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A Numerical Investigation to Enhance Latent Heat Thermal Energy Storage Through the Utilization of Spiral Fins Composed of Aluminum

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Abstract

The use of phase change materials (PCM) to store latent thermal energy is critical to bridging the disparity between energy production and consumption. This paper investigates numerically the use of double and quadruple spiral fins, spiral foam fins, and rectangular and cylindrical fins in the horizontal position to improve heat transfer in a shell-and-tube heat exchanger and compares the different types with the finless design. Comsol Multiphysics (6.0) was used to create a 3D model. In this simulation, natural convection was taken into account, and an approximately equal amount of phase change material was used in all cases. Water as the heat transfer fluid and commercial paraffin (RT-28) as the phase change material were used in this simulation. The results show that the melting time was significantly decreased when the spiral fins were used, and the heat transfer rate was significantly improved when the spiral foam fins were used.

Keywords: Phase change materials, Thermal conductivity enhancement, Thermal energy storage, Spiral fins, foam spiral fins.

تحقيق رقمي لتعزيز تخزين الطاقة الحرارية الكامنة من خلال استخدام الزعانف الحلزونية المكونة من الألومنيوم

الخلاصة

يعد استخدام مواد متغيرة الطور (PCM) لتخزين الطاقة الحرارية الكامنة أمرًا بالغ الأهمية لسد التفاوت بين إنتاج الطاقة واستهلاكها. يستقصي هذا البحث عدديًا في استخدام الزعانف الحلزونية المزدوجة والرباعية، والزعانف الحلزونية ذات المسامية الهشة، والزعانف المستطيلة والاسطوانية في الوضع الأفقي لتحسين نقل الحرارة في مبادل حراري ذو غلاف وأنبوب، ويقارن الأنواع المختلفة مع التصميم بدون زعانف. تم استخدام Comsol Multiphysics (6.0) لأنشاء نموذج ثلاثي الأبعاد. في هذه المحاكاة، تم أخذ الحمل الحراري الطبيعي في الاعتبار، و استخدام كمية متساوية تقريبًا من مادة متغير الطور في جميع الحالات. حيث تم استخدام الماء كسائل ناقل للحرارة والبارافين التجاري (RT-28) كمادة متغيرة الطور في هذه المحاكاة. أظهرت النتائج أن زمن الذوبان انخفض بشكل ملحوظ عند استخدام الزعانف الحلزونية، كما تحسن معدل نقل الحرارة بشكل كبير جدا عند استخدام الزعانف الحلزونية ذات المسامية الهشة.

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1. Introduction

With the continuous advancements in technology and the ongoing development of companies worldwide, there has been a significant increase in energy consumption. This surge in demand has been a cause for worry among scientists, who are now focused on devising innovative approaches to address this pressing dilemma. The energy crisis has been triggered by the depletion of natural resources, the increasing demand for energy among countries, and recent political disputes in the area. Efficient energy storage is a crucial aspect of addressing the energy issue and may significantly contribute to meeting energy demands. In light of this matter, it is essential to devise strategies for improving the quality and efficiency of latent heat thermal storage systems [1]. The enhancement of performance in latent heat energy storage systems has garnered significant attention in the academic literature. Numerous researchers have explored various methods to expedite the phase change process and improve the quality of energy storage. This is done with the aim of effectively managing energy resources.

Researchers have used several methodologies to optimise the effectiveness of latent heat storage systems. The aforementioned techniques include the incorporation of nanoadditives into the phase change material (PCM) [2], incorporation of porous media into phase change material (PCM) has been explored in previous studies [3–7]. Additionally, the use of several cascaded PCMs has been investigated in device configurations [8, 9]. Furthermore, alterations to the form and geometrical properties of the storage unit [10–13] Nevertheless, the predominant approach to boosting the performance of PCM units is the use of highly conductive fins, which effectively promote heat transmission [14–22]. Various forms of fins have been used in scholarly literature, including a range of factors that have been fine-tuned to get the most optimal configuration for enhancing heat transfer in storage systems.

In a recent work, [23] used an innovative fin design to improve the solidification process inside a hexagonal storage unit. The fins are composed of triangular fins with varying configurations. A significant improvement in the solidification process of phase change material (PCM) was seen by the modification of geometrical parameters and arrangements of the fins, as stated by the researchers.

This study examines the impact of spiral fins, foamed spiral fins, rectangular fins, and cylindrical fins on the phase change process of PCM. The performance of these different fin configurations is analyzed and compared.

