Investigations of optimized fin structures in a compact thermal energy storage panel

Jinlong Xie¹, Wensheng Luo¹, Wei Zhang², Zhou Wu² and Hsiao Mun Lee¹,³

¹ School of Mechanical and Electrical Engineering, Guangzhou University, 230 Wai Huan Xi Road, Guangzhou Higher Education Mega Center, Guangzhou 510006, P.R.China;
² Yangzhong Shenyang Heat Exchange Equipment Co., Ltd., Nanzi Road, Xinba Technology Park, Yangzhongshi, Zhenjiang 225000, P.R.China
³ Email: hmlee@gzhu.edu.cn

Abstract. In this paper, the thermal behaviours of phase change material (PCM) within a compact thermal energy storage panel installed with conductive metal fins were numerically investigated. To improve the thermal performance of the metal fins, a topological structure optimization approach was employed to generate innovative fin structures with the so-called best heat conduction paths. The conventional plate fin structures with the same metal volume fractions as the optimized structures were considered for comparative studies. Four metal volume fractions including 0.05, 0.10, 0.15 and 0.20 were investigated. Results showed that at the same metal volume fraction, the optimized fin design can significantly enhance the melting of PCM as compared to the plate fin design. When $f=0.9$, the average PCM melting speeds obtained by the optimized fin structures were 1.10, 1.19, 1.31, and 1.69 times as that by the plate fin designs at the melt volume fractions of 0.05, 0.10, 0.15 and 0.20, respectively. Moreover, the optimized fin structure with a metal volume fraction of 0.10 can achieve the comparable melting speed as the plate fin design with a metal volume fraction of 0.15, and the melting speed achieved by the optimized fin structure with a metal volume fraction of 0.15 is even faster than that by the plate fin design with a metal volume fraction of 0.20.

1. Introduction

Thermal energy is classified as a low-grade form of energy that usually exists as the by-product of numerous applications such as energy conversion systems, high power-consumption equipment and electronic devices [1]. Thermal energy can be stored in the forms of sensible or latent heat by heating or cooling a bulk of heat storage materials. Among them, phase change materials (PCM) are widely used to store thermal energy due to their advantages in utilizing the latent heat through phase change process at a fixed temperature. However, the major drawback of PCM is their relatively low thermal conductivity which unfortunately suppresses their applications. As a result, many efforts have been devoted to enhancing the thermal conductivity of PCM by developing composite PCM packages [2].

In the past decades, the most practical and easy way to promote the thermal conductivities of PCM based thermal energy storage (TES) systems was to incorporate non-moving thermal structures such as metal fins, porous metal structures, metal foams. Among them, straight fin structures are most popular due to its easy fabrication, functional repeatability, and less maintenance [3]. Kamkari and Shokouhmand [4] experimentally investigated the effect of embedding straight fins on the melting speed of a PCM cavity. They found that the PCM melting speed can be significantly improved by 18%
and 37% respectively when one and three fins were applied. Khan et al. [5] conducted 2D simulations to investigate thermal behavior of a novel geometrical configuration of shell and tube PCM based TES system which utilized longitudinal fins surrounding the tubes to enhance the PCM melting speed. They found that the fin length rather than fin thickness has profound impact on the melting speed. Ji et al. [6] numerically investigated the effects of straight fins with different inclined angles on the heat transfer performance of a rectangular PCM based heat sink. They noted that the inclined angle of straight fins played a significant role in the natural convection heat transfer. A proper angled fin can induce strong natural convection heat transfer which eventually improved the temperature uniformity within the PCM domain and hence accelerated the PCM melting speed. Anish et al. [7] experimentally found in a horizontal shell-and-multi-finned-tube TES system that the melting induced natural convection significantly influenced the phase change heat transfer, and they suggested that a proper utilization of natural convection should be considered in optimizing the configurations of such TES systems. Apart from straight fins, recent efforts were also devoted to studying the performance of novel fin structures in PCM based TES units. Sciacovelli et al. [8] numerically investigated the thermal performance of a latent heat TES system using PCM incorporated with innovative Y-shaped fins. The geometry of Y-shaped fins with one and two bifurcations were optimized through numerical modelling. Their results showed that the system efficiency can be improved by 24% with employing the optimized Y-shaped fins. They further suggested that the Y-shaped fins with wide angles were preferable in applications with short operating times, whereas small angles were more advantageous in applications with long operating times. Sheikholeslami et al. [9] proposed a snowflake shaped fin structure to enhance the performance of a PCM based TES unit during the discharging process. It was reported that the snowflake shaped fin structure can significantly enhance the discharging process.

