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Thermal Analysis of Solar Thermal Energy Storage in a Molten-Salt Thermocline

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ABSTRACT

A comprehensive, two-temperature model is developed to investigate energy storage in a molten-salt thermocline. The commercially available molten salt HITEC is considered for illustration with quartzite rocks as the filler. Heat transfer between the molten salt and quartzite rock is represented by an interstitial heat transfer coefficient. Volume-averaged mass and momentum equations are employed, with the Brinkman-Forchheimer extension to the Darcy law used to model the porous-medium resistance. The governing equations are solved using a finite-volume approach. The model is first validated against experiments from the literature and then used to systematically study the discharge behavior of thermocline thermal storage system. Thermal characteristics including temperature profiles and discharge efficiency are explored. Guidelines are developed for designing solar thermocline systems. The discharge efficiency is found to be improved at small Reynolds numbers and larger tank heights. The filler particle size strongly influences the interstitial heat transfer rate, and thus the discharge efficiency.

Keywords: Solar thermal energy, energy storage, thermocline, molten salt

1 INTRODUCTION

Parabolic-trough solar thermal electric technology is one of the promising approaches to providing the world with clean, renewable and cost-competitive power on a large scale. In a solar parabolic-trough plant, solar thermal energy is collected by troughs in a collector field, and then delivered by a heat transfer fluid (HTF) into a steam generator to generate steam for producing electricity in a Rankine steam turbine. In early studies and applications of this technology, synthetic oils with operating temperatures below 400°C were used as the HTF in prototype plants [1–4]. For instance, the Solar Electric Generating Stations (SEGS) I plant used Caloria® as the HTF. Caloria® is in a liquid state at atmospheric pressures at temperatures below 315°C. Although higher HTF operating temperatures are desired in order to achieve higher Rankine cycle efficiencies, currently available candidate oils are limited to operating below approximately 400°C; the pressurization needed to operate at higher temperatures is prohibitively expensive. Hence, fossil fuels need to be used to further superheat the steam generated by the oils to the higher temperatures desired in the turbine.

More viable candidates for high-temperature HTFs are molten salts, such as the commercially available HITEC (binary) and HITEC XL (ternary). Molten salts can operate up to a higher temperature range of 450 to 500°C with very low vapor pressures. The use of a molten salt as the HTF can achieve a higher output temperature from the collector field, resulting in the generation of steam at temperatures above 450°C for use in the turbine, which in turn raises the Rankine

cycle efficiency to approximately 40%. In comparison, current high-temperature oils generate steam at 393°C with a corresponding cycle efficiency of 37.6% [5]. The use of molten salts as thermal storage media allows for higher storage temperatures, thereby reducing the volume of the thermal storage unit for a given storage capacity. Moreover, molten salts are cheaper and more environmentally friendly than currently available high-temperature oils. The major disadvantage of molten salts is their relatively high melting temperature (149°C for HITEC XL relative to 15°C for Caloria[®] and 12°C for Therminol VP-1[®]), which necessitates special measures such as the use of fossil fuels or electric heating to maintain the salts above their melting temperatures in order to avoid serious damage to the equipment when solar power is unavailable at night or in poor weather conditions.

Thermal energy storage [6-8] for solar thermal power plants [9-12] offers the potential to deliver electricity without fossil fuel backup as well as to meet peak demand, independent of weather fluctuations. The current baseline design for SEGS plants uses Therminol VP-1[®] as the heat transfer fluid in the collector field. Therminol VP-1[®] has a low freezing point of 12°C, and is stable up to 400°C, which is higher than the operating temperature possible with Caloria[®]. However, it is still difficult to use Therminol VP-1[®] as the HTF near or above 400°C in practical applications, due to its undesirably high vapor pressure (> 1 MPa) which can incur significant costs for pressurization of the system. A near-term solution for thermal storage in solar-trough plants is to use indirect thermal storage wherein solar thermal energy delivered by the Therminol oil from the collector field is transferred, through a

heat exchanger, to molten salt which serves as the storage medium. The expensive heat exchanger may be eliminated by employing direct thermal storage. In a direct molten-salt thermal storage system, a single fluid, *e.g.*, the molten salt, serves as both the HTF and the storage medium, and flows directly between the collector-field pipes and the thermal storage tanks. The direct solar thermal energy storage approach is attractive for future parabolic-trough solar thermal power plants both in terms of higher efficiency and lower cost.

In both indirect and direct molten-salt thermal storage systems, there are two prevailing design options: two-tank storage, and single-tank thermocline storage. In a two-tank storage system, the molten-salt HTF flows from a cold tank, through the oil-salt heat exchange (indirect system) or the collector field (direct system), to a hot tank during a charge cycle, and flows back from the hot tank, through the steam generator, to the cold tank during a discharge cycle. The two-tank molten-salt storage design was used in the Solar Two demonstration plant [13], and was shown to be a low-risk and cost-effective approach. Compared to the two-tank storage system, single-tank thermocline storage offers the potential for significantly reducing storage costs. The thermocline storage approach uses a packed bed [14-17] in a single tank that is marginally larger than one of the tanks in a two-tank thermal storage system as used in Solar Two. Buoyancy forces help maintain stable thermal stratification between hot and cold molten salts in the same tank in this one-tank approach. A low-cost filler material is used to fill most of the thermocline tank volume and acts as the primary thermal storage medium; this helps reduce the quantity of the relatively

more expensive molten salt, and presents a significant cost advantage over the two-tank approach. It was shown using system-level models [18,19] that thermocline storage may offer the lowest-cost energy storage option, saving 35% of the cost relative to the two-tank storage system.