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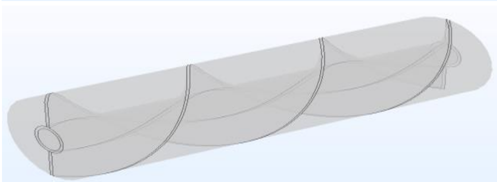
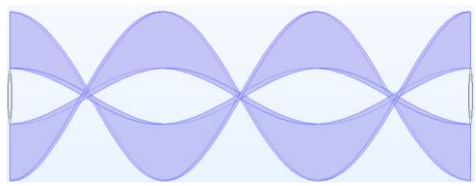
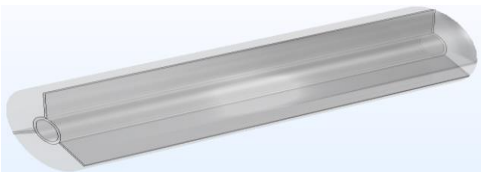
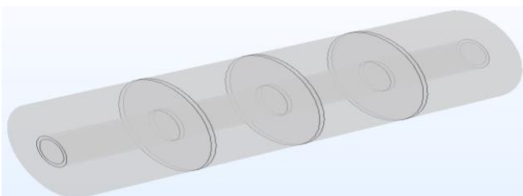
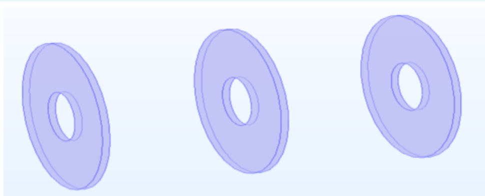
2. Numerical Modelling

The existing concept comprises a storage system enclosed within a tube casing, wherein the phase change material (PCM) is filled from the side of the casing. Simultaneously, water flows through a tube with a consistently maintained temperature. The wall that is in direct contact with the surrounding environment has been fully insulated. The inner tube walls functions as a solid medium for heat transmission during the melting process, facilitating the passage of heat from the liquid to the phase change material (PCM). Various types of fins were affixed to the solid wall in direct contact with the phase change material (PCM). These fins included double and quadruple spiral fins composed of aluminum and aluminum foam, as well as rectangular and cylindrical fins. A comparison was conducted between these configurations and a heat exchanger lacking fins. The fins have been intended to occupy a roughly identical volume. Consequently, the quantity of phase change material (PCM) employed within the shell's side is roughly equivalent across all instances. The primary aim of this study is to determine the ideal configuration for the fin design and investigate the influence of fin shape on the melting process of the phase change material (PCM). The schematic diagram of all designs employed is seen in Table (1).

Table (1): The models used for simulations

NO.	Heat exchanger	Fins / Tube
Without		
spiral fins		
rectangular fins		
cylindrical fins		

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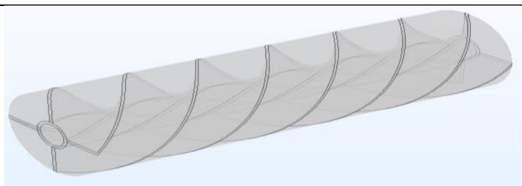
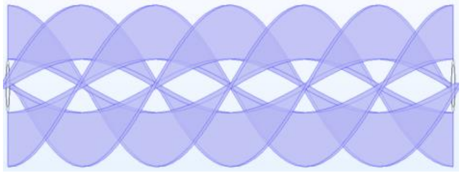
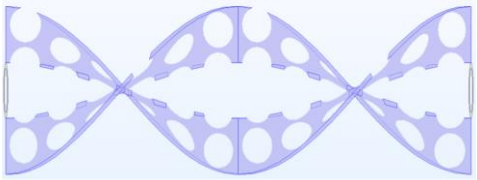
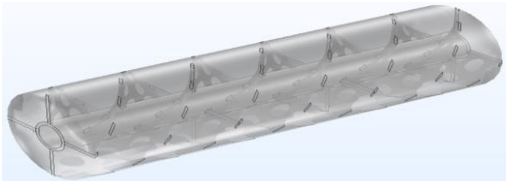
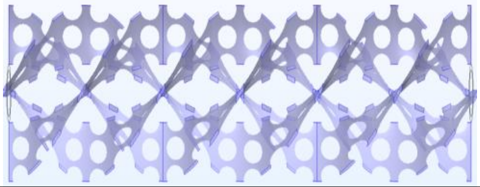
Double spiral fins		
Spiral foam fins		
Double Spiral foam fins		

In this simulation, the phase change material employed was commercial paraffin, while water was used as the heat transfer fluid. Table (2) presents the physical characteristics of paraffin RT-28.

