Structural Optimization of Heat Transfer Fins in the Energy Storage System

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Abstract. For Problem 1, this paper uses the finite-difference methods to analyze the heat transfer effect of PCMs between two rectangular fins and get the relationship among temperature, time and distance from a fin. Through establishing a physical model based on thermodynamics and adjusting the interval angle between fins, we obtain the change rule of PCM average temperature with time under different interval angles. In order to avoid overlap of fins, this paper selects 20 ° as the final angle. The time required for PCMs to rise from room temperature (273K) to phase change temperature (333K) is about 240s. For Problem 2, this paper extends the model proposed in Problem 1 to adapt to the triangular fin structure. This paper designs a multi-objective optimization model. Results show that the heat transfer effect of PCMs is better when the interval angle is 20 °, the bottom width of the triangle is 0.0025m and the height is 0.025m. In addition, it is concluded that rectangular fins are more conducive to heat transfer than triangular fins.

Keywords: Heat Transfer Fin; Genetic Algorithm; Multi-objective Optimization.

1. Introduction

The traditional energy storage capacity around the world is on the decrease, and the dual urgent situation of energy shortage and environmental degradation can no longer be ignored [1]. Due to the advantages in sustainability and environmental protection, there are more and more urgent demands on renewable energy. Among the recent renewable energy researches, phase change energy storage technology is undoubtedly a feasible and effective one. Energy storage technology is mainly divided into sensible heat energy storage, latent heat energy storage and thermochemical energy storage [2]. The phase change energy storage (PCES) utilizing the latent heat of phase change material (PCM) to absorb and release heat, has been widely applied to refrigeration, solar energy storage and waste heat recovery [3]. However, their generally low thermal conductivity factor leads to a prolonged heat storage and release process, which has become a key factor hindering the broad application of PCMs. To improve the rapid heat transfer capability of a heat storage system, we need to research the important topic to optimize the structural design and parameters of the system.

Nowadays, heat transfer fins can be used to improve the operation performance of heat storage tanks using phase change filling material (PCM) [4]. Therefore, the requirement of phase change heat storage technology becomes more and more strong. The core component of the heat storage system is a shell-and-tube heat storage tank filled with heat storage PCMs and fin structures [5-7]. When PCMs absorb heat, the high-temperature working fluid circulates in the inner tube to store and utilize its waste heat. Meanwhile, when the PCMs release heat, the cryogenic working fluid circulates in the inner tube to absorb and reuse the heat stored in the PCMs, which improves the heat transfer performance of heat storage products [8-10].

This study aims to solve the following problems:

Problem 1: For the cross-section of the heat storage tank with evenly distributed rectangular fins, optimize the interval angle θ, and illustrate the time it requires for the average temperature of PCMs to rise from room temperature (293 K) to the phase change temperature.

Problem 2: Optimize the size and distribution of triangular fins, study the influence of their size on the heating rate of PCMs, and compare and analyze their heat transfer effect with the rectangular fin structure in Problem 1.

2. Model Establishment and Solution for Problem 1

2.1. Deduction of Thermodynamic Theory

Let $u = u(x,t)$ be the temperature of a uniform object Ω at position x and time t (unit: K), c is the specific heat (unit: $J/(kgK)$), ρ is the density (unit: kg/m^3), let V be any smooth area within Ω and q is the heat flux density of the temperature field (unit: W/m^2). Additionally, $f_0(x,t)$ is the heat density generated at position$x$, time$t$, which is obtained from the law of conservation of energy:

$$\frac{d}{dt}\iiint_V c\rho u(x,t)dv + \iint_{\partial V} qndS = \iiint_V \rho f_0 dv \tag{1}$$

n Is the normal vector outside the unit of ∂V and we know that due to Gauss Formula:

$$c\rho \iiint_V \frac{\partial u(x,t)}{\partial t}dv + \iiint_V divqdv = \iiint_V \rho f_0(x,t)dv \tag{2}$$

Because of the arbitrariness of region$V$, we can get the following results:

$$c\rho \frac{\partial u(x,t)}{\partial t} + divq = \rho f_0(x,t) \tag{3}$$

According to the Fourier's law of heat conduction, the heat flow density in the temperature field is proportional to the temperature gradient, and the heat always flows from the high temperature to the low temperature.

$$q = -k\left(\frac{\partial u(x,t)}{\partial x_1}, \frac{\partial u(x,t)}{\partial x_2}, \dots, \frac{\partial u(x,t)}{\partial x_n}\right) \tag{4}$$

$k > 0$ Is the thermal conductivity of the object? After substituting the above equation, we get the thermal equation:

$$\frac{\partial u(x,t)}{\partial t} - a^2\left(\frac{\partial^2 u(x,t)}{\partial {x_1}^2} + \frac{\partial^2 u(x,t)}{\partial {x_2}^2} + \dots + \frac{\partial^2 u(x,t)}{\partial {x_n}^2}\right) = f_0(x,t) \tag{5}$$

Where $a = \sqrt{k/c\rho}$,$f(x,t) = f_0(x,t)/c$

Let the temperature distribution of each point in the object at the initial time $t = 0$.

$$u(x,0) = \varphi(x), x \in \Omega \tag{6}$$

The temperature distribution of the object boundary partial Ω at time $t > 0$ or the influence of the surrounding medium is generally divided into the following three categories:

Temperature Distribution with Known Boundary PartialΩ:

$$\mathrm{u(x,t) = g(x,t), x \in \partial\Omega, t \geq 0} \tag{7}$$

When g is constant, it means that the boundary of the object remains constant temperature.