As reviewed above, implementing conductive fins can improve the melting speed of PCM in a PCM based TES system. However, it should be noticed that by embedding metal fins, a considerable system volume could be occupied and hence the energy storage capacity is reduced. Therefore, the balance between the PCM melting speed and maximum energy storage capacity should be carefully considered for the fin design of an efficient TES system. So far, researches on optimizing the thermal performance of internal fin structure in TES systems to reduce the usage of fin materials are scarce. Recently, Xie et al. [10, 11] tried to use a thermal structure optimization method to develop the so-called best heat spreading structure in a PCM based heat sink. Results displayed great advantages of using the optimized thermal fin structures in achieving much lower heat source temperature and higher PCM melting speed in comparison to a baseline plate fin design. With this motivation, this paper aimed to explore the feasibility of using a thermal structure optimization approach to develop thermally efficient metal fins for high performance PCM based TES units with less usage of metal fins so as to reduce the weight and size of the PCM based TES units.

2. Problem descriptions

2.1. Physical model

The schematic diagram of the problem defined in the present study is shown in Figure 1(a). A compact PCM-based thermal energy storage panel is used in the thermal management of space limited electronic device whose operating period is short and the heat source temperature is constant \( (T_{\text{surf}} = 50 \degree \text{C}) \). As indicated, heat is transferred from the heat plate through the bottom wall of the thermal energy storage panel and then stored by the filled PCM. To enhance the heat absorption rate by the PCM, conductive plate fins are installed on the bottom wall to assist the heat spreading from the bottom wall into the PCM domain. For convectional plate fin designs, fins are uniformly distributed in the PCM domain, and hence the dimensions of the thermal storage panel configuration can be simply illustrated in Figure 1(b). As illustrated, the thickness of the bottom wall is 2.5 mm, the height of the PCM domain is 37.5 mm, and the spacing between the fins is 10 mm.

In order to obtain the best possible heat conduction path by the metal fins, the topological structure optimization algorithm developed by Liu and Tovar [12] was applied to redesign the shape of metal
fin, as shown in Figure 1(c). Specifically, the structure optimization is exercised by predefining the location of the heat source, metal volume fraction, thermal conductivities of the conductive metal fin and PCM, and the targeted optimization domain (10 mm × 37.5 mm for a single fin). Hereby, the metal volume fraction \(A_{\text{fin}}/A\) is defined as the ratio of the sectional area of the metal fins \(A_{\text{fin}}\) to the sectional area of the optimized domain \(A\) as indicated in Figure 1(b). In the present study, the metal fins are made of aluminum, and the PCM is a commercial paraffin RT35 that is manufactured by Rubitherm GmbH. The thermophysical properties of the metal fins and PCM are tabulated in Table 1.

After performing the thermal structure optimizations, the finally optimized metal fin structures based on the metal volume fractions of 0.05, 0.10, 0.15 and 0.20 are illustrated in Figure 1(d). To emphasize the advantages of the thermally optimized fin structures, the plate fin designs with the respective metal volume fractions are also included in the present study.

