

THERMAL PERFORMANCE INVESTIGATION AND OPTIMIZATION OF A NOVEL V-SHAPED FINNED LATENT HEAT STORAGE UNIT

by

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The low thermal conductivity of PCM limits the thermal performance of heat storage devices, hindering their widespread application in engineering. To address this challenge, this paper focused on the fin design of a horizontal triplex-tube latent heat storage device. A novel V-shaped fin with a secondary arrangement is proposed. Numerical simulation is employed to investigate the effects of fin angle, fin length, and heat transfer fluid temperature on the thermal performance of the storage system. Moreover, the fin parameter is optimized by the response surface method. The results demonstrate that, compared to traditional rectangular fins, both the primary and secondary V-shaped fins can significantly enhance the thermal performance. The secondary V-shaped fin has the most significant effects, reducing the melting time of the PCM by 48.5%, and increasing the average heat storage rate by 83.46%. Additionally, the thermal performance of the heat storage unit exhibits a trend of initially increasing and then decreasing with the increase of fin angle and length, suggesting the existence of optimal structural parameters. Consequently, the optimized secondary V-shaped fin reduces the melting time by 52.9% and increases the average heat storage rate of phase change materials by 104.48%.

Key words: PCM, V-shaped fin, melting characteristics,
latent heat storage, response surface method

Introduction

The increasing share of renewable energy, especially solar power, offers the greatest potential for CO₂ reduction and climate protection in energy industry. However, the supply of renewable energy varies greatly due to the unpredictable nature of the weather. Currently, thermal energy storage is widely recognized as a competitive solution to this challenge. It involves storing surplus thermal energy and making it available at a later time as needed. In particular, latent thermal energy storage (LTES) system with PCM has received much attention due to its high density of energy storage, stable output, and practical versatility [1]. Nevertheless, the low thermal conductivity of PCM can observably degrade the rate of heat storage and release. In this regard, various approaches to enhance the thermal performance of PCM, such as the incorporating nanoparticles [2] with high thermal conductivity into PCM and the cascade utilization

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of multiple PCM [3, 4], have been explored. Notably, embedding metal fins in PCM to increase heat exchange surface area has also been proposed, mainly because of its relative simplicity and effectiveness compared to others [5].

Since fin geometry can have insignificant effects on the thermal performance, various fin shapes with different layouts for phase change enhancement of LTES system have been introduced. Safari *et al.* [6] compared the effects of straight and bifurcated fins on melting behavior of paraffin. In their study, both the fin configuration and fin parameters at different melting temperatures were considered. It is found that the bifurcated fin outperformed the straight fin for all cases considered. Ren *et al.* [7] designed a new rotating case of Y-shaped cross-fin triplex-tube latent heat system. They found that the increase of fin length can significantly improve the thermal energy storage efficiency of TTES, however, once the fin depth, L , exceeds 13 mm, the heat transfer indices of TTES show only marginal improvement, particularly in thermal energy storage rate and density. In recent years, many scholars have endeavored to the development of bionic fin for latent heat storage (LHS) systems. For example, Ren *et al.* [8] proposed an irregular snowflake fin structure and optimized the fin structure. The results showed that the melting time using the snowflake fins was reduced by 45.92% compared with that of traditional fin. Wu *et al.* [9] numerically investigated the thermal energy release performance with the spiderweb-like fin for a LHS device by the enthalpy-porosity method. They found that the designed fin could effectively improve the overall solidification of PCM than the plate fin. However, the poor fin efficiency related to the proposed fin brought a larger radial temperature gradient of LTES device.

Among different popular fin shapes, V-shaped fin is one of the most visible options due to its favorable performance. The interesting features of this type of fin are the simple shape and easy processing. Moreover, Lorenzini *et al.* [10] found that the thermal resistance of V-shaped structure can be reduced by optimizing the fin shape. Alizadeh *et al.* [11] conducted a comparative study to assess the effectiveness of two enhancement methods, namely the using of V-shaped fins and the incorporation of nanoparticles, on enhancing the solidification process in a triple-tube energy storage system. The simulation results revealed that the optimized V-shaped fin was more reasonable to achieve the maximum capacity of energy storage. Lohrasbi *et al.* [12] utilized the multi-objective response surface method (RSM) to optimize the array and structural parameters of V-shaped fin for the solidification process of PCM in an LTES system. Taken energy storage capacity as optimal objective, the results indicated that the optimized longitudinal V-shaped fins could shorten the solidification process by 5.65 times compared with the absence of fins. Yao *et al.* [13] introduced a novel V-shaped fin structure for a triplex-tube LTES system. According to their study, the optimal fin design could reduce the melting time of the PCM by 31.92% under the optimal conditions. Guo *et al.* [14] suggested that the melting of PCM could be greatly accelerated by employing eccentric V-shaped fins with the reasonable arrangement. To uniform the heat transfer in the difficult-to-solidify region of PCM, Guo *et al.* [15] designed a novel two-stage V-shaped branch fin for a triple-tube LHS unit. They reported that fin length ratio primarily affected the evolution of phase interfaces.