The heat flowing into or out of object Ω through object boundary $\partial\Omega$ is known.

$$k\frac{\partial}{\partial n}u(x,t) = g(x,t), \mathrm{x} \in \partial\Omega, t \geq 0 \tag{8}$$

Where n is the normal vector outside the unit of$\partial\Omega$, when$g \geq 0$, it represents the heat inflow, when$g \leq 0$, it represents the heat outflow, and when$g = 0$, it represents the insulation of the object.

It is known that the heat exchange strength between the object boundary and the surrounding medium is partial.

$$k\frac{\partial}{\partial n}u(x,t) = \alpha_0(x,t)\big(g_0(x,t) - u(x,t)\big), \mathrm{x} \in \partial\Omega, t \geq 0 \tag{9}$$

Or

$$\frac{\partial}{\partial n}u(x,t)+\alpha(x,t)u(x,t)=g(x,t),\mathrm{x}\in\partial\Omega,t\geq 0 \tag{10}$$

Where n is the normal vector outside the unit of partial$\partial\Omega$, and g_0 is the temperature of the surrounding medium, $\alpha_0\geq 0$ indicates heat exchange coefficient, $\alpha=\frac{\alpha_0}{k}>0.g=\frac{g_0}{k}$

According to the data in Problem 1, the following equation is listed.

$$\begin{cases}\frac{\partial u(x,t)}{\partial t}=\alpha\frac{\partial^2 u(x,t)}{\partial x^2}\\ u(0,t)=T_1=100,u(x_d,t)=T_2=100\\ u(x,0)=20(x\neq 0)\\ \alpha=\frac{k}{\rho c},k=0.15,\rho=780,c=2500\end{cases} \tag{11}$$

For each point difference differentiation, we set N points for the x-axis and N points for each unit length of the t-axis.

$$\begin{cases}x_i=\frac{i}{N_x},i=1,2,\dots,N_x\\ t_j=\frac{j}{N_t},j=1,2,\dots,\infty\\ x_{i+1}-x_i=d_x=\frac{x_d}{N_x}\\ t_{j+1}-t_j=d_t=\frac{1}{N_t}\end{cases} \tag{12}$$

For the second order difference sub item, we use the second order central difference quotient, and the above equation can be approximately written as:

$$\frac{u(x_i,t_{j+1})-u(x_i,t_j)}{d_t}=\alpha\frac{u(x_{i+1},t_j)-2u(x_i,t_j)+u(x_{i-1},t_j)}{{d_x}^2} \tag{13}$$

This shows that$(i+1,j),(i,j)$, $(i,j+1)$ and $(i,j-1)$ can be expressed mutually. Therefore, the equation can be rewritten as follows.

$$\begin{cases}u(x_i,t_{j+1})=\left(1-2\frac{\alpha d_t}{{d_x}^2}\right)u(x_i,t_j)+\frac{\alpha d_t}{{d_x}^2}(u(x_{i+1},t_j)+u(x_{i-1},t_j))\\ u(x_i,0)=20,i=1,2,\dots,N_x\\ u(0,t_j)=100,u(N_x,t_j)=100,j=0,1,2,\dots,\infty\end{cases} \tag{14}$$

2.2. Establishment of COMSOL Model

We establish the internal section model of the heat storage tank and fill in the corresponding parameters by COMSOL software. Assume that the fins are made of aluminum, and the parameters are as follows: thermal conductivity is$0.15W/m\cdot K$, the density is $780kg/m^3$ and constant pressure heat capacity is$2500J/K$. Besides, the phase change material parameters are as follows: thermal conductivity is$214W/m\cdot K$, the density is $880kg/m^3$ and constant pressure heat capacity is$237.6J/K$. According to the set thermal insulation of the external wall of the heat storage tank, the initial temperature of the internal fluid is$373K$, and the initial temperature of the fins and PCMs is $293K$(Fig. 1).

