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Graphical abstract:

Abstract: Thermal energy storage (TES) is crucial in the efficient utilization and stable supply of renewable energy. This study aims to enhance the performance of shell-and-tube latent heat thermal energy storage (LHTES) units, particularly addressing the issue of the significant melting dead zones at the bottom, which are responsible for the long charging time. We proposed a new heat transfer enhancement technique inspired by the air channel distribution inside the root of the lotus. Numerical simulations are used to explore its melting behavior and heat storage performance, and a comparison is made with conventional shell-and-tube TES units. The results indicate that compared to the single-tube type, the bionic-lotus root type
reduced the total melting time by 89.1%, increased the average temperature by 13.2°C, and enhanced the average effective power density by 7.6 times. Subsequently, an industrial standard type was constructed and optimized according to manufacturing standards for mass production using off-the-shelf piping products. The melting time was further decreased by 5.8%, and the average effective power density increased by 6.6%. The results of this study provide a promising and prospective solution for enhancing shell-and-tube LHTES units, with the potential to increase the efficiency, reduce the footprint and manufacturing costs of energy systems equipped with TES.

**Keywords**: shell-and-tube thermal storage unit, bionic-lotus root, phase change heat transfer, enhanced heat transfer, optimization.

<table>
<thead>
<tr>
<th>Nomenclature</th>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
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<td>c</td>
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<td>f</td>
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<td>Thermal conductivity</td>
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<tr>
<td>m</td>
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<td>Constant pressure</td>
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<td>Reference</td>
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1 Introduction

Thermal energy storage (TES) supports the broader adoption of renewable energy sources [1], such as solar thermal power generation, solar photovoltaics [2], cogeneration, geothermal [3], and wind power [4]-[5]. By decoupling the heat and cold demand from the energy supply, TES helps mitigate intermittency, achieve flexible load balancing, and reduce the need for grid reinforcements [6]. According to the International Renewable Energy Agency, over 50% of global final energy consumption is in the form of heat and cold, and the installed capacity of thermal energy storage globally is expected to triple within the next decade [6]. Hence, TES has broad market potential and promising prospects for future development.
Among various TES technologies, Latent Heat Thermal Energy Storage (LHTES) has gained widespread popularity due to its small temperature fluctuation, high energy storage density, and long storage cycles [7]. This study conducted a search in the Web of Science database for 3000 publications in the field of LHTES from the past five years. The analysis of the publications’ keywords was performed using VOSviewer. Figure 1 illustrates the interconnections and groupings of these keywords within the field. It can be observed that the prominent research areas within LHTES mainly consist of three clusters: 1) thermal conductivity, 2) heat transfer, and 3) simulation and optimization. Firstly, due to the low thermal conductivity of conventional phase change materials (PCMs), which affect the charging and discharging rates of LHTES, many researchers have been actively working on improving the thermal conductivity of PCMs [8]. On the other hand, some articles focused on phase change heat transfer in TES, particularly the natural convective heat transfer of liquid PCM within tubes [9]. Moreover, employing numerical simulation methods to optimize the design of TES units is one of the primary directions in the field of LHTES. Notably, a few studies have looked into the bio-inspired optimization of TES structures [10].

The design of the structure of a LHTES unit directly affects the unit’s energy storage efficiency [11]. Among various types of LHTES, shell-and-tube units are widely used in TES due to their robust structure and ease of maintenance [12]. Due to the limited thermal conductivity of PCMs, much effort has been spent on deriving ways to enhance heat transfer in shell-and-tube LHTES [13]. Some studies focus on the compositing of PCMs with other highly conductive materials [14]-[15][16] such as high-conductive nanoparticles, which have been shown to improve the thermal conductivity of the PCMs [17][18]. The addition of fins [14],[19]-[20] to increase heat transfer surface area between the heat source and the PCMs is also a popular method to accelerate the rate of heat absorption or release by the PCMs [21]. However, the addition of fins reduces the overall thermal storage capacity of the unit [22].
Adding fins can also drive up the production cost and cause agglomeration of nanoparticles, posing challenges that have not been fully addressed [23]. Therefore, instead of adding fins, some studies propose optimizing the design of internal flow channels in LHTES units [24]-[26] to increase the contact area and contact time between the fluid and the tube, thereby improving heat transfer efficiency without reducing the thermal storage capacity. Compared to the previous two methods, optimizing the design of internal flow channels can also reduce operating and maintenance costs [27].

However, the melting dead zone, a particular problem commonly found in shell-and-tube LHTES units during the melting process of PCM, is rarely addressed. The melting dead zone is usually formed at the bottom region of the unit, where the heat transfer efficiency is extremely low. At the later stages of charging, when most of the PCM already melted, there is no direct contact between the tubes and the remaining un-melted PCM at the bottom of the unit [28]. Heat transfer to this region relies solely on the slow heating of the upper surface of the unmelted PCM by the liquid PCM above it, resulting in very low efficiency. Especially for horizontal shell-and-tube LHTES units, PCM at the bottom of the unit always melts the latest [29]. A few studies have noticed the melting dead zone. Pizzolato et al. [30] optimized the fins inside a shell-and-tube LHTES unit using topology optimization techniques. The optimized fins extended to the bottom part of the thermal storage unit, leaving only a few short fins at the top. This configuration reduced the fill time by 27% while achieving 95% of the total storage capacity. Zheng et al. [31] improved the thermal storage performance of the shell-and-tube LHTES unit by introducing an eccentricity between the inner and outer tubes. They obtained an optimal eccentricity that reduced the overall melting time. Their results indicated that vertically moving the inner tube along the center of the outer tube significantly shortened the total melting time. Kadivar et al. [32] obtained optimal radial and tangential eccentricities to minimize the thermal storage time for PCM by using response surface methodology to determine the optimal geometric configuration. They also suggested that increasing the number of tubes in the shell is another solution to enhance the heat transfer rate between PCM and the heat transfer fluid (HTF). However, the main challenge in designing multi-tube TES lies in the optimization of the arrangement of the inner tubes within the shell [33]. Among these improvement methods, the use of topology optimization for fin optimization is limited by the high computational requirements, making it difficult to be widely adopted. Therefore, internal flow channel optimization (including radial/tangential eccentricity of the inner tube, number of inner tubes, arrangement of inner tubes, etc.) would be easier to implement.