As seen from the literature overview, attaching various bifurcated or biomimetic fins in PCM has been proven to enhance thermal storage performance of LTES device. However, some of these fins are relatively complex which may result in the increased thermal resistance and fin volumes. Therefore, the design of simpler fin (*i.e.* V-shaped fin) with better thermal performance for the LTES applications is definitely meaningful. Moreover, the majority of existing studies dealt with the design of V-shaped fins for specific LHS units. Few of them

emphasized further bifurcating the V-shaped fin for thermal performance improvement, particularly taking into account the dual-heat source in the triplex-tube latent LTES device. Accordingly, this study aims to propose a novel design of primary and secondary V-shaped fin for a triplex-tube LTES device by numerical simulation. The concept of a *dual heat source* is introduced to analyze the effects of fin structural parameters on thermal storage performance, and the optimization of fin parameters is conducted through the RSM. The outcome of this study is expected to provide valuable guidance on fin optimization of the horizontal triplex-tube LTES unit.

Model establishment and numerical methods

Physical model

The novel V-shaped fins for the LTES unit are developed as shown in fig. 1(a). The straight fins are originally designed. Four baseline fins are attached to the surfaces of the inner and middle tubes, respectively. Thereafter, each straight fin is replaced with the V-shaped fin, namely the primary V-shaped fins. The thickness of each fin is half of the straight fin, while maintaining the same fin length and location to further increase the heat transfer area of fins, each primary V-shaped fin is further grown into two small secondary V-shaped fins. These two derived fins are distributed symmetrically on surfaces of tubes with the same circumferential position. For all cases, the fin volume remains fixed. The fin length is, L , and the fin thickness is W , the fin angle is denoted as, θ , with subscripts 1 and 2 representing the inner and outer sides respectively, as showed in fig. 1(b).

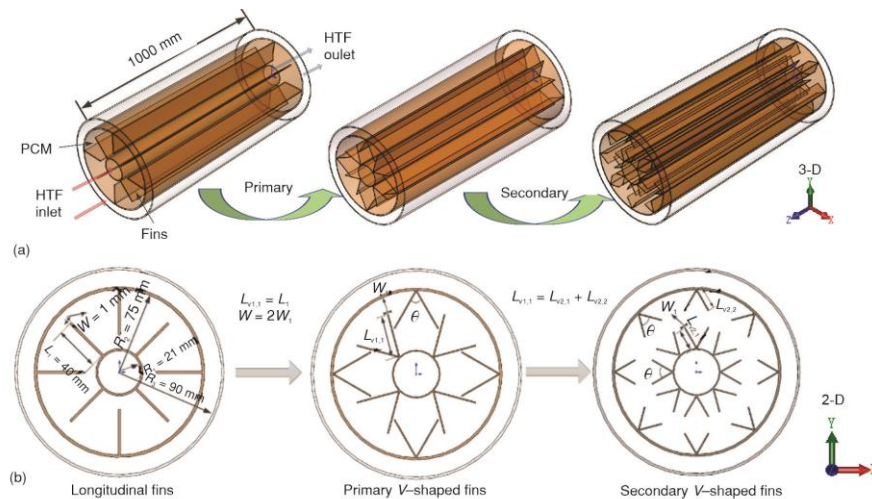


Figure 1. The V-shaped design schematic; (a) 3-D model schematic and (b) 2-D model schematic

In the present study, the triple-tube LTES unit is used for solar energy storage. It consists of three concentric copper tubes with diameters of 180 mm, 150 mm, and 42 mm for the outer, middle, and inner tubes, respectively. The PCM of paraffin RT (RT35) is encapsulated in the middle tubes. Two streams of high temperature water flow through the inner and outer tubes, which are considered as dual heat sources to transfer heat with PCM. The thermal properties of each material are shown in tab. 1.

Table 1. Physical properties of each material [16]

Physical properties	PCM (RT35)	Fin (Copper)
Density [kgm^{-3}]	770	8978
Specific heat [$\text{Jkg}^{-1}\text{K}^{-1}$]	2000	381
Thermal conductivity [$\text{Wm}^{-1}\text{K}^{-1}$]	0.2	387.6
Latent heat [kJkg^{-1}]	170	–
Dynamic viscosity [$\text{Pa}\cdot\text{s}$]	0.023	–
Thermal expansion coefficient [K^{-1}]	0.0006	–
Solid phase temperature [K]	302	–
Liquid phase temperature [K]	308	–

Governing equation

In this study, the enthalpy-porosity method [8-10] was adopted for simulating the melting process of RT35. This method utilizes enthalpy and temperature as the variables to establish uniform governing equations, which can effectively reduce the cost of numerical calculation. Based on the enthalpy-porosity model, some assumptions are made as:

- The liquid PCM is laminar and incompressible flow [8].
- The volume change of the PCM, viscous dissipation, and heat loss at the outer surface of the tube are ignored [17].
- The thermal properties of PCM remain constant at the operating temperature, while its density follows the Boussinesq approximation [18].