Table (2): Thermophysical properties of paraffin wax (RT-28)

Property	**Value**
Thermal Conductivity of PCM solid	0.28 W/(m·K)
Thermal Conductivity of PCM liquid	0.17 W/(m·K)
Heat Capacity of PCM solid	1850 J/(kg·K)
Heat Capacity of PCM liquid	2050 J/(kg·K)
Density of PCM solid	860 kg/m^3
Density of PCM liquid	820 kg/m^3
Melting Temperature	301.15 K
Temperature difference	8
Latent heat of fusion	150000 J/kg
Dynamic viscosity	0.032 kg/(m·s)
Coefficient of thermal expansion	0.000385[1/K]

2.1. Governing Equations:

It has been assumed that the PCM is Newtonian and incompressible and that its physical characteristics vary very little across the operational temperature range. In this work, the enthalpy-porosity approach has been used to mimic the phase-change process. In this method, the porosity of a given cell is determined by its volume fraction. It has a porosity of 0, which is considered solid, and a porosity of 1, which is considered liquid. The equation for determining the PCM volume fraction is as follows:

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$$\alpha = \begin{cases} 0 & if\ T < T_s \\ \dfrac{T - T_s}{T_l - T_s} & if\ T_s < T < T_l \\ 1 & if\ T > T_l \end{cases} \quad (1)$$

As in porous media, a porosity of 0 corresponds to a large pressure loss in that cell, which is indicative of the presence of solid material in that cell, whereas a porosity of 1 produces no pressure loss in that cell, indicating that the cell is filled with liquid PCM. Porosity between 1 and 0 is considered the transition zone between one state and another. The simulation of phase change is calculated by adding a term to the equation of momentum operating as the source term according to the volume fraction of melted PCM in each cell. The following equation is used to compute the source term:

$$S_D = \frac{(1-\alpha)^2}{\alpha^3 + \varepsilon} V A_{mush} \quad (2)$$

In the above equation, Amushy is a constant parameter that dictates how quickly the velocity reaches zero or one and the PCM transforms into a solid or liquid state. According to previous studies in the scientific literature, a value of 10^5 for the mushy constant parameter can yield the finest results, whereas higher values may result in undesired defects [25], [26]. In the above equation, volume fraction equal to zero ($\alpha = 0$) is associated with the solid state, which returns a large amount of the source term that completely dampens the velocity of PCM, and volume fraction equal to one ($\alpha = 1$) is associated with liquid PCM and eliminates the source term as if there were no barrier in the path of the liquid PCM. Add the source term from Equation (2) to the momentum equation. The parameter ε in the denominator of Equation (2) is a negligible constant to prevent a denominator of zero. For calculating the motion of PCM, which is assumed to be a Newtonian, incompressible fluid with constant properties and negligible volume change, the following equations apply:

Continuity equation:

$$\frac{\partial \rho}{\partial t} + \nabla \cdot (\rho V) = 0 \quad (3)$$

Momentum equation:

$$\frac{\partial (\rho V)}{\partial t} + \nabla \cdot (\rho V V) = \mu \nabla^2 V - \nabla P + S_u \quad (4)$$

Where: $S_u = S_D + S_g$ (5)

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And $$S_g = \rho_{ref} g\beta\left(T - T_{ref}\right) \quad (6)$$

Also $$S_D = \frac{(1-\alpha)^2}{\alpha^3+\varepsilon} VA_{mush} \quad (7)$$

Energy equation:

$$\frac{\partial}{\partial t}(\rho H) + \nabla \cdot (\rho V H) = \nabla \cdot (K \nabla T) \quad (8)$$

The total enthalpy, H, is defined in Equation (8) as the sum of the sensible energy and the latent energy.

$$H = H_{SE} + H_{Lh} \quad (9)$$

$$H_{SE} = H_{ref} + \int_{T_{ref}}^{T} c_p \, dT \quad (10)$$

$$H_{Lh} = \alpha L_m \quad (11)$$

In this study, buoyancy force was determined by integrating the momentum equation with a source component, as the Boussinesq approximation states that the influence of buoyancy-driven forces owing to gravity and temperature change may be determined by doing so. The Boussinesq approximation's source term has been added as S_g