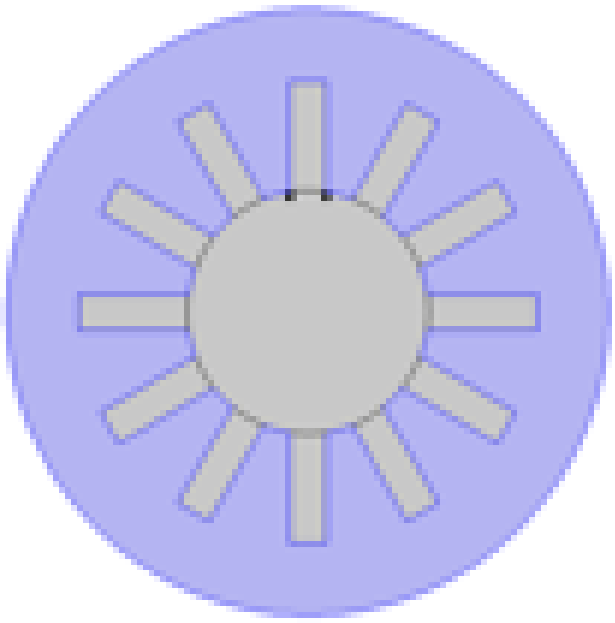
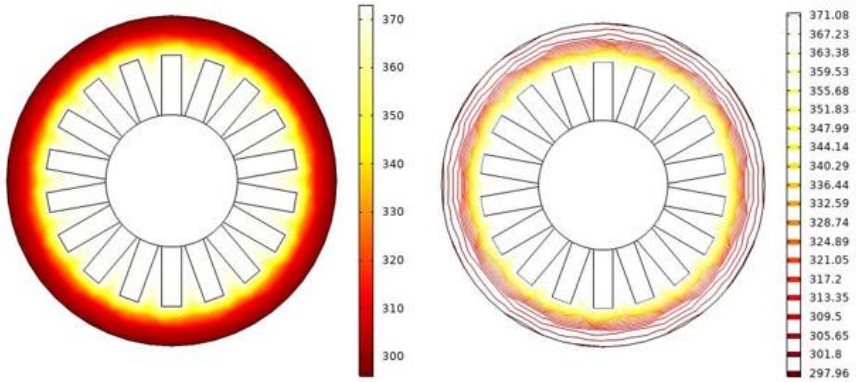


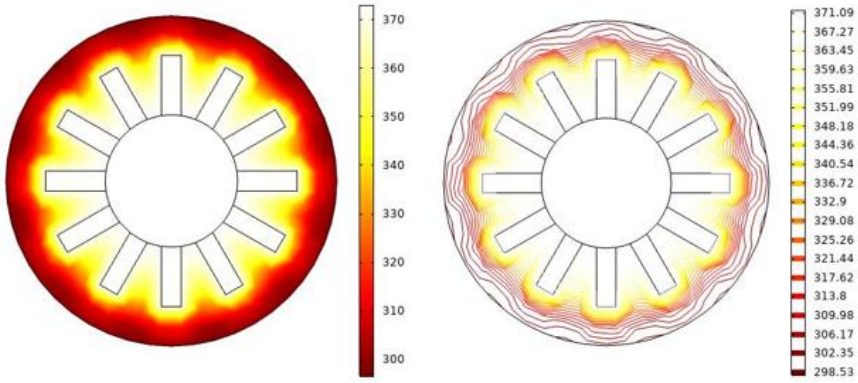
Fig 1. Internal section model of heat storage tank.

We conduct numerical simulation for these situations through COMSOL software. We set the interval angle θ respectively, which are $20°, 30°, 45°, 60°, 75°$ and $90°$, and the time is 30 minutes. We conduct simulation when the average temperature of PCMs reaches the phase transition temperature. The temperature distribution diagrams and the isotherm images inside the heat storage tank are as follows (Fig. 2).

