Heat transfer enhancement in solidification process by change of fins arrangements in a heat exchanger containing phase-change materials

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Abstract

Purpose – Reducing discrepancy between energy demand and supply has been a controversial issue among researchers. Thermal energy storage is a technique to decrease this difference to increase the thermal efficiency of systems. Latent heat thermal energy storage has interested many researchers over the past few decades because of its high thermal energy density and constant operating temperature. The purpose of this paper is to provide a numerical study of the solidification process in a triplex tube heat exchanger containing phase change material (PCM) RT82.

Design/methodology/approach – A two-dimensional transient model was generated using finite volume method and regarding enthalpy-porosity technique. After that, a detailed and systematic approach has been presented to modify longitudinal fins' configuration to enhance heat transfer rate in PCMs and reducing solidification time. The numerical results of this study have been validated by reference experimental results.

Findings – The ultimate model reduced solidification time up to 21.1 per cent of the Reference model which is a substantial improvement. Moreover, after testing different arrangements of rectangular fins and studying the flow pattern of liquid PCM during solidification, two general criteria was introduced so that engineers can reach the highest rate of heat transfer for a specified value of total surface area of fins. Finally, the effect of considering natural convection during solidification was studied, and the results showed that disregarding natural convection slows down the solidification process remarkably in comparison with experimental results and in fact, this assumption generates non-real estimation of solidification process.

Originality/value – The arrangement of the fins to have the best possible solidification time is the novelty in this paper.

Keywords Phase change material, Solidification, Fins arrangements, Latent heat storage

Paper type Research paper

Nomenclatures

Symbols

C_p = Specific heat of PCM (J/kg C);
G = Gravity acceleration (m/s^2);
h = Sensible enthalpy (J/kg);
H = Enthalpy (J/kg);
K = Thermal conductivity (W/m K);

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L = Latent heat (J/Kg);
P = Pressure (Pa);
T = Temperature (K);
u_i = Velocity component (m/s);
R = Tube radius (m);
Si = Momentum source term (Pa/m);
Ste = Stefan number; and
C = Mushy zone constant (kg/m^3 s).

Greek letters
ρ = Fluid density (kg/m^3);
γ = Liquid fraction coefficient;
β = Thermal expansion (K−1);
μ = Dynamic viscosity (kg/m s); and
ε = Constant.

Subscripts
i, j = Components;
ini = Initial;
HTF = Heat transfer fluid;
m = Melting;
ref = Reference;
s = Solid; and
l = Liquid.