Ideal filler materials for thermocline thermal energy storage should meet several requirements: low cost, wide availability, high heat capacity, and compatibility with the molten-salt HTFs. A wide range of materials, including quartzite, taconite, marble, NM limestone, apatite, corundum, scheelite and cassiterite, have been considered as candidates for the filler material in a HITEC XL molten-salt thermocline storage system [20]. Quartzite rock and silica sand were found to withstand the molten salt environment with no significant deterioration that would impact the performance or operability of a thermocline thermal storage. A demonstration on such a thermocline on a pilot-scale (2.3 MWh) was reported in Ref. [21].

Although a few studies of molten-salt thermocline thermal energy storage for parabolic-trough solar thermal plants have been reported, the thermal behavior and efficiency of these systems under different operating conditions is not yet well-understood. A model that is capable of predicting the charge/discharge efficiency is needed, as are guidelines for the design of molten-salt thermocline systems for parabolic-trough solar thermal electric plants.

The present work develops a comprehensive analysis of the discharge dynamics of molten-salt thermocline thermal energy storage for parabolic-trough solar thermal

electric plants. HITEC molten salt is considered as the HTF and quartzite rock as the filler in the computations, although the analysis methodology is valid for any combination of salt and filler. The thermal behavior, including temperature profiles and discharge efficiency, are specifically investigated. Based on results from the model, guidelines are developed for the design of thermocline thermal energy storage systems.

2 DEVELOPMENT OF A THERMOCLINE MODEL

The thermocline unit considered for analysis is schematically illustrated in Fig. 1. The height of the filler region is denoted h , h' is the height of the distributor region, d the diameter of the cylindrical tank, and d' the diameter of the ports, as shown in the figure. An axisymmetric coordinate system is used as indicated. The cylindrical thermocline thermal storage tank has inlet/exit ports at the center of the top and bottom surfaces. The bulk of the tank is occupied by a filler material, quartzite rock, at a porosity of ε . A molten-salt heat transfer fluid, HITEC, fills the pore volume as well as the unfilled portions at the top and bottom of the tank as shown in the figure. HITEC is a eutectic mixture of water-soluble, inorganic salts: potassium nitrate (53% wt), sodium nitrite (40% wt) and sodium nitrate (7% wt). It is in a liquid state above 149°C (its melting temperature) and very stable up to 538°C. Its physical properties, such as viscosity and thermal conductivity, change with temperature. HITEC is nonflammable, non-explosive and evolves no toxic vapors under recommended conditions of use, and therefore is considered a potential candidate for

molten-salt HTFs used in parabolic-trough solar thermal electric plants [23].

During the charging (heating) period, hot molten salt from the collector field enters the storage tank from the upper port, transfers heat to the cold filler material, and exits the storage tank at a lower temperature through the bottom port. Thermal energy from the collector field is thus stored in the filler medium of the storage tank. During discharge, cold liquid is pumped into the storage tank through the bottom port, heated by the hot filler medium, and drawn from the tank at a higher temperature through the upper port. The thermal energy stored in the filler medium is thus retrieved by the cold fluid for further use.

Since energy storage in a thermocline depends on buoyancy to maintain thermal stratification, a uniform flow at the inlet and outlet contributes to effective thermal stratification and for improved performance of the thermocline. Therefore, measures for maintaining a uniform flow condition at the inlet and outlet are needed in practice. In Fig. 1, two distributor regions (free of filler material) are included at the upper and lower ends of the filler region in the storage tank. Due to their low flow resistance (compared to the filler material region), the distributor regions are expected to lead to a uniform distribution of the flow at the inlet and outlet.

The volume-averaged governing continuum and momentum equations for the HTF phase are:

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

$$\frac{\partial(\rho_l \bar{u})}{\partial t} + \nabla \cdot \left(\rho_l \frac{\bar{u} \bar{u}}{\varepsilon} \right) = -\varepsilon \nabla p + \nabla \cdot \bar{\bar{\tau}} + \varphi_l \bar{g} + \varepsilon \left(\frac{\mu}{K} \bar{u} + \frac{F}{\sqrt{K}} \rho_l u_{mag} \bar{u} \right) \quad (2)$$

where $\bar{\bar{\tau}} = 2\mu \bar{S} - \frac{2}{3}\mu S_{kk} \bar{I}$, $\bar{S} = \frac{\nabla \bar{u} + (\nabla \bar{u})^T}{2}$ and $\nabla = \bar{e}_r \frac{\partial}{\partial r} + \frac{\bar{e}_\theta}{r} \frac{\partial}{\partial \theta} + \bar{e}_x \frac{\partial}{\partial x}$. In the axisymmetric coordinate system shown in Fig. 1, the problem is two-dimensional, and the velocity vector is $\bar{u} = u_r \bar{e}_r + u_x \bar{e}_x$ and its derivatives in the θ -direction are all zeros, *i.e.*, $\frac{\partial u_r}{\partial \theta} = \frac{\partial u_x}{\partial \theta} = 0$.

Since the HTF and the filler material may be at different temperatures due to their distinct thermal conductivities and heat capacities, the energy equation is applied separately to the two phases. For the HTF, the energy equation is:

$$\begin{aligned} & \frac{\partial [\varphi_l C_{p,l} (T_l - T_c)]}{\partial t} + \nabla \cdot [\rho_l \bar{u} C_{p,l} (T_l - T_c)] \\ & = \nabla \cdot (k_e \nabla T_l) - p \nabla \cdot \bar{u} + \text{tr} \left[\nabla \left(\frac{\bar{u}}{\varepsilon} \right) \cdot \bar{\bar{\tau}} \right] + \frac{\bar{u} \cdot \bar{u}}{2\varepsilon} \frac{\partial \rho_l}{\partial t} + h_i (T_s - T_l) \end{aligned} \quad (3)$$

and for the filler, it is:

$$\frac{\partial [(1-\varepsilon) \rho_s C_{p,s} (T_s - T_c)]}{\partial t} = -h_i (T_s - T_l) \quad (4)$$

The heat transfer between the HTF and the filler is accounted for with a volumetric interstitial heat transfer coefficient h_i , which appears as a source term on the right side of Eqs. (3) and (4). Heating of the HTF caused by compression work due to volume expansion/shrinkage, viscous effects and kinetic energy changes, as respectively shown in the second, third and fourth terms on the right side of Eq. (3), is small and less than 10^{-4} times the conduction or convection terms, but is included for completeness.

