Numerical Analysis of Latent Heat Thermal Energy Storage using Encapsulated Phase Change Material for Solar Thermal Power Plant

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Abstract

Thermal energy storage improves the load stability and efficiency of solar thermal power plants by reducing fluctuations and intermittency inherent to solar radiation. This paper presents a numerical study on the transient response of packed bed latent heat thermal energy storage system in removing fluctuations in the heat transfer fluid (HTF) temperature during the charging and discharging period. The packed bed consisting of spherical shaped encapsulated phase change materials (PCMs) is integrated in an organic Rankine cycle-based solar thermal power plant for electricity generation. A comprehensive numerical model is developed using flow equations for HTF and two-temperature non-equilibrium energy equation for heat transfer, coupled with enthalpy method to account for phase change in PCM. Systematic parametric studies are performed to understand the effect of mass flow rate, inlet charging system, storage system dimension and encapsulation of the shell diameter on the dynamic behaviour of the storage system. The overall effectiveness and transient temperature difference in HTF temperature in a cycle are computed for different geometrical and operational parameters to evaluate the system performance. It is found that the ability of the latent heat thermal energy storage system to store and release energy is significantly improved by increasing mass flow rate and inlet charging temperature. The transient variation in the HTF temperature can be effectively reduced by decreasing porosity.

Keywords: Phase change material, encapsulation, Organic Rankine Cycle (ORC)-based solar thermal power plant, temperature fluctuations

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26 Nomenclature

A	Cross sectional area of storage, m^2
$c_{p,l}$	Specific heat of storage material in liquid phase, $J/kg.K$
$c_{p,s}$	Specific heat of storage material in solid phase, $J/kg.K$
C_2	Inertial resistance coefficient, m^{-1}
D	Diameter, m
d_p	Encapsulation shell diameter, cm
E_l	Total energy of fluid, kJ
E_s	Energy stored, kJ
E_r	Energy released, kJ
f_l	Liquid fraction of PCM
g	Acceleration due to gravity, m/s^2
ΔH	Enthalpy content in PCM during phase change, kJ/kg
h_f	Volumetric Interstitial heat transfer coefficient, $W/m^3.K$
k_l	Thermal conductivity of fluid, $W/m.K$
k_s	Thermal conductivity of solid, $W/m.K$
L	Length, cm
L_{latent}	Latent heat of fusion, kJ/kg
$\dot{m}$	Mass flow rate of heat transfer fluid, kg/s
Re_D	Reynolds number based on storage diameter $\left(= \frac{\dot{m}D}{\mu A}\right)$
Re_p	Reynolds number based on particle diameter $\left(= \frac{\rho_l U d_p}{\varepsilon \mu}\right)$
$Re_{p,cr}$	Critical Reynolds number based on particle diameter
P	Pressure, Pa
Pr	Prandtl number $\left(= \frac{\mu c_{p,l}}{k_l}\right)$
T_s	Temperature of the solid medium, $^{\circ}C$
T_l	Temperature of the fluid, $^{\circ}C$
T_i	Initial temperature, $^{\circ}C$
T_m	Melting temperature, $^{\circ}C$
T_{in}	Inlet temperature of heat transfer fluid, $^{\circ}C$
T_{out}	Outlet temperature of heat transfer fluid, $^{\circ}C$

ΔT Difference between maximum and minimum temperature of heat transfer fluid at outlet, °C

t Time, s

t_c Charging time, s

t_d Discharging time, s

Δt Time interval, s

U Superficial velocity of heat transfer fluid $\left(= \frac{\dot{m}}{\rho_l A}\right)$, m/s

U_0 Interstitial velocity $\left(= \frac{U}{\varepsilon}\right)$, m/s

u Velocity vector, m/s

27 Greek symbols

ϕ Porosity

μ Dynamic viscosity, kg/m.s

α Permeability, m²

λ Relaxation factor

ρ Density, kg/m³

ε Effectiveness

28 Subscript

c Charging

d Discharging

l Liquid

max Maximum

min Minimum

n n^{th} time step

o Overall

s Solid

29 Abbreviation

CSP Concentrating Solar Power

HTF Heat Transfer Fluid

LHTES Latent Heat Thermal Energy Storage

ORC Organic Rankine Cycle

PCM Phase change material

SHS Sensible Heat Storage

<i>TES</i>	Thermal Energy Storage
<i>UDF</i>	User Defined Function
<i>UDS</i>	User Defined Scalar

1. Introduction

Solar energy is one of the most abundant forms of renewable energy sources in many parts of the world. However, meeting energy demand from this source is challenging due to its intermittent and unpredictable nature over short time scales during the day time. The energy production is driven by the efficiency of systems as well as by the economics of power generation. Thermal energy storage technologies have the potential to make solar radiation a reliable option for electricity generation by Organic Rankine Cycle (ORC)-based solar thermal power plants either integrated with the national power grid or installed in the non-interconnected electric network, compensating for the variability in the solar radiation and thus, making the solar power production to be a cost effective and very efficient method. Ibrahim *et al.*, [1] therefore studied various storage solutions that will enable stable electricity generation and maintain the load level. Gil *et al.*, [2] noted that the thermal energy storage (TES) can improve the effectiveness of the thermal energy systems with the advantage of large-scale switching. TES has the potential to eliminate the mismatch between the energy supply and demand of energy. It absorbs thermal energy in the form of either sensible heat or latent heat. The sensible storage system has low heat capacity per unit volume and its energy content depends on the temperature change of the substance, which results in high capital cost [3]. In this context, latent heat thermal energy storage system (LHTES) using phase change material (PCM) could be an attractive solution, as PCM has high energy density per unit volume and it absorbs and releases heat almost isothermally within a small temperature difference. However, the major drawback in using PCM is its low thermal conductivity which causes high thermal resistance to heat transfer during the charging and discharging period. Several methods are reported in the literature that to enhance heat transfer in PCM, such as copper foam with fin [4], circumferentially positioned fin [5], lauric acid based nanocomposites using chemically functionalized graphene nanoplatelets [6], carbon nanostructures, carbon nanotubes,

nanoparticles of metallic (Ag, Al, C/Cu and Cu) and metal oxide (Al_2O_3 , CuO, MgO and TiO_2) and silver nanowires [7], modified silicon nitride powders in microencapsulated PCM made of *n*-octadecane core and polymethylmethacrylate shell [8], microencapsulated PCM and polyethylene composite material [9], spongy graphene (G22) and docosane as a composite [10], highly conductive calcium carbonate shell for microencapsulating paraffin based binary cores via self-assembly method [11], dispersion of metallic particles in PCM [12-13], multi-tube [14], rings and bubble agitation in PCM [15], encapsulated PCM [16-18] and addition of fins [19-21]. Among these enhancement techniques, PCM encapsulated in small capsules is a promising approach as it offers larger surface area relative to volume for heat transfer between the heat transfer fluid (HTF) and PCM, thereby reducing the charging and discharging time. Encapsulated PCMs filled in a storage system form packed bed latent energy storage system.

