HEAT TRANSFER ENHANCEMENT IN LATENT HEAT THERMAL ENERGY STORAGE SYSTEM USING FINS FOR SOLAR THERMAL POWER PLANT

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ABSTRACT
Thermal energy storage is essential for the solar thermal power plant for the continuous generation of electricity which may be interrupted due to the intermittent nature of solar radiation. In this context, phase change materials (PCMs) can be used as the storage materials because of their high energy density and the ability to store more energy compared to sensible material with a small temperature difference. However, most of the PCMs possess very low thermal conductivity (~0.2–0.5 W/m-k), which severely affects the thermal performance of the storage system. Therefore, it becomes important to improve the effective thermal conductivity of PCMs. In the present study, fin is used as a thermal conductivity enhancer (TCE) to augment heat transfer in PCM. The study numerically investigates the thermal performance of the storage system during melting and solidification with and without PCM. The enthalpy technique is adopted for modeling convection-diffusion phase change in the storage system.

INTRODUCTION
Concentrating solar power (CSP) plant uses solar radiation to transfer heat to the fluid, which can be used for electricity generation using different power cycles. Since the solar radiation fluctuates throughout the day, thermal energy storage (TES) can provide the heat during unavailability of solar radiation to produce electricity on continuous basis. TES temporarily hold thermal energy in form of substance for later utilization. Figure 1 shows the classification of TES materials [1]. Among the TES materials, latent heat storage system is superior due to its high storage density and the ability to exchange heat within a small temperature difference. Latent TES stores energy through phase change using phase change material (PCM). This storage system can be more effectively used during charging and discharging period. During charging, heat is supplied to the PCM which melts and store the thermal energy. During discharging, heat is extracted from the PCM and it solidifies.

Though PCMs have high energy density and nearly isothermal nature of storage process, PCMs possess very low thermal conductivity, which drastically affects the performance of the unit [2]. The effect of low thermal conductivity is reflected in slow heating and cooling processes during charging and discharging of the PCM. As a result, the rate of phase change is not up to the expected level and large scale utilization of TES is unsuccessful. Therefore, it becomes important to improve the thermal conductivity of the PCM.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Description</th>
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<tbody>
<tr>
<td>a</td>
<td>[-]</td>
<td>Coefficient in the discretised energy equation</td>
</tr>
<tr>
<td>A</td>
<td>[-]</td>
<td>Porosity function for the momentum equations</td>
</tr>
<tr>
<td>b</td>
<td>[-]</td>
<td>Computational constant</td>
</tr>
<tr>
<td>c</td>
<td>[J/kg.K]</td>
<td>Specific heat</td>
</tr>
<tr>
<td>D_{i}</td>
<td>[m]</td>
<td>Outer diameter of TES</td>
</tr>
<tr>
<td>f</td>
<td>[-]</td>
<td>Latent heat function</td>
</tr>
<tr>
<td>g</td>
<td>[m/s^2]</td>
<td>Acceleration due to gravity</td>
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<tr>
<td>k</td>
<td>[J/kg]</td>
<td>Sensible enthalpy</td>
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<tr>
<td>H</td>
<td>[J]</td>
<td>Total enthalpy</td>
</tr>
<tr>
<td>k</td>
<td>[W/m.K]</td>
<td>Thermal conductivity</td>
</tr>
<tr>
<td>L</td>
<td>[m]</td>
<td>Length</td>
</tr>
<tr>
<td>M</td>
<td>[-]</td>
<td>Morphological constant</td>
</tr>
<tr>
<td>P</td>
<td>[N/m^2]</td>
<td>Effective pressure</td>
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<td>S</td>
<td>[-]</td>
<td>Source term</td>
</tr>
<tr>
<td>S_{v}</td>
<td>[-]</td>
<td>Buoyancy source term for x momentum equation</td>
</tr>
<tr>
<td>S_{t}</td>
<td>[-]</td>
<td>Source term for energy equation in terms of temperature</td>
</tr>
<tr>
<td>t</td>
<td>[s]</td>
<td>Time</td>
</tr>
<tr>
<td>T</td>
<td>[°C]</td>
<td>Temperature</td>
</tr>
<tr>
<td>T_m</td>
<td>[°C]</td>
<td>Melting temperature of PCM</td>
</tr>
<tr>
<td>u_{i}</td>
<td>[m/s]</td>
<td>Velocity components in x, y and z directions</td>
</tr>
<tr>
<td>x, y, z</td>
<td>[-]</td>
<td>Cartesian axis direction</td>
</tr>
<tr>
<td>X</td>
<td>[-]</td>
<td>Dimensionless length</td>
</tr>
</tbody>
</table>