The fluid inside the tube is an incompressible Newtonian fluid with constant thermo-physical parameters, and the following equations are employed to calculate its motion:

Continuity equation:

$$\frac{\partial \rho_f}{\partial t} + \nabla \cdot (\rho_f V) = 0 \quad (12)$$

Momentum equation:

$$\rho_f \frac{\partial V}{\partial t} + \rho_f V(\nabla . V) = \mu_f \nabla^2 V - \nabla P \quad (13)$$

Energy equation:

$$\rho_f C_{P_f} \frac{\partial T}{\partial t} + \rho_f C_{P_f} V . \nabla T = \nabla \cdot (K_f \nabla T) \quad (14)$$

2.2. Geometry, initial and boundary conditions:

In this numerical study, COMSOL 6.0 was used, where 3D geometry was selected. The following are the primary and boundary terms.

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1- Initial Conditions

When time is zero, water and PCM temperatures are both $T_{initial}$.

2- Boundary Conditions

The external surface of the inner tube. $r = r_i$,$U = V = W = 0$ *and*

$T = T_{cold}$ *or* $T = T_{hot}$.

The external surface for the outer tube. $r = r_o$, $U = V = W = 0$ *and* $\frac{\partial T}{\partial r} = 0$.

$At = r_i$ *and* $0 \leq \theta \leq 2\pi$, $U = V = W = 0, T = T_W = T_{cold}$ *or* T_{hot}

At $r = r_o$ *and* $0 \leq \theta \leq 2\pi, U = V = W = 0 , \frac{\partial T}{\partial r} = 0$

Where:

The initial temperature of the system is 298.15 K.

The system is thermally insulated.

The initial velocity of the heat transfer fluid is 0 m/s.

conditions, non-slip on pipe surfaces.

"No viscous stress"

A fixed and variable thermal surface was used.

2.3. Validation:

A study investigated the efficiency of a copper shell and tube latent heat storage unit (LTESU) with three longitudinal fins at varying angles [24]. The heat transfer fluid was water, and the shell was made of steel. The inner surface of the tube remained constant due to a minimal temperature change. The study considered factors like melting percentage, average temperature, and LTESU performance improvement. The convective portion of the momentum and energy equation was solved numerically using the Ansys Fluent 19.0 platform and a third-order MUSCL technique. The results validated the numerical technique, showing good agreement between computational and experimental results.

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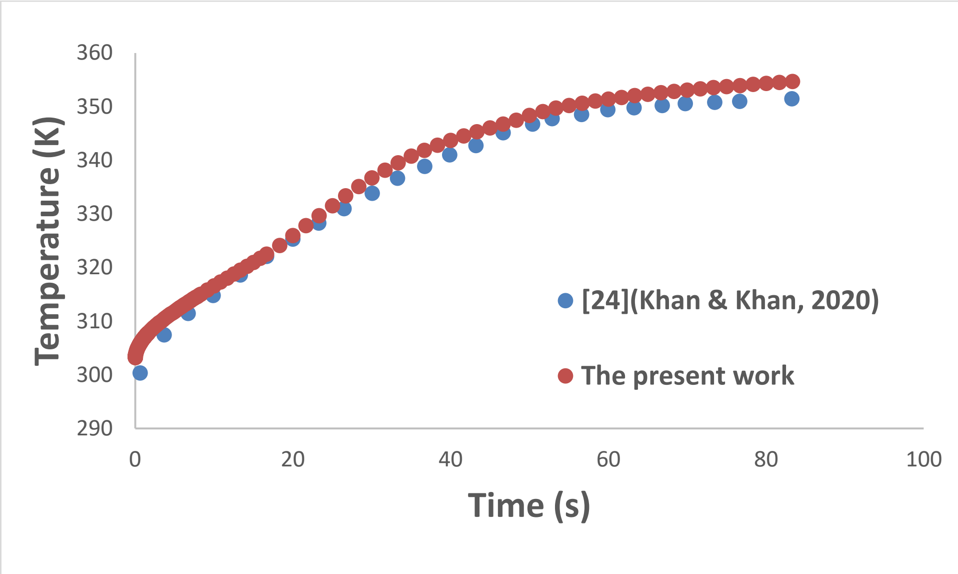
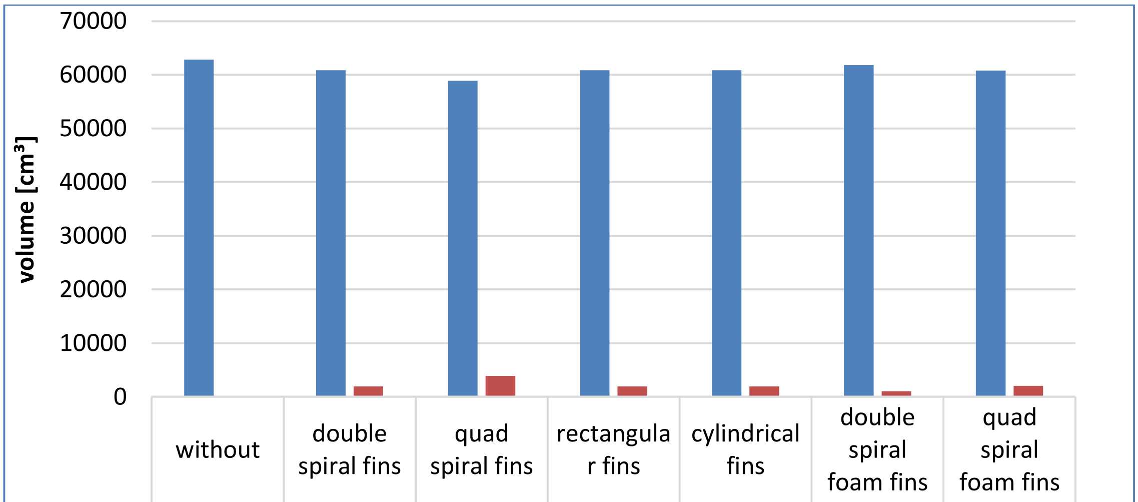