Besides traditional optimization methods, using biomimicry to improve device structures and optimize unit performance has been gaining popularity recently. Wang et al. [34] drew inspiration from the structure of the gourd and proposed a biomimetic gourd-shaped PCM storage unit. They conducted experiments and numerical simulations to optimize the heat transfer and flow characteristics of the storage unit. The results showed that the throat structure of the biomimetic gourd-shaped storage unit accelerated the melting process of the PCM. Compared to a spherical storage unit, the melting time was reduced by 18.2%, and the HTF flow drag coefficient was decreased by 34.2%. Dong et al. [35] proposed an elliptical PCM storage unit with a biomimetic egg structure to improve its flow characteristics and thermal performance. Different structures (biomimetic elliptical, spherical, and elongated elliptical) of the storage unit were prepared using 3D printing, and experimental and numerical simulations
were conducted. The results showed that compared to the spherical storage unit, the non-restricted melting time of the biomimetic elliptical storage unit was reduced by 12%, and the average Nusselt number increased by 20%. Mourad et al. [36] proposed a pear-shaped storage unit with added fins and a composite of PCM and Al₂O₃ nanoparticles to enhance the thermal storage performance. The influence of factors such as the volume fraction of nanoparticles, time, and the number and size of fins on the melting and heat transfer process was studied using numerical simulations. The research results showed that adding nanoparticles improved the melting process of PCM. Cheng et al. [37] proposed a red-blood-cell-shaped PCM thermal storage unit. Through experiments and numerical simulations, comprehensive studies and comparisons were conducted on the storage unit and its deformed structures (cylindrical, bulging, annular, and spherical), focusing on thermal properties such as liquid fraction variation, temperature distribution, and heat storage rate. The results showed that the red-blood-cell-shaped storage unit exhibited the best thermal performance, while the spherical storage unit performed the worst. The heat storage rate of the red-blood-cell-shaped storage unit was 2.12 times that of the spherical capsule. Ren et al. [38] proposed an irregular snowflake-shaped longitudinally finned structure and evaluated its impact on melting fraction, temperature field, velocity distribution, and heat storage capacity through numerical simulations. The results showed that the snowflake-shaped fins significantly shortened the melting time of a shell-and-tube LHTES unit by 45.92% compared to conventional fin structures. However, most of the above-mentioned studies applied biomimicry to the external shape of spherical storage units. In shell-and-tube LHTES, only a few studies are conducted on biomimetic optimization of fin shapes, but no research has ever been put on optimizing the internal flow channels based on biomimetic principles.

Limited natural structures are similar to the shell-and-tube configuration, with the lotus root being one of them. To adapt to its natural environment underwater, the lotus root has multiple well-distributed elliptical air channels, which significantly increase the contact area between oxygen and the inner wall of the lotus root [39] (Figure 2(a)). These characteristics are well-suited for optimizing the internal flow channels in shell-and-tube LHTES units to solve the problem of the melting dead zone. If multiple elliptical tubes can be incorporated within the shell, the heat transfer area between the HTF and the PCM can be increased to enhance heat conduction during the early melting stage and natural convection of molten PCM during the later melting stage. Hence, by drawing inspiration from the lotus roots’ natural structure, multiple parallel elliptical flow channels can offer a new potential to develop an optimal arrangement of the inner tubes for horizontally mounted LHTES units.

Therefore, this study proposed and evaluated the application of a bionic-lotus root structure to the shell-and-tube thermal storage unit. A numerical model for phase change heat transfer in LHTES is established. An evaluation index, effective power density, is introduced to evaluate the charging performance of the bionic-lotus root design, which is compared with that of the conventional shell-and-tube structures. The phase change heat transfer mechanism of the bionic-lotus root structure during the melting process is also discussed. Under the constraints of existing industrial tube dimension standards, the effects of major geometric parameters on thermal storage performance are analyzed using the response surface methodology (RSM). Finally, the bionic-lotus root LHTES is optimized according to industrial manufacturing standards, and the superiority of the optimized structure is validated. In
summary, this study proposes a promising LHTES unit structure solution with potential applications, and the research findings provide promising guidance for further developments in the industrial sector.

2 Bionic-lotus root LHTES design

Inspired by lotus roots (Figure 2(a)), a bionic-lotus root type was developed in this sector. A traditional shell-and-tube LHTES unit (single-tube type), a multi-tube type, and an inverted bionic-lotus root type (inverse bionic-lotus root) were also constructed for comparison.

The major macroscopic features of the natural shape of a lotus root can be geometrically represented by a cylindrical 3D model, as shown in Figure 2(b). In this representation, the red regions represent the flow channels for the HTF, while the blue region represents the filled PCM. The bionic-lotus root type can be defined as a combination of elliptic and circular channels, with their main geometrical parameters, as shown in Figure 2(c). A1 and B1 represent the major and minor axes of the larger ellipses, whereas A2 and B2 represent those of the smaller ellipses at the bottom. D1 represents the diameter of the outer circle, and D2 represents the diameter of the inner circle. L1 is the distance from the shell center to the center of the larger ellipses, and L2 is the distance from the shell center to the center of the smaller ellipses. The angle $\alpha$ denotes the counterclockwise angular separation between the major axis of a large ellipse relative to the positive x-axis.

