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Effect of fin number on the melting phase change in a horizontal finned shell-and-tube thermal energy storage unit

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ABSTRACT

This paper studies the enhanced heat transfer of adding longitudinal fins in a horizontal shell-and-tube heat storage unit. A two-dimensional numerical model is established and validated through comparing with experimental data in literature. Under the same ratio of fin mass to phase change materials (PCMs), the melting thermal performance is optimized by changing the fin thickness, interval and the number. Results show that adding longitudinal fins is a simple and effective method to enhance the thermal energy storage efficiency. The number of fins greatly affects the complete melting time, and the maximum time difference caused by the number of fins is as high as 70% under the same heat storage effect. At the same time, increasing the number of fins will weaken the local natural convection. In this paper, the optimal number of fins in the limited research range is given, and the effectiveness of longitudinal fins in improving melting speed is quantified, which has certain practical significance for the engineering application research of phase change energy storage.

Keywords: Longitudinal fins; Thermal energy storage; Phase change material; Fin number; Shell-and-tube heat exchanger

1. INTRODUCTION

As an ideal conversion technology to improve energy efficiency, latent heat thermal energy storage (LHTES) of phase change materials (PCMs) has been widely used in solar energy utilization, waste heat recovery, desalination, spacecraft thermal protection and other engineering fields. Common PCMs have the advantages of high latent heat of fusion, small expansion and contraction, but their low thermal conductivity seriously limits the heat storage rate of PCMs. In order to enhance heat transfer, fins, porous metals, and nanomaterials are usually added to PCMs to improve thermal conductivity. Among these enhancement methods, finned tubes are widely welcomed thanks to their convenience, simplicity and effectiveness [1,2].

A great number of studies have been carried out to improve heat transfer by adding fins to PCMs. Ismai et al. [3] experimentally studied the enhancement on PCMs in a horizontal tube with radial fins and turbulence promoters. They found that turbulence promoters can shorten the complete solidification time and accelerate the solidification process, but the effect was not as obvious as the radial fins. Kamkari et al. [4] conducted an experimental study on melting of PCMs in a transparent rectangular shell with or without horizontal fins at different wall temperatures, and directly visualized the melting process and temperature field. Yang et al. [5] carried out a numerical attempt to the PCM melting process in a shell-and-tube latent heat storage device with annular fins, and quantitatively analyzed the influence of fin parameters on the melting process. It was suggested that inserting annular fins in the PCM can shorten the melting time by 65% to the greatest extent. Tao et al. [6] proposed local longitudinal fins and studied the latent heat process of PCM in a horizontal circular tube. The results revealed that local fins can improve the uniformity of the melting of PCM.

To sum up, the theory that finned tubes can enhance heat transfer with PCMs has been demonstrated through a large number of experimental studies and numerical simulations, mainly focusing on the enhancement of heat transfer measures. Up to now, however, there are few studies on the optimal number of longitudinal fins in horizontal tubes. To this end, this paper studies the melting process of horizontal finned shell-and-tube thermal energy storage (TES) unit, trying to find the optimal number for longitudinal fins towards maximizing melting heat transfer. A numerical model with paraffin as PCM and water as HTF was established. The melting rate, full melting time and melting front evolution were quantified.

2. NUMERICAL MODEL

2.1 Physical model

The horizontal shell-and-tube TES unit with longitudinal fins is shown in **Fig. 1**. The TES device is composed of two concentric inlaid cylinders, and the diameters of the large and small cylinders are 60 mm and 20 mm respectively. The outer cylinder is made of Plexiglas, and the inner cylinder is an aluminum round tube. The longitudinal fins are uniformly arranged on the outer surface of the aluminum tube in the radial direction. Its cross section of its structure is shown in **Fig. 2**, where the height is *h*, the thickness is *t*, and the space between the fins is b.