In a thermocline using quartzite rock as the filler, the filler particles are completely surrounded by the HTF (a continuous phase) and have poor thermal

contact with neighboring particles; therefore, the filler is treated as a dispersed phase embedded in a continuous HTF phase. The effective thermal conductivity k_e of the HTF-filler mixture in Eq. (3) can then be expressed as [22]

$$k_e = k_l \frac{1 + 2\beta\phi + (2\beta^3 - 0.1\beta)\phi^2 + \phi^3 0.05 \exp(4.5\beta)}{1 - \beta\phi} \quad (5)$$

where $\phi = 1 - \varepsilon$ and $\beta = \frac{k_s - k_l}{k_s + 2k_l}$. Equation (5) provides a good estimate for the effective conductivity of liquid-saturated porous media with ϕ ranging from 0.15 to 0.85 and β from -0.499 to 1. Thermal conduction of the filler phase is, however, neglected in Eq. (4) because of the large thermal contact resistance between filler particles; conduction within the filler particles is also neglected due to their small size (< 0.1 m). Temperature changes in the filler particles are thus solely due to heat transfer from the HTF phase.

The physical properties of HITEC change with temperature, and can be calculated according to the following curvefits to experimental measurements reported by Coastal Chemical Co. [23]:

$$\text{Density: } \rho_l = 1938.0 - 0.732(T_l - 200.0) \quad (6)$$

$$\text{Viscosity: } \mu = \exp[-4.343 - 2.0143(\ln T_l - 5.011)] \quad (7)$$

$$\text{Thermal conductivity: } k_l = -6.53 \times 10^{-4}(T_l - 260.0) + 0.421 \quad (8)$$

The fits in Eqs. (6) – (8) represent the experimental data with maximum errors of 0.2%, 0.5% and 0.8%, respectively. The heat capacity of HITEC is relatively constant at 1561.7 J/kg-K according to the published data. Properties of quartzite rock are treated as constants, with a specific heat capacity of 830 J/kg-K and density

of 2500 kg/m^3 [24].

In practice, flow distributors are generally employed to ensure a uniform flow condition at the inlet/outlet of the filler region. The characteristic length of the distributor (represented by diameter d or height h' as shown in Fig. 1) is much larger than the particle size in the filler region. Also, the molten salt is pumped into the distributor through the port at a velocity much larger than that in the filler region because of the small cross-sectional area of the port relative to the open frontal area of the filler region. For these two reasons, a high-Reynolds-number turbulent flow is present in the distributor region; the flow in the filler region, in contrast, is laminar, at a much lower local Reynolds number defined based on the particle size. To model the turbulent flow in the distributor region, the standard k - ε model with a standard wall function [25] is employed. At the interface between the distributor and the filler regions, the scalars, *i.e.*, pressure and temperature, and the fluxes, *i.e.*, mass, momentum and energy, are all held continuous; the turbulent energy k_t and its rate of dissipation ε_t are set in the usual way for standard wall functions, *i.e.*, $\partial k_t / \partial n = 0$ and $\varepsilon_t = 0.09^{0.75} k_t^{1.5} / (0.42 \Delta l)$, where $\partial / \partial n$ is the gradient in the wall normal direction and Δl the distance from the wall to its adjacent cell center.

In the validation presented in the next section, it will be shown that the distributor regions do effectively render the flow uniform at the entrance and exit to the filler region. A thermocline thermal energy storage tank with distributors is thus thermally equivalent to one without distributors but with a uniform flow imposed at the inlet and outlet of the filler region.

In order to generalize the model and the corresponding results, Eqs. (1) – (4) are non-dimensionalized as follows.

Continuity equation:

$$\varepsilon \frac{\partial(\Phi_\rho)}{\partial \tau} + \tilde{\nabla} \cdot (\Phi_\rho \bar{U}) = 0 \quad (9)$$

Momentum equation:

$$\begin{aligned} \frac{\partial(\Phi_\rho \bar{U})}{\partial \tau} + Re \tilde{\nabla} \cdot \left(\frac{\Phi_\rho \bar{U} \bar{U}}{\varepsilon} \right) = & -\varepsilon \tilde{\nabla} P + \tilde{\nabla} \cdot \bar{\bar{T}} - \varepsilon \Phi_\rho Gr \bar{e}_x \\ & - \varepsilon \left(\frac{\Phi_\mu \bar{U}}{Da^2} + \frac{FRe}{Da} \Phi_\rho U_{mag} \bar{U} \right) \end{aligned} \quad (10)$$

Energy equation for the molten salt:

$$\begin{aligned} Pr \frac{\partial}{\partial \tau} (\varepsilon \Phi_\rho \Phi_{Cpl} \Theta_l) + Pr Re \tilde{\nabla} \cdot (\Phi_\rho \Phi_{Cpl} \Theta_l \bar{U}) \\ = \tilde{\nabla} \cdot (\Phi_{ke} \tilde{\nabla} \Theta_l) + Pr A \left[P \tilde{\nabla} \cdot \bar{U} + \text{tr}(\tilde{\nabla} \bar{U} \cdot \bar{\bar{T}} / \varepsilon) + \frac{UU}{2} \frac{\partial \Phi_\rho}{\partial \tau} \right] + \Phi_{kl} Nu_i (\Theta_s - \Theta_l) \end{aligned} \quad (11)$$