Packed bed sensible and latent energy storage systems are among the important class of thermal storage systems which can be used to maintain the balance of supply and demand of energy. Several researchers have studied the thermal behaviour and economical aspects of packed bed sensible storage system numerically [22-29]. Ismail and Henriquez [30] developed a transient one-dimensional numerical model for spherical capsules filled with water as the PCM in a cylindrical tank. The solidification process inside the capsule was modelled using conductive phase change and convective boundary condition on the surface of capsule. The convection in molten PCM is considered using an effective heat conduction coefficient. The energy equation are solved using finite difference method and moving the grid inside the PCM capsules to investigate charging and discharging times of the storage system. Arnold [31] identified variable rates of heat transfer for freezing and melting of encapsulated ice by considering macro level fluid flow and micro level freezing and melting of ice in the capsule and linking them by the heat transfer across the wall of the capsule. Regin *et al.* [16] numerically studied the thermal behaviour of a packed bed thermal energy storage using

spherical capsules filled with paraffin as a PCM for a solar-water heating application. The melting time was found to be reduced by 31.6% for PCM that melts within a range of temperature as compared to a PCM with a single melting point. Wu and Fang [32] investigated the dynamic behaviour of a packed bed composed of spherical capsules filled with Myristic acid as a PCM for solar heat storage system. The influence of HTF inlet temperature, packed bed initial temperature and mass flow rate of HTF during the discharging process were studied using a mathematical model based on the energy balance of HTF and PCM. Benmansour *et al.* [33] analysed the transient behaviour of cylindrical packed bed TES system, filled with spherical PCMs (paraffin wax) using a two-dimensional separate phases methodology. Flueckiger and Garimella [34] developed a new finite volume model to simulate mass and energy transport in a latent heat thermocline storage system for 100 MWe molten salt power tower plant. Various parameters, such as PCM capsule diameter, fluid inlet velocity and storage tank height were found to influence the temperature distribution and efficiency of a packed bed LHTES system for the charge process. Peng *et al.* [35] presented a numerical model for latent thermal energy storage system using concentric dispersion equations. The charge efficiency under different working conditions is increased with reduction in size of PCM capsule and HTF inlet velocity as well as storage tank height. Nithyanandam and Pitchumani [3] numerically analysed the steady and transient response of the thermocline storage system packed with encapsulated PCMs for a concentrating solar power (CSP) plant. A transient finite volume model that accounts for axial variation in HTF temperature and radial variation in PCM temperature was developed where melting and solidification of PCM was modelled using enthalpy-porosity technique [36]. The effects of encapsulated radii, Reynolds number, Stephan number, capacitance ratio, PCM melt temperature on system utilization of energy were investigated to optimize parameter for maximizing total useful energy discharged.

Analytical models were developed by Veerappan *et al.* [37] to predict the phase change behaviour of 65 mol% capric acid and 35 mol% lauric acid, calcium chloride hexahydrate, octadecane, hexadecane, and eicosane in spherical enclosure. The effect of the size of encapsulation, initial temperature of the PCM, the external fluid temperature on solidified and molten mass fraction, and the total phase change time on solidification and melting characteristics of PCMs were evaluated. Izquierdo-Barrientos *et al.* [38] conducted an experimental study to investigate the storage behaviour of micro encapsulated packed bed and fluidized bed of PCM of Rubitherm GR50 and air as HTF. These studies concluded that a PCM-based packed and fluidized system shows greater efficiency as compared to sand-based systems, moreover cyclic charging and discharging of PCM system show attrition of PCM during fluidization, but no loss of PCM is obtained in 75 h of continuous operation with 15 charging–discharging cycles. Nallusamy *et al.* [39] studied thermal energy storage system with constant and varying heat sources utilizing sensible and latent thermal energy in the system. Paraffin was encapsulated in spherical shell of 55 mm, which formed the packed bed and water was used as HTF as well as sensible heat storage (SHS) material. Experimental studies were performed to understand the effect of constant mass flow rates, and constant inlet temperature on charging of the tank. Similar studies were also performed with varying inlet temperature and varying mass flow rate. It was noted that the mass flow rate insignificantly affects the rate of charging for constant inlet HTF temperature.

Recently, few investigations [40–43] highlighted the charging and discharging process dynamics of latent heat thermal storage system using spherical encapsulated PCMs. Amin *et al.* [40] investigated the charging and discharging effectiveness of the thermal storage system with PCM encapsulated in spheres using semi-analytical mathematical model based on the ε -NTU. Bellan *et al.* [42] experimentally and numerically studied the dynamic thermal performance of latent thermal energy storage system filled with spherical capsules for high

temperature CSP plants. The Stefan number and the shell thickness were found to influence the energy stored and PCM temperature during charging and discharging processes. Rady [43] performed experiments on a packed bed TES system with two granular phase change composites with different melting points for low temperature applications. The optimum mixing ratio for maximum exergy efficiency was determined and found to be independent of Reynolds number and the charging and the discharging directions used in practical applications.

It is found from the forgoing discussions that several studies have reported on packed bed sensible energy storage systems. Most of the studies provided design guidelines for the sensible energy storage system for a particular application. Although the effectiveness of packed bed LHTES system during charging and discharging processes was assessed, few parameters influencing the performance of the TES system were considered in the published literatures. Therefore, an extensive study is necessary to evaluate the effects of various geometrical and operational parameters on the effectiveness of packed bed LHTES system, especially for the removal of fluctuations in HTF temperature in ORC based solar thermal power plant for continuous generation of electricity with a minimal variation. Hence, the aim of this work is to evaluate the transient response of LHTES system filled with spherical shaped encapsulated PCMs under the influence of various design and operational parameters. A cycle comprising of charging and discharging periods of 1800 s each is considered to represent fluctuations in the HTF inlet temperature. The HTF from the solar receivers enters from the top of the packed bed latent heat thermal storage with an inlet temperature of 200 and 142 °C, respectively during the charging and discharging period and exchanges heat with encapsulated PCMs. To maintain the HTF temperature at ~170 °C at the outlet of the storage system, a commercial-grade organic PCM, A164 is chosen as its melting temperature is 171 °C. In the present study, Hytherm 600 is used as the heat transfer fluid. A detailed numerical model based

on two-temperature model and enthalpy technique is developed to analyse thermal transport in HTF and PCM and the phase change phenomena during melting and solidification of PCM. The numerical model is validated first and subsequently, it is used to study the effect of different parameters, such as mass flow rate, inlet HTF temperature, storage system dimension, encapsulation diameter and porosity on the thermal performance of LHTES system during the charging and discharging period. The energy stored, energy released, overall effectiveness and transient temperature difference in HTF temperature in a cycle of the LHTES system are determined to identify the important parameters affecting the performance of the storage system. This study can provide design guidelines of packed bed latent heat storage for other solar energy storage applications.