In the case, the space between two adjacent fins and the fin thickness is designed to be variable, that is, to increase the number of fins. Paraffin wax is used as the PCM to fill in the annular area between two concentric cylinders, and water flows in the tube as HTF. Considering the geometric symmetry of the physical model and the uniform and constant temperature of the heating boundary, the three-dimensional physical model can be simplified to a two-dimensional one.

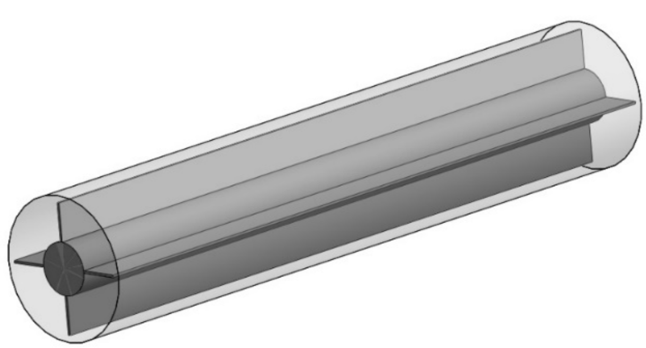

Fig. 1 The horizontal shell-and-tube TES device

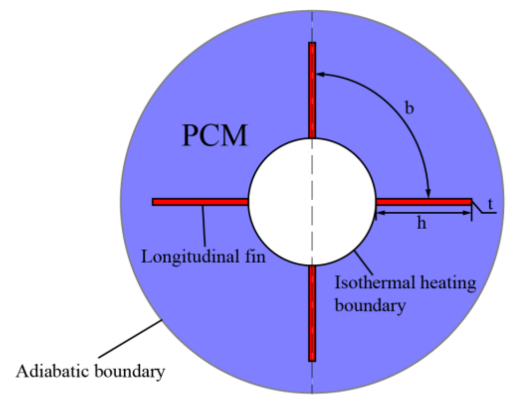

Fig. 2 The cross section of energy storage device with longitudinal fins

2.2 Governing equations

The transient heat transfer was described as follows:

$$\nabla \cdot \vec{u} = 0 \tag{1}$$

$$\rho_f \frac{\partial \vec{u}}{\partial t} + \rho_f (\vec{u} \cdot \nabla)\vec{u} = -\nabla P + \mu_f \nabla^2 \vec{u} + \rho_f \vec{g}\beta(T_f - T_m) + A\vec{u} \tag{2}$$

$$\rho_f c_{pf} \frac{\partial T_f}{\partial t} + \rho_f c_{pf} \vec{u} \cdot \nabla T_f = \nabla \cdot (k_f \nabla T_f) - \rho_f L_f \frac{\partial f_l}{\partial t} \tag{3}$$

where the coefficient *A* in Eq.(2) donated the damping coefficient to damp the velocity in solidified phase, which can be calculated through the following equation [18]:

$$A = \frac{C(1-f_l)^2}{S + f_l^3} \tag{4}$$

where *C* and *S* were the numerical coefficients, recommended to be very large (1×10^{15}) and small (1×10^{-10}); f_l was the melting fraction of PCM and it was determined by the representative temperature in the mushy zone:

$$f_l = \begin{cases} 0 & \text{at} \quad T < T_{\text{solidus}} & \text{solid} \\ \dfrac{T - T_{\text{solidus}}}{T_{\text{liquidus}} - T_{\text{solidus}}} & \text{at} \quad T_{\text{solidus}} < T < T_{\text{liquidus}} & \text{mushy} \\ 1 & \text{at} \quad T > T_{\text{liquidus}} & \text{liquid} \end{cases} \tag{5}$$

2.3 Initial and boundary conditions

The initial temperature of the PCM and the fin domain were set to 293.15K. The outer tube wall of the annular area was regarded as the adiabatic boundary, without considering the heat loss; while the inner tube wall was the isothermal heating boundary with a temperature of 343.15K.