Considering above mentioned assumptions, the governing equations for the 2-D computational domain are given as [18]:

Continuity equation

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

Momentum equation

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

The momentum source terms, \vec{S} , is represented as:

$$\vec{S} = \frac{(1 - \beta)^2}{(\beta^3 + 0.001)} A_{\text{mush}} \vec{V} \quad (3)$$

Energy equation:

$$\frac{\partial}{\partial t} (\rho H) + \nabla \cdot (\rho \vec{V} H) = \nabla \cdot (k \nabla T) - \rho L_H \frac{\partial \beta}{\partial t} \quad (4)$$

where A_{mush} is the constant for the mushy zone, it is set to 10^5 according to [19], \vec{V} [ms^{-1}] is the velocity vector, β – the liquid fraction, ξ [K^{-1}] – the thermal expansion coefficient of the PCM, T_{ref} [K] – the reference temperature, H [kJkg^{-1}] – the total enthalpy, ρ [kgm^{-3}] – the density, t [s] – the time, c_p [$\text{kJkg}^{-1}\text{K}^{-1}$] and μ [$\text{Pa}\cdot\text{s}$] are specific heat and dynamic viscosity, respectively, p [Pa] – the pressure, and L [kJkg^{-1}] – the latent heat of PCM.

The total enthalpy, H , is the summation of sensible enthalpy and latent enthalpy [20]:

$$H = h_{\text{ref}} + \int_{T_{\text{ref}}}^T c_p dT + \beta L \quad (5)$$

where β is the liquid fraction of PCM, which can be expressed by:

$$\beta = \begin{cases} 0, & T < T_{\text{solid}} \\ \frac{T - T_{\text{solid}}}{T_{\text{liquid}} - T_{\text{solid}}}, & T_{\text{solid}} < T < T_{\text{liquid}} \\ 1, & T_{\text{liquid}} < T \end{cases} \quad (6)$$

where T_{solid} [K] and T_{liquid} [K] are the temperature of solid and liquid PCM, respectively.

Initial and boundary conditions

The initial temperature of PCM and water is set to 297.15 K and 353.15 K, respectively:

$$\begin{aligned} t = 0, \quad T_{\text{PCM}} = T_{\text{ini}} = 297.15 \\ t > 0, \quad r = R_1, \quad T = T_{\text{HTF-w}} = 353.15 \end{aligned} \quad (7)$$

where T_{ini} is the initial temperature and $T_{\text{HTF-w}}$ – the wall temperature of the conduit through which the heat transfer fluid (HTF) flows.

The temperature of the outer tube wall is set to be adiabatic:

$$r = R_3, \quad \frac{\partial T}{\partial r} = 0 \quad (8)$$

The heat transfer between the fins and the PCM is considered continuous, and the contact surface is governed by a coupled boundary condition:

$$\begin{aligned} T_{\text{fin}} = T_{\text{PCM}} \\ -\lambda_E \frac{\partial T_{\text{fin}}}{\partial \vec{n}} = -\lambda_{\text{PCM}} \frac{\partial T_{\text{PCM}}}{\partial \vec{n}} \end{aligned} \quad (9)$$

Evaluation indicators

The performance evaluation indexes for the PCM include the total heat storage and the average heat storage rate, which are expressed.

The heat storage capacity is given by [19]:

$$Q_{\text{tot}} = mc_p (T - T_{\text{ini}}) + mL\beta \quad (10)$$

where Q_{tot} [kJ] is the total heat storage capacity and m [kg] – the mass of PCM.

The average heat storage rate is expressed as [21]:

$$\psi = \frac{Q_{\text{tot}}}{t} \quad (11)$$

where t [s] is the total melting time of PCM.

Model validation

To have a robust and effective result, the independent analysis of numerical simulations is firstly performed using three total mesh numbers (11386, 23880, and 42804) and time steps (0.01, 0.05, and 0.1 seconds). The verification results are shown in fig. 2. As shown in fig. 2(a), there is a change magnitude of 0.08% and a very small relative error. Hence, the grid number of 23880 provides higher accuracy. Additionally, fig. 2(b) shows that for time steps of 0.01 and 0.05 seconds, the relative error with a change magnitude of 0.016% after 500 seconds. Therefore, a grid number of 23880 and a time step of 0.05 seconds are selected.