Special characters

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Unit</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>\alpha</td>
<td>[m^2/s]</td>
<td>Thermal diffusivity</td>
</tr>
<tr>
<td>\beta</td>
<td>[K']</td>
<td>Thermal expansion coefficient</td>
</tr>
<tr>
<td>\epsilon</td>
<td>[-]</td>
<td>Liquid fraction</td>
</tr>
<tr>
<td>\Delta H</td>
<td>[J]</td>
<td>Nodal latent heat</td>
</tr>
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</table>
Influence of laminar natural-convection flow on the melting process. The advantage of this technique is that the fixed-grid solution of the coupled momentum and energy equations can be obtained without variable transformations.

In this paper, a numerical solution approach is performed to investigate the thermal performance of TES system using PCM under fluctuating inlet condition of heat transfer fluid (HTF). A numerical model using enthalpy technique is used to characterize the thermal energy storage system. Based on the operating condition of the turbine in the medium temperature (~200 °C) solar thermal power plant, a eutectic mixture of lithium nitrate (58.1 by vol %) and potassium chloride (41.9 vol %) is selected as the PCM because of its desirable melting point (166 °C). The heat transfer fluid is chosen as Hytherm 600. The effect of thermal conductivity enhancer in the form of fin on heat transfer of PCM is evaluated.

DESCRIPTION OF PHYSICAL PROBLEM

The thermal storage system investigated in this study is a shell and tube heat exchanger where heat transfer fluid is inside the inner multiple pipes and PCM is filled in the annular space of the storage system. There are 7 tubes with 800 mm length through which HTF is flowing as shown in figure 2. A eutectic mixture of LiNO₃ (58.1 by vol %) and KCl (41.9 by vol %) is used as PCM, HTF is chosen as Hytherm 600. As the PCM is corrosive in nature, the container is made of SS 304 coated with Teflon. The thermophysical properties of HTF, PCM and container material are listed in table 1. Fins are incorporated in the inner HTF pipes to enhance the heat transfer rate from HTF to PCM as shown in figure 3. The thermal energy storage system is divided into six symmetrical parts as shown in figure 2. Since these parts are symmetric, the behaviour of the thermal energy storage can be represented by one of the symmetric parts. In the present analysis, three-dimensional analysis is carried out as two-dimensional study is insufficient for the numerical domain.

Table 1 Thermophysical properties of Materials

<table>
<thead>
<tr>
<th>Material</th>
<th>( \rho ) (kg/m³)</th>
<th>( c_p ) (J/kg.K)</th>
<th>( k ) (W/m.K)</th>
<th>( \mu ) (Pa.s)</th>
<th>( T_m ) (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>HTF</td>
<td>720.9</td>
<td>3097.4</td>
<td>0.116</td>
<td>0.0195</td>
<td>-</td>
</tr>
<tr>
<td>PCM</td>
<td>2010</td>
<td>1485</td>
<td>0.5</td>
<td>0.003</td>
<td>166</td>
</tr>
<tr>
<td>Steel 304</td>
<td>8030</td>
<td>502.48</td>
<td>16</td>
<td>-</td>
<td>-</td>
</tr>
</tbody>
</table>

The thermal expansion coefficient \( \beta \) and the latent heat \( (L_{latent}) \) of PCM are taken as 0.00066 K⁻¹ and 272 kJ/kg, respectively.