Energy equation for the filler phase:

$$Pr \frac{\partial}{\partial \tau} [(1 - \varepsilon) \Omega \Phi_{\rho s} \Phi_{Cps} \Theta_s] = -\Phi_{kl} Nu_i (\Theta_s - \Theta_l) \quad (12)$$

The non-dimensional parameters included in Eqs. (9) – (12) are defined as follows:

$$\tau = \frac{t v_c}{d_s^2}, \quad X = \frac{x}{d_s}, \quad R = \frac{r}{d_s}, \quad \bar{U} = \frac{\bar{u}}{u_m}, \quad H = \frac{h}{d_s}, \quad H' = \frac{h'}{d_s}, \quad D = \frac{d}{d_s}, \quad D' = \frac{d'}{d_s},$$

$$Re = \frac{u_m d_s}{\nu_c}, \quad P = \frac{p d_s}{\mu_c u_m}, \quad Gr = \frac{g d_s^2}{\nu_c u_m}, \quad Da = \frac{\sqrt{K}}{d_s}, \quad A = \frac{u_m^2}{C_{P,l,c} (T_h - T_c)}, \quad Nu_i = \frac{h_i d_s^2}{k_{l,c}},$$

$$Pr = \frac{\nu_c}{\alpha_c} = \frac{C_{P,l,c} \mu_c}{k_{l,c}}, \quad \Theta_l = \frac{T_l - T_c}{T_h - T_c}, \quad \Theta_s = \frac{T_s - T_c}{T_h - T_c}, \quad \Omega = \frac{\rho_{s,c} C_{P,s,c}}{\rho_{l,c} C_{P,l,c}}, \quad \bar{\bar{T}} = \frac{\bar{\bar{t}} d_s}{\mu u_m}$$

$$\tilde{\nabla} = \bar{e}_x \frac{\partial}{\partial X} + \frac{\bar{e}_\theta}{R} \frac{\partial}{\partial \theta} + \bar{e}_r \frac{\partial}{\partial R}.$$

Coefficients Φ_ρ , Φ_μ , Φ_{Cpl} , Φ_{kl} , Φ_{ke} , $\Phi_{\rho s}$ and Φ_{Cps} account for the temperature

dependence of the density, viscosity, specific heat, thermal conductivity, effective thermal conductivity of the molten salt, and the density and specific heat of the filler material, respectively. These coefficients can be expressed as follows, according to the data in Ref. [23]:

$$\Phi_{\rho} = 1 - \frac{0.732(T_h - T_c)}{2084.4 - 0.732T_c} \Theta_l$$

$$\Phi_{\mu} = \frac{\exp[-4.343 - 2.0143 \ln[(T_h - T_c)\Theta_l + T_c] + 10.094]}{\exp[-4.343 - 2.0143 \ln T_c + 10.094]}$$

$$\Phi_{kl} = \frac{-6.53 \times 10^{-4} [(T_h - T_c)\Theta_l + T_c] + 0.5908}{-6.53 \times 10^{-4} T_c + 0.5908}$$

$$\Phi_{ke} = \Phi_{kl} \frac{1 + 2\beta\phi + (2\beta^3 - 0.1\beta)\phi^2 + \phi^3 0.05 \exp(4.5\beta)}{1 - \beta\phi}$$

$$\Phi_{Cpl} = 1, \quad \Phi_{Cps} = 1, \quad \Phi_{\rho s} = 1$$

According to Wakao *et al.* [26], the interstitial Nusselt number for liquid flow through particle beds can be expressed by

$$Nu_i = 6(1 - \varepsilon) \left[2 + 1.1 Re_L^{0.6} Pr_L^{1/3} \right] \quad (13)$$

where Re_L and Pr_L are the local Reynolds and Prandtl numbers, respectively.

Assuming the distributor regions are properly designed so that uniform flows at the inlet and outlet of the filler region have been achieved, the appropriate boundary conditions are as follows.

At the inlet:

$$U_x = 1, \quad U_R = 0, \quad \Theta_l = 0 \quad (14)$$

and at the outlet:

$$\frac{\partial U_X}{\partial X} = \frac{\partial U_R}{\partial X} = \frac{\partial \Theta_l}{\partial X} = 0 \quad (15)$$

Equations (9) – (13) show that heat transfer and fluid flow in a thermocline storage tank is decided by Re and material properties, *i.e.*, Φ_ρ , Φ_μ , Φ_{kl} , Φ_{ks} , Φ_{ke} , Φ_{Cpl} , Φ_{Cps} and Pr . Once the HTF and the filler particles are selected and the material properties are determined, the characteristics of the thermal energy storage process are solely determined by Re .

The computational domain is discretized into finite volumes. All the variables are stored at the centers of the square mesh cells. A second-order upwind scheme is used for the convective fluxes, while a central-differencing scheme is used for discretizing the diffusion fluxes. A second-order implicit scheme is used for time discretization. Pressure-velocity coupling is implemented through the PISO algorithm [27]. Iterations at each time step are terminated when the dimensionless residuals for all equations drop below 10^{-4} . The computations are performed using the commercial software FLUENT [28]. User-defined functions (UDFs) are developed to account for Eqs. (11) and (12). Grid and time-step dependence are checked by inspecting results from different grid densities and time intervals. Based on this, $\Delta X = \Delta R = 0.01$ and $\Delta \tau = 1 \times 10^{-3}$ are chosen as this setting results in a temperature along the line $R = 0$ throughout the discharge process that is within 1% of that for the case with $\Delta X = \Delta R = 0.005$ and $\Delta \tau = 5 \times 10^{-4}$.

3 MODEL VALIDATION

The experimental results of Pacheco *et al.* [21] are used here to validate the

numerical model. A small pilot-scale, 2.3 MWh, thermocline system was designed and built for their experiments. The storage tank was filled with a mixture of quartzite rock and silica sand resulting in a porosity of 0.22. A nearly eutectic mixture of sodium nitrate and potassium nitrate was used as the HTF. The non-dimensional parameters for the experiments were as follows: $H = 67$, $H' = 1.1$, $D = 33$, $D' = 3.3$, $Re = 220$, $Pr = 13.4$, $Gr = 9.59 \times 10^7$, $Da = 0.01$, $A = 1.21 \times 10^{-12}$.