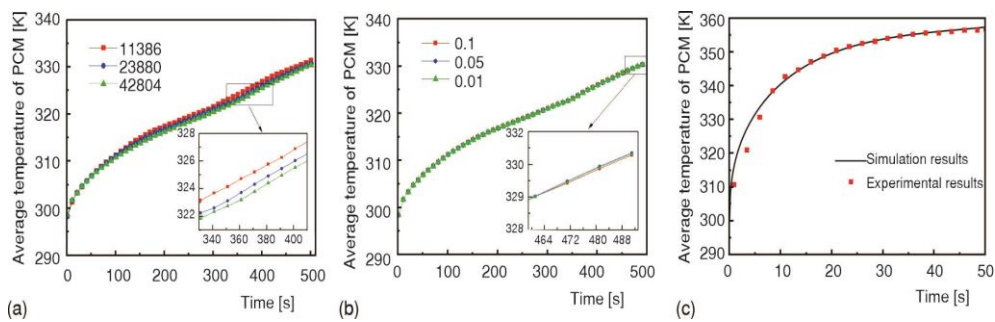


Figure 2. (a) Verification of grid independence, (b) time step independence, and (c) experimental validation of PCM melting by [22]

To validate the reliability of the simulation results, a comparison is then made between the numerical predictions from the present model and experimental data from Mat *et al.* [22]. As can be seen in fig. 2(c), the maximum relative error between simulation results and test values is 2.55%. The verification results show that the numerical model of this paper is reliable.

Results and discussion

Comparative analysis of different cases

Figure 3 shows the cloud diagram of the liquid fraction and temperature in the LTES system with different fin structures. It can be seen that at the initial duration stage ($t < 100$ seconds), since the fin has high thermal conductivity, the PCM contacted with the fin and tube wall starts to melt in all cases. However, the thickness of the melted film is smaller, so heat transfer undergoes with heat conduction. As the increase in amount of melted PCM, the heat transfer process is gradually dominated by natural convection. Since liquid PCM has a higher temperature and lower density, it moves upwards under the action of thermal buoyancy, leading to the accumulation of a large amount of liquid PCM in the top part of the heat storage unit. After $t > 200$ seconds, irregular molten zones appear in all structures containing PCM. With the proceeding of PCM melting (*i.e.*, $300 < t < 400$ seconds), the liquid fraction and temperature distribution reveals distinct characteristics for the considered fins, with the unmelted core predominantly surrounded by the fin profile. For the secondary V-shaped fin, the PCM near the adjacent fins melts more rapidly, resulting in a very small annular solid region. Although the primary V-shaped fin can effectively partition the PCM into discrete melting regions, the liquid PCM zone is still large as shown in fig. 3. This is mainly because the natural convection is weakened in the area between the adjacent fins. As expected, the PCM with conventional rectangular fin melts locally near the heated wall boundaries, maintaining a more uniform and predictable melting profile due to the low distribution density of fins. At $t = 500$ seconds, it is

found that the melting of PCM with the secondary V-shaped fin is almost completed, while the traditional rectangular fins shows worsen performance. Overall, the secondary V-shaped fin stands out in enhancing heat transfer compared with other two fin configurations.

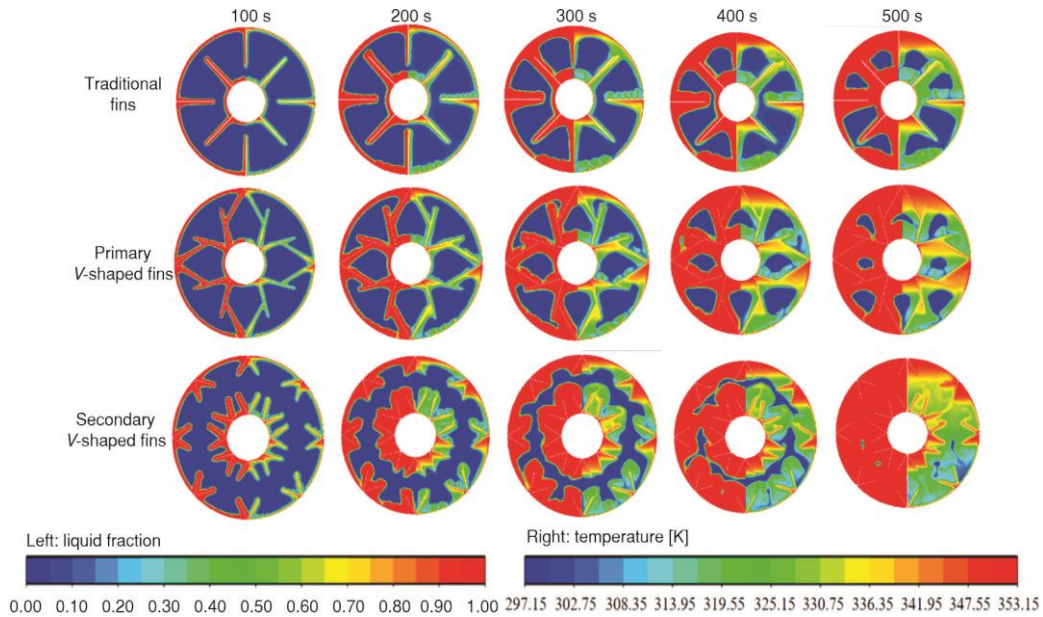


Figure 3. The liquid fraction and temperature evolution of PCM in different cases

