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

M Prabhakar[a] , S Singh[a] , S K Saha[b,*]
[a]Department of Energy Science and Engineering, Indian Institute of Technology Bombay, Powai, Mumbai- 400076, INDIA.
[b]Department of Mechanical Engineering, Indian Institute of Technology Bombay, Powai, Mumbai- 400076, INDIA.
* Corresponding author. Tel.: +91 22 25767392; fax: +91 22 2572 6875.
E-mail address: sandip.saha@iitb.ac.in

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

a	[-]	Coefficient in the discretised energy equation
A	[-]	Porosity function for the momentum equations
b	[-]	Computational constant
c	[J/kg.K]	Specific heat
D_i	[m]	Outer diameter of TES
f	[-]	Latent heat function
g	[m/s^2]	Acceleration due to gravity
h	[J/kg]	Sensible enthalpy
H	[J]	Total enthalpy
k	[W/m.K]	Thermal conductivity
L	[m]	Length
M	[-]	Morphological constant
P	[N/m^2]	Effective pressure
S	[-]	Source term
S_b	[-]	Buoyancy source term for v momentum equation
S_e	[-]	Source term for energy equation in terms of temperature
t	[s]	Time
T	[ºC]	Temperature
T_m	[ºC]	Melting temperature of PCM
u_i	[m/s]	Velocity components in x, y and z directions
x, y, z	[-]	Cartesian axis direction
X	[-]	Dimensionless length

Special characters

α	[m^2/s]	Thermal diffusivity
β	[K^{-1}]	Thermal expansion coefficient
ε	[-]	Liquid fraction
ΔH	[J]	Nodal latent heat

λ	[J/kg]	Latent heat of fusion
μ	[Pa.s]	Dynamic viscosity
ρ	[kg/m^3]	Density

Subscripts

cold	Cold
hot	Hot
int	Initial
m	Melting point
r	Reference condition
n	Iteration counter
p	p^{th} node point or control volume
i	i^{th} node point

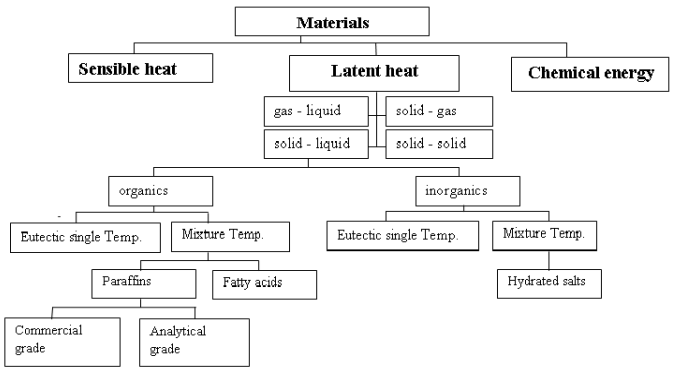


Figure 1 Classification of TES materials [1]

A detailed review of various phase change materials, which can be used to store thermal and solar energy, was performed by Kenisarin [3]. The thermophysical properties like melting temperature, thermal conductivity and heat of fusion of the salts and their compositions were reported. Experimental studies were performed by Velraj *et al.* [4] on heat transfer enhancement during solidification of paraffin. The study investigated three different heat transfer enhancement techniques including longitudinal fins, metallic rings, and bubble agitation. The study concluded that fin configuration was most effective. Seeniraj *et al.* [5] investigated the performance of the latent heat thermal storage systems employed in solar power plant or such similar energy storage applications during active phase charging mode. The tubes carrying heat transfer fluid (HTF) are provided externally with thin circumferential fins. The study showed that with increase in number of fins, the heat transfer rate increases.