The numerical results for the axial temperature profiles are compared with the experimental ones in Fig. 2. Simulation 1 uses the same conditions as in the experiment. Property parameters Φ_ρ , Φ_μ , Φ_{kl} , Φ_{ke} , Φ_{Cpl} and Pr are taken as those of HITEC salt, since properties for the exact eutectic mixture (whose composition is close to that of HITEC) used in Ref. [21] were not provided. In order to understand the effectiveness of the flow distributor in rendering uniform flow to the filler region, another case (simulation 2) is considered; conditions for simulation 2 are identical to those for simulation 1, except that the molten-salt flow field is not solved for in the distributor region, and instead, the flow is set to be uniform at the entrance and exit of the filler region.

As shown in Fig. 2, the experimental results display some scatter in the temperature profiles. This may have been caused by the contact of some of the thermocouples with the rock while others may have been located squarely in the pore centers. Results from the model show that the non-dimensional temperature difference $\Delta\theta$ between the HTF and the filler particles takes values of as much as 0.1; this is consistent with the extent of scatter in the experimental measurements and the

deviation (~ 0.1) between the experimental and the numerical temperature profiles in Fig. 2. Within the experimental uncertainty, therefore, the results from the simulations are seen to agree well with the experiments. The flow distributor regions are seen to be quite effective from the results of simulation 1, since the axial temperature profiles at different times in simulation 1 are almost identical to those for the ideal uniform inlet and exit flow assumed in simulation 2. It is clear from these results that fluid flow and heat transfer in a thermocline thermal energy storage tank with well-designed distributor regions are equivalent to those under uniform inlet and outlet flow conditions.

4 RESULTS AND DISCUSSION

The validated numerical code discussed above is employed here to systematically investigate the discharge characteristics of a thermocline energy storage unit. Based on the numerical results, a procedure is proposed for designing thermocline thermal storage systems. In the results presented in this section, it is assumed that effective flow distributors have led to the establishment of uniform flow of molten salt in the filler region in all cases. As discussed above, this condition is readily achieved in practice.

4.1 Temperature profiles

Typical temperature histories of the molten salt and the filler material during a discharge cycle are shown in Fig. 3. The temperature profiles at any given time, *e.g.*,

$\tau = 1.5$, can be divided into three zones. In the constant, low-temperature zone near the inlet ($X = 0$) at the bottom of the storage tank, both Θ_l and Θ_s take values of zero. The next zone is characterized by significant temperature changes in both phases ($0 < \Theta_l < 1$ and $0 < \Theta_s < 1$). The final zone is the constant, high-temperature zone near the exit at the top of the storage tank, where both Θ_l and Θ_s take values of unity. In the first zone, the filler is completely cooled by the cold molten salt ($\Theta_l = 0$) pumped into the storage tank, and thermal equilibrium exists between the two phases. In the second zone (referred to hereafter as the heat-exchange zone), the temperature of the quartzite rock is higher than that of the molten salt, and thermal energy is transferred to the cold salt. In the third zone, the salt is fully heated up to $\Theta_l = 1$ and is once again in thermal equilibrium with the filler material.

As time progresses, the intermediate, heat-exchange zone advances from the inlet towards the outlet, leaving behind an expanding constant-low-temperature zone and causing the constant-high-temperature zone in front to shrink, as shown in Fig. 3. It is convenient to track the location X_m , where the molten salt is at temperature $\Theta_l = 0.5$, as being representative of the heat-exchange zone. Figure 4 illustrates the change in the position of the heat-exchange zone with time at different Reynolds numbers. All the results are seen to be well represented by a single linear fit passing through the origin, when plotted against the product of τ and Re . The slope of this line obtained by linear regression is 1.29.

The slope of the line in Fig. 4 can also be obtained via a simple energy balance on a control volume that covers the molten salt and filler in the entire heat-exchange

zone, as shown in Fig. 5. The moving coordinate system has its origin fixed at the location X_m , and travels with the control volume from the bottom to the top of the storage tank. In this moving coordinate system, the filler enters the control volume at the high temperature T_h and exits at the low temperature T_c ; the molten salt enters the control volume in the opposite direction as the filler at the low temperature T_c and exits at the high temperature T_h . The net thermal energy flux of the salt and the filler changes the total thermal energy in the control volume, according to

$$\Delta E = \rho_{l,c} u' A_c \varepsilon C_{p_{l,c}} (T_c - T_h) + (1 - \varepsilon) \rho_s v A_c C_{p_s} (T_h - T_c) \quad (16)$$

where ΔE is the thermal energy change in the control volume, A_c is the cross-section area of the storage tank, u' ($= u_m / \varepsilon - v$) is the relative speed of molten salt in the moving coordinate system, and v is the speed of the traveling coordinate system which is equal in magnitude to the relative speed of the filler. Although temperature profiles in the molten salt in the heat-exchange zone change with time, they are essentially symmetric about the mid-temperature point X_m , as will be discussed later. This indicates that the thermal energy of the salt in this zone changes little with time, which is also true for the thermal energy of the filler material in this zone. Therefore, ΔE is close to zero so that Eq. (16) becomes

$$\left(\frac{u_m}{\varepsilon} - v \right) A_c \varepsilon C_{p_{l,c}} \rho_{l,c} (T_h - T_c) = (1 - \varepsilon) v A_c C_{p_s} \rho_s (T_h - T_c) \quad (17)$$

which yields

$$v = \frac{C_{p_{l,c}} \rho_{l,c} u_m}{\varepsilon C_{p_{l,c}} \rho_{l,c} + (1 - \varepsilon) C_{p_s} \rho_s} \quad (18)$$