Figure 4 illustrates the variation of PCM velocity ($t = 300$ seconds) in LTES system for different fin configurations. It can be found in fig. 4 that the velocity distribution of PCM varies significantly with various modifications of fin shape, due to the effect of natural convection during PCM melting. It can be found in fig. 4(a) that for the traditional rectangular fin, the vortexes occur mainly in the liquid PCM layer near the surfaces of fin and tube wall, where the liquid PCM content increases due to the large temperature difference between fin and PCM. Propelled by thermal buoyancy, the melted PCM moves upward and the natural convection is enhanced gradually. Consequently, the red liquid region that represents large flow velocity is enlarged in the upper part of the LTES unit, as shown in fig. 4(b). It is evident that the PCM in the middle part of the LHS unit is more prone to forming melting *dead zone* for the traditional

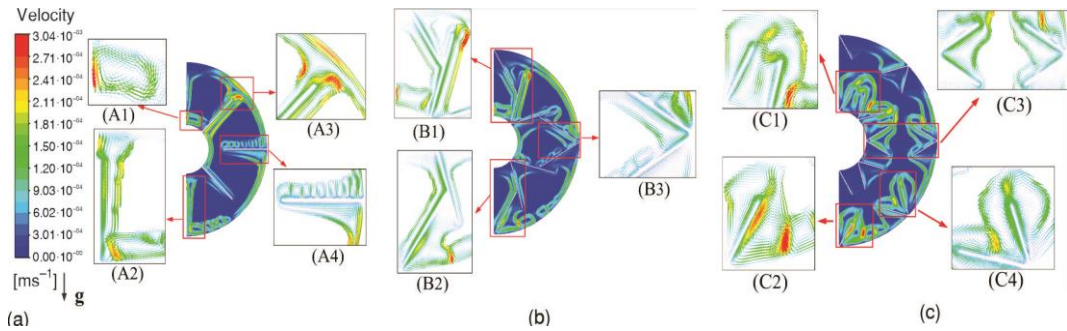


Figure 4. The flow vector diagram of PCM with different fins; (a) traditional rectangular fins, (b) primary V-shaped fins, and (c) secondary V-shaped fins

rectangular fin and the primary V-shaped fin. It implies that the long fin in these two cases may impede the development of natural convection during PCM melting. Conversely, the large velocity zones in fig. 4(c) appears throughout the LHS unit. This can be explained that the high distribution density of secondary V-shaped fin can promotes the heat transfereee towards different directions along the short fin body, and the heat reaches the fin tip more faster. Thus, both the natural convection and thermal conduction in the PCM are enhanced.

Figure 5 presents the thermal storage performance of the LTES system for three fin structures. As depicted in figs. 5(a) and 5(b), compared to the traditional rectangular fin, the primary and secondary V-shaped fins reduce the PCM melting time by 30.04% and 48.5%, respectively, and the increase in the average heat storage rate is 40.37% and 83.46%, respectively. Furthermore, the average temperature of PCM with the secondary V-shaped fin structure is the highest among the three considered cases. Compared with the primary V-shaped fins, the performance of the secondary is better with a reduction in melting time by 26.39% and an increase in the average heat storage rate by 30.7%. This phenomenon arises from the enhanced natural convection due to increased heat transfer area. Figure 5(c) illustrates the similar trends in the average heat storage rate for three fin structures. In the initial melting, the average heat storage rate decreased rapidly, and then reduced gradually with the increasing melting time. Since the secondary V-shaped fin structure exhibits superior performance, the effects of structural parameters on heat storage performance using this fin structure are investigated in the following sections.

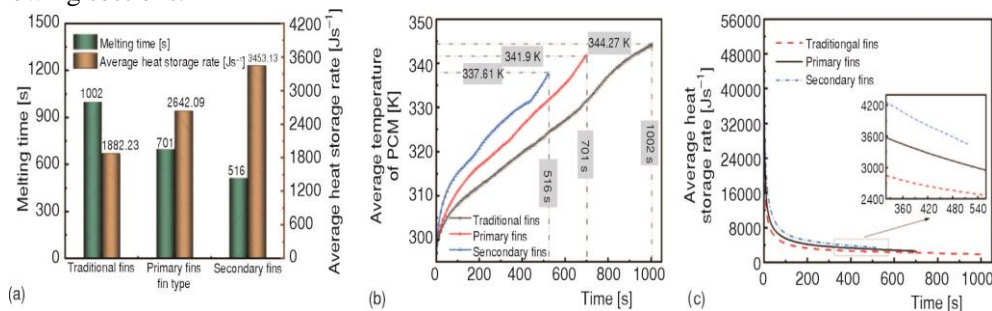


Figure 5. Melting characteristics of PCM for different fin structures; (a) thermal energy storage performance, (b) average temperature of PCM, and (c) trends in the average heat storage rate

Effect of fin angle on heat transfer performance

In this study, fin angles, θ , ranging from 30° to 90° are selected as the main design parameters. Figure 6(a) reveals a decrease in PCM melting time as the fin angle increases from 30° to 60° . However, the melting time increases when the angle further increases to 90° , suggesting that an excessively large angle for the sub-V fin is not optimal. Fin angles that are too small would result in non-uniform heat transfer and may lead to the formation of a melting *dead zone*. Since natural convection dominates the melting process in the middle and later stages, a large angle of the fins could weaken the natural convection process. Thus, the melting rate of PCM with the fin angle of 90° gradually slows down after 150 seconds. Figure 6(b) indicates that a fin angle of 60° yields the shortest PCM melting time and the highest average heat storage rate. This can be attributed to the more uniform heat distribution at 60° fin angle, resulting in a larger area for the liquid-phase zone during the early duration of PCM melting, thereby facilitating the advancement of natural convection.