$\boldsymbol{\theta = 20°}$



$\boldsymbol{\theta = 30°}$



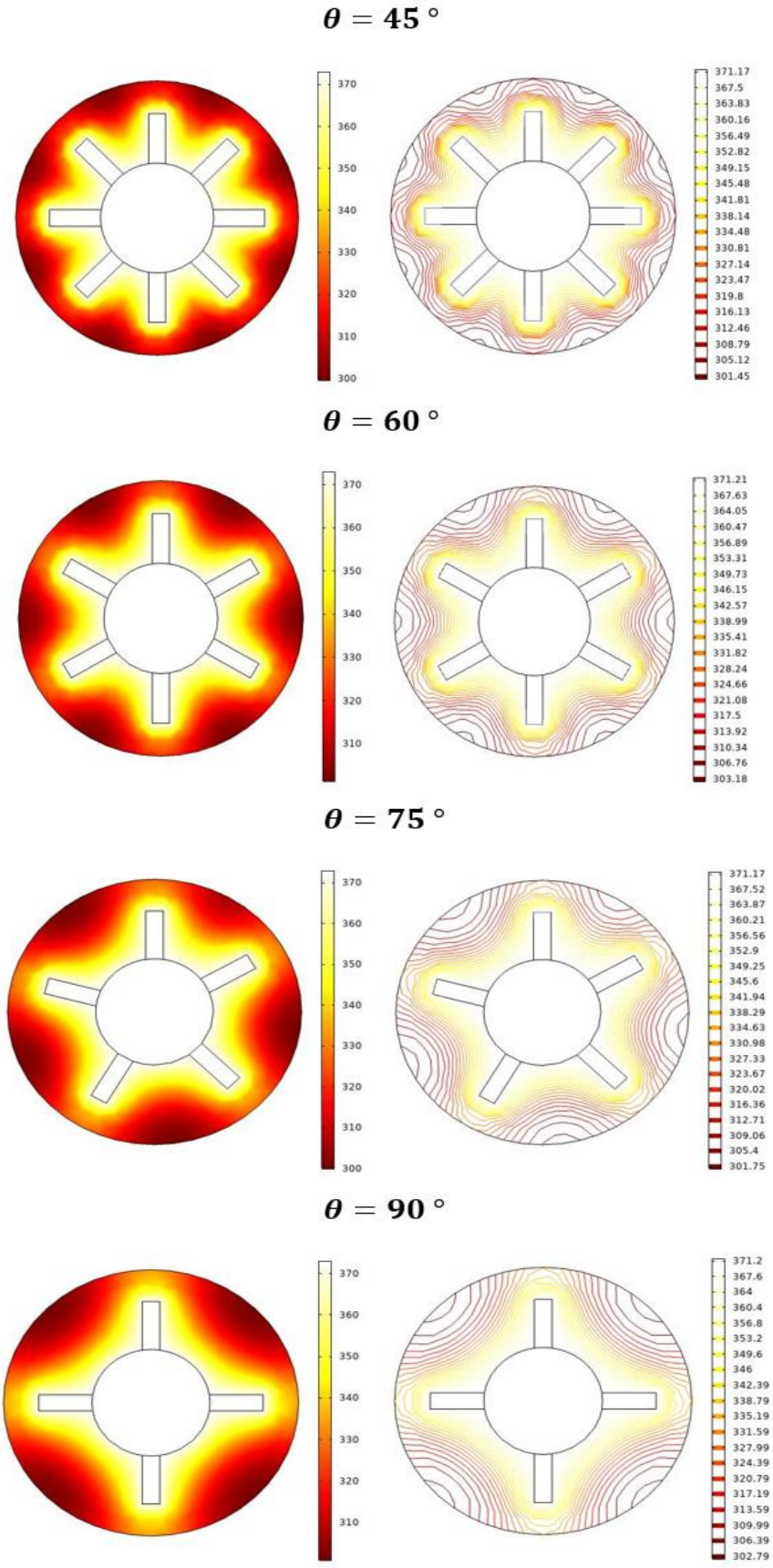


Fig 2. Temperature distribution diagrams and the isotherm images

From the previous simulation calculation results, the temperature changes fastest with time when the interval angle is 20 °. In this case, the temperature distribution in the first 5 minutes is shown in the figure, which shows that the temperature changes significantly in a short time (Fig. 3).