Since v is the speed of the traveling coordinate system with its origin fixed in the

heat-exchange zone, it is also the speed of advance of this zone, *i.e.*, $dx_m / dt = v$, where x_m is the dimensional form of X_m . The slope in Fig. 4 can then be rewritten as

$$\frac{dX_m}{d(\tau \cdot Re)} = \frac{dx_m}{d(tu_m)} = \frac{v}{u_m} = \frac{Cp_{l,c}\rho_{l,c}}{\varepsilon Cp_{l,c}\rho_{l,c} + (1-\varepsilon)Cp_s\rho_s} = 1.29 \quad (19)$$

This result depends only on the physical properties, and is identical to the slope obtained from the linear fit to the numerical data in Fig. 4. This validates the assumption made earlier that the total thermal energy in the heat-exchange zone is essentially invariant, *i.e.*, $\Delta E = 0$, during the discharge cycle. Both the analysis above and the data in Fig. 4 indicate that the heat-exchange zone advances at a constant speed from the inlet on the bottom to the outlet at the top in a thermocline thermal storage tank.

Figure 6 shows the development of the axial temperature profiles plotted in the moving coordinate system, with the horizontal axis being $(X - X_m)$. All the temperature profiles pass through the point $(0, 0.5)$ in Fig. 6, and appear to be symmetrical about this point. As the discharge process proceeds (and the position of the heat-exchange zone X_m increases in Fig. 6), the thermal energy decrease (temperature drop) in the region to the right of the point $(0, 0.5)$ is effectively compensated by an increase (temperature rise) in the region to left of the point, as shown by a comparison of the profiles at $X_m = 65$ and 322 at $Re = 1$ or 10 . This causes the thermal energy of the molten salt in the heat-exchange zone to be maintained at a near-constant level. This conclusion also holds for the filler phase. Therefore, the total thermal energy in the heat-exchange zone changes little, *i.e.*, $\Delta E \approx 0$ in Eq. (16), which supports the earlier assumption in the derivation of Eq. (17).

Increasing the Reynolds number results in an expansion of the heat-exchange zone. For instance, the temperature profiles at $Re = 10$ change more gradually in the heat-exchange zone than at $Re = 1$ when compared at the same position X_m , resulting in a wider heat-exchange zone, as shown in Fig. 6. At the higher Reynolds number, a longer flow distance is needed for the fluid to be heated by the filler phase, leading to a more gradual temperature rise and a corresponding increase in the extent of the heat-exchange zone. Since the molten salt in the heat-exchange zone is at a relatively lower temperature, an expanded heat-exchange zone can lead to significant waste of thermal energy if the salt delivered at lower temperatures is not useful for further application. This points to the important effects of Reynolds number on the design of a thermocline thermal energy storage system.

The effect of tank height is illustrated in Fig. 7 in terms of its effect on the temperature history of the molten salt. Prior to the heat-exchange zone reaching the tank outlet, salt at a constant high temperature level is available at the outlet, *i.e.*, $\Theta_l = 1$. As the heat-exchange zone arrives at the outlet, the salt temperature begins to drop, finally reaching the constant low temperature level ($\Theta_l = 0$) when the thermal energy stored in the filler particles has been completely depleted. Thermocline tanks with a larger height can effectively extend the discharge stage wherein the salt temperature is maintained at a high level. For instance, the salt temperature begins to drop at $\tau = 5$ when tank height H is 450, whereas this time is prolonged to twice the period ($\tau = 10$) with $H = 850$, as shown in Fig. 7. Since the quality of low-temperature salt is not acceptable for generating superheated steam in the

turbines, it is desired that most of the stored thermal energy be retrieved at a high-temperature level in order to meet the design conditions; this also helps to maintain higher thermal-to-electrical conversion efficiency of the turbine generator. It may be noted that only thermal energy with temperature above a certain level, *e.g.*, $\Theta_l > 0.95$ as chosen for this work and shown in Fig. 7, is usually considered as “useful” energy.

4.2 Discharge efficiency

It is of interest to quantify the amount of useful energy that a thermocline system can deliver during a discharge cycle. The discharge efficiency of a thermocline thermal energy storage system is defined in this work as follows

$$\eta = \frac{\text{Output energy with } \Theta_l > \Theta_0}{\text{Total energy initially stored in the thermocline tank}} \quad (20)$$

where Θ_0 is a threshold value determined by the application of interest. A value of 0.95 for Θ_0 is chosen in this work, implying that thermal energy delivered at temperatures greater than $(T_c + 0.95(T_h - T_c))$ qualifies as useful energy. If $T_h = 450^\circ\text{C}$ and $T_c = 250^\circ\text{C}$, for example, HITEC liquid delivered at temperatures above 440°C is considered useful in generating superheated steam for the steam turbine.

The efficiency defined by Eq. (20) varies depending on the construction and working conditions of the thermocline system. Figure 8 shows the discharge efficiency calculated for different Re for thermoclines of different heights H . It is clear that the efficiency increases with tank height H , and decreases with a rise in the Reynolds number. A tank with a larger height extends the constant high-temperature

discharge stage, as shown earlier in Fig. 7, thus increasing the fraction of initial stored energy that is recovered as high-temperature thermal energy. At a higher Reynolds number, the heat-exchange zone expands greatly, as shown in Fig. 6. For instance, the heat-exchange zone for $Re = 10$ at $X_m = 579$ extends over an $X - X_m$ of 200; for a particle diameter of 5 cm, this would imply a zone length of 10 m. Since the salt temperature in the heat-exchange zone is lower than the constant high-temperature level, an expanded heat-exchange zone reduces the amount of high-temperature molten salt delivered, and thus decreases the discharge efficiency η .

