Elsevier Editorial System(tm) for Applied Energy
Manuscript Draft

Manuscript Number:

Title: INFLUENCE OF FLOW DISTRIBUTION ON THE THERMAL PERFORMANCE OF DUAL-MEDIA THERMOCLINE ENERGY STORAGE SYSTEMS

Article Type: Original Paper

Keywords: Thermal energy storage; thermocline; dual-media; flow distribution; entropy analysis

Corresponding Author: Professor Zhen Yang,

Corresponding Author's Institution:

First Author: Letian Wang

Order of Authors: Letian Wang; Zhen Yang; Yuanyuan Duan

Abstract: Dual-media molten-salt thermocline thermal energy storage (TES) systems can be used to maintain constant power production from Concentrated Solar Power (CSP) plants independent of weather changes at less cost than traditional two-tank molten-salt storage systems. The flow distribution is a critical parameter affecting the thermal performance but has rarely been considered for dual-media TES systems in previous studies. This study analyzes the influence of the flow distribution at the inlet and outlet of a salt-rock dual-media thermocline TES tank on the thermal performance. The flow distribution is characterized by the radial and azimuthal components with a two-temperature model used to investigate the thermal performance of the thermocline tank. The model is first validated against experiment data in the literature and then used to study the discharge process of the thermocline thermal storage tank for various flow distributions. The results show that even with a large (80% area) flow blockage at the inlet, the flow distribution has limited influence on the useable energy output (<3%) of the dual-media storage tank. In fact, the flow non-uniformity reduces the thickness of the thermocline layer and slightly increases the useable energy output while that at the top outlet slightly decreases the output. An entropy generation analysis, including the effects of the diffusion and interstitial heat transfer, is given to further explain these phenomena. The interstitial heat transfer is found the main cause for the entropy generation in discharge. Flow non-uniformity reduces the entropy generation.

Zhen Yang
Associate Professor
Department of Thermal Engineering
Tsinghua University
Beijing 100084, China
Email: zhenyang@tsinghua.edu.cn
Tel/Fax: +86 10 6278 9751

Tuesday, 22 October 2013

Prof. J. Yan
Editor, Applied Energy
KTH Royal Institute of Technology, Stockholm, and Mälardalen University, Västerås, Sweden

Dear Prof. Yan:

Enclosed please find our manuscript entitled "INFLUENCE OF FLOW DISTRIBUTION ON THE THERMAL PERFORMANCE OF DUAL-MEDIA THERMOCLINE ENERGY STORAGE SYSTEMS" by Letian Wang, Zhen Yang and Yuanyuan Duan, submitted for possible publication in *Applied Energy*. This paper contains original results, and has not been published or submitted for publication elsewhere.

According to your advices to our last submission, two major improvements have been made as follows.

1. The novelty/originality of the work has now been strengthened in the abstract, introduction and conclusions of the paper, which is also listed as follows.

Effects of flow distribution on the discharge performance are first studied for dual-media molten-salt thermocline thermal energy storage tanks. Different from single-media storage, non-uniform flow does not reduce but slightly increases the useable energy output in dual-media storage. Non-uniform flow enhances heat transfer in the tank, thins the thermocline layer and reduces the entropy generation in discharge.

2. The language has been revised by an English native speaker.

We appreciated your kind advices on our work. Please feel free to contact me if I can provide any additional information. Thank you very much for your consideration of our manuscript.

Sincerely yours,

Zhen Yang
Associate Professor
Department of Thermal Engineering, Tsinghua University
Beijing 100084, China

Highlights

1. Effects of flow distribution are studied for molten-salt thermocline TES tanks
2. Non-uniform flow slightly increases the useable energy output in discharge
3. Non-uniform flow enhances heat transfer and thins the thermocline layer
4. The interstitial heat transfer causes most of the entropy generation in discharge
5. Non-uniform flow reduces the entropy generation in discharge.