![Figure 1. Description of the problem in the present study.](image)

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**Table 1.** Properties of PCM and aluminium in the present simulations.

| Material     | \(\lambda\) (W/m·K) | \(\rho\) (kg/m³) | \(C_p\) (J/kg·K) | \(T_m\) (°C) | \(\mu\) (N·S/m²) | \(h_{\text{fg}}\) (kJ/kg) |
|--------------|----------------------|------------------|------------------|--------------|------------------|-----------------|
| RT35         | 0.20                 | Solid:860        | 2000             | 34-36        | 0.001\exp(-4.25 + \frac{1790}{T_e}) | 160             |
|               |                      | Liquid:770       |                   |              |                  |                 |
| Aluminium    | 202.4                |                  | 871              | 660.4        |                  |                 |
2.2. Numerical model

According to the geometry of the thermal energy storage panel as shown in Figure 1, the configuration of each metal fin is identical. Moreover, as illustrated in Figure 1(b), the geometry of a single metal fin is symmetric. Therefore, the physical model in the present study is further simplified as shown in Figure 2(a) where the boundary conditions are indicated.

In order to characterize the melting process of PCM under different fin structures with different metal volume fractions, the enthalpy-porosity approach [13] was adopted. The effect of natural convection was considered by applying the Boussinesq approximation to take the density variation into account during the melting process. Specifically, the density of liquid PCM as a function of temperature can be presented as:

$$\rho = \rho_l / (\beta(T - T_l) + 1)$$  (1)

where $\rho_l$ is the liquid PCM density as indicated in Table 1, and $\beta$ is the thermal expansion coefficient (0.0001).

The governing equations solved in the present study include:

Continuity equation:

$$\partial \rho / \partial t + \nabla \cdot (\rho \mathbf{u}) = 0$$  (2)

Momentum equations:

$$\partial (\rho \mathbf{u}) / \partial t + \nabla \cdot (\rho \mathbf{u} \mathbf{u}) = -\nabla p + \mu \nabla^2 \mathbf{u} + \rho \mathbf{g} + S_m$$  (3)

Energy equation:

$$\partial (\rho h) / \partial t + \nabla \cdot (\rho h \mathbf{u}) = \nabla \cdot (\lambda \nabla T) - S_h$$  (4)

where $\rho$ is the density; $u$ and $v$ are the velocities in the $x$ and $y$ directions, respectively; $\lambda$ and $\mu$ are the thermal conductivity and dynamic viscosity, respectively. $h$ is the sensible enthalpy. $S_m$ is the momentum sink term, and $S_h$ is a volumetric source term considering the latent heat ($\Delta H = \beta h \mathbf{\Delta} h$).

$$S_m = (1 - \chi)^2 / (\chi^2 + \epsilon) A u$$

$$S_h = \partial (\rho \Delta H) / \partial t + \nabla \cdot (\rho u \Delta H)$$  (5)

**Figure 2.** Schematic of the simulation model: (a) physical model and boundary conditions; (b) final mesh profiles for the simulated cases at $A_{\text{fin}}/A = 0.2$.  

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where, $A$ is a constant ($10^5$) that reflects the morphology of the mushy zone; $\varepsilon$ is a small number applied to prevent the division by zero; $\chi$ is the liquid fraction at the melting front ($T_m > T > T_s$) and it is defined by,

$$
\chi = \begin{cases} 
0 & \text{if } T < T_i \\
1 & \text{if } T > T_i \\
\frac{T - T_i}{T_m - T_i} & \text{if } T_i < T < T_m 
\end{cases}
$$

To solve the governing equations above, ANSYS FLUENT 19.2 was employed as the computation platform. Prior to conduct the simulations, a mesh independent study was conducted. As shown in Figure 2(b), the final computational cells for the plate fin and optimized fin designs are in the ranges of 32657-33645 and 35689-42159, respectively. Meanwhile, a time step independent study was also conducted, and the time step for the simulation was determined at $\Delta t = 0.1s$. During the simulation, the temperatures of the solid PCM, aluminum structures at the initial stage were all fixed at 30ºC.

