquid fraction (y-axis at the left), and mean temperature (y-axis at the right) during the solidification process.

Table 9

The discharging period and the thermal recovery rate for the cases 4 and 5.

	Heat recovery rate (W)	Solidification time (s)
Case 4 (2 Revolutions of helix)	72.45	2656
Case 5 (4 Revolutions of helix)	81.05	2348

potential drawbacks. Taller voids compromise structural integrity or increase pressure drop due to greater flow disruption. Case 7's slower performance (14 mm voids) demonstrates that insufficient void height can lead to thermal bottlenecks, where heat extraction becomes limited by reduced convection and longer conduction paths. For applications prioritizing rapid energy discharge, Case 6's design is optimal, but engineers must verify that the taller voids maintain mechanical stability under operational stresses.

Fig. 21 compares temperature contours for Cases 6 (20 mm voids) and 7 (14 mm voids) during solidification up to 2400 s, revealing that taller voids (Case 6) achieve both lower temperatures and more uniform cooling. By 1200 s, Case 6 exhibits temperatures $\sim 5\text{--}7\text{ }^{\circ}\text{C}$ cooler than Case 7 in the PCM core, with this gap widening to $\sim 10\text{ }^{\circ}\text{C}$ by 2400 s. This thermal advantage stems from the taller voids enhanced convective heat transfer: (1) the 20 mm height increases fluid-PCM contact area by $\sim 43\%$ over 14 mm voids, (2) larger flow channels reduce flow resistance, enabling stronger coolant circulation, and (3) the expanded void volume shortens heat conduction paths through the PCM. While Case 6's design optimizes solidification, its taller voids require structural reinforcement to mitigate potential deformation under thermal stresses, a critical consideration for real-world applications balancing performance with durability.

Fig. 22 compares the liquid fraction evolution during solidification for Cases 5 (18 mm void height), 6 (20 mm), and 7 (14 mm), demonstrating that taller voids significantly enhance phase change rates. Case 6 (20 mm) achieves the fastest solidification, reaching $\sim 90\%$ liquid fraction reduction by 1500s, while Case 5 (18 mm) and Case 7 (14 mm) require 1700 s and 2200 s, respectively, for equivalent progress. This 16.7 % performance improvement (Case 6 vs. Case 5) arises from the taller voids' dual advantage: (1) expanded heat transfer surface area (11 % larger than Case 5) and (2) reduced thermal resistance from shorter PCM conduction paths. Case 7's delayed solidification highlights the limitations of insufficient void height, where restricted flow channels impair convective cooling. The results suggest void height has a nonlinear effect, performance gains diminish beyond 20 mm, as seen in the marginal difference between Cases 5 and 6 compared to the significant gap with Case 7. For design optimization, a void height of 18–20 mm appears optimal, balancing thermal performance with structural

practicality.

The data in Table 10 reveal a counterintuitive yet significant trend in thermal recovery performance relative to void height. While Case 6 (20 mm voids) demonstrates expected improvements over Case 5 (18 mm), with a 28.7 % higher heat recovery rate (81.05 W vs. 62.97 W) and 21.2 % faster solidification (2359 s vs. 2993 s), Case 7 (14 mm voids) unexpectedly outperforms both with a remarkable 94.60 W recovery rate and 2037 s solidification time. This anomaly suggests that smaller voids (14 mm) enhance heat transfer through intensified flow velocities and greater turbulence generation within the restricted channels, despite their reduced surface area. The 14 mm voids create higher shear rates and stronger secondary flows, which disrupt thermal boundary layers more effectively than taller voids. However, this superior performance must be balanced against potential drawbacks like increased pressure drop and pumping power requirements in the more constricted flow paths of Case 7.

The nonlinear relationship between void height and thermal recovery rate challenges conventional assumptions about heat transfer optimization. Case 7's exceptional performance with smaller voids (14 mm) indicates that flow dynamics dominate over surface area effects in certain configurations. The 50.3 % improvement in recovery rate from Case 5 to Case 7, coupled with a 31.9 % reduction in solidification time, suggests that void height optimization exists in a Goldilocks zone, where neither too large (20 mm) nor too small (14 mm) voids deliver ideal performance, but an intermediate height (between 14 and 18 mm) offer the best compromise. For practical applications, Case 7's design, while thermally superior, face challenges in maintaining structural integrity and managing pressure losses. Engineers must conduct coupled thermo-mechanical analyses to determine whether the performance gains justify these trade-offs, particularly in systems where energy efficiency outweighs operational costs.

Despite the larger exchange area of taller voids, the 14 mm configuration (Case 7) attains the highest heat-recovery rate (94.6 W) and shortest solidification time (2037 s) (Table 10) because the narrower flow passages accelerate the coolant and raise local shear rates, which strengthens secondary (Dean) vortices and erodes thermal boundary layers at the HTF wall. This shear-enhanced mixing promotes more uniform and deeper penetration of the cold front into the PCM than in the 18–20 mm cases, where wider channels lower velocities and reduce boundary-layer disruption, so the benefit of added surface area is partly offset. Consequently, void-height effects during discharge reflect a competition between area and shear-driven mixing: at 20 mm, increased area improves performance (81.05 W, 2359 s), but the shear advantage at 14 mm dominates, yielding the best overall recovery. The result is consistent with the mid-range optimum suggested by Fig. 22, where $H = 14\text{--}18\text{ mm}$ provides the steepest decay in liquid fraction while avoiding late-stage stagnation. Practically, the 14 mm geometry may also incur a higher pressure drop and pumping power, so the recommended design window is $H = 14\text{--}18\text{ mm}$, to balance recovery rate with hydraulic penalties. Future work should couple these thermal metrics with measured Δp and HTF inlet-outlet $\Delta T(t)$ to complete the trade-off map for sizing.