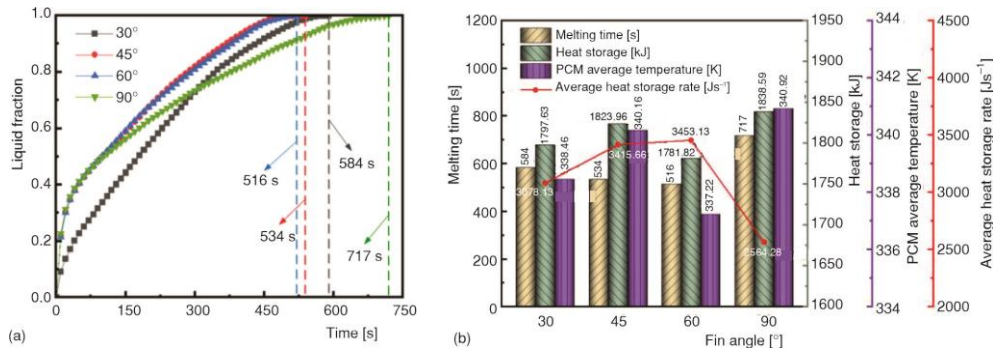


Figure 6. Melting characteristics of PCM with different fin angles; (a) liquid fraction and (b) effects of different fin angles on thermal storage performance

Effect of fin length on heat transfer performance

Figure 7 shows the influences of different fin length combinations on the thermal storage performance. Five fin length combinations, which are Case 1 (10 mm and 30 mm), Case 2 (15 mm and 25 mm), Case 3 (20 mm and 20 mm), Case 4 (25 mm and 15 mm), and Case 5 (30 mm, and 10 mm), are considered while keeping the total fin length remains constant. As can be seen in fig. 7(a), at first there is no visible difference in melting performance for all cases. However, in the later melting process, the melting rate of Case 1 is faster than Case 5, and Case 2 outperforms Case 4. Moreover, the fin structure in Case 3 exhibits the best performance. This reveals that the combination of fin lengths for the inner and outer tubes significantly influences the melting process during the middle and late stages. The improvement in melting performance can be obtained when the fin structure is configured as *short inner and long outer*. Figure 7(b) indicates that the average heat storage rate increases initially and then decreases with increasing fin length of the inner tube. Particularly, when the fin lengths of the inner and outer walls are equivalent, the average heat storage rate reaches its maximum.

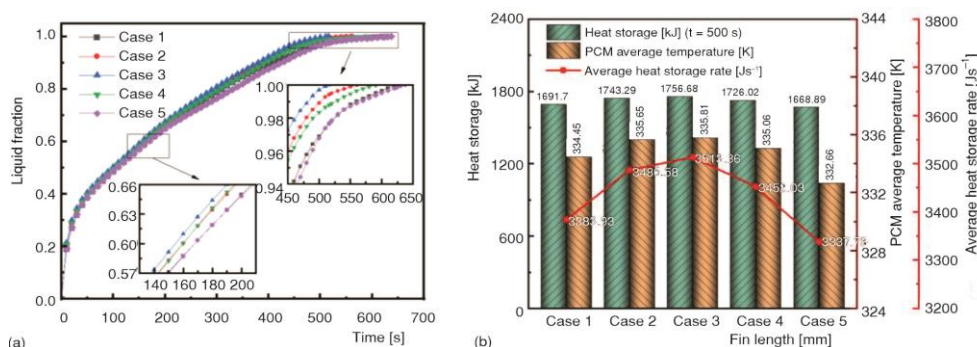


Figure 7. Effects of different fin length combinations on the thermal storage performance; (a) variations of liquid fraction and (b) variations of heat storage

Effect of HTF temperature on heat transfer performance

Figure 8 shows the impact of HTF temperature on heat storage performance. Five different HTF temperature values (343.15 K, 348.15 K, 353.15 K, 358.15 K, and 363.15 K) are considered. It can be seen that the higher the HTF temperatures, the shorter the melting times and the higher the average temperatures of PCM. This is because the fact that the higher the

HTF temperature, the greater the temperature difference for heat transfer, resulting in a stronger driving force for heat transfer. By elevating the temperature from 343.15 K to 363.15 K, the melting time reduces by 30.08%, and the average heat storage rate increases by 38.71%.