1. Introduction

Reducing discrepancy between energy demand and supply has been a controversial issue among researchers. Thermal energy storage (TES) is a technique to decrease this difference to increase the thermal efficiency of systems. Latent heat thermal energy storage (LHTES) has interested many researchers over the past few decades because of its high thermal energy density and constant operating temperature. Over the past three decades, many researchers have studied phase change materials (PCMs) in TES systems owing to its beneficial properties including high thermal energy density per unit volume and heat storage in constant temperature. Thermal storage aspect of phase change materials has considerable contribution in decreasing the difference between the energy demand and supply of the solar thermal energy applications, particularly when use of solar thermal energy is completely dependent on solar energy as the principal source. As solar thermal energy applications need constant or near constant temperature to acquire high efficiency, using phase change materials as a LHTES can supply this constant temperature which is equivalent to the phase change temperature. The underlying principle of TES systems is absorbing and release of energy during the phase transition from solid to liquid or vice versa. Indeed, thermal energy is stored during the off-peak load period and then delivered during the peak time span. The application of phase change materials has been restricted due to their low thermal conductivity. This undesirable feature protracts melting and solidification processes. To obviate this drawback, over the past three decades, researchers have tried to introduce various techniques to raise heat transfer in phase change materials and consequently improve performance of heat exchangers. These methods include either an increase in surface area by use of finned tubes and multi-tubes heat exchangers (Agyenim *et al.*, 2010; Ermis *et al.*, 2007) or improving thermal conductivity of phase change material by inserting metal matrix or high-conductivity particles into PCMs (Baby and Balaji, 2013; Ho and Gao, 2013; Dheep and Sreekumar, 2014). The use of fins owing to its simplicity and low cost of production and maintenance is considered as a rational and suitable way to improve heat transfer in phase change material, which leads to enhancement of efficiency of thermal energy systems. In the past few years, numerous studies have been carried out to understand the influence of using various types of fins to enhance the heat transfer in the phase change material. In a study (Mosaffa *et al.*, 2012), a two-dimensional analytical model was presented to simulate solidification of phase change material in a shell-and-tube heat exchanger with radial fins. Two different geometry of shells (cylindrical and rectangular) with the same volume and surface area were compared. The results proved that the PCM solidifies more slowly in the rectangular shell than in the cylindrical one. Moreover, it was found that the operational parameters such as the inlet temperature and mass flow rate of heat transfer fluid (HTF) can affect the solidification rate of the PCM. Rathod and Banerjee (2015) investigated experimentally melting and solidification of paraffin as a phase change material in vertical shell-and-tube heat exchanger with longitudinal fins. Using fins was observed to reduce solidification time up to 43.6 per cent. They found that heat transfer rate in PCM is more dependent on inlet temperature changes of HTF compared to the mass flow rate of it. In addition, designing parameters related to fins including fin shape, fin length and fin thickness have a remarkable influence on the progress of the solidification process of the PCM. Ismail *et al.* (2001) studied numerically and experimentally the effects of various design parameters of vertically axial fins including the number of fins, fin length, fin thickness and aspect ratio of the annular space on the solidification time and the solidified mass fraction. The results indicated that the annular space size, the radial length of the fin and the number of fins have a strong influence on the overall solidification time. In addition, the results proved the benefits of using fins in deferring the undesirable effects of natural convection over the solidification process. This is because natural convection makes an uneven heat distribution throughout PCM zone, which consequently prolongs solidification process. Triplex tube heat exchanger due to having more heat transfer surface than double tube heat exchanger has higher efficiency. Darzi *et al.* (2016) conducted the numerical simulation of the melting and solidification of PCM within three various horizontal annulus configurations in a circular cylinder. The geometries of container, fins and nanoparticles as influential factors on melting and solidification rate were studied as well. Mat *et al.* (2013) conducted a numerical study on the melting process of RT82 as PCM in a triplex tube heat exchanger. They used three heating methods including heating from the inside tube, outside tube and both tubes to melt the PCM. They confirmed that utilizing a triplex tube heat exchanger with internal–external fins decreases melting time to 43.3 per cent compared to that without fins. In an attempt to improve the model introduced by Mat *et al.* (2013), Eslamnezhad and Rahimi (2017) investigated changing the angle of rectangular fins and eccentricity of inner tube during the melting process. They offered five cases with different arrangement of fins with no change in total surface area of fins which were chosen randomly with no use of optimization procedure to find the configuration of fins providing least melting time. The ultimate model provided by them, decreased melting time to 17.9 per cent of the model presented in Mat *et al.* (2013). However, that study lacks a general and systematic method to predict the optimum configuration of fins for every other case. In a similar study, Al-Abidi *et al.* (2013) investigated the phase change of a paraffin

RT82 in finned triplex tube heat exchanger but this time for solidification process. They used three freezing methods including freezing from inner tube, outer tube and both of them. They stated that freezing from both sides is more favorable than two other ones. As freezing from both sides provides more heat transfer surface and consequently needs less time to complete solidification process. This result is true for melting process as well.