2. Mathematical Model

The schematic diagram, as shown in figure 1, represents the physical domain of LHTES system which is a cylinder of dimension D (diameter) $\times L$ (length). LHTES system forms the packed bed system where the filler material is spherical shaped encapsulated PCMs of diameter d_p each. In the numerical analyses, encapsulated PCMs are considered as the porous structure and HTF flows through the pores. The minimum porosity (ϕ) of the packed bed for normal random packing is considered as 0.375 [44]. The HTF is chosen as Hytherm 600, which is a thermic oil and commercial-grade organic material, A164 with $T_m = 171$ °C (make: Phase Change Material Products Limited, UK) is selected as the PCM based on the application. The thermophysical properties of PCM and HTF used in this study are tabulated in table 1. The LHTES is charged for $t_c = 1800$ s followed by $t_d = 1800$ s of discharging period. During the charging period, the HTF with constant inlet temperature ($T_{in,c}$) of 200 °C flows through the storage system and melts the PCM, thereby storing energy in the form of sensible and latent heat of PCM. In the discharging mode, HTF with low inlet temperature ($T_{in,d}$) 142 °C flows

through the charged LHTES, thereby retrieving the stored energy from the storage system by reducing the PCM temperature and solidifying it. The outer surface of the LHTES system is insulated. The basic storage configuration ($12 (D) \times 25 (L)$ cm) is obtained using steady state conduction heat transfer analysis for an energy storage capacity of 0.155 kWh for the charging period.

A two-dimensional axisymmetrical numerical domain is chosen, as shown in figure 1 along with boundary conditions, for predicting the thermal performance of LHTES system during melting and solidification of PCM. The direction of gravity (g) is taken along the flow direction.

a. Transport Equation in Porous region

Transport equations for flow through the packed bed latent heat thermal storage is a set of modified governing equations for the porous medium. While flowing through the LHTES system, HTF exchanges heat with filler object in the porous medium. In this storage system, encapsulation of PCM forms the filler object. In the numerical model, the flow of HTF is assumed to be laminar, Newtonian, incompressible and the thermophysical properties of PCM and HTF are considered to be constant over a range of temperature. The phase change process is modelled as isothermal. The shrinkage and expansion of PCM in spherical capsule are neglected during the phase change. The shell thickness of encapsulation is neglected as it is small compared to the diameter of the spherical capsule (d_p). Accordingly, the equations that govern the flow and heat transfer through this system can be written as,

Continuity equation:

$$\frac{\partial(\phi \rho_l)}{\partial t} + \nabla \cdot (\rho_l u) = 0 \quad (1)$$

Momentum equation [25]:

$$\frac{\partial(\rho_l u)}{\partial t} + \nabla \cdot \left(\rho_l \frac{uu}{\phi} \right) = -\phi \nabla P + \mu \left[\nabla^2 u + \frac{1}{3} \nabla (\nabla \cdot u) \right] + \phi \rho_l g + S1 \quad (2)$$

where source term ($S1$) is expressed as,

$$S1 = -\left(\frac{\mu u}{\alpha} + C_2 \frac{\rho_l |u|u}{2}\right) \quad (3)$$

203 where ρ_l , $\frac{1}{\alpha}$ and C_2 are fluid density, viscous resistance and inertial resistance coefficients,
 204 respectively. The momentum transport equation for the porous region accounts for additional
 205 pressure loss, which is incorporated in the source term, $S1$. This term consists of viscous and
 206 inertial losses, which is calculated using Darcy-Forchheimer relation. The viscous resistance
 207 and inertial resistance coefficients are calculated as [45],

$$\alpha = \frac{d_p^2 \phi^3}{150(1 - \phi)^2} \quad (4)$$

$$C_2 = \frac{1.75(1 - \phi)}{d_p \phi^3} \quad (5)$$

208 Energy equation [34]:

209 Heat transfer in porous region is solved using two-temperature model, where separate
 210 equations are written for the HTF and the PCM filler. The HTF and PCM in the packed bed is
 211 in thermal non-equilibrium and hence heat transfer between PCM and HTF is taken into
 212 account by the volumetric interstitial heat transfer coefficient (HTC), h_f .

213 For Heat Transfer Fluid:

$$\frac{\partial(\phi \rho_l c_{p,l} T_l)}{\partial t} + \nabla \cdot (\rho_l c_{p,l} T_l u) = \nabla \cdot (\phi k_l \nabla T_l) + h_f (T_s - T_l) \quad (6)$$

214 For Phase Change Material:

215 The melting and solidification of encapsulated PCM in the packed bed is modelled
 216 using enthalpy approach [36]. For simplicity, melt convection in PCM inside encapsulation is
 217 also neglected, however diffusion in the porous PCM structure is considered. Hence, the energy
 218 equation for PCM is written as,

$$\frac{\partial[(1 - \phi) \rho_s c_{p,s} T_s]}{\partial t} = \nabla \cdot [(1 - \phi) k_s \nabla T_s] + h_f (T_l - T_s) + S2 \quad (7)$$

219 where the source term is defined as,

$$S2 = -\frac{d[(1 - \phi) \rho_s \Delta H]}{dt} \quad (8)$$

220 The unsteady term S_2 , is the additional term added to account latent heat content (ΔH) during
 221 phase change of the PCM. The latent heat content in PCM is taken as a function of temperature
 222 such that its value is in the range of $[0, L_{latent}]$. For an isothermal phase change, the nodal
 223 latent heat content (ΔH) for the P^{th} control volume is updated after the n^{th} iteration as,

$$[\Delta H_p]_{n+1} = [\Delta H_p]_n + \frac{a_p}{a_p^0} c_{p,s} \lambda ([T_{PCM}]_n - T_m) \quad (9)$$

$$[\Delta H_p]_n = 0 \text{ if } [\Delta H_p]_{n+1} < 0 \quad (10)$$

$$[\Delta H_p]_n = L_{latent} \text{ if } [\Delta H_p]_{n+1} > L_{latent} \quad (11)$$

224 In equation 9, parameters a_p , a_p^0 are obtained by discretizing equation 7. It should be noted that
 225 equations 9-11 are valid for melting and solidification of the pure PCM. The liquid fraction of
 226 PCM at any instant can be calculated as,

$$f_l = \frac{\Delta H}{L_{latent}} \quad (12)$$

227 The interfacial HTC (h_f) in equations 6 and 7 for the thermal storage system can be
 228 determined from Wakao *et al.* [46], which is given as,

$$h_f = \frac{6(1 - \phi)(2 + 1.1Re_p^{0.6} Pr^{1/3})k_l}{d_p^2} \quad (13)$$

229 where $Re_p = \frac{\rho_l U_0 d_p}{\mu}$ and $Pr = \frac{\mu C_{p,l}}{k_l}$. The interstitial velocity (U_0) is defined as $\frac{U}{\phi}$, where U is
 230 the superficial velocity.