Sparrow *et al.* [6] reported that natural convection retards the solidification process and even can terminate the process. The retardation of the solidification process is a major deterrent to the heat transfer performance of the thermal energy storage. This decreases the rate at which the heat is extracted from the phase change material. Therefore, there is a need to enhance the heat transfer during solidification. Although the previous studies investigated the heat transfer enhancement during melting and solidification, none of the studies focus on the application prospective. There are various heat transfer enhancement techniques reported but use of finned tube is effective and reliable. Brent *et al.* [7] numerically investigated the melting of pure metal using the enthalpy-porosity approach for modelling combined convection- diffusion phase change. A two-dimensional transient model was used considering the 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 $LiNO_3$ (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 Theromophysical properties of Materials

Material	ρ (kg/m^3)	c_p (J/kg.K)	k (W/m.K)	μ (Pa.s)	T_m (°C)
HTF	720.9	3097.4	0.116	0.0195	-
PCM	2010	1485	0.5	0.003	166
Steel 304	8030	502.48	16	-	-

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

NUMERICAL MODELLING

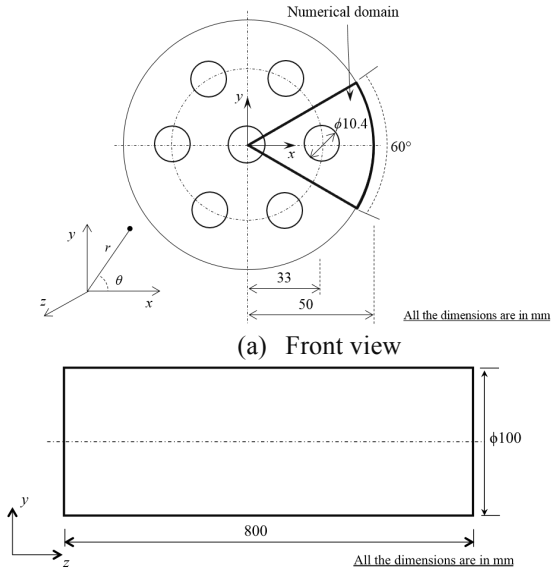
A numerical study was carried out using the enthalpy porosity approach for modelling combined convection-diffusion phase change given by Brent *et al.* [7]. The governing equations for TES filled with PCM are written using a single domain approach. The energy equation can be written in terms of sensible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

$$\frac{\partial(\rho h)}{\partial t} + \frac{\partial(\rho u_i h)}{\partial x_i} = \nabla.(k\nabla T) + S_e \quad (1)$$

The conservation of momentum and mass equations under the assumption of Newtonian fluid can be written as,

$$\frac{\partial(\rho u_i)}{\partial t} + \frac{\partial(\rho u_i u_j)}{\partial x_i} = \nabla.(\mu \nabla u_i) - \frac{\partial P}{\partial x_i} + S_i + \rho_r g \beta (T - T_r) \quad (2)$$

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\rho u_i)}{\partial x_i} = 0 \quad (3)$$



(a) Front view

(b) Side view

Figure 2 Schematic diagram of thermal energy storage

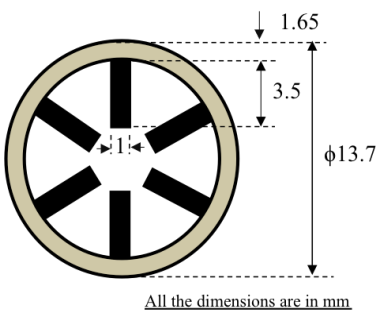


Figure 3 Fins in the inner HTF pipe

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,

$$S_i = A u_i \quad (4)$$

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

$$A = -\frac{M(1-\varepsilon)^2}{\varepsilon^3 + b} \quad (5)$$

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.

$$H = h + \Delta H \quad (8)$$

Substituting equation 8 in equation 1, one can obtain,

$$S_e = \frac{\partial(\rho \Delta H)}{\partial t} + \frac{\partial(\rho u_i \Delta H)}{\partial x_i} \quad (11)$$

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

$$\Delta H = \lambda \; for \; T > T_m \quad (12)$$

$$\Delta H = 0 \; for \; T < T_m \quad (13)$$

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 $r = \frac{D_i}{2}$.

