Simulation of Triplex Tube Thermal Energy Storage System Using High Temperature Phase Change Material
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Abstract:
To increase the capacity factor of a concentrated solar thermal power plant (CSTPP) beyond the hours of sunlight, the use of thermal energy storage systems (TES) can be a promising solution. Phase change materials (PCMs) can store latent thermal energy in the course of the melting process and release it when solar energy is not available. Generally, PCMs have low thermal conductivity. One of the most commercially promising solutions is the application of an extended heat transfer surface inside the PCM container. Moreover, the distance of the heat transfer fluid (HTF) to the core of the PCM in a TES system can affect the storage performance. Accordingly, a triplex tube heat exchanger with eight fins is considered in this paper, to investigate the impact of the different velocity of HTF and different entrance pattern in a vertical PCM container. Notably, the middle enclosure of the triplex tube is filled with PCM. Numerical analysis using an enthalpy porosity technique revealed that increasing HTF velocity reduces the charging time. Also, when the HTF enters from the bottom of the container, the storage time will diminish owing to a natural convection side-effect, but if the HTF flows downward, the amount of sensible thermal energy storage is higher than in the other cases.

Keywords: Thermal Energy Storage, PCM, Triplex Tube Heat Exchanger, CSP

Nomenclature

\[ C \] Mushy zone constant \( \frac{(kg/m^3)s}{(kg/m^3)s} \)
\[ C_p \] Specific heat of PCM \( (J/kg°C) \)
\[ g_i \] Gravity acceleration in the i-direction \( (m/s^2) \)
\[ h \] Sensible enthalpy \( (J/kg) \)
\[ H \] Enthalpy \( (J/kg) \)
\[ L \] Latent heat fusion \( (J/kg) \)
\[ P \] Pressure \( (Pa) \)
\[ T \] Temperature \( (°C or K) \)
\[ r \] Tube radius \( (m) \)
\[ K \] Thermal conductivity \( (W/m K) \)

Greek letters

\[ \mu \] Dynamic viscosity \( (kg/m s) \)
\[ \epsilon \] Constant

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\( \rho \)

Fluid density (kg/m³)

1. Introduction

In light of the environmental pollution caused by fossil fuels, the use of renewable energy has made significant progress in the field of power generation, and electricity generation specifically. Solar power is a renewable energy that allows the harvest of thermal energy for a heat engine.

Since renewable energies are not permanently available, it is necessary to think about ways to increase capacity factors. Often, this involves the use of a thermal energy storage system (TES). Thermal energy storage systems play an essential role in improving the reliability and performance of solar energy. In variant TES systems, the latent thermal energy system employs phase change materials (PCMs) to store a massive amount of thermal energy with a rather monotonic constant temperature. These systems have various applications in diverse industries, including the construction industry, power plants, air and space, and cold and cold-run systems [1].

The most considerable deficiency of these materials is their low thermal conductivity [2]. This increases the time taken for charging and discharging processes in the TES. In this regard, the proposed system should be considered, to improve thermal conductivity [3]. Means to increase thermal conductivity include the use of expanded surfaces, encapsulated phase change materials, and adding high conductivity materials to the PCM [4-7].

Because of the high temperatures required for concentrated solar power (CSP) systems, eutectic salts are used as the phase change materials. Molten salts have been used as phase change material to reduce the system’s costs. It should be noted that molten salts have low thermal conductivity, which limits the heat transfer of HTF and PCM during the charging process [1,2]. Some researchers have used the encapsulated PCM in their work [9]. In other studies, increasing the heat transfer surface, adding heat pipes, adding fins to the geometry of the storage system are considered [10, 11]. In experimental work performed by [3][12] in a cylindrical storage system with transverse fins, the natural convection has an effective role in improving the storage system. It is concluded that increasing the inlet temperature of the fluid decreases the melting process time. Also, in the case of discharging, increasing HTF flow rate does not have a significant effect on discharging process. This result derives from this reality; when the PCM freezes on the wall, it prompts a new thermal resistance in front of the heat transfer surface and limits natural convection.

Tay et al. [13] represented a 4-tube heat storage tank to simulate in two different ways. In the first case, the effect of natural convection was neglected, and in the second case, the combination of natural convection and the thermal conductivity was simulated. The numerical results were completely different from the experimental results. In the simulation, the phase change was started from the top and bottom of the PCM container, and it was completed in the middle of the tube. In the experimental case, the phase change began at the end of the container and was completed at the top of the reservoir.
According to simulation accomplished by Tay et al. [13], where the melting process formed in a shell and tube heat exchanger, it was revealed that if the only heat transfer mechanism is conduction, the melting process will take longer to complete than when the conduction combined with the convection mechanism. In addition, increasing the height of the geometric models of the storage system would increase the PCM melting rate. Numerical simulation by Tao et al. [14] was used in a vertical storage tank using enthalpy techniques, and it was shown that increasing the velocity of the HTF would increase the PCM melting rate.