The superior solidification performance of Case 7 ($H = 14\text{ mm}$) is attributed to a mechanism that prioritizes convective intensity over surface area during discharge. While taller voids increase the heat transfer area, they permit broader, more stratified natural convection, resulting in a thicker thermal boundary layer at the cooled wall. In contrast, the narrower flow passages of Case 7 accelerate the liquid PCM and generate stronger shear forces, which more effectively erode the thermal boundary layer and maintain a steeper temperature gradient. This shear-enhanced mixing proves more decisive for heat extraction than a larger surface area, making the flow-constricting geometry of Case 7 optimal for the solidification process.

5.2.4. Influence of Reynolds number on solidification dynamics

Fig. 23 compares the solidification dynamics of Case 6 (20 mm voids)

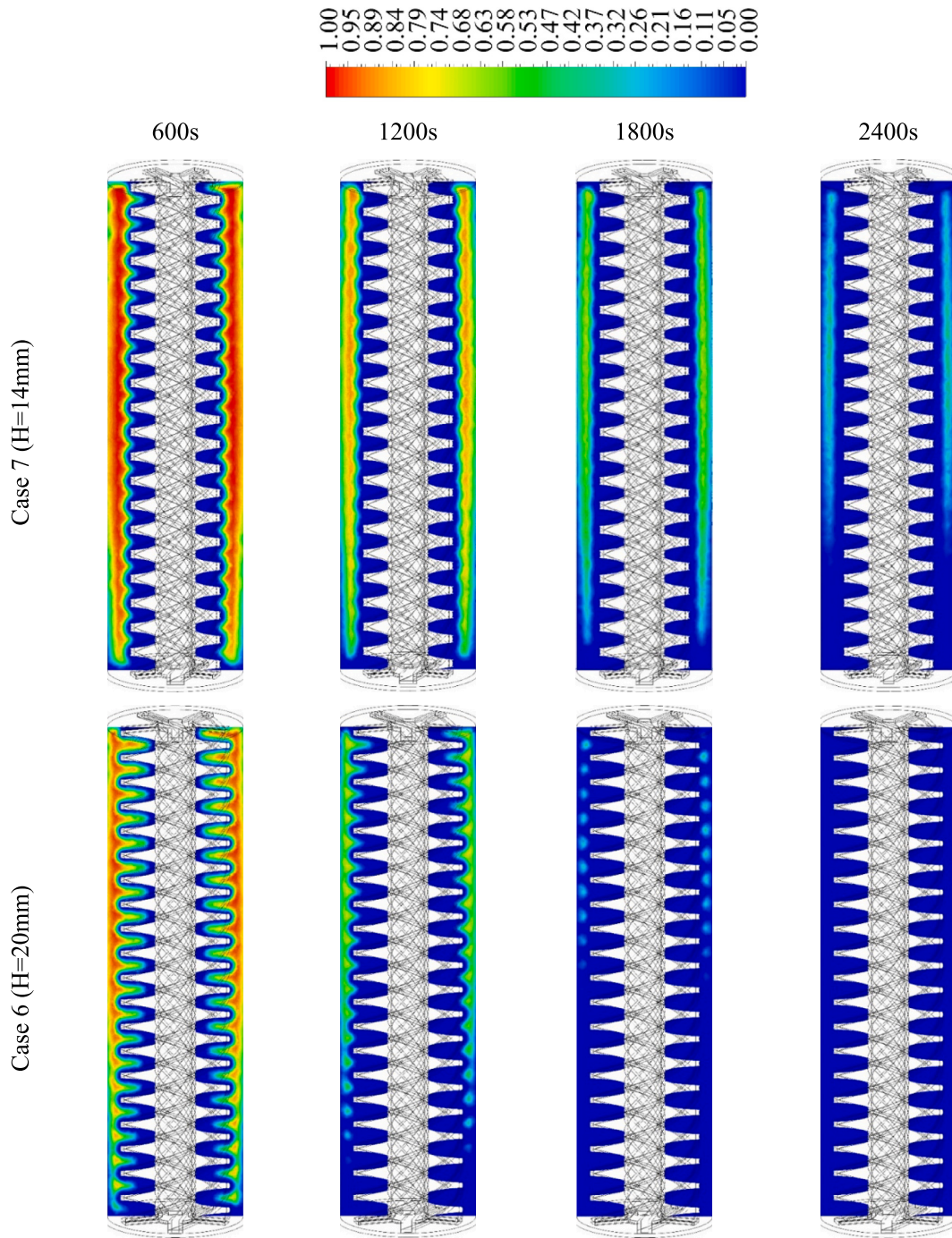


Fig. 20. Liquid fraction contours for cases 6 (20×4.4) and 7 (14×6.15) for the time steps up to 2400 s, number of voids is 5, number of revolutions is 4.

across varying Re by tracking liquid fraction (Fig. 23(a)) and temperature (Fig. 23(b)). The results demonstrate a strong positive correlation between Re and solidification efficiency higher Re values (e.g., $Re = 2000$) accelerate PCM and improve thermal uniformity compared to lower Re (e.g., $Re = 1000$). For instance, at $Re = 2000$, the PCM achieves 80 % solidification by 1000 s with a steeper temperature decline (26°C), while $Re = 1000$ exhibits delayed solidification (80 % at 1100 s) and higher residual temperatures (26.3°C). This enhancement stems from intensified turbulent mixing at elevated Re , which disrupts thermal stratification and enhances convective heat transfer. However, the performance gains diminish marginally beyond $Re = 1500$, suggesting a threshold where further increases in flow velocity yield limited thermal benefits while disproportionately raising pumping power requirements.