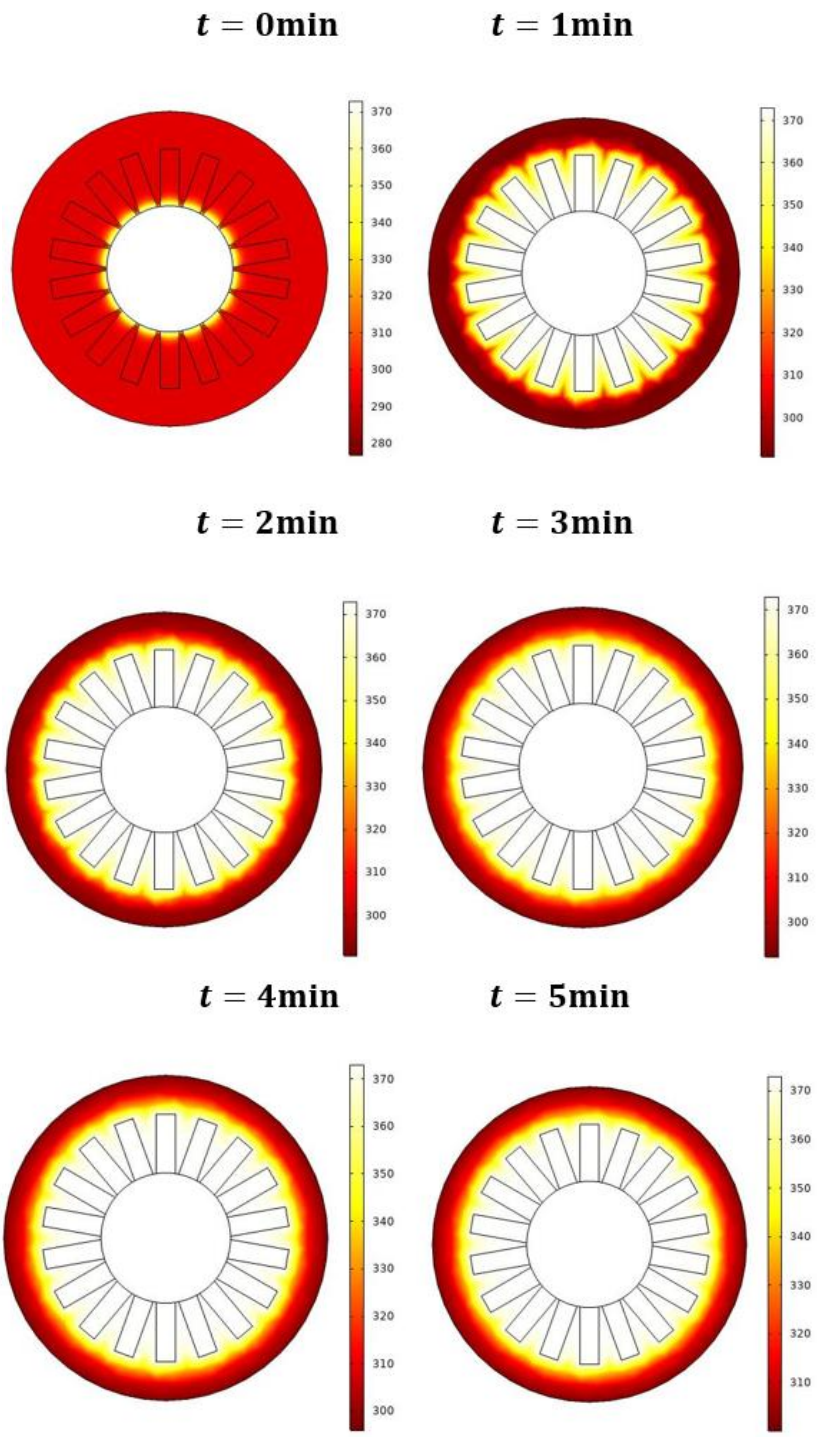


Fig 3. Temperature distribution diagrams

2.3. Finite Difference Method and its Solution in MATLAB

Based on the above data, temperature of the phase change material between the two fins changes most obviously with time, so we deduce the formula based on the theory of finite difference method and use MATLAB to program. We change the interval angle ($\theta = 20°, 30°, 45°, 60°, 75°$ and $90°$)to explore the relationship between temperature and time. Finally, we get the rule of temperature change of some phase change materials between two fins with time, and the curve is shown in the Fig. 4.

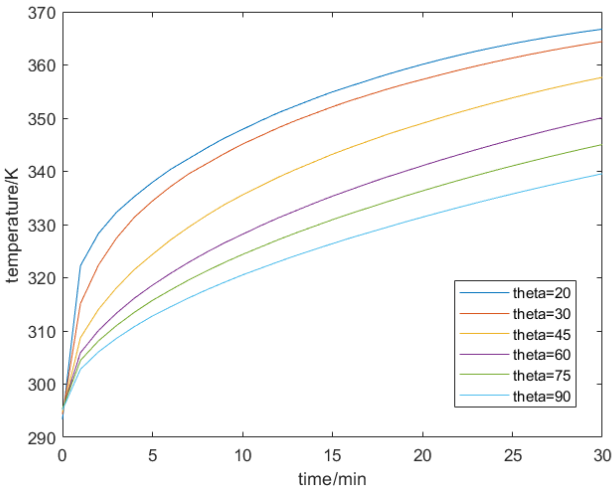


Fig 4. Average temperature versus time curve of phase change materials

From the images above, we come to the conclusion: the smaller the interval angle is, the faster the average temperature changes with time. In addition, the average temperature changes fastest with time when $\theta = 20°$ and it takes 4 minutes to reach$333K$. However, if the angle is too small, the fins will not be distributed uniformly, causing the uneven heat transfer of PCMs. Therefore, $\theta = 20°$ is considered as the appropriate interval angle between the fins. It can be seen from the figure that when$\theta = 20°$, it takes about $240\ s$ for PCMs between the fins to reach $333K$(Fig. 5).