3. RESULTS AND DISCUSSION

In this paper, ANSYS-Fluent 18.2 was used as the numerical simulation software. On the basis of finite volume method (FVM), the SIMPLEC algorithm with pressure-velocity coupling and the pressure correction equation in PRESTO! scheme were used to carry out two-dimensional simulations. Gradient's discretization method adopted the Least Squares Cell-Based method, while the momentum and energy equations were discretized by the second-order upwind scheme. The thermophysical properties of PCM and fin material (aluminum) were shown in **Table 1**.

Table 1
Thermo-physical properties of PCM and aluminum

Material	Property	Value
Paraffin	Latent heat ($J{\cdot}Kg^{-1}$)	232400
	Density ($Kg{\cdot}m^{-3}$)	771.2
	Special heat capacity ($J{\cdot}Kg^{-1}{\cdot}K^{-1}$)	2176
	Thermal conductivity ($W{\cdot}m^{-1}{\cdot}K^{-1}$)	0.089
	Dynamic viscosity ($kg{\cdot}m^{-1}{\cdot}s^{-1}$)	0.002508
	Thermal expansion coefficient	0.00075
Aluminum	Density ($Kg{\cdot}m^{-3}$)	2719
	Special heat capacity ($J{\cdot}Kg^{-1}{\cdot}K^{-1}$)	871
	Thermal conductivity ($W{\cdot}m^{-1}{\cdot}K^{-1}$)	202.4

3.1 Liquid fraction

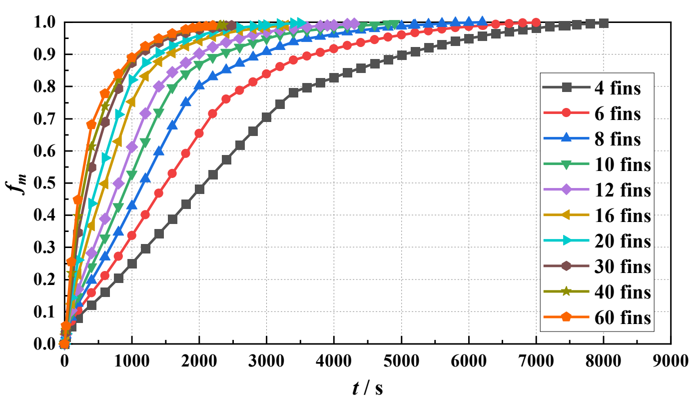

Fig. 3 Melting fraction of different fin numbers

Firstly, the influence of the number of fins on the melting process was studied. **Fig. 3** showed the melting fraction curves with different numbers of fins. Obviously, the melting fraction increased monotonously with the fin number, and the whole curve showed a steep first and then slow trend. As time going by, more PCMs changed from solid to molten, which made the thermal resistance between the solid-liquid interface and the heat transfer surface larger and the melting rate slower. In order to find the optimal number of fins, when the aluminum mass was constant, that is, when the volume ratio of fins to PCMs kept constant (2.53%), the height of fins was fixed (16mm) but the thickness and number changed.

In **Fig. 4**, it can be seen that the number of fins greatly affected the complete melting time, and the maximum time difference was 5240 s and the maximum time difference caused by the number of fins was as high as 70% under the same heat storage effect. In a given number of fins, when the number of fins was ranging from 52 to 60, the melting time was the shortest and the heat transfer was the fastest. This indicated that adding fins reasonably can effectively improve the heat transfer during melting. In the case of a small number of fins, the complete melting time basically decreased with the increase in fin number. In the later stage, when the fin number is large, the complete melting time basically did not increase with the fin number. If one blindly increased the fin number and reduce the fin interval, the adjacent fluid flow between the fins will slow down, or even stagnate, leading to the weakened natural convection effect between local areas. Therefore, when the number of fins was 52-60, the complete melting time was basically the same.