Note that HTF-side pressure drop and pumping power are not computed in the present model (isothermal inner wall; HTF not resolved); any mention of these penalties is qualitative to indicate expected trade-offs, while the reported metrics are PCM-side thermal only. The figure underscores the need to balance Re selection with energy efficiency in real-world systems, prioritizing values (e.g., $Re = 1500$ – 2000) that optimize solidification rates without excessive operational costs.

The trends in Fig. 23 arise from classical convection–conduction competition in PCM solidification. Stronger core flow generates tighter Dean vortices around the twisted inner tube, which (i) thin the thermal boundary layer at the PCM interface, (ii) sweep cold fluid into the near-wall region, and (iii) disrupt thermal stratification. These effects raise the interfacial heat-flux and the local Nusselt number, accelerating

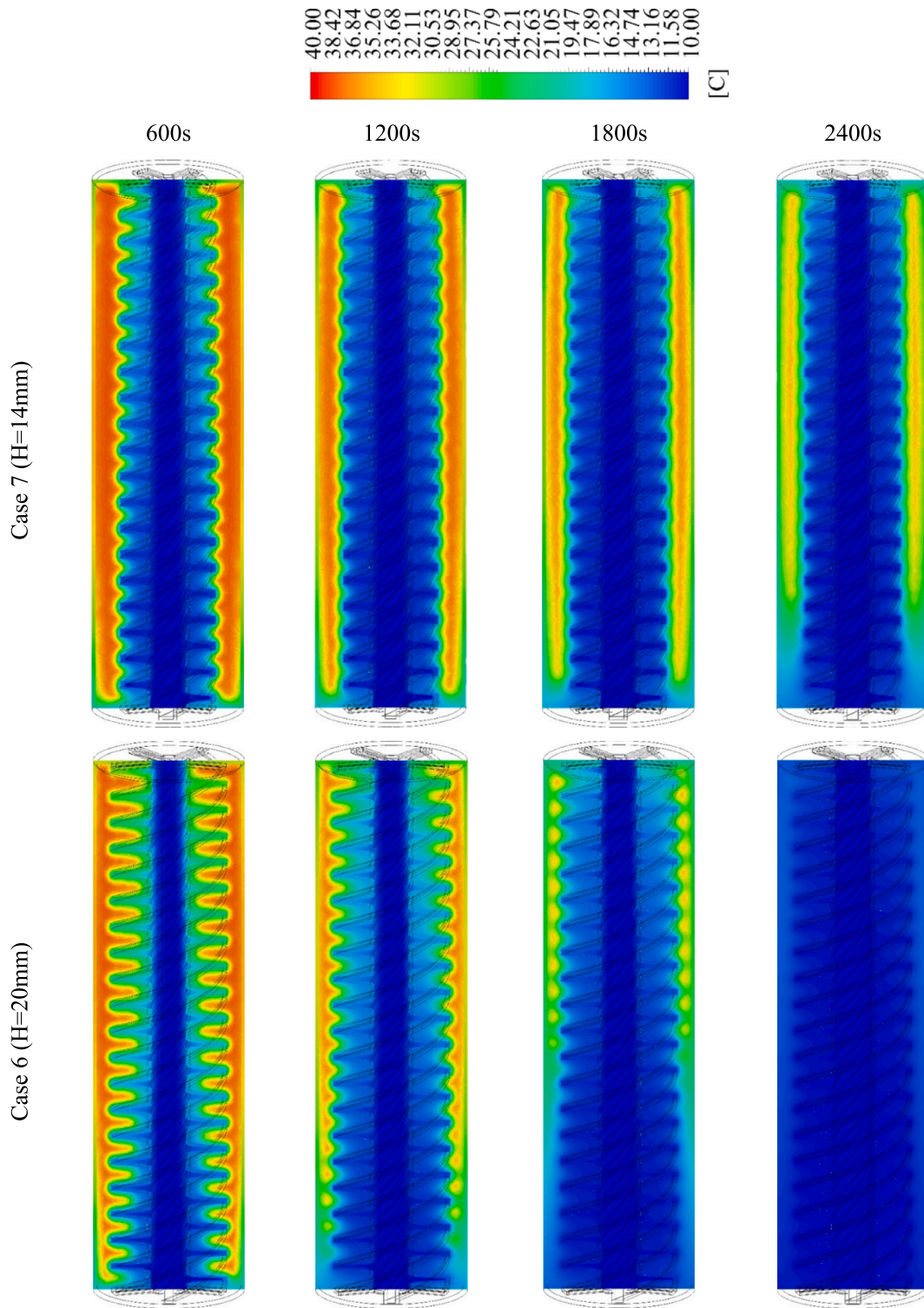


Fig. 21. Temperature contours for cases 6 (20×4.4) and 7 (14×6.15) for the time steps up to 2400 s, number of voids is 5, number of revolutions is 4.

latent heat removal and driving the steeper liquid-fraction decay observed for $Re = 1500$ – 2000 .

At $Re \approx 1000$, mixing is weaker; the boundary layer remains thicker and heat extraction is conduction-limited, yielding a slower decline of liquid fraction and a higher mean temperature. As Re increases to 1500 – 2000 , enhanced secondary flows distribute isotherms more uniformly, reducing internal thermal resistance within the mushy zone and advancing the solid front more rapidly. The diminishing gap between the $Re = 1500$ and $Re = 2000$ curves indicates the onset of a transport-

limited regime where further boundary-layer thinning yields smaller gains, while the required pumping power increases roughly. The data therefore reflect a balance between heat-transfer enhancement by vortex-induced mixing and the energetic cost of higher Re . In our geometry, $Re \approx 1500$ captures most of the convective benefit (Fig. 23), with $Re = 2000$ providing incremental improvements consistent with the smaller residual liquid fraction at late times.