INFLUENCE OF FLOW DISTRIBUTION ON THE THERMAL PERFORMANCE OF DUAL-MEDIA THERMOCLINE ENERGY STORAGE SYSTEMS

Letian Wang, Zhen Yang * and Yuanyuan Duan **

Key Laboratory for Thermal Science and Power Engineering of Ministry of Education

Beijing Key Laboratory for CO_2 Utilization and Reduction Technology,

Department of Thermal Engineering, Tsinghua University

Beijing 100084 China

Corresponding Authors:

* Zhen Yang: +86 10 6278 9751 (Tel/Fax), zhenyang@tsinghua.edu.cn (Email)

** Yuanyuan Duan: +86 10 6279 6381 (Tel/Fax), yyduan@tsinghua.edu.cn (Email)

Abstract

Dual-media molten-salt thermocline thermal energy storage (TES) systems can be used to maintain constant power production from Concentrated Solar Power (CSP) plants independent of weather changes at less cost than traditional two-tank molten-salt storage systems. The flow distribution is a critical parameter affecting the single-media thermocline thermal performance but has rarely been considered for dual-media TES systems in previous studies. This study analyzes the influence of the flow distribution at the inlet and outlet of a salt-rock dual-media thermocline TES tank on the thermal performance. The flow distribution is characterized by the radial and azimuthal components with a two-temperature model used to investigate the thermal performance of the thermocline tank. The model is first validated against experiment data in the literature and then used to study the discharge process of the thermocline thermal storage tank for various flow distributions. The results show that even with a large (80% area) flow blockage at the inlet, the flow distribution has limited influence on the useable energy output (<3%) of the dual-media storage tank. In fact, the flow non-uniformity reduces the thickness of the thermocline layer and slightly increases the useable energy output while that at the top outlet slightly decreases the output. An entropy generation analysis, including the effects of the diffusion and interstitial heat transfer, is given to further explain these phenomena. The interstitial heat transfer is found the main cause for the entropy generation in discharge. Flow non-uniformity reduces the entropy generation.

Keywords

Thermal energy storage, thermocline, dual-media, flow distribution, entropy analysis

Nomenclature

C_p	specific heat [J/kg-K]
d	thermocline tank diameter [m]
d_s	filler particle diameter [m]
F	inertial coefficient, $F = \frac{1.75}{\sqrt{150\varepsilon^3}}$
g	gravity [m/s^2]
h	thermocline tank height [m]
h_{in}	interstitial heat transfer coefficient [W/m$^3\cdot$ K]
h_{conv}	wall heat transfer coefficient [W/m$^2\cdot$ K]
K	permeability, $K = \frac{d_s{}^2\varepsilon^3}{175(1-\varepsilon)^2}$ [m^2]
k	thermal conductivity [W/m-K]
P_{CSP}	concentrated solar power plant power [W]
Q_v	volumetric flow rate of molten-salt [m3/s]
η_p	thermal to electricity conversion efficiency
A_f	flow area at tank boundary
p	pressure [Pa]
T	temperature [K]
t	time[s]
x	tank axial coordinate [m]
y	tank radial coordinate [m]
$\mathbf{u}$	velocity vector [m/s]

Δh thermocline thickness

Greek

ε porosity [-]

η efficiency [-]

Θ non-dimensional temperature, $\Theta = \dfrac{T - T_{\mathrm{C}}}{T_{\mathrm{H}} - T_{\mathrm{C}}}$ [-]

μ viscosity [Pa-s]

ν kinematic viscosity [m^2/s]

ρ density [kg/m^3]