Longeon et al. [15] discussed the location of the HTF for inputting to the storage tank. Accordingly, if the HTF enters from the upside of the container, it would be suitable for the charging process, whereas if the entrance is located on the downside, it is proper for the discharging process. Investigations of different charging and discharging process in different initial conditions have been conducted by Mastani Joybari et al. [16]. In this study, they surveyed different thermal conditions for internal and external pipes under heat and cold thermal loads. Al-Abidi et al. [17] examined a triplex tube heat exchanger in two models. First, they used fins in their investigation, and secondly a simple model was considered. After an experimental investigation, it was revealed that the use of fins could reduce melting time significantly. Regarding the effect of mass flow rates and HTF temperature, Al-Abidi et al. [18] held that the effect of the input temperature is greater than the effect of the input velocity, and reduced the melting time.

In the study of the layout of fins in three-pipe heat exchangers that was presented by Eslamnezhad & Rahimi [19], it was shown that the ideal layout should prepare conditions for heat flow to move upwards. Mahfuz et al. [20] claimed that by using thermal energy storage systems at concentrating solar thermal power plant of Shiraz, Iran, exergy will increase by up to 30%. Using PCMs H190 and H250 would lead to an increase in the exergy of the system to 17% and 21%, respectively. On the other hand, PCMs with higher melting temperature give higher exergy in the system.

According to the aforementioned studies, latent thermal energy storage systems show promise in use as TES systems in CSP plants and have the potential to increase efficiency and performance. In this paper, a triplex-annulus-tube heat exchanger with fins is considered as a latent thermal energy storage system. The intermediate layer is filled with PSM, and the HTF can flow in the inner and outer layers of the triplex tube. In addition to the conduction heat transfer mechanism, since the natural convection has a significant effect on melting process time, the purpose of this study is to investigate the melting rate in a different patterns of HTS flow direction. Notably, the flow direction and mass flow rate of HTF can affect the inception point of PCM melting, to influence the natural convection heat rate in the TES system.

2. System Description and PCM Properties

A triplex tube heat exchanger is considered in this numerical simulation. The schematic of a 2D model of the triplex tube heat exchanger is depicted in Figure 1, and the characteristics are demonstrated in Table 1. Tubes are made of carbon steel 1.0425 [21], which will prevent the corrosion of PCM and tubes. In this triplex tube system, the middle tube is filled with PCM and
HTF passes through two internal and external tubes. In this way, the HTF surrounds the PCM, and it can influence the PCM from two sides during charging and discharging of the PCM.

Table 1. Geometrical dimensions of the 2D simulation model

| Parameters                  | Dimensions (mm) |
|-----------------------------|-----------------|
| Internal pipe radius ($r_i$) | 25.4            |
| Internal pipes thickness    | 1.2             |
| External tube radius ($r_o$) | 100             |
| Middle tube radius ($r_m$)  | 75              |
| Middle and internal pipes thickness | 2             |
| Fins length                | 8               |
| Fins thickness             | 1               |

Molten salts are a promising candidate as PCM at concentrating solar power plants, as their decomposition temperatures are about 500° C. Using different mixtures of molten salts, PCMs with selective melting temperature can be attained [22]. Notably, in a study to achieve a lower melting temperature in eutectic PCM (below 200° C), LiNO₃ and Ca(NO₃)₂ is added to other nitrate salt mixtures [23]. Typically, eutectic salts of sodium nitrate and potassium nitrate have been considered as PCM. Moreover, in the high-temperature CSP systems [24], molten salts are used as HTF, too. It is reported that using molten salts as PCM and HTF would increase the annual efficiency of the conversion of solar energy to electricity by up to 18% [25]. In this numerical study, the molten salts as PCM comprise 60% NaNO₃– 40% KNO₃, and the HTF is composed of 40%KNO₃ – 60%KNO₃. Table 2 shows the thermal properties of the PCM and HTF.