NUMERICAL MODELLING

A numerical study was carried out using the enthalpy porosity approach for modelling combined convection-diffusion phase change given by Brent et al. [7]. The governing equations for TES filled with PCM are written using a single domain approach. The energy equation can be written in terms of sensible enthalpy \( h \).

\[
\frac{\partial (\rho h)}{\partial t} + \nabla \cdot (\rho u h) = \nabla \cdot (k \nabla T) + S_e
\]

(1)

The conservation of momentum and mass equations under the assumption of Newtonian fluid can be written as,
Substituting equation sensibly, the total enthalpy can be written as the sum of heat and latent heat content.

where \( \Delta H = f(t) \), the latent heat content of the cell is a function of temperature of the cell.

\[
\Delta H = \lambda \text{ for } T > T_m \\
\Delta H = 0 \text{ for } T < T_m
\]

The details of the numerical formulation can be found elsewhere [8]. The governing equations are solved iteratively using finite volume method (FVM) according to the SIMPLE algorithm. Coefficients in the momentum and energy equations are determined by the power law. A commercial software Fluent 14 is used to solve the governing equations.

**Boundary and Initial Conditions**

The boundary conditions adopted for solutions of conservation equations are:

(a) No slip and impermeability condition at the walls, i.e. \( u_i = 0 \)

(b) At inlet, \( z = 0 \), \( u_z = u_{in} \) and \( T_z = T_{in} \)

(c) At outlet, \( z = L \), pressure outlet condition with \( p = 0 \)

(d) Insulated sidewalls and outer surface of the TES at \( r = \frac{D_1}{2} \).

(e) Symmetric boundary conditions are taken for the symmetry planes at \( \theta = 0 \) and \( \theta = 60^\circ \).

The inlet temperature \( (T_{in}) \) of HTF is maintained at 200°C for 1800 s during charging period and at 132°C for the same duration during discharging period. The appropriate initial condition to the physical situation is,

(f) At \( t = 0 \), \( T(r,\theta,z,0) = 165.9 °C \) which is slightly below \( T_w \).

**Validation of Numerical Model**

The validation of the present numerical model is performed by comparing the results with Brent et al. [7]. In the present analysis, the simulation is carried out in Fluent 14 to study the two dimensional melting of pure gallium in a rectangular cavity. A rectangular cavity of \( 8.89 \times 6.35 \text{ cm} \) is selected. The heated wall temperature \( (T_{hot}) \) and cold wall temperature \( (T_{cold}) \) are 38 and 28.3 °C, respectively and the initial temperature of the cavity is \( T_{int} = 28.3 °C \). For the numerical solution, after refining the grid size, a uniform grid of \( 84 \times 64 \) and a constant time step of 0.1 s is used. The contour plot of streamlines and isotherms are compared at 3 and 6 min as shown in figure 4.

In the momentum equation (equation 2), the viscosity is set to very high value (~10^8) in fin and pipe regions.

The source terms in the momentum equation (equation 2) are written as,

\[
S_i = A u_i
\]

where the flow resistance \( (A) \) is defined where porous media can be imitated as,

\[
A = \frac{M (1 - \varepsilon)^2}{\varepsilon^2 + b}
\]

where the value of \( M \) is sufficiently large (~10^8) and \( b \) is used to avoid division by zero.

In the enthalpy-porosity approach, the latent heat evolution is accounted for on defining the source term in the energy equation. The total enthalpy can be written as the sum of sensible heat and latent heat content.

\[
H = h + \Delta H
\]

Substituting equation 8 in equation 1, one can obtain,

\[
S_e = \frac{\partial (\rho H)}{\partial t} + \frac{\partial (\rho u_i H)}{\partial x_i}
\]
As observed from the figure at 3 min, the natural convection is just started and a convection cell sets up. At 6 min, the natural convection is intensified and it can be seen that upper section of the melt front is moving faster than the lower section because of the impingement of the warm fluid. It can be noted that the streamlines and the isotherms predicted by the present model agree well with the results reported in the literature.

**Grid Independence Study**

The grid independence study is performed to reduce the spatial discretisation errors. The study involves performing simulation on successively coarser grids keeping the time step (0.1 s) constant. The successive grid coarsening is performed by increasing the number of elements by a factor of 1.5 times. In the present study, three cases are considered, viz: (i) Grid 1 is the coarse mesh (ii) Grid 2 is the fine mesh (iii) Grid 3 is the finer mesh. The total number of elements in the domain for Grid 1 is 455712. Temperature of the heat transfer fluid at the outlet of the pipe is monitored with time for the three cases as shown in Figure 5. The TES is kept at an initial temperature of 165.9 °C.