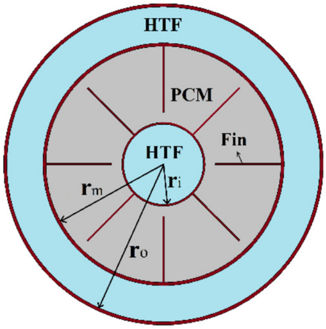
Many available articles in the literature have studied solidification process in the steady-state condition. However, in this study, it has been tried to investigate solidification process at different times so as to achieve a profound understanding of flow pattern of fluid (melted PCM) to optimize fin's configuration and maximizing heat transfer throughout the solidification process. Also, the number of studies conducted to optimize fins is very few according to the literature (Guelpa *et al.*, 2013; Sciacovelli *et al.*, 2015; Lohrasbi *et al.*, 2017), and there is a lack of general and effective method to optimize fins arrangement to achieve the least solidification or melting time. Both Sciacovelli *et al.* (2015) and Lohrasbi *et al.* (2017) investigated the optimization process of solidification process in triplex-tube heat exchanger with tree-shaped fins. In those studies, the optimization of fins' length and position was carried out using response surface method (RSM) and assuming no natural convention mechanism. Although this assumption makes a solution simpler and reduces computation time, it generates error. Because natural convection plays an important role in both melting and solidification processes (more on melting), particularly in early stages and ignoring this, deviates numerical results from experimental data according to Al-Abidi *et al.* (2013), Mahdi and Nsofor (2016). This assumption also made researchers of studies (Sciacovelli *et al.*, 2015; Lohrasbi *et al.*, 2017) able to use identical design for each tree-shaped fin throughout the cross-section of the tube as RSM method does not provide reasonable and accurate results when natural convention is considered. Therefore, a need of new approach far from these limits for optimization of fins' configuration during solidification process is essential. Use of phase change material in heat transfer enhancement studies in cavities, over cylinder, wavy surfaces and also using nano-fluids were presented by Ghalambaz *et al.* (2017a, 2017b), Chamkha *et al.* (2016), Doostanidezfuli *et al.* (2017), Kashani *et al.* (2012), Esmaeilpour and Abdollahzadeh (2011), Abdollahzadeh and Esmaeilpour (2015).

This study investigates the effect of considering natural convention during fins modification process which is a novel matter. In fact, almost all previous studies have modified fins' shape or arrangement without natural convention assumption and just considering conduction. Here, to raise heat transfer and reduce the solidification time, we tried to find a systematic and general method to correct the length and position of the rectangular fins while the total surface area of the fins is kept constant. Furthermore, the ultimate model of the present study reduces solidification time of model in (Al-Abidi *et al.*, 2013) up to 21 per cent with no addition in fins' total area or volume which increases the efficiency of the system considerably.

2. Numerical analysis

2.1 Physical model

The schematic model of a triplex tube heat exchanger of Al-Abidi *et al.* (2013) is shown in Figure 1. The model has inner tube radius of 25.4 mm (r_i) and a thickness of 1.2 mm. Also, the radius of middle tube (r_m) and outer tube (r_o) are 75 and 100 mm, respectively, with a thickness of 2 mm. All pipes' material was selected as cooper to ensure high thermal conductivity. As can be seen in this figure, eight fins in length of 42 mm and a



Source: Al-Abidi *et al.* (2013)

Figure 1. Schematic model of triplex tubes heat exchanger

thickness of 1 mm have been put on the middle and the inner tubes. Water flows inside the outer and the inner tubes as a heat transfer fluid, while the phase change material which its commercial name is RT82, is located in the middle tube. Physical and thermodynamic properties of the phase change material (RT82) and cooper are presented in Table I.

2.2 Governing equations

The standard Navier–Stokes and energy equations govern the equations pertaining to the fluid motion and the temperature distribution of phase change materials. The flow in annulus during solidification is supposed to be laminar, incompressible and transient. The viscose dissipation is negligible. The thermo-physical properties of the HTF and PCMs are independent of temperature. The effect of natural convection is considered according to the Boussinesq approximation and just density variation of the PCM linked by body-force term in momentum equation is considered. The Boussinesq approximation is defined as:

$$\rho = \frac{\rho_l}{(\beta(T - T_l) + 1)} \tag{1}$$

Property	RT82	Cooper
Density of PCM, liquid, ρ_1 (kg/m^3)	770	–
Density of PCM, solid, ρ_s (kg/m^3)	950	8,978
Specific heat of PCM, C_P (J/kg K)	2,000	381
Melting temperature, T_m (K)	350.15-358.15	–
Latent heat, L (J/kg)	176,000	–
Dynamic viscosity, μ (kg/m s)	0.03499	–
Thermal conductivity, k (W/m K)	0.2	387.6
Thermal expansion coefficient, β (1/K)	0.001	–