The data in Table 11 demonstrate a clear positive correlation between Re and thermal recovery performance for Case 6 (20 mm voids).

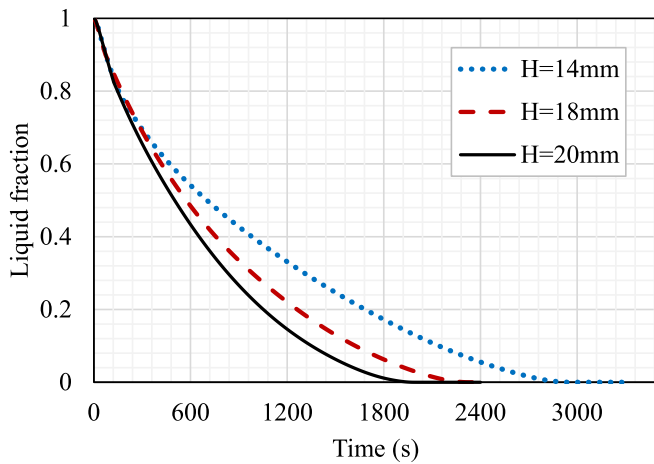
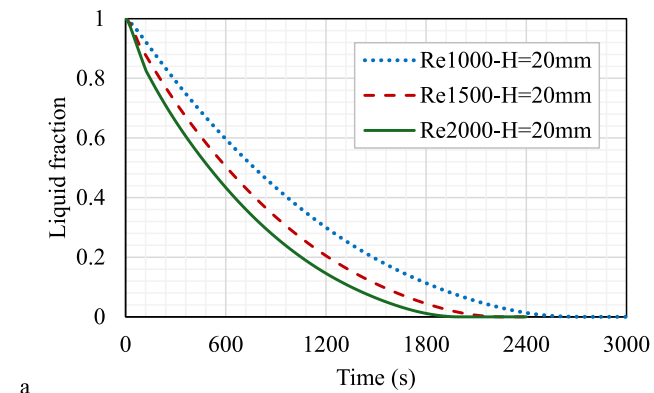


Fig. 22. Comparison of the cases 5, 6, and 7 regarding liquid fraction during the solidification process.

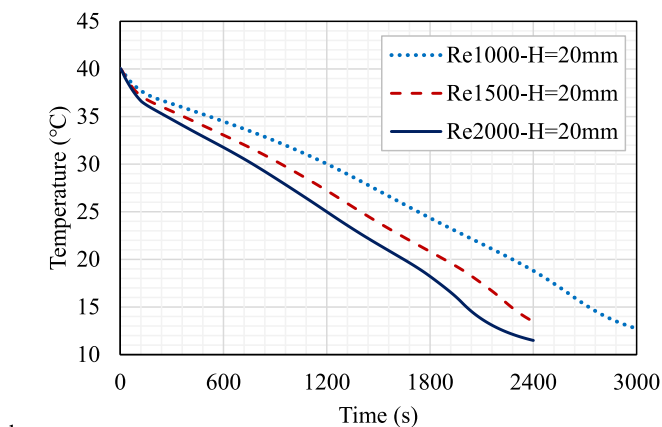
Table 10

The discharging period and the thermal recovery rate for the cases 5, 6 and 7.

	Heat recovery rate (W)	Solidification time (s)
Case 5 (H = 18 mm)	62.97	2993
Case 6 (H = 20 mm)	81.05	2359
Case 7 (H = 14 mm)	94.60	2037



a



b

Fig. 23. Comparison of various Re of case 6 regarding a) liquid fraction, and b) temperature during the solidification process.

As Re increases from 1000 to 2000, the heat recovery rate improves by 36.9 % (from 69.08 W to 94.60 W), while the solidification time

Table 11

The discharging period and the thermal recovery rate for the case 6 at different Re .

	Heat storage rate (W)	Solidification time (s)
$Re = 1000$	69.08	2804
$Re = 1500$	83.29	2322
$Re = 2000$	94.60	2037

decreases by 27.3 % (from 2804 s to 2037 s). This enhancement occurs because higher Re values promote turbulent flow conditions, which intensify convective heat transfer through three key mechanisms: (1) disruption of thermal boundary layers, (2) improved mixing of the cooling fluid, and (3) more uniform temperature distribution throughout the PCM. The transition from laminar ($Re = 1000$) to turbulent flow ($Re \geq 1500$) is particularly impactful, as evidenced by the 20.6 % performance jump between $Re = 1000$ and 1500, compared to the 13.6 % gain between $Re = 1500$ and 2000. This nonlinear improvement indicates diminishing returns in the PCM-side thermal metrics. In a full conjugate analysis, higher Re would also incur larger HTF pressure drops, reinforcing the trade-off, but those hydraulic effects are not evaluated here.

The results highlight that $Re = 1500$ represents a critical threshold where turbulent flow begins to significantly enhance solidification performance. While $Re = 2000$ delivers the highest recovery rate (94.60 W), the marginal gain over $Re = 1500$ (83.29 W) would need to be weighed against the expected increase in HTF pumping power in a conjugate thermo-hydraulic analysis; HTF pumping power is not evaluated in the present study. For many applications, $Re = 1500$ offer an optimal balance, providing 80 % of the maximum performance improvement (relative to $Re = 1000$) while avoiding the excessive energy costs of $Re = 2000$. The data also imply that void-enhanced heat exchangers are particularly effective in turbulent regimes, where their geometry can fully exploit chaotic flow patterns to maximize heat extraction.