Figure 2 Bionic structure design (a) The natural structure of the lotus-root; (b) 3D illustration of the lotus-root inspired shell-and-tube LHTES unit; (c) Simplified 2D model.

The bionic-lotus root was constructed using the semi-minor axis of the small ellipses as the basic length unit. The parameters for the geometric dimensions are as follows: the semi-minor axis of the small ellipses is “B2 = 4.66 mm,” the semi-major axis of the small ellipses is “A2 = 2 × B2,” the semi-minor axis of the large ellipses is “B1 = 3 × B2,” the semi-major axis
of the large ellipses is “A1 = 5 × B2,” and the diameter of the inner circle is “D2 = 2.5 × B2.”

Detailed dimensions of the model are provided in Table 1.

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<th>Parameters</th>
<th>Value</th>
<th>Parameters</th>
<th>Value</th>
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<td>A1</td>
<td>23.30 mm</td>
<td>D2</td>
<td>11.65 mm</td>
</tr>
<tr>
<td>A2</td>
<td>9.32 mm</td>
<td>L1</td>
<td>35.00 mm</td>
</tr>
<tr>
<td>B1</td>
<td>13.98 mm</td>
<td>L2</td>
<td>40.00 mm</td>
</tr>
<tr>
<td>B2</td>
<td>4.66 mm</td>
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<td>45°</td>
</tr>
<tr>
<td>D1</td>
<td>100.00 mm</td>
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The elliptical curves in the bionic-lotus root are calculated by Equation (1):

\[
\frac{(x - m) \cos \alpha + (y - n) \sin \alpha)^2}{a^2} + \frac{(m - x) \sin \alpha + (y - n) \cos \alpha)^2}{b^2} = 1
\]  

(1)

The centers of both the outer and the inner circles are also the origin of both the x- and y-axis. The outer and inner circular curves are calculated by Equation (2):

\[ x^2 + y^2 = D^2/4 \]  

(2)

The distance between the center of the ellipse and the center of the inner circle is calculated by Equation (3):

\[ L = \sqrt{m^2 + n^2} \]  

(3)

where, \( m \) is the displacement of the ellipse along the x-axis. \( n \) is the displacement of the ellipse along the y-axis. \( D \) represents the diameter of a circle.

The main parameters have to satisfy the constraint conditions given by Equation (4).

\[
\begin{align*}
\frac{D_2}{2} + 2.5B_2 &< L_1 < \frac{D_1}{2} - 2.5B_2 \\
\frac{D_2}{2} + B_2 &< L_2 < \frac{D_1}{2} - B_2
\end{align*}
\]  

(4)

Three different TES units (single-tube, multi-tube, and bionic-lotus root) were established by keeping the outer circle diameter, unit length, and volume of PCM equal in all three units but only changing the shape and distribution of internal flow channels. The three-dimensional structures of the three units are shown in Figure 3(a-c). Since the axial temperature gradient of the 3D models is much smaller than the radial temperature gradient, the geometric models were simplified into 2D computational models for this research. The specific geometric parameters are illustrated in Figure 3(e-g), and the simplification of the 3D models has been validated in previous studies [40][41]. Additionally, considering the asymmetry of the bionic-lotus root, an inverted bionic-lotus root type was also simulated to study its melting process, as shown in Figure 3(d) and Figure 3(h).
Figure 3 3D structure and cross-sectional dimensions of the (a)(e) single-tube type unit; (b)(f) multi-tube type unit; (c)(g) bionic-lotus root type unit; (d)(h) inverse bionic-lotus root type unit.

This study used n-octadecane as the PCM with the thermal properties [42] shown in Table 2. Considering the assumption given in Section 3.1, the values of the density, specific heat capacity, and thermal conductivity of the PCM in the simulation process were chosen as the average of the solid and liquid phases [43], which are 817.5 kg m\(^{-3}\), 2325 J kg\(^{-1}\) K\(^{-1}\), and 0.25 W m\(^{-1}\) K\(^{-1}\), respectively.

Table 2 Thermophysical properties of PCM

<table>
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<td>Solid phase density (kg m(^{-3}))</td>
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<tr>
<td>Liquid phase density (kg m(^{-3}))</td>
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<td>Solid phase specific heat capacity (J kg(^{-1}) K(^{-1}))</td>
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<td>Liquid phase specific heat capacity (J kg(^{-1}) K(^{-1}))</td>
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<td>Liquid phase thermal conductivity (W m(^{-1}) K(^{-1}))</td>
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<tr>
<td>Dynamic viscosity (m(^{2}) s(^{-1}))</td>
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<tr>
<td>Latent heat (kJ kg(^{-1}))</td>
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<tr>
<td>Phase change temperature (°C)</td>
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<tr>
<td>Coefficient of thermal expansion (K(^{-1}))</td>
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3 Numerical model

In this sector, a numerical model is developed to simulate the performance of the LHTES units.

3.1 Governing equations

The following assumptions were considered for modeling the flow, heat transfer, and phase change process inside the TES unit:

1) The molten PCM is an incompressible Newtonian fluid, and its flow is laminar.
2) PCM properties are assumed to be uniform and isotropic. Except for the density, all other thermal properties are constant.
3) Density variation is considered only when calculating buoyancy, and the Boussinesq...
approximation is used to account for the natural convection during the phase change process. The gravity acceleration is directed along the negative y-axis with a magnitude of 9.81 m s\(^{-2}\).

4) The outer shell of the thermal storage unit is thermally insulated, neglecting any heat losses to the surroundings. The inner tube wall is assumed to have a constant temperature, and the heat transfer resistance of the tube wall and the convective heat transfer process inside the tube are disregarded.

5) Volume change of the PCM, wall heat losses, and viscous dissipation effects are neglected. The no-slip boundary condition is considered for fluid flow.