Table 2. PCM and HTF thermodynamic properties

| Properties                          | PCM [17] | RT82 | PCM, Molten salts [10] | HTF [26] |
|-------------------------------------|----------|------|------------------------|----------|
| Density (kg/m³)                     | 950      | 1750 | 1820                   |          |
| Specific thermal capacity (J/kg.K)  | 2000     | 1425 | 1553                   |          |
| Thermal conductivity (W/m.K)        | 0.2      | 0.8  | 0.52                   |          |
| Viscosity (Pa.s)                    | 0.3499   | 0.00259 | 0.00326              |          |
| Thermal distribution coefficient (1/K)| 0.001    | 0.002 | -                      |          |
| Melting point (K)                   | 358.15   | –    | 493                    | -        |
|                                     | 350.15   |      |                        | -        |
| Phase change enthalpy (kJ/kg)       | 176      | 108.67 | -                      |          |
3. Numerical Simulation and Governing Equations

The assumptions presumed in the problem definition, the governing equations and the modeling method, as well as the boundary conditions, are expressed as follows:

- The fluid flow is laminar and incompressible.
- The thermodynamic properties of PCM and HTF are not dependent on the temperature except density.
- The non-slip boundary condition is applied to the walls.
- The direction of gravitational acceleration (9.82 m.s⁻²) is aligned with the tube axis in the vertical downward.

Fig. 1 Schematic view of the problem from a top view [18]

Continuity, momentum, and thermal energy equations are as follows [27]:

Continuity equation:

\[ \nabla \cdot \vec{v} = 0 \]  \hspace{1cm} (1)

Momentum equation:

\[ \frac{\partial \vec{v}}{\partial t} + \vec{v} \cdot \nabla \vec{v} = -\frac{1}{\rho} \nabla p + \mu \nabla^2 \vec{v} + \rho g \beta (T - T_{ref}) + \vec{s} \]  \hspace{1cm} (2)

Energy equation:

\[ \frac{\partial h}{\partial t} + \frac{\partial H}{\partial t} + \nabla \cdot (\vec{v} h) = \nabla \cdot \left( \frac{k}{c_p} \nabla h \right) \] \hspace{1cm} (3)

The total enthalpy of the material (sensible and latent):

\[ H = h + \Delta h \] \hspace{1cm} (4)
The latent heat term can be written as follows:

$$h = h_{ref} + \int_{T_{ref}}^{T} c_p dT$$  \hspace{1cm} (5)$$

The latent heat term can be written as follows:

$$\Delta H = \lambda L$$  \hspace{1cm} (6)$$

Where $\lambda$ varies from zero to 1. Zero is when the matter state is completely solid, and 1 is for the completely liquid state.

$$\lambda = \begin{cases} 
\frac{\Delta H}{L} = 0 & \text{if } T < T_s \\
\frac{\Delta H}{L} = 1 & \text{if } T > T_{liq} \\
\frac{\Delta H}{L} = \frac{(T - T_s)}{T_{liq} - T_s} & \text{if } T_s < T < T_{liq}
\end{cases}$$ \hspace{1cm} (7)$$

In Eq. (2), due to the natural convection in the system, Darcy’s law is used. Darcy’s law is introduced as follows:

$$s_i = \frac{c(1-\gamma)^2}{\gamma^3 + \varepsilon} u_i$$  \hspace{1cm} (8)$$

Where $c$ is the $A_{mushy}$ coefficient. Typically, it is a large constant coefficient, varying between $10^4$ and $10^7$. Here in this simulation, $10^5$ is considered for this parameter. The $\varepsilon$ is used in this equation to prohibit the denominator from becoming zero and gives the value of 0.001. The density is obtained using the Boussinesq approximation and then applied in the equations.

$$\rho = \rho_i [1 - \beta(T - T_i)]$$  \hspace{1cm} (9)$$

The above methodology is defined in Ansys-Fluent software. Therefore, this software is implemented to simulate the transient behavior of the latent thermal energy storage phenomenon. The quad mesh is used in this simulation. The boundary conditions, such as velocity inlet, isolated wall, and outflow, are defined. To solve the equations of melting and solidification, the pressure-based solver model, applied for solving nonlinear equations, is considered. SIMPLE algorithm was selected to solve the equations. To connect the momentum and energy equations, QUICK method is used, and to correct the pressure equations, PRESTO is implemented. The values for the under-relaxation factor for pressure, velocity, energy, and liquid fraction are 0.3, 0.6, 1, and 0, respectively. To check the independence of of time step on liquid fraction, the three-time steps of 1, 0.5, and 0.1 are examined, which means that in the simulation with time-step 0.5, the calculation is rational.