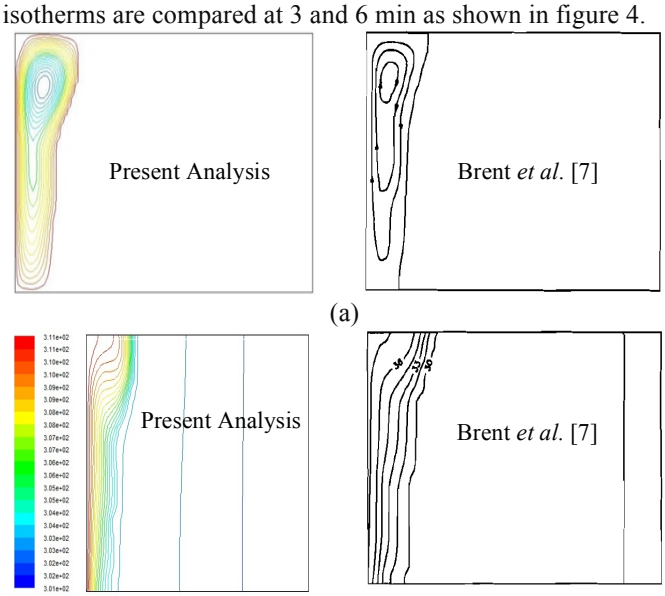
(e) Symmetric boundary conditions are taken for the symmetry planes at $\theta = 0$ and $\theta = 60°$.

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_m.

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 × 6.35 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 84×64 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.



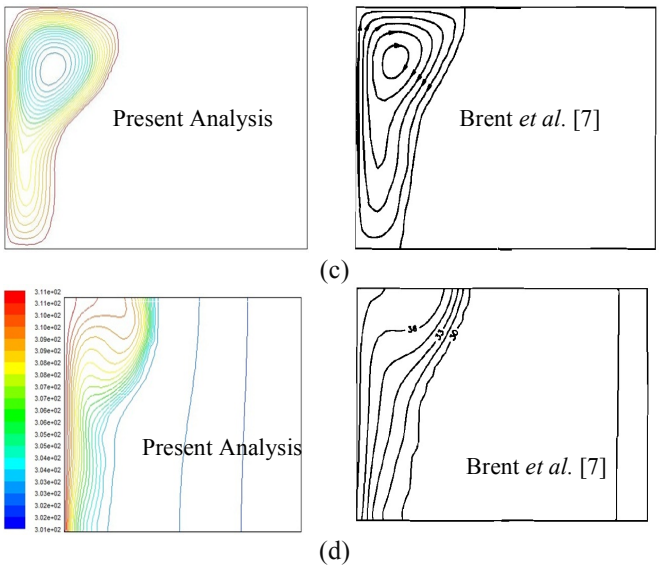


Figure 4 At 3 min (a) streamlines (b) isotherms; at 6 min (c) streamlines (d) isotherms

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 agrees 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.

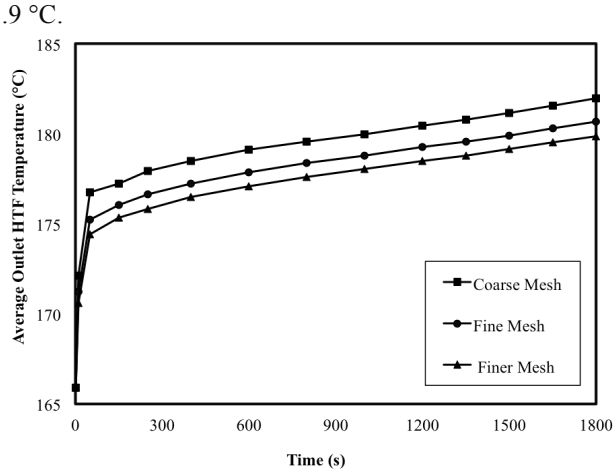


Figure 5 Grid independence study

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.