Based on the above assumptions, the enthalpy-porosity method is used to simulate the phase change process. The following are the governing equations, including the continuity equation, momentum equation, and energy equation, among others:

Continuity equation:
\[
\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0
\] (5)

Momentum equation in the x-direction:
\[
\rho \left( \frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} \right) = -\frac{\partial p}{\partial x} + \mu \left( \frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) - A_{mush} \left( \frac{(1 - f)^2}{f^3 + \varepsilon} \right) u
\] (6)

Momentum equation in the y-direction:
\[
\rho \left( \frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \right) = -\frac{\partial p}{\partial y} + \mu \left( \frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) - A_{mush} \left( \frac{(1 - f)^2}{f^3 + \varepsilon} \right) v + \rho g \beta (T_l - T_m)
\] (7)

Energy equation:
\[
(\rho c_p) \left( \frac{\partial T_l}{\partial t} \right) + \left( \rho c_p \right) \left( u \frac{\partial T_l}{\partial x} + v \frac{\partial T_l}{\partial y} \right) = k \left( \frac{\partial^2 T_l}{\partial x^2} + \frac{\partial^2 T_l}{\partial y^2} \right) - L_h \rho \frac{\partial f}{\partial t}
\] (8)

In the above control equations, \( u \) and \( v \) represent the velocities in the \( x \) and \( y \) directions, respectively. \( p \) is the pressure, \( \rho_l, \mu_l, T_l, \) and \( k_l \) are the density, dynamic viscosity, temperature, and thermal conductivity of the liquid phase PCM, respectively. \( T_m \) is the phase change temperature of the PCM, \( \beta \) is the volumetric expansion coefficient, \( c_p \) is the specific heat capacity at constant pressure, and \( L_h \) is the latent heat of phase change.

In the enthalpy-porosity method, the solid-liquid interface is referred to as the “mushy region,” where both solid and liquid phases coexist. In this region, the fluid velocity varies between zero in the solid phase and increases to the natural convection velocity in the liquid phase. To account for these changes and distinguish between the solid and liquid phases of the PCM, momentum source terms are introduced in the momentum equations. The definition of the momentum source term is as follows:

\[
S_i = A_{mush} \left( \frac{(1 - f)^2}{f^3 + \varepsilon} \right) u_i
\] (9)

The term \( A_{mush} \) represents the mushy zone coefficient, which describes the degree of velocity slowdown at the melting front. In most studies, the value of \( A_{mush} \) typically ranges between \( 10^4 \) and \( 10^7 \). Tang et al. [44] investigated the influence of \( A_{mush} \) on the PCM melting.
process, and the results showed good agreement with experimental data when $A_{\text{mush}}$ was set to $10^5$. Therefore, in this study, $A_{\text{mush}}$ is chosen to be $10^5$. The variable $f$ represents the liquid fraction of the PCM, indicating the degree of melting. It is defined as follows:

$$f = \begin{cases} 
0 & , \ T < T_s \\
\frac{T-T_s}{T_l-T_s} & , \ T_s < T < T_l \\
1 & , \ T > T_l 
\end{cases}$$ \hspace{1cm} (10)

$T_1$ and $T_s$ represent the liquidus and solidus temperatures of the PCM. To avoid division by zero when $f=0$ during the calculation, a small constant $\varepsilon$ is added to the denominator. Typically, $\varepsilon$ is set to a sufficiently small value, such as $10^{-3}$. The total enthalpy of the PCM, $H_{\text{tot}}$, is composed of two parts: sensible enthalpy, $H$, and latent enthalpy, $\Delta H$:

$$H_{\text{tot}} = H + \Delta H$$ \hspace{1cm} (11)

$$H = H_{\text{ref}} + \int_{T_{\text{ref}}}^{T} c_p dT$$ \hspace{1cm} (12)

$$\Delta H = f L$$ \hspace{1cm} (13)

where $H_{\text{ref}}$ is the sensible enthalpy at the reference temperature, and $T_{\text{ref}}$ is the reference temperature.

The finite volume method is employed to solve the above mass, momentum, and energy conservation integral equations. The Pressure-Implicit with Splitting of Operators (PISO) algorithm is used for velocity-pressure coupling, and the Pressure-Ratio Explicit Simple Transport Operator (PRESTO!) scheme is considered for solving the pressure correction equation. Second Order Upwind differencing is applied to discretize the energy and momentum equations. The convergence criteria for continuity, momentum, and energy equations are set to $10^{-3}$, $10^{-3}$, and $10^{-7}$. The under-relaxation factors for pressure, momentum, energy, and liquid fraction are set to 0.3, 0.7, 1, and 0.2, respectively. The commercial software ANSYS Fluent 2020 R2 is utilized to solve the control equations with appropriate initial and boundary conditions.

### 3.2 Boundary conditions and evaluation index

The outer pipe is assumed to be adiabatic, neglecting any heat loss to the environment. The internal flow passages are treated as constant wall temperature at 45°C. The initial temperature of the bulk of PCM is 27°C (1 K below the PCM’s melting temperature). The boundary conditions are as follows:

- The heated inner tube walls' surface temperature is constant at $T_{\text{hot}}=318.15K$;
- The outer tube wall is adiabatic: $\frac{\partial T}{\partial n} + \frac{2}{r^2} = 0$;
- The initial temperature is $T(x,y,0) = 300.15K$.

The effective power density, $\rho_{ts}$, is proposed in this study to evaluate the performance of each type. It is defined as:

$$\rho_{ts} = \frac{\frac{dq}{dt} V_{\text{PCM}}}{V_{\text{tot}}}$$ \hspace{1cm} (14)

where $\frac{dq}{dt}$ represents the latent heat and sensible heat absorbed by the unit mass of phase change energy storage material over time, $V_{\text{PCM}}$ represents the volume of the phase change
energy storage material, $V_{tot}$ represents the total volume of the thermal storage unit, which is the sum of the volume of the phase change energy storage material and other components. Therefore, maximizing the $\rho_{ts}$ requires enhanced heat transfer without over-increasing the sizes of the fluid tunnels and the TES unit, which results in optimized flow channel design without increasing the manufacturing costs and unit footprint.