Fig. (1): Average temperature for Case 1

3. Results and Discussion

The use of fins is one of the most common ways to improve thermal performance. In order to compare the performance of the spiral and spiral foam fins with the traditional rectangular and cylindrical fin shapes, fins of approximately equal sizes were used in all cases, as in Figure (2).



	without	double spiral fins	quad spiral fins	rectangular fins	cylindrical fins	double spiral foam fins	quad spiral foam fins
PCM	62792	60820	58859	60826	60834	61783	60770
Fin	0	1959.9	3919.8	1959.3	1958.7	1014.2	2028.4

Fig. (2): PCM volume used vs. fins volume

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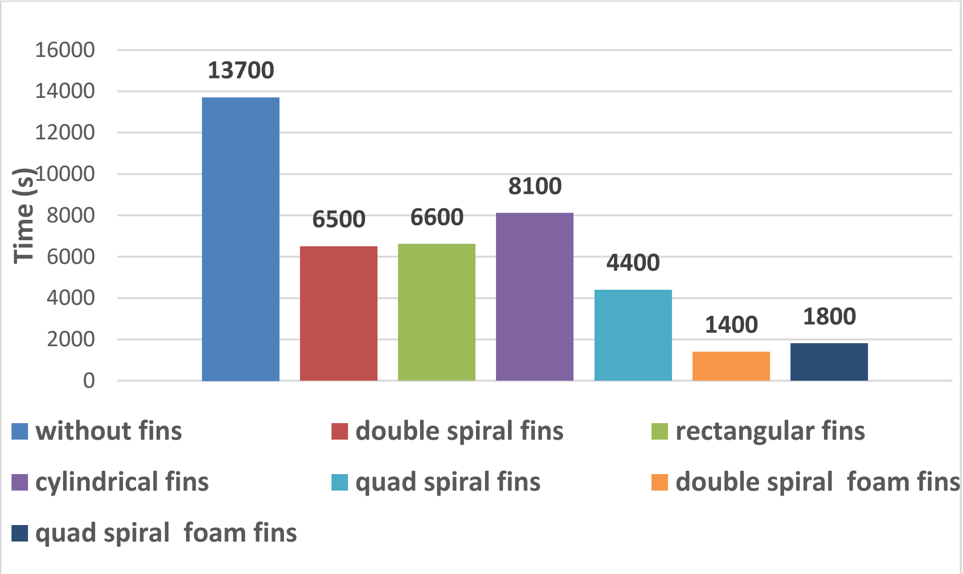
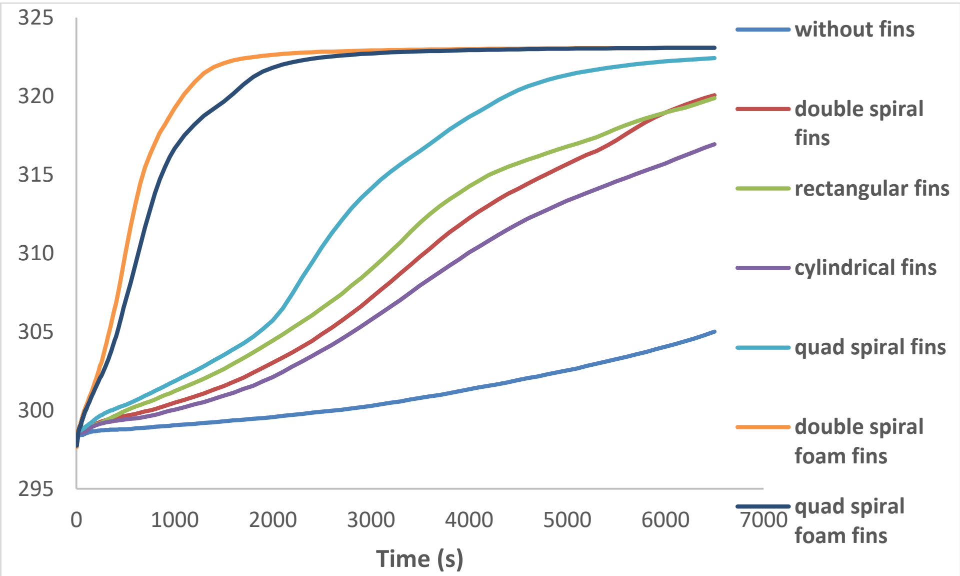