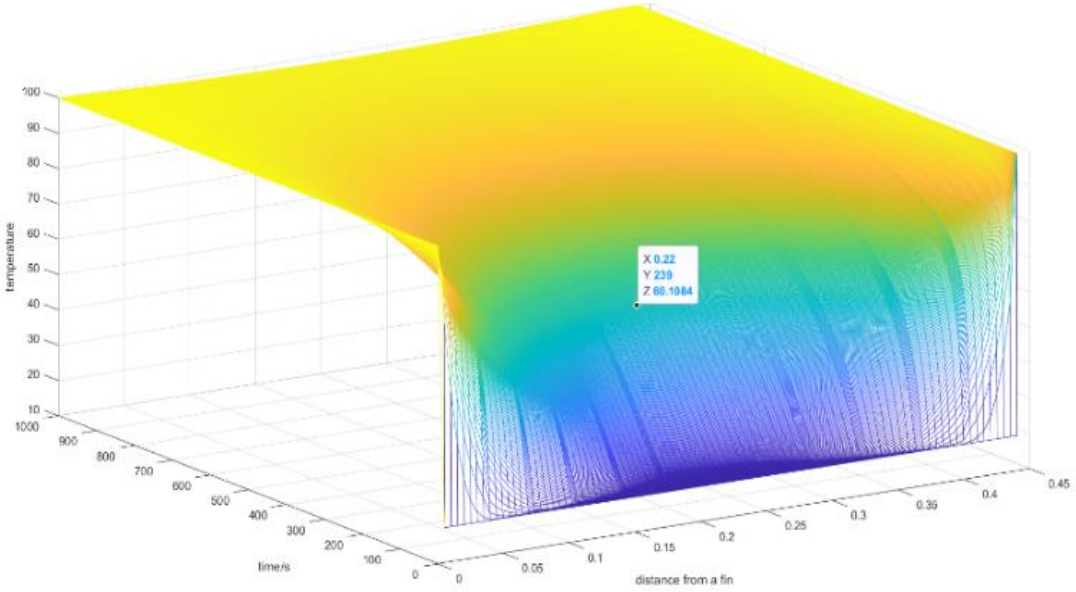


Fig 5. Variation of temperature between two fins with time when $\theta = 20°$

3. Model Establishment and Solution for Problem 2

3.1. Model Optimization

Based on the model of Problem 1, we change rectangular fins into triangular fins. Then we set the following data: The angle θ are 20 °,30 °,45 °,60 °, the triangular height is 0.015m,0.018m,0.02m, and the triangular bottom width is 0.001m 0.0015m,0.002m,0.0025m. According to the permutation and combination, there are 48 cases, for each case the time required to reach 333K from 293K is calculated by COMSOL software. We use the partial least squares method to fit the data and the steps are as follows:

(1) Refer to the maximum principal component by variance explanation and Variable Importance in Projection.

(2)Get the composition of the principal components through the composition matrix table.

(3)The importance of variables is obtained from the factor load factor table.

(4)Get the standardized formula of partial least squares regression.

Finally, we get the standardized formula of the model:

$$t = 22.9 - 1258 * h - 80 * w + 0.3 * \theta \tag{15}$$

For the established multi-objective programming model for the parameter optimization design of triangular fins, we require the interval angle between fins, the width and height of the triangle bottom edge. However, the range of optional values for solving the target is not fixed and the search space is large, which leads to the complexity of solving the optimization problem of triangular fin parameter design and makes it difficult to find an absolute optimal solution. Genetic algorithm uses stochastic optimization technology, which does not require continuity and differentiability. Therefore, this paper uses genetic algorithm to solve the optimization of triangular fin parameter design.

The variables in Problem 2 are the bottom edge width, height of a triangular fin and the interval angle between fins and we set the variables as follows: the height of each triangular fin is$h$, the bottom edge width is$w$, and the interval angle between fins isθ. At the same time, in order to ensure that the triangular fins height does not exceed the difference between the inner and outer pipe diameters. There should be no overlap between fins. To sum up, the variables meet the following constraints:

$$\theta > 0, 0 < h < 0.03m, 0 < w < 0.004m \tag{16}$$

The optimization steps are as follows (Fig. 6).

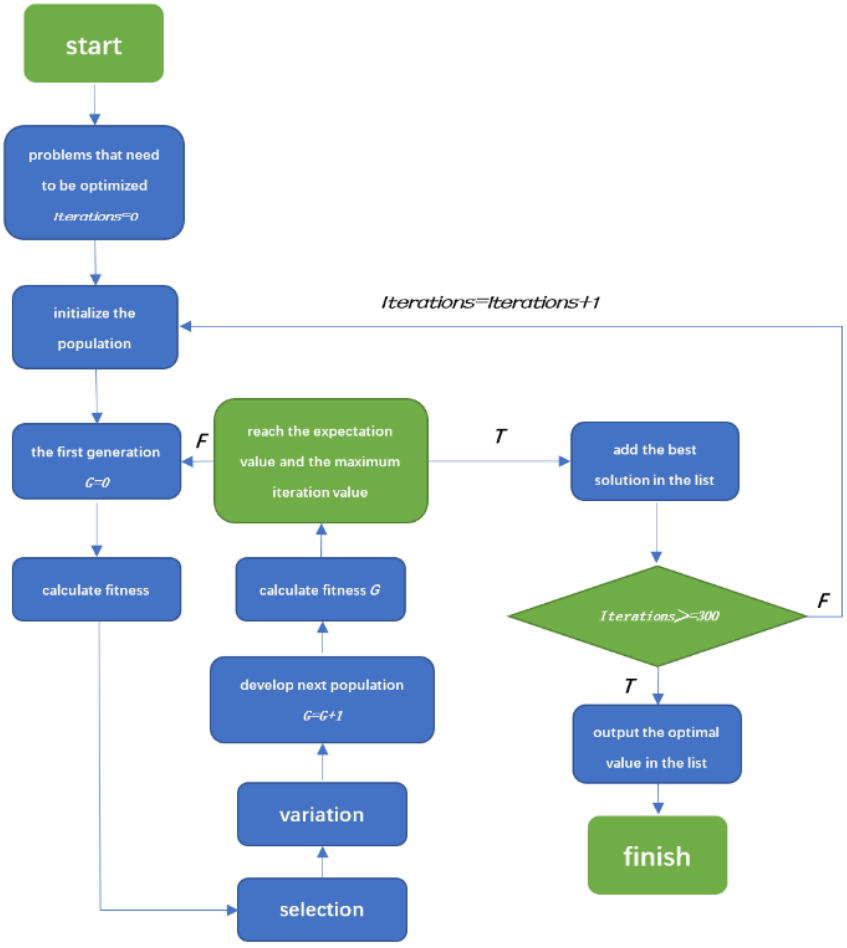


Fig 6. Optimization flow chart

We use the obtained model standardization formula and the relationship between the variables obtained by data fitting to build the planning model, and the equation is as follows:

$$min\ Z = 22.9 + 0.3 * \theta - 80 * w - 1258 * h \tag{17}$$

$$s.t. \begin{cases} 22.9 + 0.3 - 80 * w - 1258 * h > 0 \\ 29.1 + 56 * w - 1222 * h > 0 \\ 21.9 + 0.3 * \theta - 1206 * h > 0 \\ -3.5 + 0.38 * \theta - 25 * w > 0 \\ 0.00035 * \theta - w - 0.001 > 0 \end{cases} \tag{18}$$

$$\theta \in (10,90), w \in (0,0.004), h \in (0,0.03) \tag{19}$$

We add constraints so that the two fins will not coincide and the fin height will not exceed the difference between the inner and outer pipe diameters. Then we use Genetic Algorithm to optimize it in SPSSPRO software.

Considering the intensity and fabrication of aluminum materials in reality, we come to the conclusion that the optimal angle θ between fins is 20 °, and the optimal size of triangular fins is 0.0025m wide at the bottom edge and 0.025m high. In this case, it takes about 2 minutes for PCMs to reach 333K from 293K, which is shown in the following temperature versus time curve (Fig. 7).

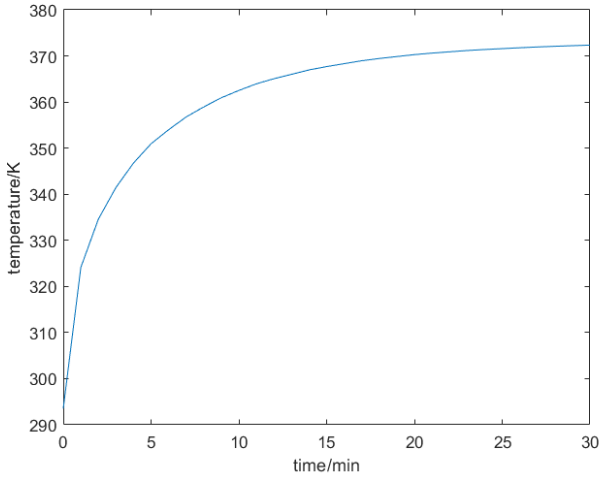


Fig 7. Average temperature versus time curve

The temperature distribution diagram and isotherm image at 2 minutes are as follows (Fig. 8).

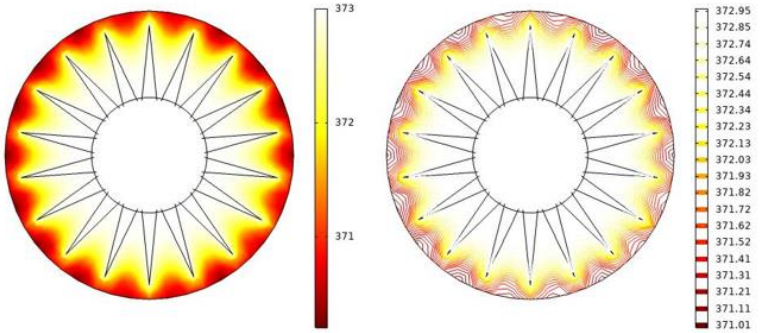


Fig 8. Temperature distribution diagram and the isotherm image

3.2. Effect on Heating Efficiency of PCMs

In order to conveniently explore the influence of triangular fin size on PCMs heating rate, we adopt the control variable method. We take the angle of 30 ° and the triangle height of 0.018m as the same, meanwhile, the triangle bottom width is 0.001 m, 0.0015 m and 0.002 m respectively. Then calculate and draw the curve of temperature change with time within 30 minutes (Fig. 9).

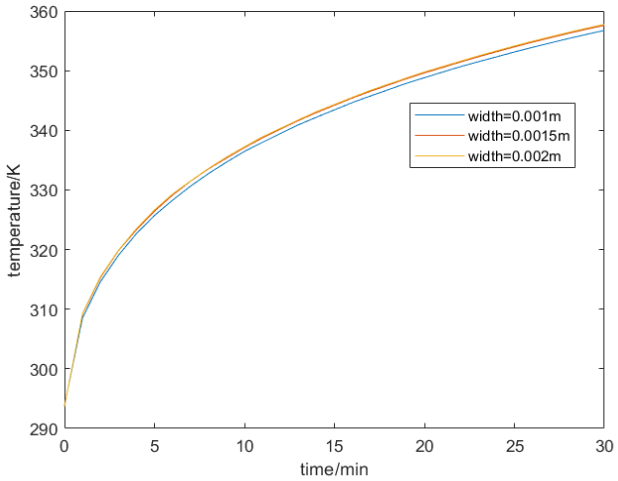


Fig 9. Temperature versus time curve at the same height

Similarly, we control the width of the bottom edge of the triangle to 0.0015m and set the triangle height to 0.01m, 0.018m and 0.025m respectively. Besides, we calculate and draw the curve of temperature change with time within 30 minutes as follows (Fig. 10).