Table I. Thermo-physical properties of the PCM and copper

The continuity, momentum and the energy equations are, respectively, defined as follows:

Continuity equation:

$$\partial_t(\rho) + \partial_i(\rho u_i) = 0 \tag{2}$$

Momentum equation

$$\partial_t(\rho u_i) + \partial_i(\rho u_i u_j) = \mu\,\partial_{jj}u_i - \partial_i p \;+\; \rho g_i + S_i \tag{3}$$

Energy equation:

$$\partial_t(\rho H) + \partial_t(\rho \Delta H) + \partial_i(\rho u_i H) = \partial_i(k\partial_i T) \tag{4}$$

Here, ρ is the density of the PCM, u_i is the fluid velocity, μ is dynamic viscosity, p is the pressure, g is the gravity acceleration, k is thermal conductivity, T is the fluid temperature of PCM and h is sensible enthalpy of the phase change material. Moreover, sensible enthalpy can be expressed as:

$$h = h_{ref} + \int_{T_{ref}}^{T} c_p\ \Delta T \tag{5}$$

The total enthalpy is defined as follows:

$$H = h\ +\ \Delta H \tag{6}$$

where href is the reference enthalpy at the reference temperature (T_{ref} = 273 K), C_p is the specific heat at constant pressure and ΔH is the latent heat of phase change material. Furthermore, γ is the liquid fraction varied during the phase transition when temperature is $T_l > T > T_s$, which can be defined as:

$$\gamma = \Delta H/L \tag{7}$$

$$\gamma = \begin{cases} 0 & if \quad T < T_s \\ 1 & if \quad T > T_1 \\ (T - T_s)/(T_l - T_s) & if \quad T_l > T > T_s \end{cases} \tag{8}$$

The source term S_i in momentum equation, equation (2), is described as:

$$S_i = C(1-\gamma)^2 \frac{u_i}{\gamma^3 + \varepsilon} \tag{9}$$

where C is the constant describing how steeply the velocity is reduced to 0 when the material solidifies. The value of this constant changes from 104 to 107. Here, the value of this constant is regarded to be 105 according to Al-Abidi *et al.* (2013). Furthermore, ε is a small number (0.001), which is explained in Ye *et al.* (2011).

2.3 Initial and boundary conditions

The initial temperature of phase change material is 363.15 K (90°C) and PCM is in liquid phase at this temperature. Also, the temperature of the tube walls is assumed to be equal to the heat transfer fluid temperature at about 338.15 K (65°C). To have more accurate solution, the temperature of the walls and fins was supposed to be variable and as a result, walls and fins zones were considered in numerical computation. Because of the heat transfer from both inner and outer tubes, the boundary conditions are defined as follows:

$$r = 25.4\,mm \rightarrow \; T = T_{HTF} \tag{10}$$

$$r = 77\,mm \; \rightarrow \; T = T_{HTF} \tag{11}$$

2.4 Numerical approach

The transient two-dimensional problem has been solved numerically using finite volume method. Enthalpy–porosity approach was used to model the problem (Patankar, 1980). Moreover, a pressure-based model was selected because of the low velocity flow of PCM. For the pressure correction equation, the pressure staggering option method, and for pressure–velocity coupling, semi-implicit pressure-linked equation method was used. To solve the momentum and energy equations Quadratic upwind differencing scheme was adopted. The values of under-relaxation factors for pressure, velocity, energy and volume fraction were 0.4, 0.2, 1 and 0.9, respectively. The convergence was checked at each time step, with convergence criterion of $10 - 3$ for continuity and momentum equations and $10 - 6$ for energy equation.