Fig. (3): Complete melting time for each case



(a)Temperature (K)

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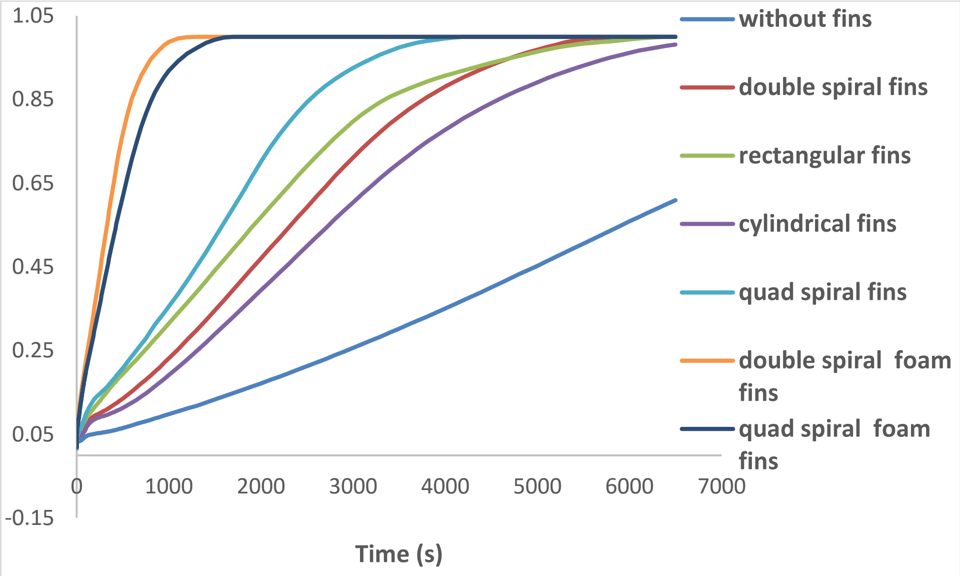
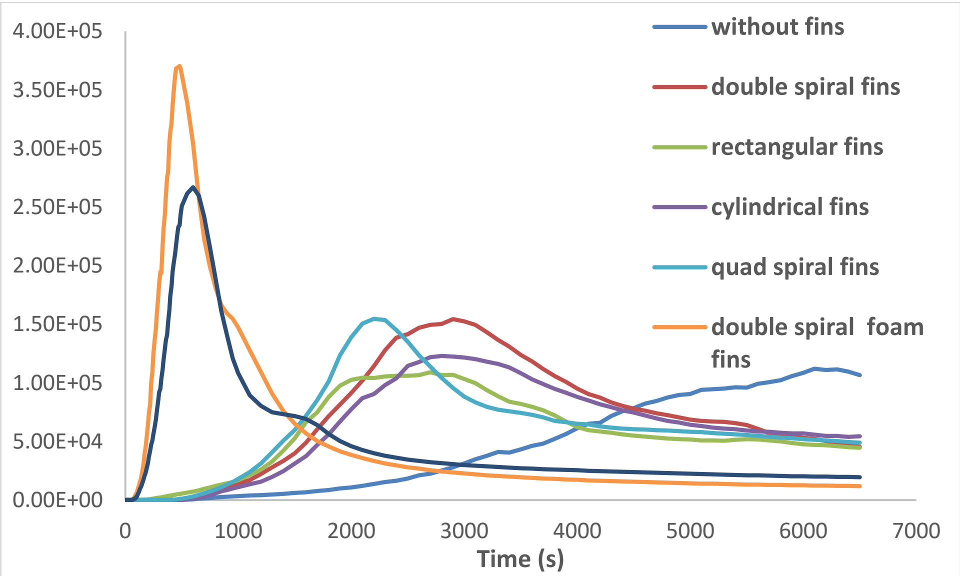