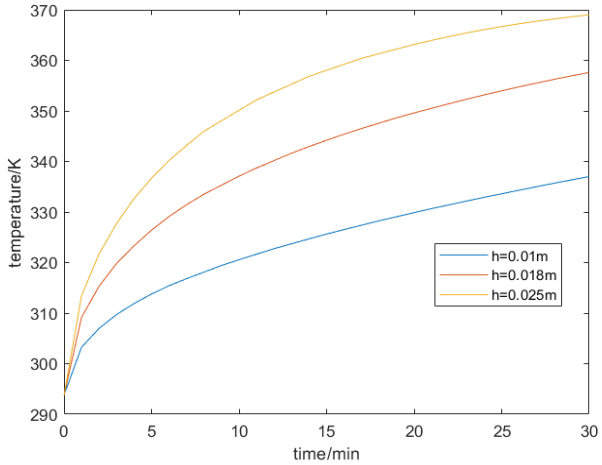


Fig 10. Temperature versus time curve at the same width

It can be concluded that the width of the bottom edge of the triangular fin has little effect on the PCMs heating rate, while the height has a significant effect on it. The higher the triangular fin is, the faster the PCMs heat. The importance ratio of angle, height and width is shown in the figure below (Fig. 11).

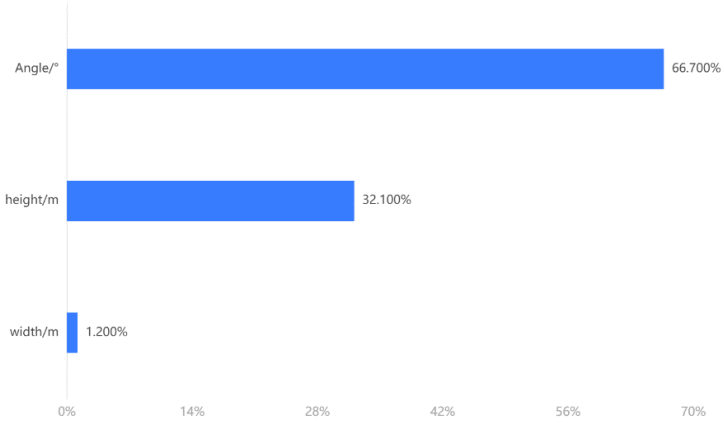


Fig 11. Importance ratio of each variable

3.3. Comparison with Rectangular Fin Structure

According to the previously established model and conclusion in Problem 1, the heat conduction effect is the best when θ=20 °. Therefore, we set the height of triangular fin the same as that of rectangular fin in Problem 1, which is 0.018m. Then we compare and analyze the data of Problem 1 and Problem 2 when θ=20 °, and draw the curve of temperature versus time as shown in the figure below (Fig. 12).

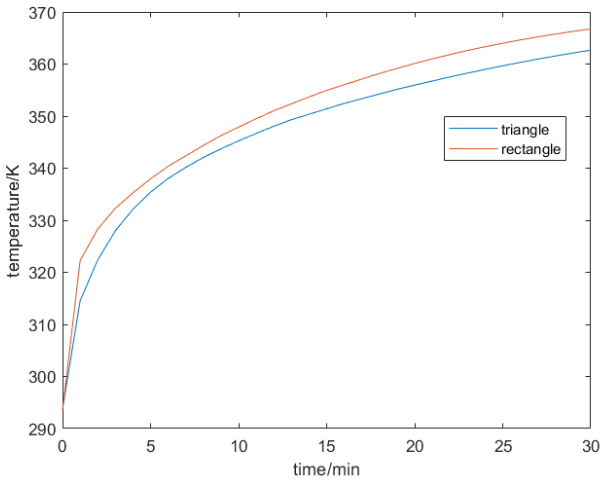


Fig 12. Average temperature versus time curve of different shape fin structure

4. Conclusions

For Problem 1, this paper uses the finite-difference methods to analyze the heat transfer effect of PCMs between two rectangular fins and get the relationship among temperature, time and distance from a fin. Through establishing a physical model based on thermodynamics and adjusting the interval angle between fins, we obtain the change rule of PCM average temperature with time under different interval angles. In order to avoid overlap of fins, this paper selects 20 ° as the final angle. The time required for PCMs to rise from room temperature (273K) to phase change temperature (333K) is about 240s. For Problem 2, this paper extends the model proposed in Problem 1 to adapt to the triangular fin structure. This paper designs a multi-objective optimization model. Results show that the heat transfer effect of PCMs is better when the interval angle is 20 °, the bottom width of the triangle is 0.0025m and the height is 0.025m. In addition, it is concluded that rectangular fins are more conducive to heat transfer than triangular fins.

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