Non-dimensional variables

Re Reynolds number $\mathrm{Re} = \dfrac{u_m d_s}{\nu_C}$

Nu_i Interstitial Nusselt number $Nu_i = \dfrac{h_i d_s^{\ 2}}{k_{l,C}}$,

Pr Prandtl number $\mathrm{Pr} = \dfrac{\mu C_{p,l}}{k_l}$,

Nu_W Wall Nusselt number $Nu_W = \dfrac{h_i D}{\lambda_{\mathrm{air}}}$

Subscripts

C cold end of the tank

H hot end of the tank

l	molten salt phase
s	solid filler phase
I	case number
b	tank bottom
t	tank top

1 Introduction

Molten salts are viable candidates for high-temperature (>400°C) heat transfer fluids (HTFs) in Concentrated Solar Power (CSP) plants due to their lower costs and operating pressures relative to current high-temperature oils. Molten salts have stable heat transfer properties and their heat transfer characteristics can be well predicted by experimentally verified correlations [1-3]. One major disadvantage of molten salts is their relatively high solidification temperatures, which causes difficulties when filling the salts into the heat transfer loops. As a hot molten salt flows into a cold tube, the salt first cools in the region close to the tube wall, then solidifies and is later re-melted by fresh hot salt [4]. The air-salt interface first quickly increases, then fluctuates and finally becomes stable [5]. The salt solidification greatly increases the flow pressure loss and extends the filling time [6]. Preheating of the tubes may be necessary to eliminate the risk of tube blockage when filling tubes with molten salts.

Molten-salt thermal energy storage (TES) systems have been widely used in Concentrated Solar Power (CSP) plants [7] to produce electricity independent of the weather conditions. Of the various TES technologies, the dual-media molten-salt thermocline TES has been recognized as a promising approach due to its relatively higher energy efficiency and lower cost [8]. A dual-media molten-salt thermocline TES system is primarily a tank filled with a filler material (rock and sand) as the main storage medium with a molten-salt HTF in the pores between the filler particles.

The flow and heat transfer characteristics has been carefully studied to have a better understanding of the thermal performance of dual-media molten-salt thermocline TES systems. Yang and Garimella [9] developed a two-temperature numerical model of a thermocline TES system with the heat transfer between the salt and the filler particles represented by an interstitial Nusselt number. They also studied the effects of the thermal conditions at the tank wall and found that a non-adiabatic wall only slightly reduced the efficiency for wall Nusselt numbers smaller than 10^4 [10]. Later, Flueckiger *et al*. [11] investigated the thermal and mechanical performance of a discharging thermocline tank; and the mechanical stress caused by thermal ratcheting was found to be well predicted by a simplified one-dimensional model using Hook's Law. Xu *et al.* [12] presented a parametric study of a molten salt thermocline TES system showing that the mean inlet velocity had negligible effect on the discharge efficiency and thermocline thickness, with only a slight increase in the efficiency, and suggested the reason to be a trade-off between the two counteracting effects of the increasing velocity which increased the thermocline thickness due to less heat transfer time between the solid and liquid, and less time for conduction within the particles which reduced the thermocline thickness. Yang and Garimella [13] studied the cyclic operation of molten-salt thermocline tanks and found that for a specific power cycle, an increase in the tank height increased the thermal efficiency. Recently, Flueckiger and Garimella [14] reviewed the numerical studies of thermocline storage systems.

All these studies used a uniform flow velocity in the thermocline tank with no consideration of the flow distribution effects. Consequently, the flow distribution effect is not clearly understood. However, flow distribution is a concern in practical TES systems. For instance, the flow enters a storage tank through a distribution system as shown in Fig. 1 with multiple manifolds. The flow rates in these manifolds may differ due to different flow distances and pressure losses, resulting in non-uniform flow across the diffuser. In addition, the manifolds may leak or be blocked during operation, leading to flow rate variations across the diffuser.

Flow distribution has been found to be important in water (single-medium) thermal storage systems [17-22]. A recent experimental study [23] on hot water solar energy storage validated that the inlet flow distribution could lead to a change of 40% in the effective discharge efficiency. Therefore, flow distribution may also be important to dual-media thermocline TES systems and needs to be carefully considered. Till now, few studies have been found on the influence of the flow distribution on the thermal performance of dual-media thermocline TES systems.

This study aims to investigate the influence of the flow distributions at the bottom and top of a thermocline TES tank on the thermal performance. The flow distribution is characterized by radial and azimuthal components with the influence of each component analyzed in this study. A two-temperature model is used to investigate the effects of the flow distribution on the discharge process of the thermocline tank. An entropy generation analysis is given to include the effects of the diffusion and interstitial heat transfer on the energy degradation in the tank.

2 Numerical model

2.1 Problem description

A schematic illustration of a TES thermocline tank is shown in Fig. 2. The tank is packed with quartzite rock with a molten salt filling the pores between the rock particles. Initially, the rock and the molten salt are at 450 ºC[9] to provide thermal energy in the tank for later power generation. The thermal energy is discharged from the tank by pumping cold molten salt into the tank through the bottom. As the cold salt flows upward through the tank, it is heated by the hot rock to high temperatures suitable for high-efficiency electricity generation when released from the top of the tank. The hot molten salt is then used to generate steam in a Rankine cycle. This study focus on the discharge process of the thermocline TES with non-uniform flow.