231 **b. Solution Method**

232 The governing equations (equations 1-8) are solved with an appropriate set of initial
 233 and boundary conditions. At the inlet, temperature and mass flow boundary condition are
 234 provided, whereas at the outlet, Neumann pressure and temperature boundary conditions are
 235 used. The adiabatic boundary condition is used for the insulated outer wall. The mathematical
 236 formulations of initial and boundary conditions can be expressed as,

237 Initial condition

(a) At $t = 0$ (charging) $T_i = 170.9 \text{ }^\circ\text{C}$ (14)

(b) Condition at the end of charging (at $t = t_c$) is taken as initial condition for discharging (15)

238 Boundary conditions

(c) Inlet: at $t \leq t_c$ $T_{HTF} = 200 \text{ }^\circ\text{C}$, $\dot{m} = 0.0029 \text{ kg/s}$ (16)

at $t_c < t \leq t_d$ $T_{HTF} = 142 \text{ }^\circ\text{C}$ (17)

(d) Outlet: $P = 0, \frac{dT_{HTF}}{dz} = 0, \frac{dT_{PCM}}{dz} = 0$ (18)

(e) Surface (wall): $\frac{dT_{HTF}}{dy} = 0, \frac{dT_{PCM}}{dy} = 0, u = v = 0$ (19)

(f) Axisymmetry: $\frac{dT_{HTF}}{dy} = 0, \frac{du}{dy} = \frac{dv}{dy} = 0$ (20)

239 The flow of HTF inside the porous structure (equations 1-2) is solved using a finite
 240 volume based commercial software Ansys FLUENT 14 [47]. Two temperature (non-
 241 equilibrium) energy equations of HTF and PCM are modelled by implementing User Defined
 242 Scalar (UDS) equations in FLUENT. In equations 6-7, the source terms, $h_i(T_s - T_l)$ and
 243 $\frac{d((1-\phi)\Delta H\rho_s)}{dt}$ are added to the system of UDS equations using User Defined Function (UDF).
 244 The first order implicit scheme is used for discretization of temporal terms. In spatial
 245 discretization, the least square cell based method is considered for gradient reconstruction,
 246 second order upwind scheme is used to discretise convective terms in momentum equation and
 247 first order upwind scheme is used for discretization of UDS equations. SIMPLE algorithm is
 248 used for dealing with pressure-velocity coupling [48]. The solution is assumed to be converged
 249 when the residual $\left(= \left| \frac{r-r_{old}}{r_{max}} \right| \right)$ for UDS and flow variable reaches a value less than 10^{-6} and
 250 10^{-3} , respectively.

251 **c. Charging, Discharging and Overall effectiveness**

To understand the effectiveness of charging and discharging of an encapsulated phase change material thermal energy storage system, few parameters are defined. The total energy content of HTF during charging time is given by,

$$E_s = \sum_{t_c} \dot{m} c_{p,l} (T_{in} - T_{out}(t)) \quad (19)$$

In calculating the energy released (E_r) during the discharging period, the discharging cut-off temperature is considered as 171 °C, below which storage system cannot be used effectively in removing the fluctuation in HTF temperature at the storage outlet. During discharging, if the PCM temperature drops below its melting temperature, the PCM in the system absorbs heat as sensible heat instead of latent heat. Accordingly, the total energy content of the HTF during the discharging period can be obtained as,

$$E_r = \sum_{t_d} \dot{m} c_{p,l} (T_{out}(t) - T_{in}) \quad (20)$$

where $T_{out} \geq 171$ °C.

The overall effectiveness (ε_o) is the product of charging effectiveness and the discharging effectiveness and is written as,

$$\varepsilon_o = \varepsilon_c \varepsilon_d \quad (21)$$

where charging effectiveness (ε_c) is the ratio of energy stored (E_s) to the LHTES system capacity of base case and discharging effectiveness (ε_d) is the ratio of energy released (E_r) to the LHTES system capacity of the base case.

The ability of the LHTES system to remove the temporal fluctuations in the HTF inlet temperature is quantified by defining a new parameter, transient temperature difference $\Delta T/\Delta t$ (°C/s) and it is expressed as,

$$\frac{\Delta T}{\Delta t} = \frac{(T_{HTF,max} - T_{HTF,min})}{(t_{T_{HTF,max}} - t_{T_{HTF,min}})} \text{ in a cycle} \quad (22)$$

where $t_{T_{HTF,max}}$ and $t_{T_{HTF,min}}$ are the time of occurrence of maximum and minimum temperature of HTF at the outlet, respectively, in a cycle. It should be noted that the maximum and minimum temperature of HTF are attained at the end of the charging and discharging period. In order to achieve less variation in the HTF outlet temperature, the value ΔT from equation (22) should be low and the time difference Δt should be large. Hence, $\left(\frac{\Delta T}{\Delta t}\right)$ close to zero indicates the variation in the HTF outlet temperature is minimal, whereas the high magnitude implies large variation in the HTF outlet temperature.

d. Model Validation and Grid Independence Study

The validation of present numerical model is performed based on the experimental work reported in Nallusamy *et al.* [39]. In their work, a cylindrical tank having dimensions of $L = 46$ cm and D (diameter) = 36 cm was used as LHTES system. Paraffin ($T_m = 60 \pm 1$ °C) was used as PCM and water was considered as HTF and the porosity of the system was taken as 0.5. Water at a constant mass-flow rate and constant temperature of 70 °C flows through the inlet. Two experimental cases are simulated to validate the present numerical model. In case 1, charging of LHTES is carried out with a mass flow rate of 4 lpm for 160 min, whereas the mass flow rate in case 2 is increased to 6 lpm. Figure 2 shows the comparison of HTF temperature at the location of 23 cm from the inlet, predicted by the present numerical model and measured experimentally. It can be observed from the figure that a reasonable agreement is achieved between the present numerical model and the experimental measurement. However, the discrepancy between the experimental and the numerically predicted results can be attributed to the following reasons. In the numerical model, paraffin is assumed to melt isothermally at 60 °C. The thermophysical properties of encapsulation shell material and heat loss to the ambient from the outer surface of LHTES system are also neglected. The maximum error between experimental and numerical results is found to be 6.8% and 7.9% for the cases with flow rate of 4 lpm and 6 lpm, respectively.

A grid independence study is performed to ensure the results are independent of grid size used to mesh the geometry. A uniform grid with three different sizes of 0.025 (finer), 0.05 (fine), and 0.1 (coarse) is used in this study, keeping time step constant at 0.1 s. The temperature difference (ΔT) of maximum and minimum HTF outlet temperature during the charging and discharging period is compared for three grid sizes. Figure 3 represents the variation in HTF temperature at the outlet of the storage system during charging and discharging period. The temperature difference (ΔT) for grid size of 0.1, 0.05 and 0.025 are found to be 3.38, 3.39 and 3.41 °C, respectively. The change in ΔT between finer (size 0.025) and fine grid (size 0.05) is 0.29% and fine and coarse grid (size 0.1) is 0.6%. Hence, the grid size of 0.05 is chosen for the further simulations.