4 Results and discussion

4.1 Model validation

4.1.1 Grid and time step independence validation

The independence verification on the grid quantity and time step was partitioned using the MultiZone Quad/Tri method, and the simulations were carried out for grid quantities of 70,000, 58,800, and 35,000, respectively. The results of the grid quantity independence verification are presented in Figure 4, indicating that when the grid quantity exceeded 58,800 (with a grid size of 0.3 mm), the temperature curve remained almost unchanged, with a mere 0.27% error in total melting time.

![Figure 4 Mesh independence analysis.](image)

To verify the time step independence of the simulation, this study conducted simulations with time steps set at 0.05 s, 0.1 s, and 0.2 s, respectively. The results are illustrated in Figure 5. It was observed that when the time step was less than 0.1 s, the temperature curve exhibited little variation with changes in the time step size. By analyzing the model’s total melting time, it was found that a time step less than 0.1 s resulted in an error of less than 0.01%. Based on these analyses, the selection of a grid size of 0.3 mm and a time step of 0.1 s for the calculations in this study is considered reasonable.
4.1.2 Reliability validation

To validate the accuracy of the numerical simulations in this study, experimental results from Dhaidan et al. [45] were selected for comparison. Figure 6 shows the photographs of the melting front of PCM recorded by Dhaidan et al. [45] in their experiments. The numerically predicted melting front by Dhaidan et al. [45] (blue lines) and the current study (red lines) are overlaid on the photographs for comparison. It can be observed that the red contour of the simulation results in this study agrees well with the experimental and the numerically predicted melting front by Dhaidan et al. [45] of the results. This good agreement is seen at different stages of the melting process (20 – 70 min).

Figure 6 Comparison of the experimental results of Dhaidan et al. [45] with the numerical prediction obtained in the current study.
Additionally, the numerical model of the current study was validated against the experimental results of Tan [46], the numerical simulation results of Hosseinizadeh et al. [42], and the experimental results of Idris et al. [47]. The comparisons are presented in Figure 7. Tan [46] used paraffin n-octadecane as the PCM filled in a spherical thermal storage unit made of glass with a relatively high thermal conductivity. The PCM was initially set at 27°C, and a constant-temperature water bath at 45°C was used to heat the thermal storage unit. Based on Tan’s experiment [46], Hosseinizadeh et al. [42] constructed a numerical model to predict the experimental results. From Figure 7, it can be seen that the numerically predicted overall melting trend using the numerical model by the current study is in good agreement with Tan’s experiment [46], with a total melting time error of only 2.5% obtained from the numerical simulation in this study. Compared to the experimental results of Idris et al. [47], the melting trend predicted by the numerical model in this study also aligns very well with their experimental findings. The root mean square error (RMSE) of the temperature throughout the entire melting process is 0.33°C. Thus, it can be concluded that the model constructed in this study is capable of accurately predicting the PCM melting process.

4.2 Charging performance analysis of the bionic-lotus root type

4.2.1 Heat transfer performance comparison

Figure 8 depicts the predicted liquid fraction distribution maps and streamline plots for single-tube, multi-tube, bionic-lotus root, and inverse bionic-lotus root. Overall, multi-tube, bionic-lotus root, and inverse bionic-lotus root exhibit a more uniform overall melting characteristic, effectively reducing the time required for complete melting. However, they also display significant differences in un-melted areas at different stages. Further analysis through streamline plots reveals the distinct influence of natural convection differences on the melting process.
In the first 600 seconds of the melting process, all four units rely primarily on heat conduction for heat transfer. This can be seen by the melting boundaries are aligned concentrically with the internal ellipses or circular pipes in all four units. During this stage, there are only thin layers of molten PCM surrounding the outer walls of the inner tubes, and heat convection is limited to the regions near the heated wall. At this point, the multi-tube type experiences a relatively even overall melting, while in the bionic-lotus root type and inverse bionic-lotus root type, there is a more concentrated un-melted PCM between the central channel and the two small elliptical channels on the sides. Figure 9 shows the time variation of the liquid fraction for all four types. During this stage, the multi-tube type demonstrates the fastest melting rate (defined as the slope of the Liquefaction curve versus time in Figure 9, followed by the bionic-lotus root type and the inverse bionic-lotus root type. As the melting progresses, heat conduction is gradually suppressed, and the melting rates of all types begin to decrease, with the single-tube type being the most affected.

After 1200 seconds, the melting boundary of the single-tube type still exhibits a concentric circle with the central pore. However, in the multi-tube, bionic-lotus root, and inverse bionic-lotus root types, the melting boundaries gradually merge due to the mutual influence of the internal pores. This enhances the natural convection, and the heat transfer mode shifts gradually from local convection to global convection, with heat convection gradually becoming the dominant mode of heat transfer. In the regions near the heated wall and at the boundaries where melting occurs, the liquid-phase PCM has a higher flow velocity compared to other liquid-
phase areas, and by 2400 seconds, a complete convection has already formed. Figure 10 shows a local velocity vector map in the bionic-lotus root type, where the elliptical pores facilitate the longitudinal convection of the liquid PCM on the outer side compared to circular pores. As depicted in Figure 9, it can also be observed that at around 1000 seconds, the liquid fraction of the bionic-lotus root type surpassed that of the multi-tube type and maintained the fastest melting rate until the completion of the melting process.