2.5 Mesh independency and time-step independency

To check uniqueness of the numerical results of this study, mesh independency and time-step independency given by Al-Abidi *et al.* (2013) have been performed. Three different grid sizes of 13,334, 7,760 and 2,446 cells for the same time-step of 0.1 s, were tested to ensure the independency of the grid size for the numerical solution. The results are shown in Figure 2. As seen in this figure, cases with 7,760 and 13,334 cells produce nearly the same values of liquid fraction. As a result, to reduce the calculations, the number of cells were chosen as 7,760. Furthermore, testing time-step independency was done for different time steps which include 0.05, 0.1 and 0.5 s using 7,760 cells. The result is shown in Figure 3, which presents

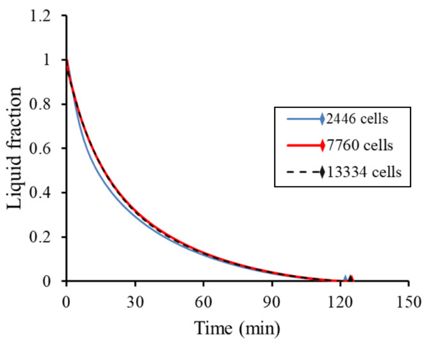


Figure 2.
Grid size dependence of the numerical solution

that cases with the time steps of 0.1 and 0.05 s have approximately identical results. Consequently, the time-step 0.1 s was used to reduce calculations.

2.6 Validation of the numerical model

To validate present numerical results, the experimental results of the model presented in Al-Abidi *et al.* (2013) have been produced as shown in Figure 4. According to this figure, numerical result of this study has an acceptable agreement with experimental ones of Al-Abidi *et al.* (2013). As seen in this figure, at the end of solidification, the temperature fall in the experimental curve was larger than that of the numerical one. This can be due to the heat loss to the environment as explained in Al-Abidi *et al.* (2013).

3. Results and discussions

3.1 The effect of considering natural convection on heat transfer during solidification

In this part, the influence of consideration of natural convection during solidification process is investigated. The previous numerical results (Al-Abidi *et al.*, 2013) have been produced again both with consideration of natural convection and without considering this assumption. The results show that natural convection accelerates solidification process through creating convection cells, which in turn increases heat transfer rate and consequently reduces solidification time. As can be seen in Figure 5, in the case with consideration of natural convection, the measure of liquid fraction is less than the other case at any times during the process and the overall time of solidification is also less than the case without convection assumption up to 28.1 min, which represents 22.6 per cent error. As the case with convection assumption has been validated with experimental results and a good

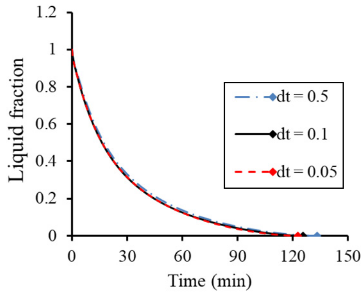


Figure 3. Time step dependence of the numerical solution

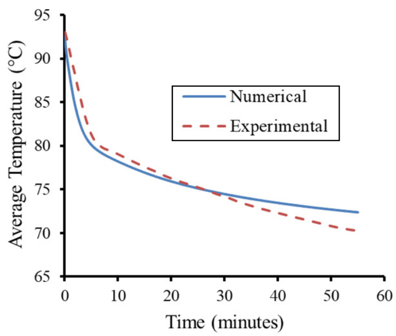


Figure 4. Comparison the values of average temperature of the present numerical solution with experimental work of Al-Abidi *et al.* (2013)

agreement between numerical and experimental results has been obtained, it can be concluded that consideration of natural convention for simulating solidification is necessary to gain accurate results. In addition, according to Figure 6, disregarding natural convection makes the simulation of solidification process too artificial and far from reality. Because when natural convection is disregarded, only conduction is considered as heat transfer