### 4. Validation
For validation, the simulated sample was compared with the experimental model, which is a triplex tube heat exchanger with eight fins, of which four are internal and the others external fins. The geometry of the present study is according to this experimental work [18]. The PCM used in that model was RT82, and HTF was water. The PCM melting temperature range was 77 - 85 °C, whereby 77° C is considered as solidus temperature and 85 °C as liquidus temperature. The thermal properties of this PCM are shown in Table 2. The HTF was water with 90 °C as its temperature and volume flow rate of 8.3 lit/min.

Melting process is simulated with coarse, medium, and fine meshes to examine mesh independency of 2D simulation analysis. The results of the liquid fractions during the melting process are expressed in terms of the number of meshes, at Table 3. From these data, it can be deduced that model with 37168 meshes can be addressed as the satisfactory model to obtain rational and good results.

Consequently, this model is used to validate the results obtained in this simulation. Here, the verification of simulation is performed comparing our results with numerical [17] and experimental [18] works. In Figures 2 and 3 the results obtained in our simulation are compared with the numerical and experimental investigation, respectively. It can be seen that present simulation results have good consistency with other works.

| Mesh Nodes | 10 min | min 30 | min 45 |
|------------|--------|--------|--------|
| 17538      | 0.2332 | 0.7861 | 0.9741 |
| 37168      | 0.2637 | 0.8144 | 0.9981 |
| 43536      | 0.2655 | 0.8192 | 0.999  |
Figure 2. Validation of the model with numerical results

Figure 3. Verification of the model with the experimental results
5. Results and Discussion

For this study, the triplex tube heat exchanger as a latent thermal energy storage tank with 4 external longitudinal fins and 4 internal longitudinal fins are considered in a 3D model. The axis of the proposed heat exchanger is in a direction to earth gravity with 100 mm vertical length. Heat transfer fluid (HTF) can flow in the core and outer annulus of the triplex tube, and the annulus between the HTF flow is filled with PCM.

In the first step, the mesh independence of the 3D model should be confirmed.

Table 4 shows the results of mesh independence, checking the 3D model at three different mesh numbers. As can be seen, the model with 625982 number of meshes can be selected as the base of the 3D numerical simulation model.

| Mesh nodes | min 10 | min 20 | min 30 |
|------------|--------|--------|--------|
| 399326     | 0.161  | 0.57   | 0.90   |
| 625982     | 0.1800 | 0.6180 | 0.930  |
| 770301     | 0.1779 | 0.6149 | 0.9279 |

The aim of this study is to determine the effect of HTF flow direction on the heat transfer performance in the triplex storage tank. Therefore, three flow configurations are considered here. In the first case (“Case A”), both of the entrances are from the top of the storage tank, in the direction of the gravitational force. In the second one (“Case B”) both of entrances are from the bottom of the tank. In the third case (“Case C”), one HTF enters from the top of the tank, and the other one enters from the bottom. Furthermore, to express the effect of the flow velocity, three different velocities (1, 3, and 5 mm/s with the temperature of 565 °C) are applied in the simulation. The initial temperature of the PCM at the solid phase is 443 °C.

“Case A”

In this case, both HTF enter from the top side of heat exchangers. To discuss the results obtained in this case, first the for three mass flow rates, the temperature counters during the melting process are described. Figure 4 shows the temperature contours in the course of the melting process. The melting process is initiated from the top side of the heat exchanger. Therefore, the natural convection inside the PCM enclosure occurs on the liquid side of PCM at the top of the heat exchanger without affecting the lower side significantly. Furthermore, the graphs show that the increasing of HTF flow rate results in increasing PCM temperature rates. This is because of lower temperature differences between inlet and outlet of the HTF and hence, rising mid-temperature of the heat exchanger. Consequently, it will cause an increase in the melting rate of PCM.
The trend of the mean temperature of the PCM and the liquid fraction of three different HTF flow rate are depicted in Figures 5 and 6. It is seen that when the HTF flow rate increases, the time of the melting process is significantly reduced.