The thermocline tank has an inner diameter of *d* and a height of *h*, and is packed with quartzite rock as the filler. In this study, *d* is set to 36 m and *h* to 12 m, which are similar with the tank (*d*=37.2 m, *h*=10.4 m) being built in the Solana Generating Station, USA, as well as the one operating in the Andasol Solar Power Station, Spain (*d*=36 m, *h*=14 m) [25-27]. A feasibility study for the Barstow area determined that the tank height should be limited by the ground conditions in that region to 11.9 m [28]. Thus, the tank height used in this study is set to 12 m as representative of most designs with the tank wall assumed to be thermally adiabatic as in previous studies [10, 12].

The molten salt chosen here is HITEC [29]. The lower temperature, T_L, of the molten salt during operation is set to 200℃ which is greater than the HITEC salt melting point (142℃) with the high temperature, T_H, set to 450℃ to give a high Rankine cycle efficiency (>40%). The physical properties of HITEC are calculated using the following correlations [9]:

$$\rho_l = 1938.0 - 0.732(T_l - 200.0) \quad (1)$$

$$\mu = \exp\left[-4.343 - 2.0143(\ln T_l - 5.011)\right] \quad (2)$$

$$k_l = -6.53 \times 10^{-4}(T_l - 260.0) + 0.421 \quad (3)$$

The specific heat of HITEC is 1561.7 $J \cdot kg^{-1}K^{-1}$. Quartzite rock is used as the filler with a density of 2201 $kg \cdot m^{-3}$ and specific heat of 964 $J \cdot kg^{-1}\ K^{-1}$. The porosity, ε, of the packed rock bed is 0.22 and the average rock particle size is 0.05 m [10].

2.2 Governing equations

The continuity and momentum equations for laminar flow of the molten salt in the tank are:

$$\frac{\partial(\varepsilon\rho_l)}{\partial t} + \nabla \cdot (\rho_l \mathbf{u}) = 0 \quad (4)$$

$$\frac{\partial(\rho_l \mathbf{u})}{\partial t} + \nabla \cdot \left(\rho_l \frac{\mathbf{uu}}{\varepsilon}\right) = -\varepsilon\nabla p + \nabla \cdot \tilde{\boldsymbol{\tau}} + \varepsilon\rho_l \mathbf{g} + \varepsilon\left(\frac{\mu}{K}\mathbf{u} + \frac{F}{\sqrt{K}}\rho_l |\mathbf{u}|\mathbf{u}\right) \quad (5)$$

Since the molten salt and the rock bed may not be in thermal equilibrium due to their different thermal properties, heat is transferred between them with separate energy equations used for each phase [9]. For the molten salt, the energy equation is:

$$\frac{\partial\left[\varepsilon\rho_l C_{P,l}(T_l - T_c)\right]}{\partial t} + \nabla \cdot \left[\rho_l \mathbf{u} C_{P,l}(T_l - T_c)\right] = \nabla \cdot (k_e \nabla T_l) + h_i(T_s - T_l) \quad (6)$$

For the rock bed, the energy equation is:

$$\frac{\partial\left[(1-\varepsilon)\rho_s C_{P,s}\left(T_s - T_c\right)\right]}{\partial t} = -h_i\left(T_s - T_l\right) \tag{7}$$

The last terms on the right sides of the two equations are for the interstitial heat transfer between the two phases. The viscous dissipation term in the molten salt energy equation is neglected due to its negligible influence as indicated by small non-dimensional number *N* defined by Nield [30].

$$N = Ec \cdot \mathrm{Pr}/Da \tag{8}$$

N defines the importance of viscous dissipation, μΦ, with respect to the fluid energy transport and entropy generation. For the current geometry and molten salt inlet velocity, *N* is on the order of 10^{-21} [31]; thus, viscous heating is negligible.