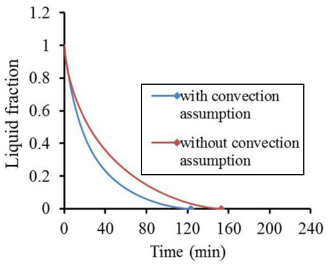


Figure 5.
Comparison of liquid fraction values between cases with and without consideration of natural convection

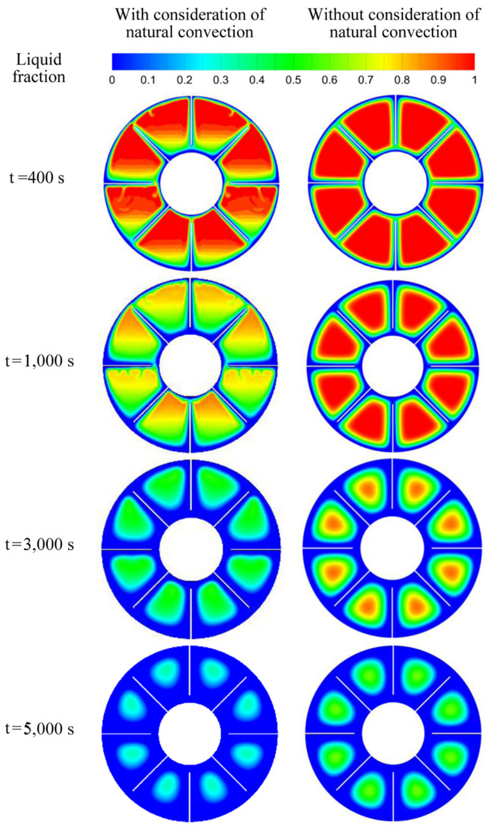


Figure 6.
Liquid fraction contours for different cases of natural convection at three selected times

mechanism and the melted PCM is treated like a solid, which is an inaccurate approximation.

3.2 First step to increase solidification rate

As it can be seen in Figure 7, in a previous model (Al-Abidi *et al.*, 2013), at 1,000 s, the area near the fins freezes before other areas far from fins, and in these cases, a noticeable deal of liquid PCM is trapped in the areas between each two adjacent fins. Another problem with this model is that because a quite large angular distance exists between fins, the cold cannot penetrate enough in angular direction, although it is able to penetrate in radial direction well, which in turn helps trapping liquid materials in zones between fins. Thus, to obviate this problem, consecutive divisions of fins were done by making gradual reduction in the length of the fins and simultaneously increasing their numbers (the total area of fins is kept constant). As seen in Figure 7, first, this work causes heat transfers both in radial and angular directions more balanced in comparison with the reference model. However, by continuing this work, the solidification rate decreases due to creation of gap between the fins of the inner and outer tubes. In fact, this time, the cold penetration decreases in radial direction unlike the reference model. To make comparison between different cases more conveniently and also accurately, in each case, all the fins are situated at the same distance from each other with similar arrangement. The details of divisions have been brought in Table II. It must be mentioned that throughout this study, all the fins are perpendicular to the tubes. According to the contours shown in Figure 7, the case with division factor $n = 1.5$ has a minimum solidification time compared to others and consequently is the most