(b)Fraction of liquid phase (1)



(c) Convective heat flux magnitude (W/m^2)

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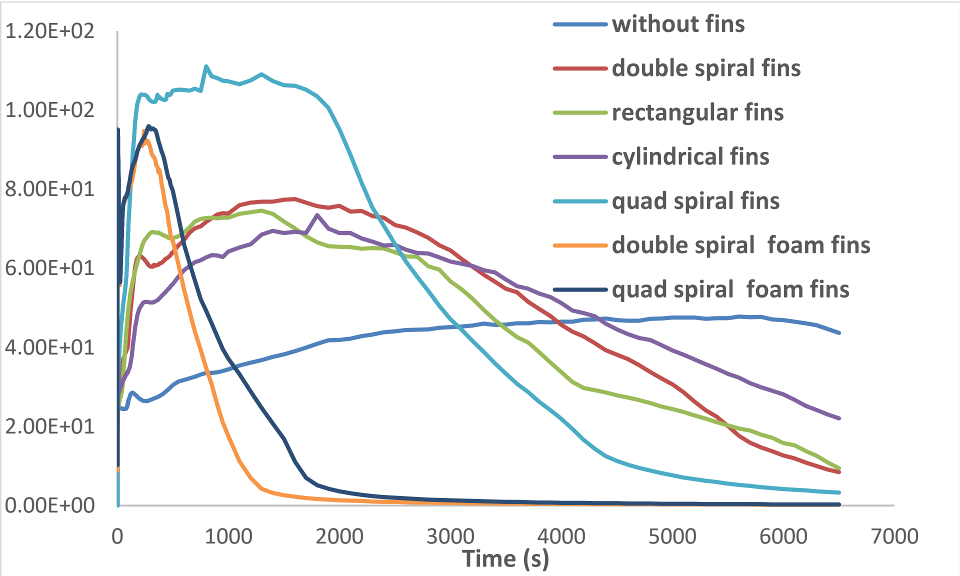

(d) Conductive heat flux magnitude(W/m^2)

Fig. (4): Compare performance for all cases

In the first stage of melting, thermal conduction is dominant, as the quadrilateral spiral fin has the most heat conduction because it has a larger surface area with PCM. After time passes, a liquid layer of PCM forms between the surface of the inner tube and the solid PCM. This layer forms resistance that reduces heat conduction. However, the melting part of the PCM increases its movement, and there becomes a difference between the density of the solid PCM and the liquid. Because of the buoyancy force, the liquid PCM rises to the top and begins to transfer heat by means of natural convection, which controls the melting process until all of the PCM melts.

In order to obtain ideal heat, transfer between the inner tube wall and the PCM, the heat conduction and natural convection phases must be activated. A spiral fin made of metal foam is designed to maintain heat conduction and activate natural convection.

Figures (4), (5), and the Figure (3) show that the best thermal performance was achieved with double spiral foam fins. The performance was better (52.56%, 51.83%, 40.88%, 67.88%, 89.78%, and 86.87%) for the double spiral, quadruple spiral, rectangular, cylindrical, double foam spiral, and four-foam spiral fins, respectively, compared to the heat exchanger that doesn't have fins.