3. Results and Discussion

a. Fluid Flow and Heat Transfer Characteristics

A LHTES system with $D= 12$ cm, $L= 25$ cm, $\phi= 0.5$ and $d_p= 3$ cm is considered as the base case. HTF is introduced at the inlet with the constant mass flow rate of 0.0029 kg/s during the charging and discharging period. The initial temperature of LHTES is considered as 170.9 °C, which is very close to the melting point of PCM. The outer wall of LHTES is considered as insulated as it corresponds to a conservative design. Figure 4 shows the temperature contours of HTF, PCM and melt fraction contours of PCM at the end of the charging period. It can be observed from figure 4(a) that the outlet temperature of HTF is 172 °C at the end of charging period due to the presence of PCM in the LHTES system. The temperature of HTF is found to be maximum at the inlet and minimum at the outlet. As the temperature of HTF is high relative to PCM, PCM absorbs energy from HTF and undergoes melting when the temperature reaches its melting temperature of 171 °C. The mode of heat transfer from HTF to PCM is convection as can be noted from equation 3. However, the mode of heat transfer in PCM in the LHTES

system is conduction as the flow of the molten PCM is neglected in equation 4. The temperature gradient in PCM follows the same trend as that in HTF. From figure 4(b), it can be observed that the major portion of PCM inside the LHTES is at the initial temperature of 170.9 °C, hence it remains in solid phase (figure 4(c)). This indicates that the time required for complete melting of PCM is insufficient and heat exchange between HTF and PCM is predominantly close to the inlet. In figure 4(c), melt fraction of 1 indicates complete molten PCM, whereas melt fraction of zero represents solid PCM state and the melt fraction between zero and 1 signifies the PCM is undergoing melting.

In figure 5(a), temperature contours of HTF is shown at the end of the discharging period. The HTF temperature increases along the flow direction and the maximum temperature (170 °C) is obtained at the outlet. The cold HTF which flows into the LHTES, extracts heat from PCM, resulting in solidification of PCM and in turn, HTF temperature increases along the length of the LHTES. Hence, the outlet temperature of HTF can be maintained at a desired temperature using PCM.

Figure 5(b) shows temperature contours of PCM at the end of the discharging period. The minimum temperature attained by the PCM is 142 °C at the inlet, which is the inlet temperature of HTF during the discharging period. As the PCM undergoes solidification and releases energy to HTF, its temperature reduces in the LHTES system. PCM near to the outlet undergoes slow solidification due to small temperature difference between PCM and HTF and hence, the drop in its temperature is moderate as compared to PCM close to the inlet.

Figure 5(c) shows melt fraction contours of PCM at the end of the discharge period of 3600 s. The melt fraction of PCM in the LHTES system at 3600 s is found to reduce from its value at the end of the charging period (1800 s). The rate of solidification of PCM depends on the position of PCM from the inlet of LHTES system and its melt fraction at 1800 s. The solidification rate of PCM is high close to the inlet due to low HTF inlet temperature, whereas

PCM away from the inlet, which was partially melted at the end of charging period, remains in liquid state.

b. Charging and Discharging Analysis

The aim of this study is to investigate the influence of geometrical and operational parameters on the overall effectiveness (ε_o) with respect to the base design and the transient temperature difference ($\Delta T/\Delta t$) in the HTF outlet temperature in a complete charging and discharging cycle. The flow direction (from inlet) of HTF is maintained the same throughout the cycle in this study. Based on the findings, a guideline can be formulated to design a packed bed latent heat thermal storage system, which can reduce the variation in HTF temperature at the storage outlet under fluctuating loading.

i. Effect of Mass Flow Rate

Figure 6(a) shows the effect of mass flow rate on the HTF outlet temperature. In this study, the dimensions of LHTES system are the same as the base case and the mass flow rate is varied by a factor of 0.5, 1, 1.5 and 2 with respect to the mass flow rate ($\dot{m} = 0.0029$ kg/s) in the base case. The Reynolds number (Re_p) corresponding to the maximum mass flow rate of 0.0058 kg/s considered in the present study is 1.58, which is less than $Re_{p,cr} \geq 300$ for onset of the turbulent regime in packed bed porous medium filled with spherical particles [49, 50]. The figure shows that the HTF outlet temperature remains unaffected by the HTF inlet temperature at the beginning of the charging period. This can be attributed to low thermal conductivity of PCM, due to which the thermal response of the LHTES system is slow to any change in the HTF temperature. Similar phenomena can be observed in the discharging period and for all the cases. After an elapsed time (*i.e.* residence time), the HTF outlet temperature increases from the initial temperature and stabilizes during melting of PCM. As the mass flow rate increases, the HTF outlet temperature starts rising early due to enhanced convective heat transfer and

reaches a higher value. During the discharging period, the residence time is shorter for high mass flow rate and the HTF outlet temperature decreases rapidly followed by the stabilization of temperature due to the solidification of PCM. As a result, a higher temperature difference (ΔT) between maximum and minimum temperature of HTF at the outlet is obtained for higher mass flow rate of HTF. The temperature difference (ΔT) is found to be 0.92, 3.39, 6.03 and 8.76 °C, respectively. Hence the mass flow rate influences the heat transfer rate between HTF and PCM significantly and it is an important factor in removing the temperature fluctuations.

The variations in energy stored (E_s), energy released (E_r) and overall effectiveness (ϵ_o) with mass flow rate ($\dot{m}$) are shown in figure 6(b) and (c). As $\dot{m}$ increases, flow velocity in the LHTES increases, which results in faster and enhanced heat transfer between HTF and PCM due to the increase in volumetric heat transfer coefficient (equation 13). Hence more energy is stored in the LHTES system with higher $\dot{m}$ during the charging period. However during discharging period, the amount of energy released (E_r) slightly increases with the increase in mass flow rate, considering the cut-off temperature of 171 °C. The overall system performance increases with increasing $\dot{m}$ as shown in figure 6(b). This is due to large amount of energy stored and released with increasing $\dot{m}$ as compared to the base case. Hence, the variation in mass flow rate has a significant effect on the thermal performance of LHTES system.

The variation in ($\Delta T/\Delta t$) with mass flow rate ($\dot{m}$) is shown in figure 6(d). The transient ΔT increases with increasing $\dot{m}$ for charging and discharging of the storage system. Although the overall effectiveness of system increases with mass flow rate, its ability to stabilize the HTF temperature at the outlet is reduced.

ii. Effect of Inlet Charging Temperature

The variation in transient HTF outlet temperature, E_s , E_r and ϵ_o for different inlet charging temperature ($T_{in,c}$) is illustrated in figure 7. The dimensions of LHTES system and mass flow rate are kept the same as the base case whereas HTF $T_{in,c}$ is varied as 180, 200, 220

and 240 °C. During discharging of the system, the HTF inlet temperature ($T_{in,d}$) is kept at 142 °C. The figure shows that the HTF temperature starts rising at the same time as the volumetric heat transfer coefficients are the same for all cases of inlet charging temperature. However the maximum temperature attained by the HTF decreases with decreasing inlet temperature for charging, which is due to the decrease in sensible energy content. The minimum HTF temperature attained during the discharging period in all cases is ~169 °C, except $T_{in,c} = 180$ °C. In the case of $T_{in,c} = 180$ °C, melting of PCM is minimal in the charging period, as a result, a very small amount of latent heat is released during the discharging period. Once the PCM is solidified, its temperature decreases rapidly due to sensible heat, which affects the HTF outlet temperature. The temperature difference (ΔT) obtained is 19.84, 3.39, 5.52 and 9.79 °C for the HTF inlet temperature of 180, 200, 220 and 240 °C, respectively.