Around 3000 seconds, influenced by natural convection, the melting boundary of the single-tube type exhibits a pronounced upward bulging shape. However, a significant amount of un-melted PCM is still present near the inner wall of the outer tube. In the multi-tube, bionic-lotus root, and inverse bionic-lotus root types, natural convection almost dominates the entire region, and only a small amount of PCM remains unmelted in the central region. The bionic-lotus root type can melt the upper portion more rapidly. Due to the advantage of internal vertical natural convection, the unmelted PCM in the central region can also be melted faster.

In the early stages, the melting rates of the bionic-lotus root and inverse bionic-lotus root
types are similar. However, during the transition period from heat conduction to heat convection, the bionic-lotus root type exhibits a faster melting rate. Therefore, it can be concluded that the convective heat transfer intensity inside the bionic-lotus root type is stronger than that of the inverse bionic-lotus root type, and among the four types, the bionic-lotus root type has the fastest melting rate.

4.2.2 Melting performance comparison

Figure 11 presents a comparison of the complete melting times among the four types. In comparison to the single-tube type, the other three types exhibited drastic reductions in complete melting time of over 87.7%. Notably, the bionic-lotus root type had the shortest melting time, being 10.7% and 5.9% shorter than that of the multi-tube type and inverse bionic-lotus root types, respectively.

![Figure 11](https://ssrn.com/abstract=4648752)

**Figure 11** Comparison of the total melting time for the four types.

Figure 12 shows the average temperature versus time for the four types during the melting process. As can be seen from the figure, the average temperature of the single-tube type is always the lowest. In the early stages of melting, the average temperature of the multi-tube type rises the fastest among the other three types. This is because in the multi-tube type, the PCM is distributed more evenly and heated more uniformly, while in the bionic-lotus root type, there are regions with a larger area of PCM, making it difficult for heat to be conducted to the interior of those regions. However, with the development of heat transfer from conduction to convection, the average temperatures of the three types gradually become closer, but the average temperature of the inverse bionic-lotus root type remains the lowest. Around 2200 seconds, the bionic-lotus root type began to exhibit higher average temperatures. This is attributed to its unique structure that enhances heat transfer in the bottom melting dead zone of the thermal storage unit. Additionally, it benefits from stronger natural convection advantages. Until complete melting of the bionic-lotus root type, the average temperatures of the single-tube, multi-tube, bionic-lotus root, and inverse bionic-lotus root types are respectively 304.3 K, 316.8 K, 317.5 K, and 316.6 K, with the bionic-lotus root type having a 13.2 K higher average temperature than the single-tube type.
Figure 12 Time-dependent curves of the average temperature for the four types.

Figure 13 shows the temperature and logarithmic effective power density (\(\lg \rho_{ts}\)) curves for the four types, and only the data for the first 8500 seconds is shown here due to the focus on the multi-tube, bionic-lotus root, and inverse bionic-lotus root types.

The \(\lg \rho_{ts}\) of the multi-tube type is the highest at the beginning of melting until around 120 seconds, when it gradually falls below that of the bionic-lotus root and inverse bionic-lotus root types. At this point, the effective power density of the multi-tube, bionic-lotus root, and inverse bionic-lotus root types are all around \(180 \text{ W kg}^{-1}\). Before 2400 seconds, the \(\lg \rho_{ts}\) of the bionic-lotus root and inverse bionic-lotus root types are almost the same, with the bionic-lotus root curve slightly higher, and the single-tube type has the lowest value.

After 3000 seconds, the \(\lg \rho_{ts}\) of the multi-tube, bionic-lotus root, and inverse bionic-lotus root types all begin to decrease significantly, with the bionic-lotus root type being the first to show this phenomenon. By observing the temperature curves at corresponding times, it can be seen that the \(\lg \rho_{ts}\) of these three types all decrease significantly when the average temperature reaches around 315 K. This phenomenon may be related to the small temperature difference during the later stages of melting and the large convective heat transfer resistance.

After melting is complete, the \(\lg \rho_{ts}\) curves of the multi-tube, bionic-lotus root, and inverse bionic-lotus root types all show a downward trend with a constant slope. This is because after melting is complete, the PCM absorbs heat entirely through sensible heat storage, and the heat absorbed is no longer affected by the latent heat term. At this stage, the heat transfer is almost entirely due to convective heat transfer, and all liquid PCM convects to absorb the heat provided by the heating wall and convert it into sensible heat stored in the PCM.
Figure 13 Time-dependent curves of the average temperature and logarithmic effective power density for the four models. The dashed line represents the complete melting time for the corresponding colored curve.

Taking the integral of the effective power density from the initial time to the complete melting time of each type and dividing by the total melting time, we obtain the average effective power density, as shown in Figure 14. The average effective power density of the bionic-lotus root type is the highest, with an increase of 761.1%, 15.1%, and 6.1% compared to the single-tube, multi-tube, inverse bionic-lotus root types.

Figure 14 Average effective power density of four types.

4.3 Optimization of the Bionic-lotus root type

4.3.1 Optimization of oval tube TES according to industrial standards

In order to further explore the potential of the bionic-lotus root concept in actual engineering practice and reduce costs, this study constructed a bionic-lotus root type with off-the-shelf piping products that follow the industrial manufacturing standards of stainless steel pipes in GB/T 3094-2012 and GB/T 17395-2008. To make the industrial standard type similar
to the bionic-lotus root structure and to have the same PCM volume and total flow channel cross-sectional area, the size parameters of the elliptical tube were determined according to the industrial standards, under the premise of the outer tube diameter remaining unchanged. Figure 15 shows the specific dimensions of the industrial standard type.

![Figure 15 Dimensional parameters of the bionic-lotus root type under industrial manufacturing standards.](image)

A set of elliptic tube parameters with the smallest error in industry standards was selected, and the error of PCM volume was 0.64%. The selected parameters are as follows: the major axis of the large ellipse is $A_1=24\text{mm}$, the minor axis is $B_1=12\text{mm}$, the major axis of the small ellipse is $A_2=12\text{mm}$, the minor axis is $B_2=6\text{mm}$, and the inner circle diameter is $D_2=18\text{mm}$. The two symmetrical ellipses above are defined as ellipse 1, the two ellipses in the middle are ellipse 2, and the two ellipses below are ellipse 3. Other parameters are the same as those mentioned earlier.