The numerical results for the efficiency in Fig. 8 are well represented by the following correlation:

$$\eta = 1 - 0.1807Re^{0.1801}(H/100)^m \quad (21)$$

$$\text{where } m = 0.00234Re^{-0.6151} + 0.00055Re - 0.485$$

This correlation can predict the numerical data within a maximum error of 1% for Reynolds number between 1 and 50 and H between 10 and 800, as shown by the solid-line predictions from the equation included in Fig. 8.

Three other important parameters which capture the performance of the thermocline system, *i.e.*, discharge power per unit cross-sectional area (P/A), useful thermal energy per unit cross-sectional area (Q/A), and total stored thermal energy per unit cross-sectional area (Q_t/A), may be defined as follows:

$$\frac{P}{A} = u_m \rho_{l,c} C_{p,l,c} (T_h - T_c) = k_{l,c} \frac{T_h - T_c}{d_s} \cdot RePr \quad (22)$$

$$\begin{aligned}\frac{Q}{A} &= [\varepsilon \rho_l C p_l + (1 - \varepsilon) \rho_s C p_s]_{T_h} (T_h - T_c) h \eta \\ &= [\varepsilon \rho_l C p_l + (1 - \varepsilon) \rho_s C p_s]_{T_h} (T_h - T_c) d_s \cdot H \eta\end{aligned}\quad (23)$$

$$\begin{aligned}\frac{Q_t}{A} &= [\varepsilon \rho_l C p_l + (1 - \varepsilon) \rho_s C p_s]_{T_h} (T_h - T_c) h \\ &= [\varepsilon \rho_l C p_l + (1 - \varepsilon) \rho_s C p_s]_{T_h} (T_h - T_c) d_s \cdot H\end{aligned}\quad (24)$$

where Q is the useful thermal energy delivered at a temperature above Θ_0 , and the thermal properties included in the square brackets subscripted by T_h are calculated at temperature T_h . The importance of parameters (P / A) and (Q / A) in designing a thermocline storage system will be demonstrated in the following.

From Eqs. (22) – (24), a non-dimensional discharge power may be defined as $RePr$, a non-dimensional useful energy as $H\eta$, and a non-dimensional total energy as H (equaling the non-dimensional height of the storage tank). Figure 9 plots the efficiency under conditions of different discharge power and total energy, which can serve as a guideline for the design of thermocline storage systems. The discharge efficiency is seen to increase with an increase in the total stored thermal energy and decrease with an increase in discharge power. If high discharge efficiency is desired, a thermocline storage unit should be designed to have a large height and operate at a low discharge power. However, this may not be practically feasible as the discharge power would need to be maintained above a certain value and the tank height would need to be limited for cost considerations as well as to reduce heat loss from the correspondingly higher tank surface area. The discharge power and the amount of

useful thermal energy would typically be determined by the application, leaving other parameters to be decided during the design of a storage unit. To facilitate the design under such conditions, Fig. 10 shows the total thermal energy H under conditions of different discharge power and useful thermal energy. From this figure, the total thermal energy (or tank height) required to meet the discharge power and thermal energy output demands may be determined. For instance, a thermocline storage unit with a non-dimensional discharge power of 600 and a non-dimensional useful thermal energy of 400 calls for a non-dimensional total thermal energy (or non-dimensional tank height) of 470.

4.3 Design procedure and examples

The analyses presented thus far are used in this section to develop procedures for designing a thermocline thermal energy storage system with HITEC as the molten salt and quartzite rock for the filler particles, although the procedure itself is generally applicable to other materials systems. It is assumed that discharge power P and useful thermal energy Q are predetermined by the application, and that the rock can be packed to a porosity of 0.22 in the filler region [21]. The recommended design procedure follows.

1. Choose tank diameter d and filler particle size d_s based on practical requirements.
2. Calculate the cross-sectional area of the storage tank $A = 0.25\pi d^2$, and then the discharge power per unit cross-sectional area (P / A) and total thermal

energy per unit cross-sectional area (Q/A).

3. Calculate the non-dimensional discharge power $RePr$ and useful thermal energy $H\eta$, using Eqs. (22) and (23), respectively.
4. Calculate Re from the value of $RePr$ and assuming $H = H\eta$.
5. Use the Re and H obtained in step 4 to calculate η from Eq. (21).
6. Obtain H by dividing $H\eta$ with the efficiency η obtained in step 5.
7. Repeat steps 5 and 6 until the difference between the newly obtained H and that in last iteration is smaller than 0.1%.
8. The final H is the required height for the thermocline storage tank; also obtained is the discharge efficiency η .
9. The dimensional height of the tank h is calculated as $h = d_s * H$.

Table I shows some examples of thermocline designs based on this procedure.

The storage tank is initially at 450°C, and cold HITEC at 250°C (T_c) is fed into the tank. The output HITEC is at a temperature level of 450°C (T_h) during the early discharge phase, and later drops in temperature as the thermal energy stored in the tank is depleted. Thermal energy delivered at a temperature exceeding 440°C ($\Theta_0 = 0.95$) is regarded as useful energy.

It is observed that an increase in discharge power P decreases the discharge efficiency η , as evident from a comparison of cases 1, 2, 3, and 4 in Table I with cases 5, 6, 7, and 8, respectively. This trend is due to expansion of the heat-exchange zone at larger powers (also larger Re values) as shown in Fig. 6, which reduces the amount of useful thermal energy delivered. For a specified amount of useful thermal energy

Q , choosing a larger tank diameter can effectively reduce the required tank height; however, this also decreases the discharge efficiency. This decrease in discharge efficiency is related to the importance of the relatively large extent of the tank occupied by the heat-exchange zone in a short storage tank. Increasing the useful thermal energy Q , with the discharge power, the tank diameter and the filler size being fixed, needs an increase in the tank height, as can be seen from a comparison of cases 1 - 8 with cases 9 - 16, respectively. The discharge efficiency is also increased due to an increase in the storage tank height (as shown in Fig. 8). The efficiency of a design with a small tank diameter can be increased by using a larger height. However, such a design can be more expensive in terms of materials cost, and also offers more surface area for heat loss to the environment. These practical considerations would further inform design trade-offs.