### 3. Results and discussion

Although the proposed numerical method has been widely verified by many researchers, a validation study has been still conducted in the previous work [10]. Good agreement was found between the simulations and experiments, ensuring the capability of the numerical model employed in the present study.

To evaluate the amount of PCM being melted during the melting process, melt fraction ($f$) defined as the ratio of the mass of liquidus PCM to the mass of solid PCM at the initial condition of each simulation was characterized. Figure 3 presents the evolution curves of PCM melt fractions for the simulated cases. As can be seen, by increasing the metal volume fraction, the required time duration to melt the PCM is significantly reduced, particularly when the metal volume fraction is in the small range (i.e., $A_{\text{fin}}/A$ increases from 0.05 to 0.10). This is expected. First, as the metal volume fraction increases, the heat spreading within the PCM domain is improved and hence heat transfer is enhanced. Second, by increasing the metal volume fraction, the volume of PCM is reduced and the required melting time is reduced as results. From Figure 3, it is noted that the slope of the melt fraction curve of the optimized fin design was much steeper than that of its respective plate fin design, indicating a much faster PCM melting speed. This suggested that the optimized fins with the so-called best heat conduction paths exhibit obvious advantages in enhancing the PCM melting speed. For instance, at $f = 0.90$, the melting times required for the plate fin designs at $A_{\text{fin}}/A = 0.05, 0.10, 0.15$ and 0.20 are 740s, 525s, 425s and 355s, respectively. Whereas, for the optimized fin designs, the required melting times are significantly reduced to 675s, 440s, 325s and 210s, respectively. Therefore, the average melting speeds to achieve $f = 0.90$ by the optimized fin structures are 1.10, 1.19, 1.31, and 1.69 times as that by the plate fin designs at $A_{\text{fin}}/A = 0.05, 0.10, 0.15$ and 0.20, respectively. Besides, it is also observed that the optimized fin structure with $A_{\text{fin}}/A = 0.10$ can achieve a comparable melting speed as the plate fin design with $A_{\text{fin}}/A = 0.15$, and much faster melting speed by the optimized fin structure with $A_{\text{fin}}/A = 0.15$ than the plate fin design with $A_{\text{fin}}/A = 0.20$. It indicates that the thermally optimized fin structure has prominent advantages over the conventional plate fin in designing efficient thermal energy storage units with minimum usage of the thermal conductive material.

To further characterize the PCM melting speed, the curves of the historical melting speed defined as $df/dt$ are plotted in Figure 4. As can be seen, for all the simulated cases, the melting speed achieved by the optimized fin design is initially higher than that by the plate fin design with the same metal volume fraction. As the melt fraction of the optimized fin design exceeds a certain value (i.e., $f = 0.83-0.86$), the melting speed obtained by the plate fin design overtakes the optimized fin design possibly due to the fact that the plate fin is impregnated deeper into the PCM domain as shown in Figures 5 and 6. As compared to the optimized fin designs, the tips of plate fins are much closer to the remanent solid PCM at the later PCM melting stage. As a result, the melting speeds yielded by the plate fin
designs is faster at the later PCM melting stage. However, the absolute melting speed of PCM is very slow at the later PCM melting stage.

Figure 3. Evolution curves of the melt fractions of the simulated cases.

Figure 4. Historical melting speeds of the simulated cases.

Figure 5. Characteristics of PCM melting at \( t=200s \).
(a) \( A_{\text{fin}}/A = 0.05 \): \( f = 0.39\) @Conventional, \( f = 0.44\) @Optimized; (b) \( A_{\text{fin}}/A = 0.10 \): \( f = 0.50\) @Conventional, \( f = 0.58\) @Optimized; (c) \( A_{\text{fin}}/A = 0.15 \): \( f = 0.58\) @Conventional, \( f = 0.71\) @Optimized; (d) \( A_{\text{fin}}/A = 0.20 \): \( f = 0.65\) @Conventional, \( f = 0.89\) @Optimized.