5.3. Comparative evaluation of void-enhanced and conventional double-pipe configurations

As shown in Fig. 24, the conventional double-pipe design without voids (Case 1) demonstrates significantly slower thermal performance, requiring over 3600 s for solidification and showing minimal melting progress by 1200 s, due to reliance on inefficient conductive heat transfer. In stark contrast, the optimized Case 6 (20 mm voids, 5 voids, 4 helical revolutions, $Re = 2000$) achieves full melting by 1200 s and complete solidification by 2037 s, leveraging enhanced convective mixing from void-induced turbulence and helical flow. Case 6's heat storage rate (153.79 W) and thermal recovery rate (94.60 W) outperform Case 1 by 80 % and 50 %, respectively, proving that strategic void integration and flow optimization can halve phase-change times while dramatically improving energy transfer efficiency in thermal storage systems.

Fig. 25 clearly illustrates the dramatic improvement in thermal performance achieved by Case 6 (5 voids, continuous lines) compared to the conventional Case 1 (no voids, dotted lines) during both melting (red) and solidification (black) processes. For melting, Case 6 reaches a liquid fraction of 0.8 by approximately 800 s, while Case 1 barely achieves 0.25 in the same timeframe. This stark difference highlights the inefficiency of Case 1, which relies solely on conductive heat transfer, versus Case 6's superior convective heat transfer enabled by void-induced turbulence and enhanced flow dynamics. Similarly, during solidification, Case 6 completes the phase change (liquid fraction near 0) by around 1900 s, whereas Case 1 lags significantly, requiring over 5400 s to reach the same state. The accelerated phase change in Case 6 underscores the critical role of voids in optimizing heat exchange efficiency.

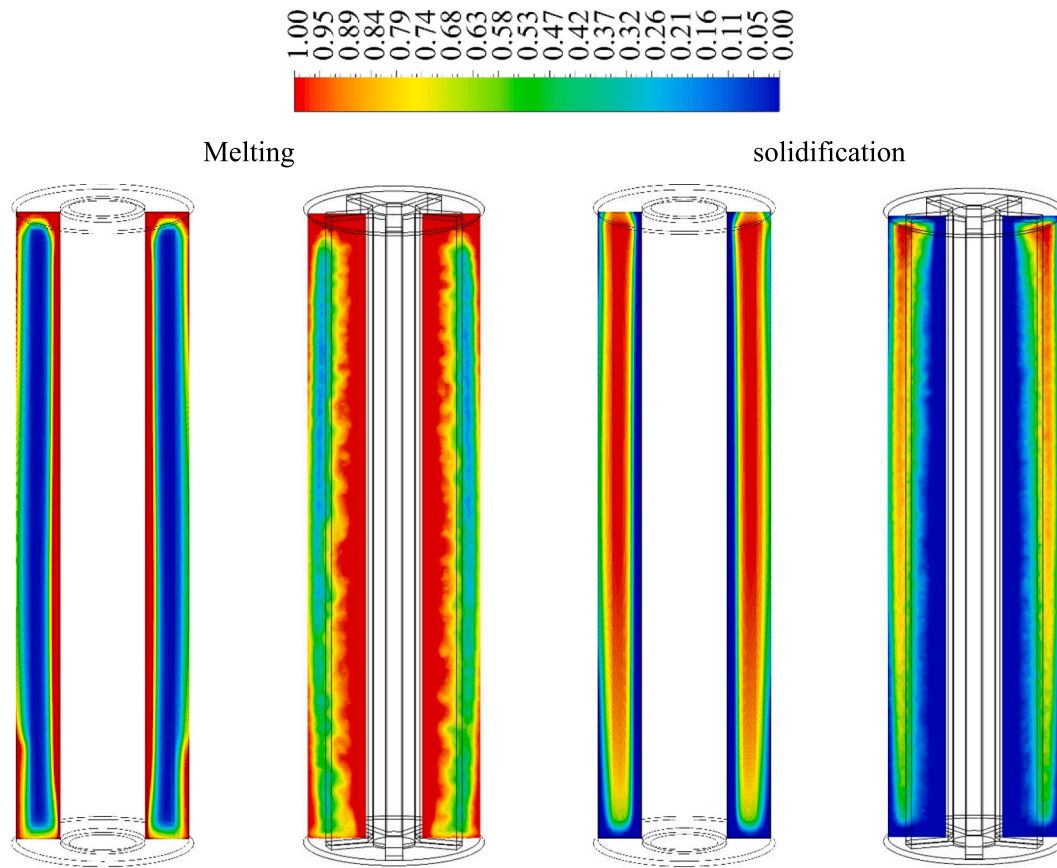


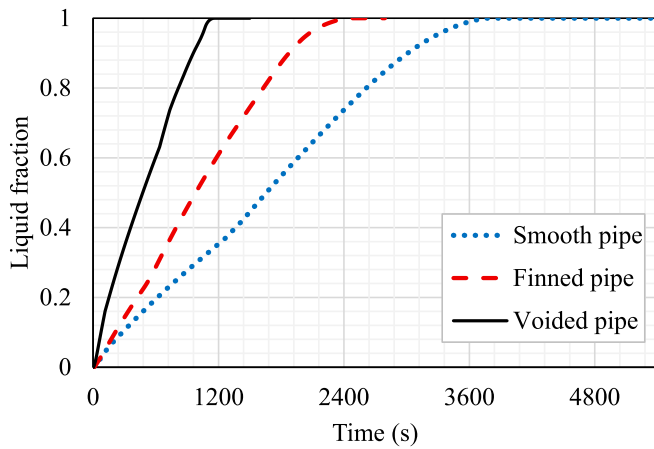
Fig. 24. Liquid fraction of the PCM in case 1 (no voids) during the melting and the solidification for $t = 1200$ s.