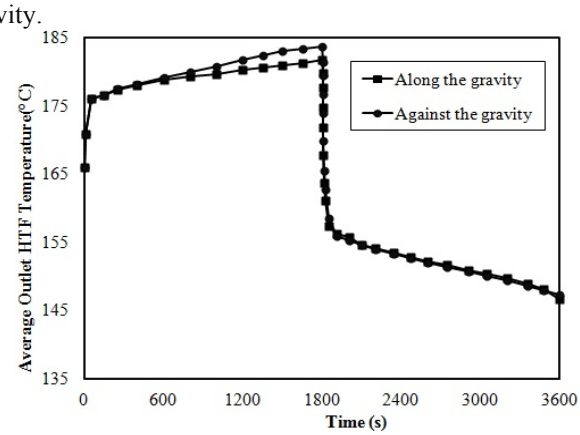
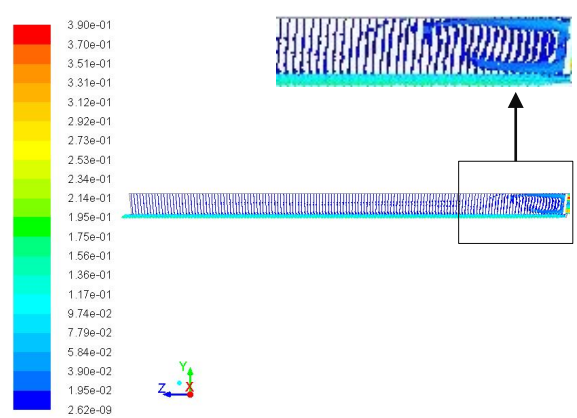
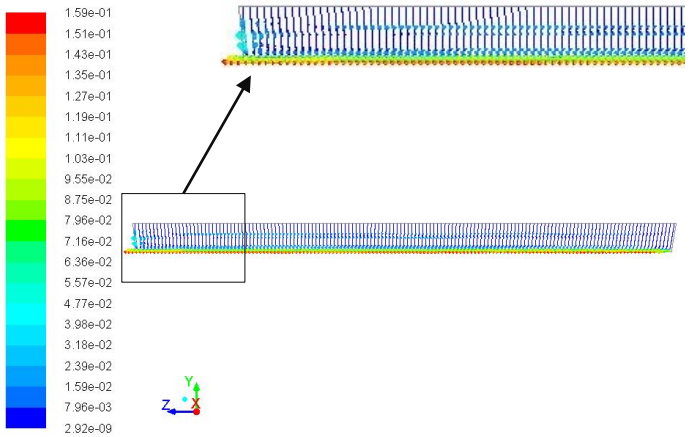


Figure 6 Effect of gravity on HTF outlet temperature



(a)



(b)

Figure 7 Velocity vector plots for (a) HTF flow along the gravity (b) HTF flow against the gravity (flow direction is along positive *z* direction) at 1800 s with enlarge view

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 (ΔT) is 42.85 °C for the convection case which is less compared to the conduction case (44.53 °C). Lower temperatures during melting signify that the heat transfer from HTF to the phase change material is more which can be attributed to the convection cells. Lower temperatures during solidification signify that the heat transfer from the phase change material to the HTF is less which indicates that convection is suppressed during solidification and the mode of heat transfer is conduction. Based on these findings, it can be concluded that during melting convection is dominant and during solidification conduction is dominant. It is also evident from figure 8 that the PCM reduces the fluctuation effectively. The ΔT applied during charging and discharging period is 68 °C and using PCM the same is 42.85 °C, which indicates reduction in the effect of applied fluctuation.

Thermal conductivity enhancer (TCE) in the form of fin is incorporated in the TES to study its effect on the performance of the TES. In the present study, the effect of two types of fin *viz.* (i) longitudinal fin and (ii) annular fin are considered. The longitudinal and annular fins used in the storage system are drawn for a given percentage of TCE volume of 7%, 6 numbers of fins and a fin height of 8.7 mm. The width of longitudinal fin is 1.2 mm and that of annular fin is 14 mm. The temperature of the HTF at the outlet for TES using longitudinal fin, annular fin and PCM alone is plotted in figure 9. The temperature difference (ΔT) 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.