This study uses RSM for the optimization design of LHTES. In order to improve the overall melting performance, four factors, namely, the distance from the center of ellipses 1, 2, and 3 ($H$), the rotation angle of ellipse 1 ($\theta_1$), ellipse 2 ($\theta_2$), and ellipse 3 ($\theta_3$), were selected as the variable factors to determine the target. Here, $H \in [31, 37]$, $\theta_1$, $\theta_2$, and $\theta_3 \in [-45, 45]$. At the same time, it is considered that the left half and right half of the model are symmetrical, and the clockwise direction is taken as the negative direction for the self-rotation direction of the ellipses on the right half. The average effective power density of the PCM in the unit is taken as the optimization target, and the Box-Behnken design (BBD) is used to design the influencing factors, with specific parameters shown in Table 3. According to the suggestion of BBD, 29 trial runs were carried out, and the RSM analysis scheme was used for simulation and simulation.

<table>
<thead>
<tr>
<th>Factors</th>
<th>Symbol</th>
<th>Factor levels</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rotation angle of ellipse 1</td>
<td>$\theta_1$</td>
<td>-45, 0, 45</td>
</tr>
<tr>
<td>Rotation angle of ellipse 2</td>
<td>$\theta_2$</td>
<td>-45, 0, 45</td>
</tr>
<tr>
<td>Rotation angle of ellipse 3</td>
<td>$\theta_3$</td>
<td>-45, 0, 45</td>
</tr>
<tr>
<td>Distance from the center of ellipses 1, 2, and 3</td>
<td>$H$</td>
<td>31, 34, 37</td>
</tr>
</tbody>
</table>
Through variance analysis and model diagnosis, it was found that the current prediction model has statistical significance, and the P-value of the model is less than 0.0001. The correlation between variables and indicators is high, indicating that the model is significant. The adjusted R² of the model fitting degree is 96.69%, indicating that the actual results and the predicted results match well. This suggests that it can be used for further research and optimization analysis of LHTES.

Figure 16 Response surface plot of the influence of the factors on the optimization objective.

Figure 16 shows the relationship between four optimization parameters and the average
effective power density of PCM, where the red region represents the area with the maximum average effective power density.

First, a cross-comparison of Figure 16(a), (b), and (c), Figure 16(a), (d), and (e), and Figure 16(a), (d), and (f) was conducted to study the impact of $\theta_1$, $\theta_2$ and $\theta_3$ on the average effective power density, respectively. It was found that all three factors have a common characteristic: the average effective power density of PCM first increases and then decreases as the self-rotation angle increases. This trend is more significant for $\theta_2$ and $\theta_3$, while the impact of $\theta_1$ is relatively small. This indicates that there exists an optimal value for these three factors in the middle, and they need to be optimized.

Next, a cross-comparison of Figure 16(c), (e), and (f) was conducted to study the impact of $H$ on the average effective power density of LHTES. The results showed that $H$ is between 31 and 37, and the average effective power density of PCM is larger on both sides and smaller in the middle. It can be seen that the change in $H$ has a more significant effect on the average effective power density of the unit than the other three factors. As mentioned earlier, due to the effect of thermal convection, the temperature of the upper part of the PCM is much higher than that of the lower part. Changing $H$ can make the melting process more uniform and reduce the difference in the melting process between the upper and lower parts. Therefore, it needs to be optimized.

The RSM also established a fluid-structure coupling function for optimizing each research parameter:

$$\bar{\rho}_{ts} = 65.38 - 0.4457\theta_1 - 1.34\theta_2 + 0.4684\theta_3 + 6.31H$$

$$+ 1.48\theta_1\theta_2 - 0.3691\theta_1\theta_3 + 0.3392\theta_2H + 1.08\theta_2\theta_3 + 0.0599\theta_2H + 0.1632\theta_3H$$

$$- 0.6357\theta_1^2 - 1.96\theta_2^2 - 3.60\theta_3^2 - 10.34H^2$$  \hspace{1cm} (15)

Based on the target equation fitted by data analysis, it can be found that the quadratic coefficient of the factor $H$ is the largest, followed by $\theta_3$ and $\theta_2$, and finally $\theta_1$. Therefore, the significance levels of the four factors affecting the average effective power density of the LHTES unit are as follows: $H > \theta_3 > \theta_2 > \theta_1$. The main reason for this result is that the change in elliptical distance significantly affects the convection of the phase-change material region. A suitable distance can make heat conduction penetrate deeper into the outer region of the LHTES, leading to a shorter melting time of the phase-change material.

The multi-objective optimization results based on the RSM showed that the recommended solution for achieving the maximum average effective power density of the LHTES unit is: $\theta_1 = -41.5^\circ$, $\theta_2 = -31.1^\circ$, $\theta_3 = 2.7^\circ$, $H = 34.9$mm. The average effective power density of the PCM obtained by this solution is 68.4 W kg$^{-1}$. Numerical simulations using the recommended dimensions showed that the error between the maximum average effective power density predicted by RSM and the simulation data is 1.0%, indicating that the optimization solution is reasonable. The average $\bar{\rho}_{ts}$ of the industrially standard optimized type exceeds that of the bionic-lotus root type by 6.6%, and the complete melting time is reduced by 6.1%, as shown in Figure 17. The optimization effect of the RSM is notably significant.

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Figure 17 Comparison of average $\rho_{ts}$ and complete melting time between bionic-lotus root type and industrial standard optimized type LHTES.