The superior performance of Case 6 can be attributed to its ability to disrupt thermal boundary layers and promote uniform temperature distribution through strategically placed voids, which are absent in Case 1. During melting, the voids in Case 6 facilitate faster energy absorption by increasing the surface area for heat transfer and creating turbulent flows that prevent localized hot spots. For solidification, the voids similarly enhance heat dissipation by improving fluid circulation and reducing thermal resistance. While Case 6's design offers clear advantages in terms of speed and efficiency, it is essential to consider potential trade-offs, such as increased manufacturing complexity or pressure drop. Nevertheless, the substantial reduction in phase-change time (by nearly 50 %) makes Case 6 a compelling choice for applications requiring rapid thermal cycling, demonstrating the transformative potential of void-enhanced designs over conventional systems.

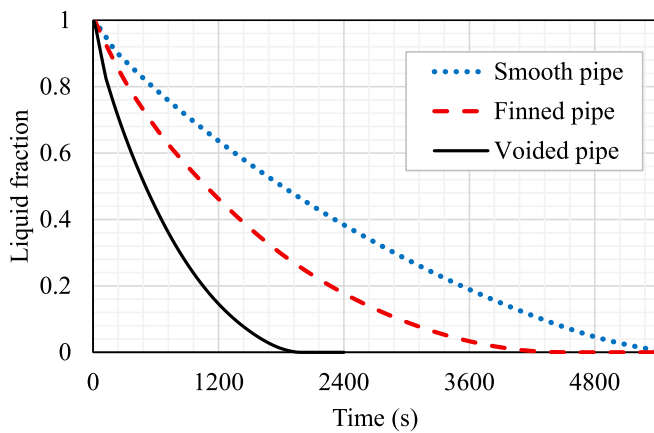
The comparative data between Case 6 (5 voids) and Case 1 (no voids) in Fig. 26 demonstrates the transformative impact of void-enhanced design on thermal performance, with Case 6 achieving melting and solidification rates of 153.80 W and 94.60 W respectively - representing 212 % and 191 % improvements over Case 1's 49.25 W (melting) and 32.56 W (solidification). This dramatic enhancement stems from the void structure's ability to convert inefficient conductive heat transfer into optimized convective heat exchange through increased surface area and turbulent flow generation, effectively tripling thermal performance metrics. While these improvements make Case 6 ideal for applications requiring rapid thermal cycling, the design's increased complexity and potential pressure drop effects necessitate careful cost-benefit analysis for specific implementation scenarios, particularly where simpler systems might suffice despite lower efficiency. The results conclusively validate void integration as a powerful strategy for advanced thermal energy storage systems, though further optimization required to balance performance gains with practical manufacturing and operational considerations.

The proposed void-enhanced twisted tube is most advantageous where high power density and fast cycling are required, such as (i) solar thermal charging/discharging, (ii) building thermal energy storage for peak-shaving, (iii) industrial batch waste-heat recovery, and (iv) compact thermal buffers for electronics or battery packs. Relative to a plain HTF tube, the design increases effective heat-transfer area, intensifies secondary flows, and suppresses thermal stratification, which consistent with Tables 3–6, Tables 8–10, reduces melting/solidification times and raises storage/recovery rates at fixed volume. Compared with finned inserts, the voided geometry offers similar heat-transfer enhancement with lower added mass. The benefits are offset by increased pressure drop and greater fabrication complexity; hence, for low-power, quasi-steady applications a plain tube may suffice, whereas high-throughput systems justify the enhanced design.

Table 1 presents a comparative analysis of the hydraulic performance, specifically pressure drop and pumping power, for three different heat exchanger pipe configurations: the proposed voided pipe, a conventional smooth pipe, and a finned pipe. The data reveals that the voided pipe introduces a moderate hydraulic penalty compared to the smooth pipe baseline. The pressure drop for the voided pipe (462.72 Pa) is approximately 1.2 % higher than that of the smooth pipe (457.02 Pa), resulting in a correspondingly slight increase in pumping power. This indicates that while the embedded voids disrupt flow to enhance thermal performance, they do so without a substantial increase in flow resistance, making their hydraulic cost relatively low. The most significant finding, however, emerges from the comparison with the finned pipe. The finned pipe exhibits a much higher pressure drop (510.87 Pa), which is about 10.4 % greater than the voided pipe and 11.8 % greater than the smooth pipe. Despite this, the pumping power for the finned pipe (0.005165 W) is less than half that of the other two designs. This apparent contradiction suggests a critical difference in the experimental or modelling conditions, most likely a significantly lower volumetric



a)



b)

Fig. 25. Comparison of the case 1 (no voids) and case 6 (5 voids) during the melting and solidification process.

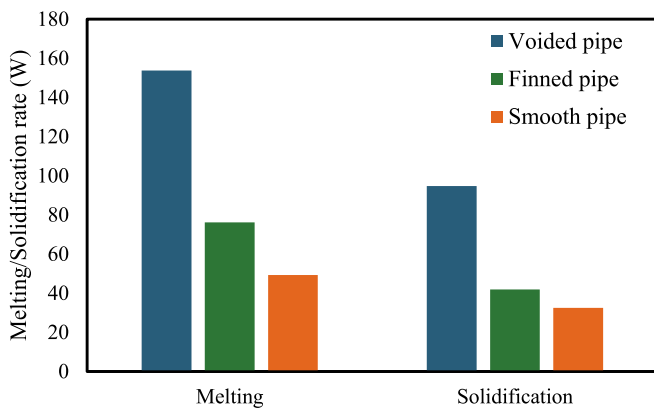


Fig. 26. Comparison of melting and solidification rates between the conventional double-pipe (Case 1, no voids) and the optimized void-enhanced configuration (Case 6, 5 voids).

flow rate used for the finned pipe simulation. This highlights that while fins can create substantial flow obstruction, their operational pumping cost is highly dependent on the system's flow rate.