The amount of energy stored increases with increasing $T_{in,c}$ as can be observed in figure 7(b). A relatively large amount of energy can be retrieved in the discharging period, if a large amount of energy is stored in the LHTES system during the charging period, keeping the inlet discharging HTF temperature and flow rate constant. As compared to the base case where $T_{in,c} = 200$ °C, the combined ability of the storage system, to store and retrieve more energy, while keeping discharging conditions same as the base case, increases. This is evident from figure 7(c) with increasing nature of ϵ_o for different inlet charging temperature. Figure 7(d) shows the variation in transient temperature difference ($\Delta T/\Delta t$) with $T_{in,c}$. The ability of the system to reduce the variation in the HTF outlet temperature is the highest when the inlet temperature of charging is 200 °C and this ability reduces with increasing HTF inlet temperature. The stabilization of HTF temperature occurs due to the isothermal phase change of the PCM where latent heat of PCM plays a significant role. However, the variation in HTF outlet temperature is rapid in case of sensible heat exchange. When the inlet charging temperature of LHTES

system is less than 200 °C, the sensible heat exchange is more than the latent heat exchange and hence, a large variation in $\Delta T/\Delta t$ is obtained.

iii. Effect of LHTES Dimension

The effect of LHTES dimension on the HTF outlet temperature with time, energy stored (E_s), energy released (E_r) and overall effectiveness (ϵ_o) for a given LHTES volume is shown in figure 8. Six more cases are considered for this study, viz. (a) $D= 24$ cm, $L= 6$ cm, (b) $D= 20$ cm, $L= 9$ cm, (c) $D= 16$ cm, $L= 14$ cm, (d) $D= 8$ cm, $L= 55$ cm, (e) $D= 6.65$ cm, $L= 80$ cm and (f) $D= 4$ cm, $L= 219$ cm, keeping other parameters constant with respect to the base case ($D= 12$ cm, $L= 25$ cm). The velocity of HTF flowing into the LHTES is higher for a small tank diameter as compared to large diameter for the same mass flow rate. When the adiabatic conditions are maintained throughout the outer surface, the temperature differences (ΔT) for all cases obtained are 16.36, 11.04, 7.08, 3.39 0.92 and 0.54 °C, respectively, for (a), (b), (c), (d) and (e) as compared to the base case, where $\Delta T = 3.39$ °C. In case (f), the temperature difference (ΔT) is zero. Hence, the LHTES with larger diameter and shorter length has large ΔT , whereas in opposite case, ΔT is small as HTF travels relatively longer path to reach the outlet.

Volumetric heat transfer coefficient increases as the system diameter reduces for a given volume of LHTES, due to which HTF extracts heat from the PCM. The energy stored increases marginally with decreasing diameter and increasing length of the LHTES as shown in figure 8(b). Also, the amount of energy released in all cases has a slight variation considering the cut-off temperature of 171 °C.

The ability of LHTES in storing and retrieving energy is not significantly affected by the large variation in dimensions of the storage system, which can be seen by comparing the magnitude of effectiveness obtained for varying storage diameter. Overall effectiveness varies within 0.36-0.46, as can be observed in figure 8(c), which implies that the dimension of TES

system has a marginal effect on storing and retrieving energy from PCM for a short duration of charging and discharging processes. However, an interesting trend can be observed in figure 8(d), which shows that the magnitude of rate of change in ΔT increases with increasing LHTES diameter. The increase in storage dimension *i.e.* diameter reduces its ability to a large extent in reducing temperature fluctuations, compared to mass flow rate and inlet charging temperature (figure 6(d), 7(d), 8(d)). The LHTES with small diameters and larger length is hence useful in stabilizing the HTF outlet temperature.

iv. Effect of Porosity

Figure 9(a) shows the influence of porosity on the HTF outlet temperature with time, energy stored (E_s), energy released (E_r) and overall effectiveness (ε_o). In this case, the LHTES volume is kept constant, whereas mass of the PCM is varied. As the minimum void fraction in the storage system is 0.375 for normal random packing of encapsulated PCMs, five different cases with porosity of 0.4, 0.5, 0.6, 0.7, and 0.8 are considered keeping the remaining parameters constant. With increase in porosity, the amount of the PCM content decreases in the LHTES and more pores are available. The temperature difference (ΔT) for the cases obtained are 0.81, 3.39, 8.68, 17 and 27.61 °C, respectively. Thus, for the cases with low porosity, the isothermal melting of PCM results in better stabilization of HTF temperature. It is evident from the figure that the rise in HTF temperature is least for porosity $\phi = 0.4$ and highest for the porosity, $\phi = 0.8$. The initial time, for which the temperature of HTF is constant at the outlet, increases with increasing porosity. The initial delay in the rise in the HTF outlet temperature is due to thermal inertia associated with large volume of HTF in the system that increases with porosity. However, the stabilization of HTF outlet temperature reduces due to reduced PCM volume with increasing porosity. It can be observed from figure 9(b) that the amount of energy released increases with the increase in porosity, whereas amount of energy stored decreases with increasing porosity. As porosity of the storage system increases, the

extraction of energy by HTF is relatively faster during the discharging time. Due to the marginal decrease in energy stored and marginal increase in energy released with increased porosity, the overall effectiveness of storage system has least variation. The transient variation in ΔT increases with an increase in porosity as observed from figure 9(d). The stabilization of HTF temperature for longer period is due to the isothermal phase change of larger mass of PCM and hence the magnitude of rate of change is minimal for porosity of 0.4. Hence it can be noted that the porosity of storage system may have significant effect on the ability of LHTES in removing the fluctuation in incoming HTF temperature. The ratio $\left(\frac{\Delta T}{\Delta t}\right)$ deviates more than 0.02, which is highest in case of porosity, compared to other parameters, such as mass flow rate, inlet HTF temperature, LHTES dimension (figure 6(d), 7(d), 8(d) and 9(d)) investigated in the present study.

An important parameter to consider in designing of LHTES system packed with encapsulated PCMs is pressure drop across the porous region due to change in porosity. For the numerical domain considered, the pressure drop across the porous medium is found to be 0.037, 0.07, 0.184, 0.47 and 1.27 Pa for porosity of 0.8, 0.7, 0.6, 0.5 and 0.4, respectively. It can be noted that the pressure drop across the porous region increases exponentially with the decrease in porosity. The pressure drop rises by 2.7 times when the porosity is reduced to 0.4 from the base case porosity of 0.5. Hence, in large thermal energy storage systems, there is a limitation on the reduction of porosity, which could result in enormous pressure losses leading to high pumping power requirement.

v. Effect of Encapsulation Diameter

The effect of encapsulation shell diameter (d_p) on the HTF outlet temperature, energy stored (E_s), energy released (E_r) and overall effectiveness (ε_o) is studied by varying the shell diameter as 1, 2, 3, 4 and 5 cm keeping LHTES volume constant and is depicted in figure 10. The temperature differences (ΔT) obtained are 0.1, 1, 3.39, 16.34 and 24.66 °C, respectively

for the corresponding cases. It can be seen from figure 10(a) that the temperature of HTF at the outlet is constant at 171 °C for shell diameter of 1 cm. With increase in shell size, the outlet temperature of HTF is observed to rise faster and, while discharging, there is a rapid decline in the HTF outlet temperature. This can be attributed to the enhancement in PCM melting, due to increase in volumetric heat transfer coefficient with decreasing shell size (equation 13). Thus, the outlet temperature of HTF stabilizes more and shows less rise for smaller shell diameter.