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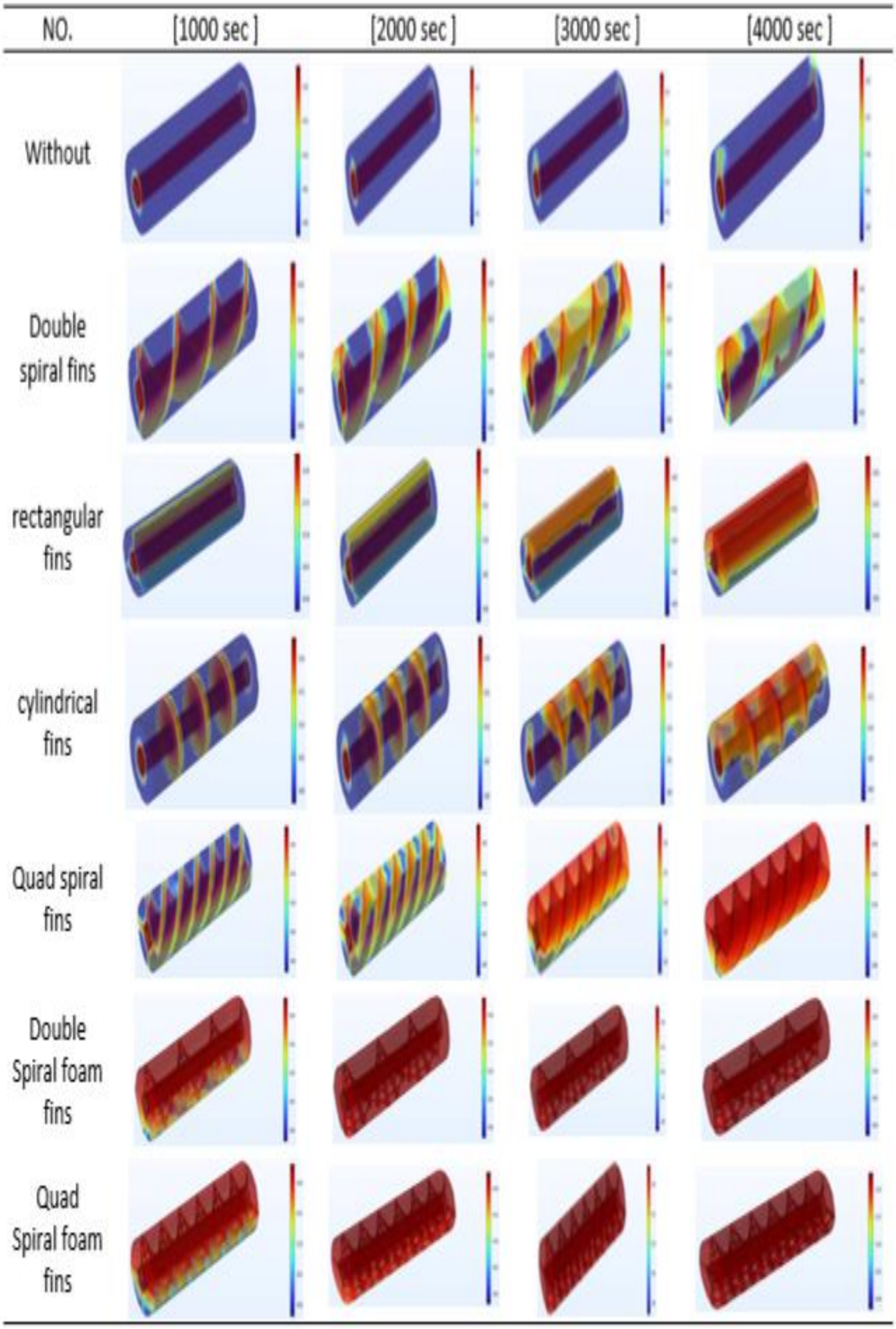

Fig. (5): Contours of temperature

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4. Conclusions

In this study, the effect of different spiral fins on the melting of PCM inside the LHTESS tube casing was measured and compared to the effects of rectangular and cylindrical fins. The study included seven simulated and compared cases. The thermal porosity method was used to simulate the phase. A constant convective flow of the heat transfer fluid was used. The fins are set to have an equal size within the heat exchanger in all cases to make it possible for them to be compared. The data were evaluated in terms of fractional solubility evolution, total solubility time, and temperature evolution within the sphere.

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