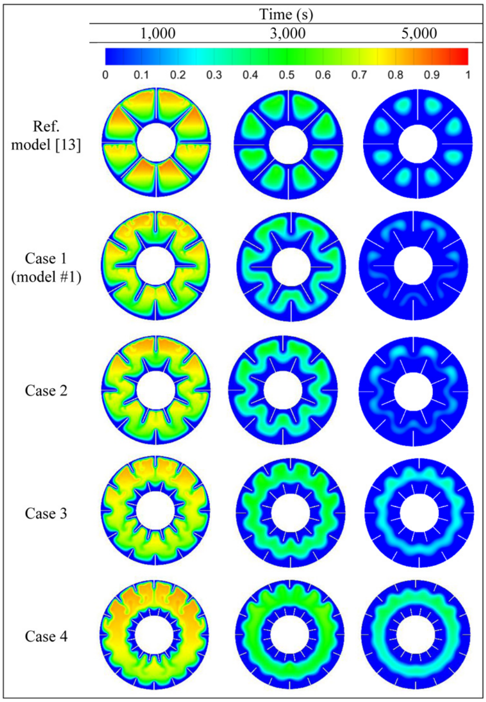


Figure 7.
Liquid fraction contours for different cases at three selected times

appropriate case for implementation of further changes in fins' length and distance between fins so as to continue decreasing solidification time more which will be explained in the next section. The total solidification time related to the Case 1 is 102 min, which provides 17.8 per cent reduction in solidification time compared to the reference model. This case consists of 12 fins with the length of 28 mm and placed at positions with 30° difference. Moreover, this case, compared to other ones, has less number of fins, which lead to reduction in production costs. From now on, we call Case 1 as Model 1.

3.3 Second step to modify rectangular fins

In this part, distances between the fins of Model 1 are modified so as to reduce the solidification time more. At first, Model 1, which consists of 12 fins, 6 of them on the outer tube and 6 on the inner tube each at 30° centrally from each other, is considered as shown in Figure 8. According to Figure 9, in Model 1, because of the upward movement of liquids which have higher temperature (less density) and downward movement of liquids which have lower temperature (more density), heat is accumulated more in upper half of the tube which brings about an unbalance situation in solidification rate in the upper and lower half of the tube and consequently engendering an increase in overall solidification time. Therefore, to raise solidification rate in the upper half of the tube, angular distances between

Model	Division factor	Length of the fins (mm)	Number of the fins	Overall solidification time (min)
Model in Ref. (Al-Abidi *et al.*, 2013)	1	42	8	124.1
Case 1 (model #1)	1.5	28	12	102.0
Case 2	2	21	16	110.1
Case 3	3	14	24	130.3
Case 4	4	10.5	32	163.4

Table II. Successive divisions of the fins for selected division factors

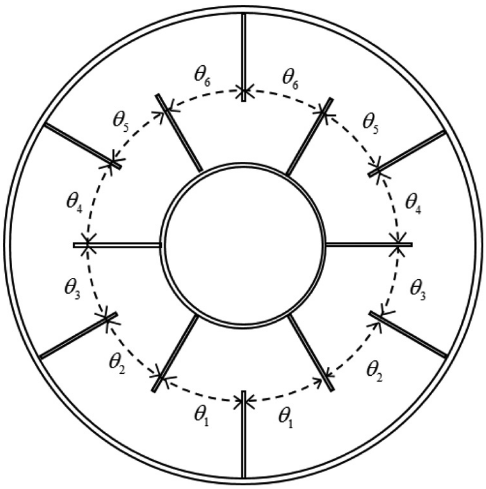


Figure 8. The geometry of Model 1 and angular distances between fins

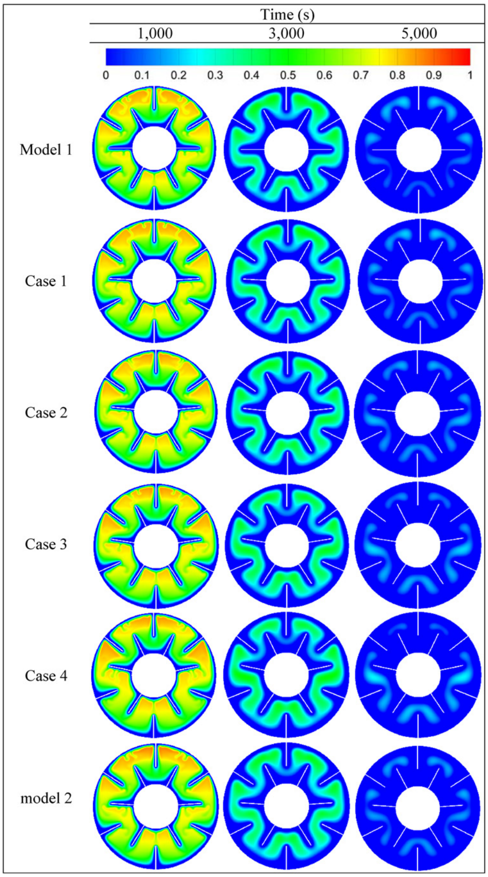


Figure 9.
Contours of liquid fraction for different cases at three different times