When the heat transfer fluid at 200 °C flows through the pipe, the temperature increases steeply as observed from the figure and then stabilizes when the phase change of the PCM starts. After the charging process of 1800 s, the HTF outlet temperature of Grid 1, Grid 2, Grid 3 are found to be 181.96, 180.6 and 179.87 °C, respectively. It can be noted that the temperature difference between finer and fine grids (0.73 °C corresponds to change in 0.405%) is lowered compared to that between fine and coarse grids (1.36 °C corresponds to change in 0.753%). Therefore, the result enables the use of fine grid (Grid 2) for further analyses.

**Effect of Gravity**

The effect of gravity during melting of the PCM for 1800 s is studied. Two cases are considered where the direction of gravity is along the HTF flow and in another case, the direction of HTF flow is opposite to the direction of gravity. Figure 6 shows the average temperature plots at the HTF outlet plane. It can be observed from the figure that the HTF outlet temperature is lower by 2.2 °C at 1800 s for the direction of gravity along the HTF flow compared to that opposite to the HTF flow. This can be attributed to the stronger melt convection in molten PCM for the direction of gravity along the HTF flow. Figure 7 shows the velocity vector plots for two cases at 1800 s. The maximum molten PCM velocity in case of HTF flow along the gravity is found to be 0.389 m/s and that in case of HTF flow against the gravity is 0.061 m/s. Therefore, the direction of HTF flow is chosen along the direction of gravity.

![Figure 5 Grid independence study](image)

![Figure 6 Effect of gravity on HTF outlet temperature](image)
RESULTS AND DISCUSSIONS

The effect of mode of heat transfer on the thermal performance of latent heat thermal storage is investigated by considering with and without melt convection in PCM. Equations 1-3 are solved for simulating melt convection whereas only the energy equation (equation 3) with $u_i = 0$ is considered for conduction heat transfer. The temperature variation of heat transfer fluid at the outlet of TES filled with PCM is plotted with time as shown in figure 8. It can be observed from the figure that temperature of HTF 1 is higher than that of HTF 2. This is due to the arrangement of the pipes, in which HTF 1 is surrounded by other pipes while HTF 2 is near the circumference of the storage system. During melting, it can be observed from the figure that temperatures are lower for convection case than the case in which conduction is considered. The HTF 1 and HTF 2 temperatures at outlet after 1800 s for convection as mode of heat transfer are 184.71 °C and 183.79 °C, respectively, whereas 189 °C and 188.3 °C, respectively for conduction as mode of heat transfer. The maximum temperature difference at the end of charging and discharging process at the outlet of HTF is minimum for longitudinal fin (35.06 °C) compared to annular fin (39.2 °C) and PCM only (42.85 °C). Hence, the longitudinal fins perform better than the annular fins. In case of annular fins, PCM between two adjacent fins are isolated along the temperature variation of HTF i.e. along the direction of flow, which causes localized heating of PCM depending on the temperature difference. The longitudinal fins are placed along the variation of HTF temperature in pipe, which leads to more heat transfer to PCM. This is evident from figure 10.

Further the effect of fin width on the performance of TES for longitudinal fin configuration is investigated. In this study, the number of fins and fin height are kept constant at 6 and 8.7 mm, respectively and the width of the fin is changed. Three sizes of fin width are chosen viz. 1.2, 2.4 and 3.6 mm. The width cannot be increased further from 3.6 mm due to
manufacturing constraints. Figure 11 shows the temporal variation of HTF temperature at the outlet with change in fin width. The temperature difference (ΔT) after melting and solidification processes for width 1.2, 2.4 and 3.6 mm is 35.06, 33.97 and 32.49 °C, respectively. It can be noted that the temperature difference decreases as the width of the fin increases. This can be attributed to the increase in heat transfer area due to increase in fin width.

CONCLUSION
A numerical study is performed to investigate the thermal performance of TES system using PCM. A numerical model using enthalpy technique is used to characterize thermal energy storage system. The effect of fin as thermal conductivity enhancer on heat transfer of PCM is evaluated. Numerical results indicate that heat transfer is dominated by convection during melting and conduction during solidification. The temperature distribution is improved using thermal conductivity enhancer which augments the heat transfer rate in PCM. For the same volume percentage of TCE longitudinal fins perform better than annular fins. The heat transfer increases with increase in fin width. Therefore, fin width should be optimized for a given application.

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REFERENCES