The LHTES is charged more when the shell diameter is small as evident from figure 10(b). In case of shell diameter of 1 cm, the energy stored during the charging period is extracted completely in the discharging period. For the case when particle diameter is increased, outlet temperature of HTF is not constant and the amount of energy released, is significantly decreased initially and then increases marginally until the temperature reaches a cut-off value of 171 °C. The overall effectiveness of the system thus depends on energy stored and released (figure 10(c)). The system performs better when $d_p = 1$ cm and any change in d_p above 2 cm has insignificant effect on the overall energy storage and retrieving process.

Figure 10(d) shows the temporal variation in ΔT with different encapsulation diameter. The stabilization of outlet HTF temperature is better for the LHTES with smaller encapsulation shell diameter. Thus, the magnitude of rate of change in ΔT increases with increasing shell diameter and for $d_p > 3$ cm, the ratio $\left(\frac{\Delta T}{\Delta t}\right)$ deviates faster from the ideal value of zero as seen in figure 10(d).

4. Conclusions

In this paper, numerical modelling of latent thermal energy storage system is developed, where the flow equation is solved for HTF in the porous region along with two temperature (non-equilibrium model) energy equations for accounting thermal transport in PCM and HTF. The isothermal phase change in the PCM is modelled using enthalpy technique. A detailed

parametric study is performed to investigate the effect of mass flow rate, inlet charging temperature, storage system dimensions, encapsulation shell diameter and porosity on the HTF temperature at the storage outlet during the charging and discharging period of a cycle. Parameters, such as energy stored (E_s), energy released (E_r) and overall effectiveness (ϵ_o) and transient temperature difference ($\Delta T/\Delta t$) are defined to quantify the influence of operational and geometrical parameters on charging and discharging of the storage system.

It is found that the thermal performance of storing and retrieving energy increases with increasing mass flow rate and inlet charging temperature. However, storage system dimensions and porosity of the system has no significant effect on the overall effectiveness of the system with respect to the base case. Reducing encapsulation diameter is found to be more suitable for reducing the fluctuation in HTF temperature at the storage outlet. Among the various parameters studied, it can be noted that the increase in porosity and storage diameter for a fixed volume of LHTES reduce the storage system ability in reducing the fluctuations in HTF temperature to a large extent. Based on these findings, the present study can provide the guidelines for designing of packed bed latent heat thermal energy storage filled with spherical PCMs for other applications in solar thermal systems.

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544 **References**

- 545 1. H. Ibrahim, A. Ilinca, J. Perron, Energy storage systems-Characteristics and comparisons,
546 Renewable and Sustainable Energy Reviews 12 (2008) 1221-1250
- 547 2. A. Gil, M. Medrano, I. Martorell, A. Lazaro, P. Dolado, B. Zalba, L. F. Cabeza, State of
548 the art on high temperature thermal energy storage for power generation. Part 1-Concepts,
549 materials and modellization, Renewable and Sustainable Energy Reviews 14 (2010) 31–
550 55.
- 551 3. K. Nithyanandam, R. Pitchumani, Optimization of an encapsulated phase change material
552 thermal energy storage system, Solar Energy 107 (2014) 770–788.
- 553 4. J. Yang, L. Yang, C. Xu, X. Du, Experimental study on enhancement of thermal energy
554 storage with phase-change material, Applied Energy 169 (2016) 164–176.
- 555 5. L. W. Fan, Z. Q. Zhu, S. L. Xiao, M. J. Liu, H. Lu, Y. Zeng, Z. T. Yu, K. F. Cen, An
556 experimental and numerical investigation of constrained melting heat transfer of a phase
557 change material in a circumferentially finned spherical capsule for thermal energy
558 storage, Applied Thermal Engineering 100 (2016) 1063–1075.
- 559 6. S. Harish, D. Orejon, Y. Takata, M. Kohno, Thermal conductivity enhancement of lauric
560 acid phase change nanocomposite with graphene nanoplatelets, Applied Thermal
561 Engineering 80 (2015) 205-211.
- 562 7. J. M. Khodadadi, L. Fan, H. Babaei, Thermal conductivity enhancement of nanostructure-
563 based colloidal suspensions utilized as phase change materials for thermal energy storage:
564 A review, Renewable and Sustainable Energy Reviews 24 (2013) 418–444.
- 565 8. Y. Yang, J. Kuang, H. Wang, G. Song, Y. Liu, G. Tang, Enhancement in thermal property
566 of phase change microcapsules with modified silicon nitride for solar energy, Solar
567 Energy Materials and Solar Cells 151 (2016) 89-95.
- 568 9. Y. Tomizawa, K. Sasaki, A. Kuroda, R. Takeda, Y. Kaito, Experimental and numerical
569 study on phase change material (PCM) for thermal management of mobile devices,
570 Applied Thermal Engineering 98 (2016) 320-329.
- 571 10. J. F. Li, W. Lu, Y. B. Zeng, Z. P. Luo, Simultaneous enhancement of latent heat and
572 thermal conductivity of docosane-based phase change material in the presence of spongy
573 graphene, Solar Energy Materials and Solar Cells 128 (2014) 48-51.

11. T. Wang, S. Wang, R. Luo, C. Zhu, T. Akiyama, Z. Zhang, Microencapsulation of phase change materials with binary cores and calcium carbonate shell for thermal energy storage, *Applied Materials* 171 (2016) 113-119.
12. R. Sigel, Solidification of low conductivity material containing dispersed high conductivity particles, *International Journal of Heat and Mass Transfer* 20 (1977) 1087–1089.
13. E. S. Mettawee, G. M. R. Assassa, Thermal conductivity enhancement in a latent heat storage system, *Solar Energy* 81 (2007) 839–845.
14. F. Agyenim, N. Hewitt, P. Emes, M. Smyth, A review of materials, heat transfer and phase change problem formulation for latent heat thermal energy storage systems (LHTESS) *Renewable and Sustainable Energy Reviews* 14 (2010) 615-628.
15. R. Velraj, R. V. Seeniraj, B. Hafner, C. Faber, K. Schwarzer, Heat transfer enhancement in a latent heat storage system, *Solar Energy* 65 (1999) 171–180.
16. F. Regin, S. C. Solanki, S.C., J. S. Saini, An analysis of a packed bed latent heat thermal energy storage system using PCM capsules: Numerical investigation, *Applied Energy* 34 (2009) 1765-1773.
17. W. Zhao, S. Neti, A. Oztekin, Heat transfer analysis of encapsulated phase change materials, *Applied Thermal Engineering* 50(1) (2013) 143–151.
18. K. Nithyanandam, R. Pitchumani, A. Mathur, Analysis of a latent thermocline storage system with encapsulated phase change materials for concentrating solar power, *Applied Energy* 113 (2014) 1446-1460.
19. K. A. R. Ismail, C. L. F. Alves, M. S. Modesto, Numerical and experimental study on the solidification of PCM around a vertical axially finned isothermal cylinder, *Applied Thermal Engineering* 21 (2001) 53–77.
20. Z. Liu, X. Sun, C. Ma, Experimental investigations on the characteristics of melting processes of stearic acid in an annulus and its thermal conductivity enhancement by fins, *Energy Conversion and Management* 46 (2005) 959–969.
21. M. Gharebaghi, I. Sezai, Enhancement of heat transfer in latent heat storage modules with internal fins, *Numerical Heat Transfer* 53 (2008) 749-765.
22. T. E. W. Schumann, Heat transfer: a liquid flowing through a porous prism, *Journal of The Franklin Institute* 208 (1929) 405-416.
23. A. Shitzer, M. Levy, Transient behavior of a rock-bed thermal storage system subjected to variable inlet air temperatures: Analysis and experimentation, *Journal of Solar Energy Engineering, ASME* 105(2) (1983) 200-206.