Table 12

Comparative hydraulic performance of voided, smooth, and finned pipes.

	Pressure drop	Pumping power (W)
Voided pipe	462.72	0.011234
Smooth pipe	457.02	0.011102
Finned pipe	510.87	0.005165

5.4. Practical considerations, operational challenges, and comparative discussion

While the proposed void-embedded twisted-tube configuration demonstrates substantial thermal enhancement and weight reduction, practical implementation in large-scale thermal energy storage systems may involve several technical challenges. Fabrication of the multi-lobed helical geometry requires precision forming and joining processes that increase manufacturing complexity and cost relative to smooth or finned tubes. Moreover, the inclusion of embedded voids alters flow distribution and may elevate pressure drop, thereby increasing the required pumping power for the heat-transfer fluid. These effects, although moderate at laboratory scale, must be optimized through future studies that integrate thermal-hydraulic performance and techno-economic analysis. Nevertheless, the advantages of the proposed design (lighter structure, faster melting/solidification, and higher power density) justify its exploration for medium-temperature storage systems where high cycling efficiency and compactness are prioritized.

From a material perspective, the lightweighting benefit can also be contextualized relative to conventional fin materials such as aluminum and its alloys, which are known for their high thermal conductivity and low density. Unlike aluminum fins that rely primarily on conductive enhancement, the proposed void-embedded structure achieves heat-transfer improvement through enhanced convective mixing and flow steering within the PCM, without the additional solid mass that fins introduce. A key limitation of the present study is the lack of quantitative data on pressure drop and pumping power, as the HTF side was modelled using an isothermal wall boundary condition rather than a fully resolved conjugate simulation. While the thermal enhancements from increased helical revolutions, void height, and Reynolds number are clearly demonstrated, their associated hydraulic penalties could not be quantified. The mentioned increases in pressure drop are based on established fluid dynamics principles where greater flow path tortuosity, surface disruption, and flow velocity inherently elevate flow resistance. Future work must incorporate a conjugate HTF-PCM model to perform a complete thermo-hydraulic optimization, balancing the reported reductions in phase-change time against the required pumping power to determine the true operational efficiency and economic viability of these designs.

6. Conclusion

This study introduced a novel void-enhanced double-pipe latent heat thermal energy storage (LHTES) unit featuring a twisted inner tube with strategically embedded voids. Unlike conventional fin-based designs, the innovation lies in employing voids as lightweight geometric enhancers that promote convective mixing while reducing structural mass. The aim was to overcome the low thermal conductivity limitation of phase change materials (PCMs) by optimizing void number, dimensions, and helical revolutions to accelerate both melting (charging) and solidification (discharging) processes. The objective was to provide a scalable and efficient design framework for medium-temperature applications such as solar thermal systems, building energy management, and industrial waste heat recovery. The key outcomes of this study include:

- Increasing void count from 3 to 5 raised the heat storage rate from 84.6 W to 119.4 W and reduced melting time from 2269 s to 1570 s.

- Optimizing void height (20 mm) achieved the best performance, with a heat storage rate of 153.8 W and a melting time of 1185 s.
- Increasing the number of voids (from 3 to 5) intensifies internal PCM circulation and secondary flows, thereby strengthening convective heat transfer and contributing directly to the accelerated melting rates observed in multi-void configurations.
- Solidification recovery rates improved from 46.2 W (3 voids) to 72.5 W (5 voids), with discharge time shortened from 4218 s to 2656 s.
- Enhanced helical revolutions (2 to 4 turns) increased heat storage rate by 5.0 % and reduced melting time by 3.3 %.
- Reynolds number escalation from 1000 to 2000 boosted heat storage from 98.6 W to 153.8 W, while reducing melting time from 1814 s to 1185 s.
- Compared to the baseline smooth tube, the optimized Case 6 achieved a 212 % higher melting rate and 191 % higher solidification rate, confirming the transformative role of voids.
- Void-enhanced twisted tube achieves its large gains in melting/solidification rate with only a marginal hydraulic penalty relative to the smooth tube configuration, confirming that the enhanced design remains energetically efficient for high-throughput LHTES applications.

These results demonstrate that void integration can be a practical and sustainable alternative to fins, offering superior energy storage density, faster thermal cycling, and reduced material use. *By shortening conduction paths and promoting secondary flows and convective mixing within the PCM*, voids fundamentally shift PCM-based storage from conduction-dominated to convection-enhanced systems. Nevertheless, several drawbacks and challenges warrant attention: (i) increased geometric complexity (twist forming with embedded voids) may introduce manufacturing tolerances, local wall thinning, and joining/sealing risks; (ii) voids can elevate flow resistance in a conjugate design and may trap gas or impurities, affecting start-up and repeatability; and (iii) long-term reliability under thermal cycling (fatigue, fouling, PCM leakage, and property drift) requires experimental verification. Accordingly, future work should include a conjugate HTF-PCM model to assess thermo-hydraulic trade-offs, targeted experiments to validate performance and durability, scaling studies for industrial deployment and a techno-

economic analysis to benchmark cost per kWh-throughput against finned and foam-based alternatives. Additionally, while this study focuses on the comparative performance within our proposed configurations, a broader numerical comparison with other advanced PCM heat exchanger geometries (e.g., finned, porous, or helical designs) would require separate modelling under identical boundary and material conditions to ensure fairness which can be included in future studies.