Fig. 4 Temperature contours for “Case A” (a: after 3 minutes, b: after 5 minutes, c: after 15 minutes)
Figure 5. “Case A” in three different velocities
Figure 6. The liquid fraction in three velocities in “Case A”

In Figure 7, different modes of PCM melting are given at different speeds (after 15 minutes). As is shown by increasing inlet velocity and because of natural convection, the speed of circulation of molten PCM will increase, which causes the melting process to complete more quickly.
“Case B”

The temperature contours for different HTF flow rates are shown in Figure 8. In this case, both HTF inlets are from the bottom side of the heat exchanger. Therefore, the PCM melting process is initiated from the bottom. This phenomenon has a positive side effect on the total heat transfer processes, due to the motivation of natural convective heat transfer. The buoyancy force moves hot liquid PCM upwards to increase the melting process rate in the top side. The temperature variation and liquid fraction in the different HTF flow rate in the course of melting process are shown in Figures 9 and 10, respectively. Here, similar to “Case A”, the rate of the melting process is faster when the HTF flow rate is higher.
Figure 9. Temperature in different velocity for “Case B”
Figure 10. The liquid fraction at different velocities for “Case B”

In this case, the effect of velocity on the flow is shown in Fig. 11. It can be seen that at the same 15-minute period, the natural flow formed within the system is more than in the others, so the melting time is less.

Fig. 11 velocity contours in three different models, after 15 minutes for “Case B” (a: 1 mm/s, b: 3 mm/s, c: 5 mm/s)

“Case C”

In this case, the direction of the HTF flow is on the opposite side. In the core of the triplex tube, the HTF flows into the system from the bottom, and in the outer annulus, the HTF flow direction
is downward. As shown in Figure 12, the PCM melting initiates from the bottom and the top of the heat exchanger, owing to the HTF flow direction. As the PCM melting process continues, the natural heat transfer convection inside the PCM medium becomes stronger. So, after a couple of times, the liquid PCM formed at the bottom joins to the PCM liquid formed at the top. Similar to previous cases, the temperature variation and liquid fraction during the melting process are shown in Figures 13 and 14. The faster melting process occurs when the HTF flow rate is higher.

Figure 12. Temperature contours for “Case C” (a: after 3 minutes, b: after 5 minutes, c: after 15 minutes)
Figure 13. Temperature in different velocities for “Case C”
Figure 14. The liquid fraction in different velocities for “Case C”

Increasing inlet velocity of the HTF will cause more natural convection to be accrued at the PCM container, which will play an important role in circulating the melting PCM in the system. According to Figure 15, by increasing inlet velocity, the rate of melting will increase.
Comparing all cases

Table 5 shows the results that have been obtained for melting time in all the cases, with different boundary conditions. According to this table, in all flow rates, “Case B” has the lowest melting time. In this case, the effect of velocity on the flow is shown in Fig. 11. It can be seen that at a snapshot (15 minutes), the natural convection formed within “Case B” is more than the others due to the high melting rate. Although upward flow with entrance from the bottom can reduce the melting period, it will also diminish the amount of sensible heat storage. As can be seen in Figure 16, the PCM mean temperature at the end of the melting process in “Case A” is the highest, while the flow direction is downward. Hence, in “Case A” more thermal energy, including higher sensible heat, is kept in the storage system. Nonetheless, since the less free convection occurs in “Case A” (see Figure 15), it has a longer melting period.

### Table 5: Melting time for three cases at different velocities

| Velocity | 1 mm/s | 3 mm/s | 5 mm/s |
|----------|--------|--------|--------|
| “Case A” | 35 Min | 27 Min | 24 Min |
| “Case B” | 33 Min | 24 Min | 22 Min |
| “Case C” | 34 Min | 25 Min | 23 Min |
6. Conclusions

Concentrated solar power plants produce electrical power from solar thermal energy with a low capacity factor, if they are not equipped with the energy storage system. To increase the availability of CSP, a latent thermal energy storage system (LTES) can be an efficient solution. The LTES system can store huge amounts of thermal energy in the PCM during normal operation, and release it in the absence of sunlight. Generally, the PCM has a low thermal conductivity ratio, which compromises the performance of the system. Many solutions have been suggested, but for the conventional system, the best solution is to apply an extended surface. In this study, a triplex tube heat exchanger with 8 fins has been considered in the numerical analysis.
Due to the importance of the natural convection inside the PCM container during thermal charging of the system, different HTF flow patterns have been considered to investigate the role of natural convection in charging time of the heat storage tank. Results obtained from this study are as follows:

- When both HTF flows’ direction is upward from the bottom of the tank, the melting time decreases (Figure 16). It was observed that, by increasing the inlet velocity, the melting period is diminished. When both HTF enter from the top of the tank, the melting period increases considerably.
- For all HTF flow rates, the hierarchy of the elapsing melting time is “Case B”, “Case C” and then “Case A”.
- In “Case B”, since both flow directions are upward, the start of natural convection is sooner. This influences the upper solid PCM and leads to the melting process occurring within a shorter time.

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