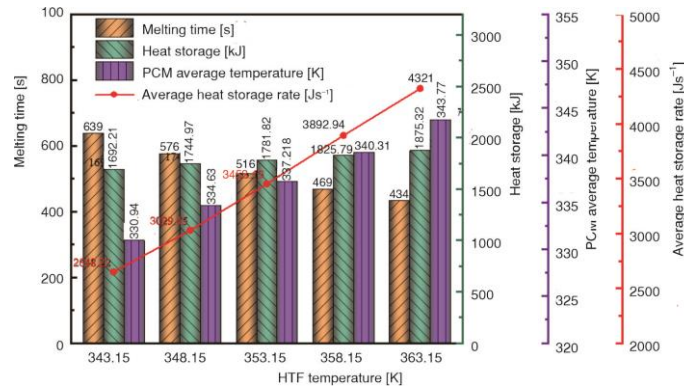


Figure 8. Effects of different temperatures of HTF on heat storage performance

Optimization of fin structure based on RSM

In this paper, the fin parameters of the secondary V-shaped fin were optimized by using RSM, including fin thickness (0.5~1 mm), fin length (15~25 mm), and fin angle (35°~65°). The total melting time of PCM is considered as the optimization objective. Box-Behnken design was employed to conduct 17 experiments.

Table 2 shows the results of variance analysis for the melting time of PCM. It can be seen that $F = 20.41$ and $P < 0.01$, indicating that the prediction model has the statistical significance. Moreover, the F -value of the fin length, B , is the biggest (*i.e.*, 36.30) among three factors, which means the PCM melting time is more sensitive to the change in fin length. This is because of the dominant influence of natural convection during the PCM melting, while fin length exerts main effects on this phenomenon.

Table 2. Results of variance analysis on PCM melting time

Source	Sum of squares	DoF	Mean squares	F -value	P -value	Significance
Model	27261.62	9	3029.07	20.41	0.0003	significant
Fin thickness, A	3951.60	1	3951.60	26.63	0.0013	
Fin length, B	5387.22	1	5387.22	36.30	0.0005	
Fin angle, C	57.24	1	57.24	0.3857	0.5542	
AB	23.52	1	23.52	0.1585	0.7024	
AC	8.12	1	8.12	0.0547	0.8217	
BC	78.32	1	78.32	0.5277	0.4911	
A^2	677.78	1	677.78	4.57	0.0699	
B^2	6682.22	1	6682.22	45.03	0.0003	
C^2	8924.01	1	8924.01	60.13	0.0001	

Considering the non-linear relationship between fin parameters and optimization objective, a regression equation for the melting time is fitting from data analysis, as shown in eq. (12). Based on the regression equation, the significance levels of various factors can also be

obtained. It can be seen that the coefficient of fin length, L , is the largest, followed by fin thickness, W , and fin angle, θ . It means that the fin length has the most significant impact on the melting time of PCM, which is consistent of the F -value results shown in tab.2.

$$\begin{aligned} \text{Melting time} = & 483.5 - 22.22W + 25.95L - 2.67\theta + 2.42WL + 1.43W\theta + 4.42L\theta + 12.69W^2 \\ & + 39.84L^2 + 46.04\theta^2 \quad (0.5 \leq W \leq 1, 15 \leq L \leq 25, 35 \leq \theta \leq 65) \end{aligned} \quad (12)$$

Figure 9 illustrates the results of response surface test with fin geometry and PCM melting time. The blue zone in fig. 9 has the smallest melting time of PCM. According to the response surface diagram, the reasonable values of fin structure for practice implement can be determined. It is evident in figs. 9(a) or 9(b) that when the fin angle (or fin length) is fixed, the melting time decreases with the increase in fin thickness. This trend mainly results from the reduction in PCM volume, therefore the heat storage capacity. The large value of fin thickness is suggested under this condition. Conversely, when fin length (or fin angle) increases, the melting time initially decreases, and then increases as show in figs 9(a) and 9(b). This means the there is an optimal value of fin length (or fin angle) which is required to be optimized. The fin length and fin angle are expected to be in the range of 15.3-20.5 mm and 49.0°-55.2°, respectively. Figure 9(c) demonstrates that the blue area is located almost at the center of the response surface diagram when the fin thickness remains constant. Accordingly, the favorable values for fin length and fin angle is narrowed to 17.6-19.3 mm and 48.0°-53.1°, respectively. This indicates that too large or too small values for the fin length and fin angle would lead to weak natural convection during the melting.