The effective thermal conductivity, k_e, of the molten salt embedded in the rock bed is given by [20]:

$$k_e = k_l \frac{1 + 2\beta\phi + \left(2\beta^3 - 0.1\beta\right)\phi^2 + \phi^3 0.05\exp\left(4.5\beta\right)}{1 - \beta\phi} \tag{9}$$

where $\phi = 1 - \varepsilon$ and $\beta = \left(k_s - k_l\right)/\left(k_s + 2k_l\right)$.

According to Wakao and Kaguei [32], the interstitial Nusselt number for heat transfer between the two phases is:

$$\mathrm{Nu}_i = 6(1-\varepsilon)\left[2 + 1.1\mathrm{Re}_l^{0.6}\,\mathrm{Pr}_l^{1/3}\right] \tag{10}$$

where Re_l and Pr_l are the local Reynolds and Prandtl numbers of the molten salt. Then, the interstitial heat transfer coefficient h_i in Eqs. (6) and (7) is given by

$$h_i = Nu_i k_l / d_s^2 \tag{11}$$

The thermal energy stored in a thermocline TES tank decreases during the discharge process, eventually resulting in a decrease of the molten salt exit temperature. An entropy analysis is needed to investigate this phenomenon. The governing equations for the entropy in the molten salt and the rock were given by Flueckiger and Garimella [31].

$$\frac{\partial\left[\varepsilon\rho_l C_{P,l}\ln\left(T_l\right)\right]}{\partial t}+\nabla\cdot\left[\rho_l\mathbf{u}C_{P,l}\ln\left(T_l\right)\right]=-\nabla\cdot\left(\frac{\mathbf{q}}{T_l}\right)+\frac{\dot{q}'''}{T_l}+\dot{S}'''_{gen,l} \qquad (12)$$

$$\frac{\partial\left[\varepsilon\rho_s C_{P,s}\ln\left(T_s\right)\right]}{\partial t}=\frac{\dot{q}'''}{T_s}+\dot{S}'''_{gen,s} \qquad (13)$$

These equations include two source terms, $\dot{S}'''_{gen,l}$ and $\dot{S}'''_{gen,s}$, for the entropy generation arising from the heat dissipation in each phase, but ignore that caused by the interfacial heat transfer between the two phases, which leads to underestimates of the entropy generation. In this paper, the entropy generation due to the interfacial heat transfer is also included as.

$$\dot{S}'''_{gen,\mathrm{int}er}=\frac{\dot{q}'''}{T_l}-\frac{\dot{q}'''}{T_s} \qquad (14)$$

As indicated by Flueckiger and Garimella [12], the sum, $\dot{S}'''_{gen,l+s}$, of the two source terms in Eqs. (12) and (13), *i.e.*, $\dot{S}'''_{gen,l}$ and $\dot{S}'''_{gen,s}$, is

$$\dot{S}'''_{gen,l+s}=\frac{k_{eff}\left(\nabla T\right)^2}{T_l^2}+\frac{\mu\Phi}{T_l}\geq 0 \qquad (15)$$

Since $\dot{S}'''_{gen,l+s}$ comes from the thermal diffusion in the molten salt and the rock bed, it is termed the *diffusion entropy generation*. The total entropy generation, $\dot{S}'''_{gen}$, including the diffusion entropy generation, $\dot{S}'''_{gen,l+s}$, and the interstitial heat transfer entropy generation, $\dot{S}'''_{gen,inter}$, is then calculated as

$$\dot{S}'''_{gen} = \frac{k_{eff}(\nabla T)^2}{T_l^2} + \frac{\mu\Phi}{T_l} + (\frac{\dot{q}'''}{T_l} - \frac{\dot{q}'''}{T_s}) \geq 0 \tag{16}$$

Then, the corresponding exergy loss is readily calculated as

$$\dot{X}'''_{dest} = T_0 \cdot \dot{S}'''_{gen} \tag{17}$$

where T_0 is the temperature of the reference environment.

2.3 Boundary conditions

The boundary conditions at the bottom and top of the tank are as follows.