the fins of the upper half of the tube ($\theta_4, \theta_5, \theta_6$) are gradually decreased in a regular way and instead added to the angular distances between the fins of the lower half of the tube ($\theta_1, \theta_2, \theta_3$).

According to Table III and Figure 8, the central angles between the upper half fins of the tube is gradually decreased by step size of 1° and the same amount is added to the distance between the fins in the lower half. As shown in Figure 9, this change step by step causes an increase in solidification rate in upper half of the tube, which results from an increase in total

Table III.
The values of the different angles between fins for showing distances

Model	θ_1	θ_2	θ_3	θ_4	θ_5	θ_6	Total solidification time
Model #1	30	30	30	30	30	30	102.0
Case 1	31	31	31	29	29	29	100.2
Case 2	32	32	32	28	28	28	102.5
Case 3	33	33	33	27	27	27	106.3
Case 4	34	34	34	26	26	26	110.7
Model 2	32	31	31	29	29	28	98.4

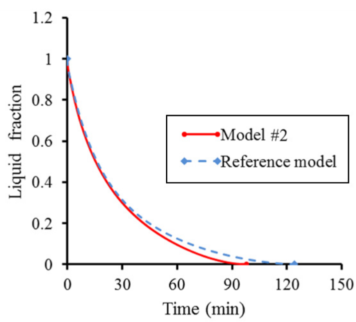
heat transfer area of the fins in upper half of the tube. Simultaneously, the total heat transfer area of the fins and the solidification rate reduces in the lower half. These changes continue until the quantity of liquids in the upper half section becomes negligible. After that, by comparing contours of Cases 1-4 at different times and regarding the key point that all solid material at final stages of phase transitions process must be melted at the same time throughout the annulus area, the ultimate model is selected which has the minimum complete solidification time compared to others which is 94.8 min. This model is named Model 2 and is shown in Figure 9.

4. Conclusions

In this study, a systematic and innovative method for modifying rectangular fins of triplex tube heat exchanger to enhance heat transfer over solidification process has been proposed. The total area and or (volume) of the fins is kept unchanged to find their most appropriate configuration. The central result of this study is that if the arrangement and dimension of the rectangular fins are selected in a way that, first, keeps a balance in heat penetration both in radial and angular directions according to Section 3.2, and in the second place, all liquid PCMs become solid simultaneously throughout the cross-section at final stages of the solidification process, according to Section 3.3, the least solidification time will be achieved. In other words, using the other arrangements or lengths of fins (with the constant area of fins) will provide the same minimum solidification time if the fins are modified by considering two above-mentioned rules. This result is general and independent of dimensions of the triplex tube exchanger and the type of PCM, which enables engineers to design much more efficient heat storage systems. Furthermore, as shown in Table IV and Figure 10, Model 2 reduces overall solidification time to 21.1 per cent of the model presented in Al-Abidi *et al.* (2013). In addition, investigating the effect of natural convection consideration during solidification showed that disregarding natural convection during solidification for the reference model generates quite large amount of error in comparison to

Model	Complete solidification time (minutes)
Model in Ref. (Al-Abidi *et al.*, 2013)	124.1
Model #1	102.0
Model # 2	97.9

Table IV. Comparison of complete solidification time for different models



Source: Al-Abidi *et al.* (2013)

Figure 10. Comparison of liquid fraction between Model 2 and reference model

the experimental results, and therefore, natural convention should be considered during solidification process in numerical simulations if obtaining accurate and realistic results matters.

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