Figure 6. Characteristics of PCM melting at \( f = 0.9 \).
(a) \( A_{\text{fin}}/A = 0.05 \): \( t = 740s\) @Conventional, \( t = 675s\) @Optimized; (b) \( A_{\text{fin}}/A = 0.10 \): \( t = 525s\) @Conventional, \( t = 440s\) @Optimized; (c) \( A_{\text{fin}}/A = 0.15 \): \( t = 425s\) @Conventional, \( t = 325s\) @Optimized; (d) \( A_{\text{fin}}/A = 0.20 \): \( t = 355s\) @Conventional, \( t = 210s\) @Optimized.
Figure 5 presents the convection patterns and temperature contours of the simulated cases at $t = 200s$. It is obvious that the melt fraction $f$ varies largely as the metal volume fraction increases, indicating a significant role of metal fins during the PCM melting process. Besides, for the counterparts at the same metal volume fraction, the optimized fin design obtains a higher $f$. This could be due to a few reasons. First, as illustrated in Figure 5, the contact surface between the fin structure and PCM is larger for the optimized fin designs, which is benefit for heat transfer. Second, the convection is more intensified for the optimized fin designs, especially when the metal volume fraction is higher, as shown in Figures 5(c) and (d). By comparing the convection flow patterns under different fin structures, it is obvious that the optimized fin structures are able to trigger more convection vortices in comparison to their plate fin counterparts. The convection vortices generated by the fin structures tend to drive the hot liquid arising from the metal fin walls to impinge on the solid PCM, and hence the melting of PCM is promoted [14]. Apart from that, it is also observed that the convection vortices are benefit for thermal mixing of the melted PCM (uniform temperature), which is in the favour of PCM melting as well. Third, as the optimized fin structure is generated in terms of the best possible heat conduction path, the temperature propagation by the optimized fin structure is more efficient and hence the phase change heat transfer is enhanced.

Figure 6 presents the the convection patterns and temperature contours of the simulated cases at $f = 0.9$. As discussed previously, at this melting stage, the melting speed obtained by the plate fin design is higher than that by the optimized fin design. As can be seen, since the optimized fin structures are less impregnated into the PCM domain, the convection invoked near the remanent solid PCM is less intensified as the temperature gradient that is the driving force for natural convection is smaller.

4. Conclusions
To enhance the melting speed of PCM in a compact thermal energy storage panel, effort was made to develop highly conductive metal fin structures using a thermal structure optimization method. A two-dimension transient numerical model was built to simulate the PCM melting process based on the enthalpy-porosity approach with considering the effects of natural convection. The thermal performance of four metal volume fractions were studied both in the convectional plate fin and optimized fin designs. Results indicated that increasing the metal volume fraction tended to accelerate the PCM melting process due to the improved overall thermal conductivity. The optimized fin structure with the so-called best heat conduction path generally outperformed its counterpart with the plate fin structure at a fixed metal volume fraction. To reach the stage when $f=0.9$, the average PCM melting speeds attained by the optimized fin structures were 1.10, 1.19, 1.31, and 1.69 times as that of the plate fin designs at the melt volume fractions of 0.05, 0.10, 0.15 and 0.20, respectively. However, at the later stage of PCM melting (i.e., $f>0.83$), the instantaneous PCM melting speed achieved by the optimized fin design was slower than that by the plate fin design. This was because the optimized fin structure was less impregnated into the PCM that was far away from the heated bottom wall. Besides, results also revealed that the optimized fin structure with the metal volume fraction of 0.10 can achieve the similar melting speed as the plate fin design with the metal volume fraction of 0.15, and the optimized fin structure with the metal volume fraction of 0.15 can achieve even higher melting speed than the plate fin design with the metal volume fraction of 0.20. Hence, the optimized fin structure proposed in the present study is preferable for designing high performance thermal energy storage units with minimum usage of the thermal conductive material.

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