It is also noted that the filler particle size strongly affects the efficiency. The use of small-sized filler particles increases the efficiency greatly, as can be seen by comparing cases 1, 3, 5, 7 with cases 2, 4, 6, 8, respectively. The contact area between HITEC and quartzite rock is increased with smaller particles, which increases the heat exchange rate between HITEC and quartzite rock, leading to increased discharge efficiency.

5 CONCLUSIONS

A two-temperature model is developed for investigating energy discharge from a thermocline thermal energy storage system using molten salt as the heat transfer fluid

and inexpensive rock as the filler. Thermal characteristics, including temperature profiles and discharge efficiency of the storage tank, are systematically explored.

During discharge, the heat-exchange zone expands with time and Reynolds number, and its rate of travel is constant and can be precisely predicted by Eq. (18).

Discharge efficiency of the thermocline storage tank is well predicted by the correlation developed in Eq. (21) for Reynolds numbers in the range of 1 to 50 and nondimensional tank heights of 10 to 800. The efficiency increases with tank height and decreases as Reynolds number increases.

Procedures for designing thermocline storage tanks are proposed. The use of smaller filler particles can greatly increase the discharge efficiency. For instance, a thermocline storage unit (2 MW, 5 MWh and $d = 5$ m) with a filler particle size of 5 cm has a discharge efficiency that exceeds that with a particle size of 10 cm by 12.9%.

Nomenclature

C_P	specific heat, $\text{J}\cdot\text{kg}^{-1}\text{K}^{-1}$
d	diameter of thermocline tank, m
d'	diameter of connecting tube at the inlet and outlet of thermocline tank, m
d_s	diameter of filler particles, m
\bar{e}_r	unit vector in the r direction, –
\bar{e}_x	unit vector in the x direction, –
F	inertial coefficient, -

g	acceleration due to gravity, m/s^2
h	thermocline tank height, m
h'	distributor region height, m
h_i	interstitial heat transfer coefficient, $\text{W}\cdot\text{m}^{-3}\text{K}^{-1}$
K	permeability, m^2
k	thermal conductivity, $\text{W}\cdot\text{m}^{-1}\text{K}^{-1}$
p	pressure, Pa
T	temperature, K
t	time, s
\bar{u}	velocity vector, ms^{-1}
u_{mag}	velocity magnitude, ms^{-1}
u_m	mean velocity magnitude at the inlet of filler region, ms^{-1}

Greek

α	thermal diffusivity of molten salt, m^2s^{-1}
ε	porosity, -
μ	viscosity of molten salt, $\text{kgm}^{-1}\text{s}^{-1}$
ν	kinematic viscosity of molten salt, m^2s^{-1}
ρ	density, kgm^{-3}
$\bar{\tau}$	stress tensor, Nm^{-2}

Subscripts

c	at the inlet low temperature
h	at the outlet high temperature

<i>l</i>	molten salt phase
<i>s</i>	solid filler phase
<i>e</i>	effective

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FIGURE CAPTIONS

Figure 1 Schematic diagram of the thermocline thermal energy storage system under analysis and the axisymmetric coordinate system used.

Figure 2 Comparison between the numerical and experimental [21] axial temperature profiles during discharge of a thermocline thermal energy storage unit (2.3 MWh): Simulation 1 – with distributors, Simulation 2 – with uniform flow at the inlet and outlet of the filler region.

Figure 3 Axial temperature profiles at different times during a discharge cycle at $Re = 50$ and $H = 250$: Θ_l – molten salt temperature, Θ_s – filler temperature.

Figure 4 Representative location X_m in the heat-exchange zone for different Reynolds numbers at different times. The results are well represented by a single straight line passing through the origin.

Figure 5 Control volume fixed at the mean-temperature location of $\Theta_l = 0.5$.

Figure 6 Molten salt temperature profiles in the heat-exchange zone. Sharper changes in temperature profile occur at lower Reynolds number and small X_m .

Figure 7 Output temperature history of the molten salt with $Re = 50$ at different tank heights. A thermocline tank of a larger height exhibits a prolonged constant-high-temperature-discharge stage.

Figure 8 Discharge efficiency η of a thermocline at different H and Re . η increases with H and decreases as Re increases.

Figure 9 Discharge efficiency at different discharge powers ($RePr$) and total thermal energies H . A high discharge efficiency occurs at a low discharge power

and a high total thermal energy.

Figure 10 Total thermal energy H at different discharge power $RePr$ and useful thermal energy $H\eta$. The value of the useful thermal energy is always lower than that of the total thermal energy.

Tables

Table I. Results for various thermocline design examples.

Case No.	Q (MWh)	P (MW)	d (m)	d_s (m)	η (-)	h (m)
1	5	1	2	0.05	83.6%	15.2
2	5	1	2	0.1	75.4%	16.8
3	5	1	5	0.05	73.4%	2.77
4	5	1	5	0.1	61.4%	3.31
5	5	2	2	0.05	81.6%	15.6
6	5	2	2	0.1	72.4%	17.5
7	5	2	5	0.05	70.5%	2.88
8	5	2	5	0.1	57.6%	3.52
9	10	1	2	0.05	88.0%	28.8
10	10	1	2	0.1	81.6%	31.1
11	10	1	5	0.05	80.1%	5.07
12	10	1	5	0.1	70.5%	5.76
13	10	2	2	0.05	86.4%	29.4
14	10	2	2	0.1	79.1%	32.1
15	10	2	5	0.05	77.8%	5.22
16	10	2	5	0.1	67.3%	6.03