4.3.2 Performance enhancement by the industrial standard optimized type

As shown in Figure 18, each subfigure shows a liquid fraction distribution map on the left and a streamline plot on the right. From a geometric perspective, the main difference between the two types mentioned above lies in the size of the internal channels and the rotation angle of the holes, with the industrial standard optimized type having downward-rotating holes.

Figure 18 Liquid fraction distribution maps (on the left) and streamline plots (on the right) for the bionic-lotus root and industrial standard optimized type.
During the first 600 seconds of the melting process, the melting interface of both types exhibited a concentric state with the internal pipes. After about 1200 seconds, more liquid PCM formed, and the heat convection became more intense. It was clear that the bionic-lotus root melted faster, which is attributed to the larger cross-sectional area of the internal channels in the large elliptical holes of the bionic-lotus root and the larger heat transfer area, which can transfer heat more deeply into the central area. By 2400 seconds, global convection had already formed, and the melting rate of the bionic-lotus root was still faster. The temperature distribution map also showed that the temperature of the bionic-lotus root was higher.

By 3000 seconds, the upper part of the bionic-lotus root type had basically melted completely, leaving only a concentrated solid PCM at the bottom. However, small amounts of unmelted PCM were at the upper edge and bottom of the industrial standard optimized type. Natural convection almost occupied the entire area, indicating that the overall uniformity of the melting of the PCM on the right side was better. By 3600 seconds, the melting trend of both types was still similar to that at 3000 seconds. The industrial standard optimized type had a similar degree of melting, but the solid PCM that had not melted was dispersed more evenly in the LHTES, and its degree of melting tended to surpass that of the bionic-lotus root type (which will be confirmed in Figure 19). The main reason for this may be that the tube distribution of the industrial standard optimized type makes the distribution of the dead volume more dispersed, thereby improving the average melting speed.

The industrial standard optimized type melted slower than the bionic-lotus root type in the early stages of the melting process. However, after parameter optimization, the industrial standard optimized type had a shorter total melting time and more uniform melting. At the same time, it also accelerated the melting speed of the lower un-melted area in the later stages of the melting process, providing a promising solution to the long-standing problem of slow melting speed in the lower part of the shell-and-tube LHTES.

Figures 19 and 20 show the curves of the liquid fraction, average temperature, and logarithmic effective power density of the two types. From Figure 19, it can be seen that the complete melting times for the bionic-lotus root and industrial standard optimized type were 4062s and 3827s, respectively. In the early stage, the liquid fraction curve of the bionic-lotus root had a steeper slope, but its slope decreased faster. As the melting progressed, the slope of the liquid fraction curve of the industrial standard optimized type gradually surpassed that of the bionic-lotus root at a certain point, which also indicated that the melting speed of the industrial standard optimized type was faster in the later stage of melting.
Figure 19 Time-dependent curves of the liquid fraction for the bionic-lotus root and industrial standard optimized type.

From Figure 20, it can be seen that the average temperature of the bionic-lotus root was higher. However, in terms of average liquid fraction and logarithmic effective power density, the industrial standard optimized type exhibited superior thermal storage performance compared to the bionic-lotus root, especially after about 3000s. The result indicates that the bionic lotus-root-inspired shell-and-tube LHTES unit concept can achieve satisfactory performance in real industrial applications.

Figure 20 Time-dependent curves of the average temperature and logarithmic effective power density for the bionic-lotus root and industrial standard optimized type.

5 Conclusion

This research aimed to enhance the thermal storage performance of shell-and-tube phase change thermal storage units by addressing the issue of dead zones. The proposed solution, named bionic-lotus root type, was compared with three other shell-and-tube thermal storage unit types. The findings revealed significant advantages of the bionic-lotus root type across multiple dimensions. It showcased a reduced melting time, higher average phase change
material temperature post-storage, and increased average effective power density. Furthermore, following the industrial standards and employing the RSM, optimization efforts led to the creation of an industrial standard optimized type using off-the-shelf piping products. Notably, this optimized type displayed the highest average effective power density. In summary, the primary conclusions drawn from this study encompass:

- Compared to the single-tube type, the melting time of the other three types (multi-tube, bionic-lotus root, and inverse bionic-lotus root) was reduced by over 87.7%, with the bionic-lotus root having the shortest melting time, which was 10.7% and 5.9% shorter than that of the multi-tube and inverse bionic-lotus root, respectively.
- When the bionic-lotus root type fully melted, the average temperatures of the four types were 304.3 K, 316.8 K, 317.5 K, and 316.6 K, respectively, with the bionic-lotus root having a 13.2 K higher temperature than the single-tube type.
- The average effective power densities of the single-tube, multi-tube, bionic-lotus root, and inverse bionic-lotus root were 7.5 W kg\(^{-1}\), 55.7 W kg\(^{-1}\), 64.1 W kg\(^{-1}\), and 60.5 W kg\(^{-1}\), respectively. Compared with the other three types, the bionic-lotus root increased the average effective power density by 761.1%, 15.1%, and 6.1%, respectively.
- For the industrial type, based on the multi-objective optimization of the RSM, the recommended scheme for the maximum average effective power density of the unit is \(\theta_1=41.5^\circ\), \(\theta_2=31.1^\circ\), \(\theta_3=2.7^\circ\), and \(H=34.9\) mm. The average effective power density obtained by this scheme is 68.4 W kg\(^{-1}\), which is 6.6% higher than that of the bionic-lotus root, 818.12% higher than that of the single-tube type, and 22.7% higher than that of the multi-tube type, with a total melting time 5.8% lower than that of the bionic-lotus root type.

The results prove that the bionic-lotus root concept is a promising solution to melting dead zones at the bottom of the shell-and-tube LHTES unit. Future studies can focus on experimental validation of the design and work on more types of PCMs, such as water-based PCMs that may increase the density while melting. A techno-economic analysis can also be carried out to evaluate its cost-effectiveness, especially in specific industrial applications.

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