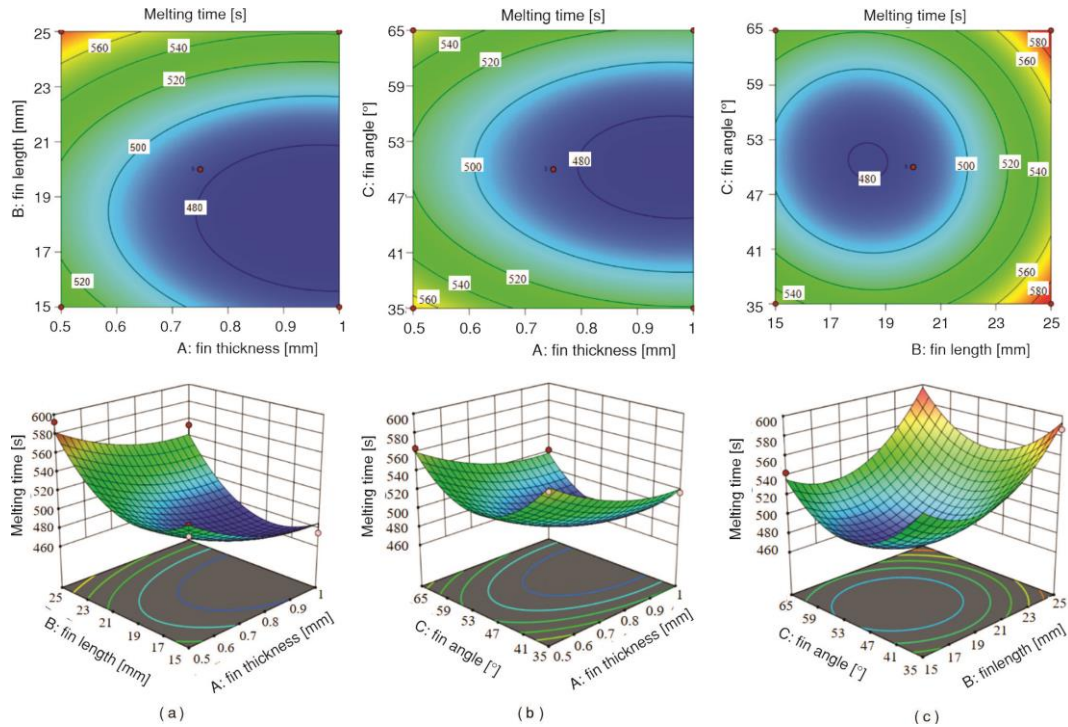


Figure 9. The results of response surface test; (a) fin angle is constant, (b) fin length is constant, and (c) fin thickness is constant

According to the aforementioned analysis, the optimal design for the fin angle, fin thickness, and fin length are 52.88° , 0.9 mm, and 19.1 mm, respectively. At the optimal conditions, the melting time and heat storage are 471.896 seconds and 1816.19 kJ, respectively, and the average heat storage rate is 3848.71 J per seconds. Compared with the traditional and pre-optimization fin configuration, the increase in the average heat storage rate is 104.48% and 11.46%, respectively, and the melting time reduction is 52.9% and 8.5%, respectively.

Conclusions

In this study, the numerical investigation of melting processes in a novel V-shaped finned triplex-tube LTES system is presented. The optimal structural parameters of the proposed fin are determined by employing RSM, and compared with the traditional rectangular fins to demonstrate its effectiveness. The main conclusions are as follows.

- Compared to the traditional rectangular fins, the primary and secondary V-shaped fin reduces the melting time by 30.04% and 48.5%, respectively, and boosts the average heat storage rate by 40.37% and 83.46%, respectively.
- The fin angle and fin length play crucial roles in determining the heat storage performance of the heat storage unit. Excessively long fins would impede the natural convection process, while unsuitable fin angles may disrupt the uniformity of heat transfer. Making a good trade-off between these parameters is essential for maximizing the heat storage efficiency.
- Under optimal conditions, the secondary V-shaped fins could reduce the PCM melting time by 52.9% and increase the average heat storage rate by 104.48% compared to traditional rectangular fins. Moreover, the melting time is reduced by 11.46% compared to the secondary V-shaped fins without optimization.

The outcomes of this study can provide valuable guidance on design of the horizontal triplex-tube LTES system. Further research (*i.e.*, the multi-objectives optimization of fin geometry, and small-scale experimental study, *etc.*) is still required for enhancing the performance of the proposed system.

Nomenclature

c_p	– specific heat, [$\text{Jkg}^{-1}\text{K}^{-1}$]
ΔH	– latent heat enthalpy, [J]
L	– latent heat, [Jkg^{-1}]
m	– mass, [kg]
p	– pressure, [Pa]
Q	– heat storage, [kJ]
S	– source term
T	– temperature, [K]
t	– time, [s]
W	– fin thickness, [mm]

Greek symbols

β	– liquid fraction
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θ	– fin angle, [$^\circ$]
ζ	– thermal expansion coefficient, [K^{-1}]
μ	– dynamic viscosity, [Pas]
ψ	– average heat storage rate, [Js^{-1}]
λ	– thermal conductivity, [$\text{Wm}^{-1}\text{K}^{-1}$]

Acronyms

HTF	– heat transfer fluid
LTES	– latent thermal energy storage
LHS	– latent heat storage
TTES	– tank thermal energy storage
RSM	– response surface method

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