At the bottom, the molten salt flows into the tank:

$$u_x = \frac{Q_V}{A_{f,b}}, u_y = 0, T_l = T_C \text{, for the flow area } A_{f,b} \tag{18}$$

$$u_x = u_y = \frac{\partial T_l}{\partial x} = \frac{\partial T_s}{\partial x} = 0 \text{, for the remaining no-flow areas} \tag{19}$$

where Q_v is the volumetric flow rate of the molten salt and $A_{f,b}$ is the flow area at the bottom. At the outlet, the salt flows out of the tank with the boundary conditions:

$$\frac{\partial u_x}{\partial x} = \frac{\partial u_y}{\partial x} = \frac{\partial T_l}{\partial x} = \frac{\partial T_s}{\partial x} = 0 \text{, for the flow area } A_{f,t} \tag{20}$$

$$u_x = u_y = \frac{\partial T_l}{\partial x} = \frac{\partial T_s}{\partial x} = 0 \text{, for the remaining no-flow areas} \tag{21}$$

where $A_{f,t}$ is the flow area at the top.

The volume flow rate, Q_v, in Eq. (18) is given by

$$Q_V = \frac{P_{CSP}}{\eta_P \rho_{L,C} C_{P,m} \left(T_H - T_C\right)} \tag{22}$$

where P_{CSP} is the electrical power of the plant with the thermocline TES tank, η_P is the thermal to electrical conversion efficiency, $\rho_{L,C}$ is the molten salt density at the low temperature, $C_{p,m}$ is the molten salt specific heat, T_H is the high temperature (450 ºC) and T_C is the low temperature (200 ºC).

As in Ref. [10], the tank wall is assumed to be well insulated with no flow slipping, *i.e.*,

$$u_x = u_y = \frac{\partial T_l}{\partial x} = \frac{\partial T_s}{\partial x} = 0 \quad (23)$$

2.4 Flow non-uniformity

The molten salt flow in a thermocline TES tank is affected by the flow distributions at the bottom and top of the tank. Since the bottom and top are both circular planes, the flow distribution can be divided into two components, in radial and in azimuthal directions. A flow annulus is used to indicate the radial distribution with flow through the flow annulus at the bottom and top of the tank and no flow in the remaining area. The flow annulus geometry is described by parameters *P* and *D*, where *P* denotes the percentage of the annular area relative to the total bottom/top area and *D* is the inner radius of the annulus indicating the distance towards center. The flow annulus used here represents the radial flow distribution that results from the diffuser configuration. Both the radial disk diffuser and the octagonal tube diffuser can be regarded as consisting of multiple annuli as shown in Fig. 3. If some of the annuli are blocked, the flow will be non-uniformed in the radial direction. The annulus has different locations as *D* changes as shown in Figs. 4a-c and reduces to a circle when *D* is zero. The azimuthal non-uniform flow treated in this paper is represented by a non-concentric circle, as shown in Fig. 4d. Table 1 shows the geometric details and operation parameters of thermocline TES tanks with various flow distributions. A total of 8 cases are investigated to show the influence of the flow distribution on the thermal performance of thermocline TES tanks.

2.5 Solution

Equations (4)-(7) were solved using the double precision solver of the commercial CFD software FLUENT 12.0.16 [33]. The axisymmetric flow cases with radial flow uniformity were solved using the 2D axisymmetric solver while cases with azimuthal non-uniformed flow were solved using the 3D solver due to the non-axisymmetric flow. The thermocline TES tank was discretized into 37340 cells (2D) and 112450 cells (3D) with the finite volume method used to solve Eqs. (4)-(7). The convections terms were discretized with the second-order [12] upwind method. The transient terms were discretized with a first-order implicit formulation [9]. The PISO algorithm was used for the pressure-momentum coupling [33]. An adaptive time step was used to capture the different stages of the flow into the tank. The time step, Δt, was initially set to 0.01 s and then gradually increased to 4 s as the thermocline formed and kept at 4 s until the end of the calculation. The time independence of the results was checked using variant time steps increasing from 0.001 s to 1 s, with deviations in the outflow temperature of less than 0.1%. The grid independence was checked by comparing the results to those with larger meshes of 73300 cells (2D) and 163730 cells (3D) with differences of less than 0.6%. The calculation was regarded as converged when all the non-dimensional residuals dropped below $10^{-3.}$