CRediT authorship contribution statement

Alsamani A.M. Salih: Writing – review & editing, Writing – original draft, Methodology, Investigation, Formal analysis, Conceptualization. **Hayder I. Mohammed:** Writing – review & editing, Writing – original draft, Validation, Methodology, Investigation, Formal analysis. **Abdul Amir H. Kadhum:** Writing – review & editing, Writing – original draft, Investigation, Formal analysis. **Bhupendra Singh Chauhan:** Writing – review & editing, Writing – original draft, Investigation, Formal analysis. **Damodharan Iranian:** Writing – review & editing, Writing – original draft, Investigation, Formal analysis. **Nashmi H. Alrasheedi:** Writing – review & editing, Writing – original draft, Supervision, Methodology, Investigation, Formal analysis. **Khalil Hajlaoui:** Writing – review & editing, Writing – original draft, Supervision, Project administration, Methodology, Investigation, Formal analysis, Conceptualization. **Nidhal Ben Khedher:** Writing – review & editing, Writing – original draft, Project administration, Methodology, Investigation, Formal analysis.

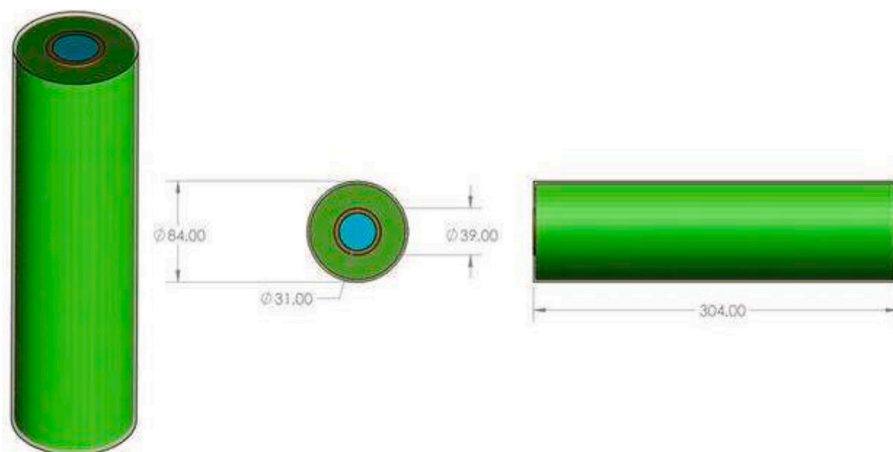
Funding statement

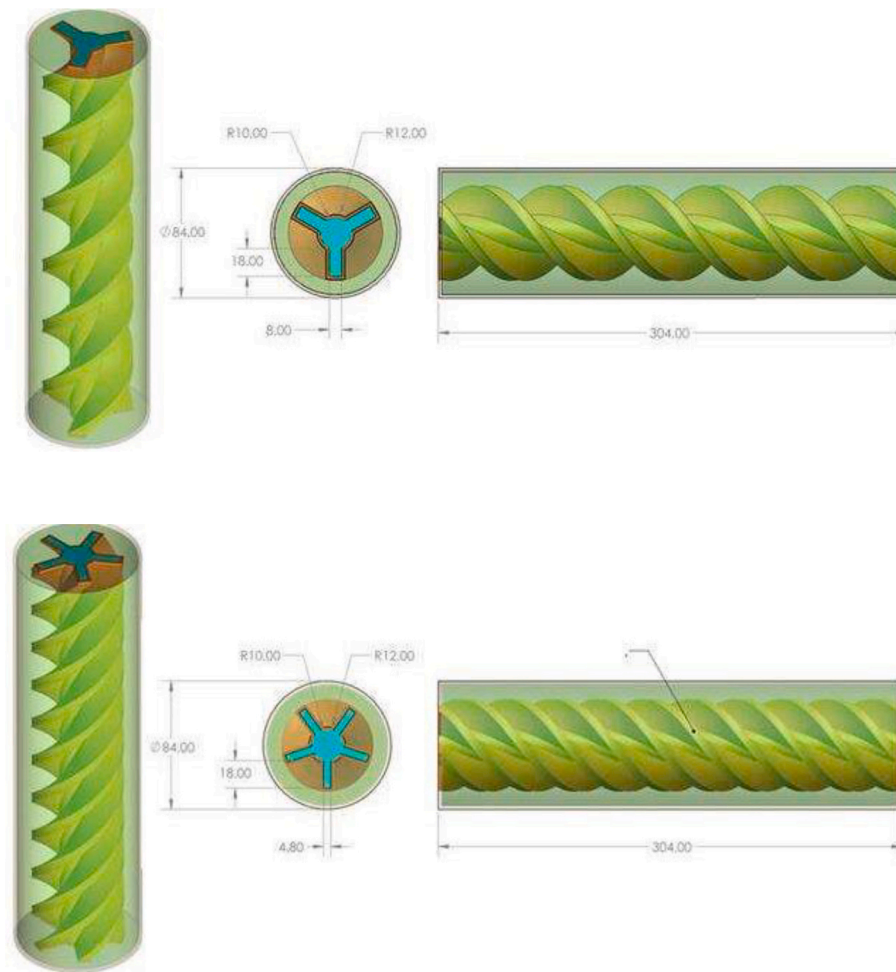
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Declaration of competing interest

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.

Appendix A





All units in mm.

Data availability

Data will be made available on request.

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