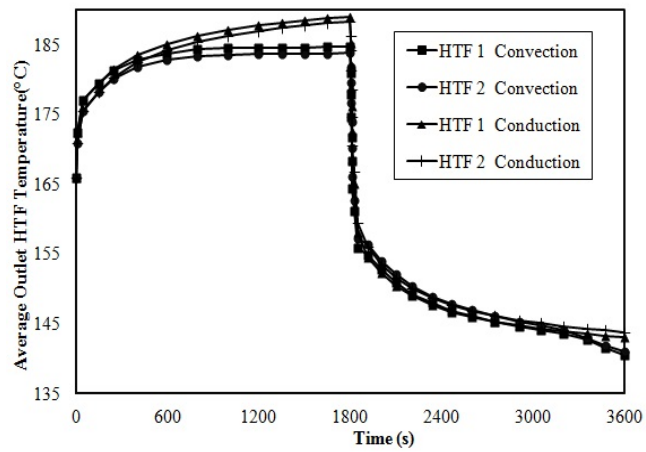


Figure 8 HTF outlet temperature *vs.* time for convection and conduction

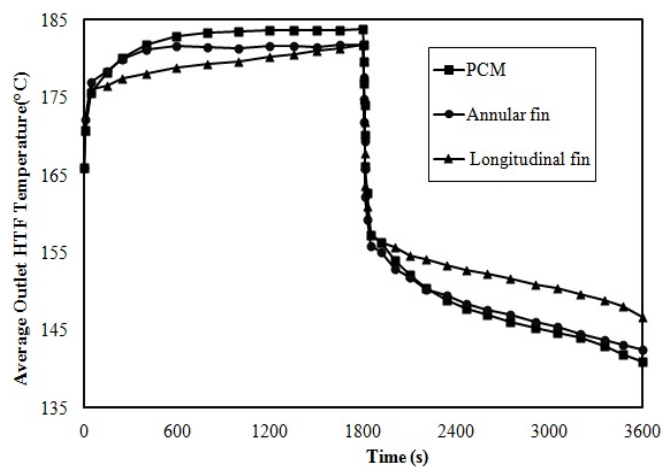
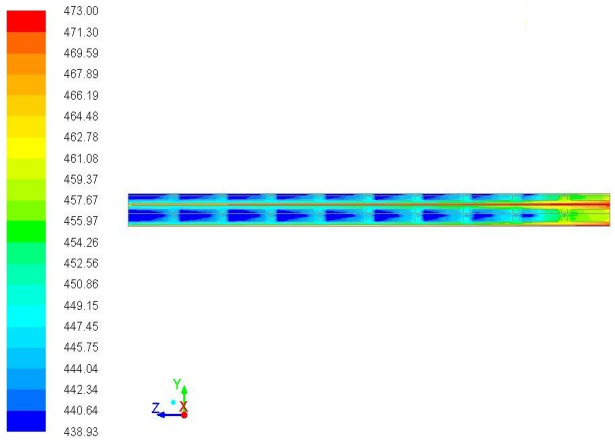


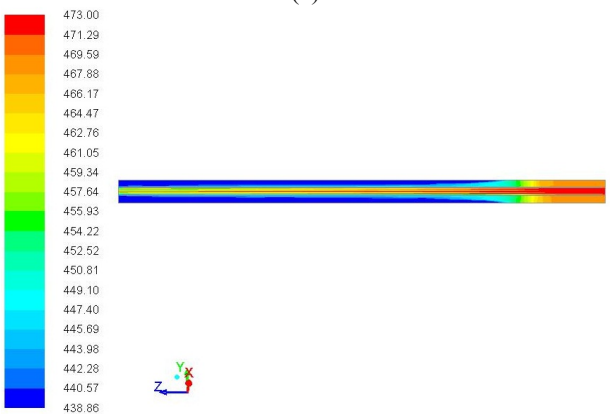
Figure 9 Effect of TCE on HTF outlet temperature

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.



(a)



(b)

Figure 10 Temperature distribution at 1800 s for (a) annular fin (b) longitudinal fin

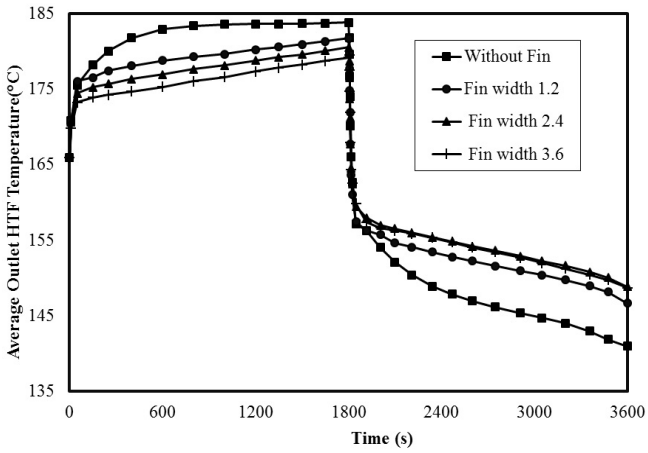


Figure 11 HTF outlet temperature plot for different fin width

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.

ACKNOWLEDGEMENT

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