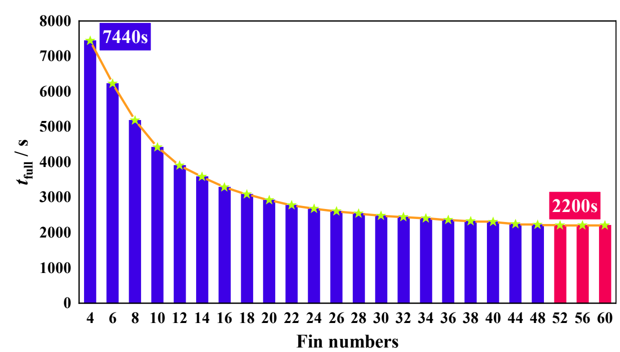

Fig. 4 Complete melting time of different fin numbers

3.2 Melting front evolution

Fig. 5 depicted the melting phase interface of the TES unit with different numbers of fins at four different typical times, 300 s, 900 s, 1600 s, and 2300 s. It can be seen that due to the presence of fins, the melting rate of the finned PCM was higher than that of the non-finned part. At the initial stage of melting, PCM was in solid phase, only the solid paraffin near the fin was melted, and the heat transfer mechanism was mainly heat conduction. With the passage of time, more and more paraffin wax was melted into liquid. The closer to the fin area was, the faster the paraffin wax melted. In this stage, the natural convection was the main action. Although the fins in the horizontal heat storage unit were uniformly arranged, due to the convection of the liquid PCM, the high temperature liquid paraffin will flow upwards, and the melting rate of the upper half of the cross section will be higher than that of the lower half. It was reflected in 900 s in **Fig. 5**. It began to manifest at 900 s, and it can be concluded that natural convection can enhance the melting process of PCMs. In addition, it can also be seen that from 4 to 20 fins, although the number of fins increased uniformly, the difference in the evolution process of the phase interface melting peak was gradually decreasing, that is, the melting speed was more uniform. This was due to the fact that the increase in the number of fins led to smaller distance between adjacent fins, weakened local natural convection and discounted heat transfer enhancement effect.

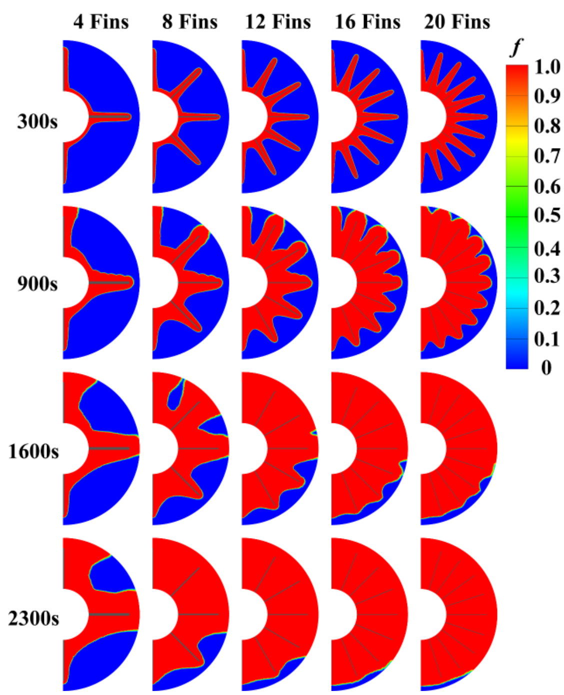

Fig. 5 Solid-liquid interface with different number of fins at four times

4. CONCLUSIONS

In this paper, numerical simulation method is used to study the effect of embedding longitudinal fins in TES on the heat storage process of PCMs. When the volume ratio is constant, the fin thickness is changed to find the optimal number of fins in the TES heat storage process. The main concluding remarks are as follows:

1) Reasonable addition of fins can effectively improve the heat transfer process, and the number of fins greatly affects the total melting time, with the maximum time difference of 5240 s, which shortens the total melting time;
2) Natural convection can enhance the melting process of PCMs, but the local natural convection will be weakened by increasing the number of fins;
3) The heat storage amount does not change due to the change of the number of fins. The optimal number of fins involved in this study is ranging from 52 to 60 to maximally enhance the heat transfer performance.

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