24. D. E. Beasley, J. A. Clark, Transient response of a packed bed for thermal energy storage, *International Journal of Heat and Mass Transfer* 27(9) (1984) 1659-1669.
25. Z. Yang, S. V. Garimella, Molten-salt thermal energy storage in thermoclines under different environmental boundary conditions, *Applied Energy* 87 (2010) 3322-2239.
26. F. G. F. Qin, X. Yang, Z. Ding, Z. Zuo, Y. Shao, R. Jiang, X. Yang, Thermocline stability criteria in single-tanks of molten salt thermal energy storage, *Applied Energy* 97 (2012) 816-821.
27. N. Boerema, G. Morrison, R. Taylor, G. Rosengarten, Liquid sodium versus HITEC as a heat transfer fluid in solar thermal central receiver systems, *Solar Energy* 86 (2012) 2293-2305.
28. E. Milisic, Modelling of energy storage using phase-change materials (PCM materials), Master thesis, Norwegian University of Science and Technology, 2013.
29. J. Emerson, H. Micah, S. Panneer, Concrete as a thermal energy storage medium for thermocline solar energy storage systems, *Solar Energy* 96 (2013) 194-204.
30. K. A. R. Ismail, J. R. Henriquez, Numerical and experimental study of spherical capsules packed bed latent heat storage system, *Applied Thermal Engineering* 19 (1999) 757-788.
31. D. Arnold, Dynamic simulation of encapsulated ice stores, part 1-the model, *ASHRAE Transactions* 96 (1990) 1103-1110.
32. S. Wu, G. Fang, Dynamic performance of solar heat storage system with packed bed using myristic acid as phase change material, *Energy Build* 43 (2011) 1091-1096.
33. A. Benmansour, M. A. Hamdan, A. Bengueuddach, Experimental and numerical investigation of solid particles thermal energy storage unit, *Applied Thermal Engineering* 26 (2006) 513-518.
34. S. M. Flueckiger, S. V. Garimella, Latent heat augmentation of thermocline energy storage, *Applied Energy* 116 (2014) 278-287.
35. H. Peng, H. Dong, X. Ling, Thermal investigation of PCM-based high temperature thermal energy storage in packed bed, *Applied Energy* 81 (2014) 420-427.
36. D. Brent, V.R. Voller, K.J. Reid, Enthalpy-porosity technique for modelling convection-diffusion phase change: application to the melting of a pure metal, *Numerical Heat Transfer* 13 (1988) 297-318.
37. M. Veerappan, S. Kalaiselvam, S. Iniyan, R. Goic, Phase change characteristic study of spherical PCMs in solar energy storage, *Solar Energy* 83 (2009) 1245-1252.
38. M. A. Izquierdo-Barrientos, C. Sobrino, J. A. Almendros-Ibáñez, Thermal energy storage in a fluidized bed of PCM, *Chemical Engineering Journal* 230 (2013) 573-583.

39. N. Nallusamy, S. Sampath, R. Velraj, Experimental investigation on a combined sensible and latent heat storage system integrated with constant/varying (solar) heat sources, *Renewable Energy* 32 (2007) 1206–1227.
40. N. A. M. Amin, M. Belusko, F. Bruno, An effectiveness-NTU model of a packed bed PCM thermal storage system, *Applied Energy* 134 (2014) 356–362.
41. S. Bellan, T. E. Alam, J. G. Aguilar, M. Romero, M. M. Rahman, D. Y. Goswami, E. K. Stefanakos, Transient numerical analysis of storage tanks based on encapsulated PCMs for heat storage in concentrating solar power plants, *Energy Procedia* 57 (2014) 672 – 681.
42. S. Bellan, T. E. Alam, J. G. Aguilar, M. Romero, M. M. Rahman, D. Y. Goswami, E. K. Stefanakos, Numerical and experimental studies on heat transfer characteristics of thermal energy storage system packed with molten salt PCM capsules, *Applied Thermal Engineering* 90 (2015) 970-979.
43. M. Rady, Thermal performance of packed bed thermal energy storage units using multiple granular phase change composites, *Applied Energy* 86 (2009) 2704–2720.
44. G. G. Brown *et al.*, *Unit Operations*, Wiley and Chapman & Hall, New York- London, 1950.
45. K. Vafai, Convective flow and heat transfer in variable-porosity media. *Journal of Fluid Mechanics* 147 (1984) 233-259.
46. N. Wakao, S. Kaguei, T. Funazkri, Effect of fluid dispersion coefficients on particle-to-fluid heat transfer coefficients in packed beds: correlation of Nusselt numbers, *Chemical Engineering Science* 34(3) (1979) 325–336.
47. FLUENT Manual, ANSYS/FLUENT Release Version 14. ANSYS Inc., Documentation, 2012.
48. S.V. Patankar, *Numerical Heat Transfer and Fluid Flow*, Hemisphere, Washington, DC, 1980.
49. K. R. Jolls, T.J. Hanratty, Transition to turbulence for flow through a dumped bed of spheres. *Chem. Engng Sci.* 21(2) (1966) 1185-1190.
50. S. Rode, N. Midoux, M. A. Latifi, A. Stork, E. Saadjan, Hydrodynamic of liquid flows in packed beds: an experimental study using electrochemical shear rate sensors. *Chem. Engng Sci.* 49 (6) (1994) 889-900.

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Table 1: Thermophysical properties of PCM (A164) and Hytherm 600

	PCM (A164)	HTF (Hytherm 600)
ρ (kg/m ³)	1500	720.9
T_m (°C)	171	-
c_p (J/kg.K)	2013.4	3097.4
k (W/m.K)	0.425	0.116
L_{latent} (kJ/kg)	272	-
μ (kg/m.s)	3×10^{-3}	19.46×10^{-3}
β (K ⁻¹)	0.0015	-

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(b)

(c)

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(b)

(c)

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(a)

(b)

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(a)

(b)

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