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Simulation of thermocline storage for solar thermal power plants: From dimensionless results to prototypes and real-size tanks

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ABSTRACT

A single-phase one-dimensional model called CIEMAT1D1SF has been developed for characterizing the behaviour of thermocline tanks with an effective storage medium formed by either a liquid or a liquid and a packed-bed. Despite its simplicity, this model has been validated with experimental data and the results of tank performance are similar to those obtained by other authors using more complex simulation models. In order to obtain general results the thermal equation has been nondimensionalized and the resulting expression only depends on the parameter called dimensionless velocity, υ^*. It has been observed that thermocline thickness decreases as υ^* increases attaining a minimum value when $\upsilon^* \geqslant 2350$ while tank efficiency increases with υ^* up to a maximum of about 87% also for $\upsilon^* \geqslant 2350$. From these results the design equation for building thermocline storage tanks with maximum theoretical efficiency has been established. Since this design equation depends on tank dimensions and thermal power, small thermocline tanks and hence prototypes are not expected to behave in the same way as large or real-size tanks. Therefore maximum efficiency guideline plots for thermocline tanks with different storage media have been presented for various temperature intervals. In these plots thermal power has proven to be the critical design parameter because the larger the power the higher the degree of freedom for choosing tank dimensions and hence storage capacity and charging/discharging time. Therefore, we strongly recommend the use of these guideline plots in the design process of thermocline prototypes.

1. Introduction

Thermal storage in Solar Thermal Power Plants (STPP's) makes it possible to overcome transients and extend the operation time in order to deliver electricity when there is no solar irradiation as well as to meet peak demand independently of weather fluctuations. Up to now the storage option for extending operation time implemented in commercial STPP's is the molten salt two-tank system [1,2], whose estimated cost is about 30–50 US$ per thermal kW h [3]. Some studies have shown that these thermal storage systems have quite a significant cost reduction potential which could be between 38% and 69% by 2020 [4]. In this way, since the cost of molten salt two-tank storage systems is dominated by the tanks (30%) and the molten salt inventory (44%), one of the concepts that nowadays is being seriously considered is the molten salt based thermocline single tank with a packed-bed of low-cost solid filler. Actually the Electric Power Research Institute (EPRI) recently conducted a study where the estimated engineering, procurement and construction (EPC) costs were compared for both two-tank and packed-bed thermocline storage cases [5]. This study concluded that the thermocline option would have a potential to reduce around 33% the cost of thermal energy storage, which could not only greatly increase the utility of STPP's but also lead to a wider adoption of this technology around the world [6]. Therefore it seems that the next step for the thermocline systems is the construction of either pilot plants or small commercial unities that can demonstrate the operation of this kind of storage out of laboratory. Additionally, standardized design and modelling procedures will be required for validating the experimental results of field tests and hence for improving the design process and speed up both production and deployment of thermocline storage systems.

The majority of models previously developed for simulating thermocline storage tanks for STPP's consider packed-bed systems and are based on Schumann's one-dimensional model [7]. This model includes two heat transfer equations because it assumes that fluid and packed-bed particles have different temperatures. However, it neglects heat conduction in the fluid, heat exchange between the packed-bed particles and also thermal losses to the environment. Pacheco et al. [8] used Schumann model for simulating a thermocline tank containing molten solar salt as fluid and quartzite rock and sand as solid filler. Some years later Kolb et al. [9] improved Pacheco's model allowing thermal conduction

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between control volumes and including thermal losses at top, bottom and tank walls. This model was implemented in TRNSYS® as Type 502 in the STEC library in order to simulate the whole performance of a STPP with a thermocline storage system [10]. Other authors proposed similar models and the main difference between them was either the procedure for solving the governing equations or the software applied. In this way both Bharathan et al. [11] and Yang et al. [12] used the commercial CFD software FLUENT® whereas Xu et al. [13] solved the heat transfer differential equations with a self written simulation code and Van Lew et al. [14] applied the numerical method of characteristics. For the particular case of liquid fluids, some of these models have shown there is little difference between fluid and solid filler temperatures since heat transfer between them is very efficient [12–14]. In such case the same temperature can be assumed for both liquid and filler particles and hence a single-phase model can be formulated for which only one heat transfer equation has to be solved. In a previous work [15], we already presented a single-phase one-dimensional model (called CIEMAT1D1SF) for simulating thermocline storage tanks with an effective storage medium formed by either a liquid or both a liquid and a packed-bed of solid filler.

In this work we have neglected thermal losses and the heat transfer equation has been expressed in dimensionless coordinates for simplifying the solving process and obtaining general results in terms of performance parameters. This new model has been successfully validated with various experimental data found in the literature for different kinds of thermocline storage tanks [8,16,17].

Simulations with CIEMAT1D1SF model have demonstrated that the performance of a thermocline tank strongly depends on design parameters like tank height and liquid velocity. Therefore small prototype tanks are not expected to behave in the same way as scaled-up thermocline tank, which means that a similarity analysis cannot be directly applied. In this way this paper presents the guidelines for building thermocline prototype tanks whose performance is representative of real size tanks.

2. Model description

2.1. Development of CIEMAT1D1SF model

The model CIEMAT1D1SF considers a single effective storage medium inside a thermocline tank (liquid or liquid plus solid filler) at a certain temperature, *T*, which varies with time, *t*, (unsteady), along the tank height, *z* (one-dimensional). This model also takes into account thermal losses to the environment and considers average values of thermophysical properties that are independent on temperature. The energy balance equation describing this storage system is:

$$(\rho C_p)_{eff}\frac{\partial T}{\partial t}+\varepsilon(\rho C_p)_{liquid}v_{liquid}\frac{\partial T}{\partial z}=k_{eff}\frac{\partial^2 T}{\partial z^2}-U_w a_w(T-T_\infty) \tag{1}$$

ρ	density (kg/m^3)
C_p	heat capacity (J/kg K)
ρC_p	volumetric heat capacity (J/m^3 K)
k	thermal conductivity (W/m K)
v_{liquid}	velocity of the liquid (m/s)
U_w	coefficient of thermal losses to the environment (W/m^2 K)
a_w	ratio between thermal losses area and tank volume (1/m) [$a_w = 4/D$]
D	tank diameter (m)
T_∞	ambient temperature (°C or K)
ε	porosity of the storage medium (dimensionless)
eff	sub index that refers to the effective storage medium
liquid	sub index that refers to the liquid

The porosity of a packed-bed, ε, is defined as the fraction of total volume that remains free for fluid circulation and its value is imposed by the kind of packing and the relative size of solid particles. From the theoretical point of view, ordered structures of identical spherical particles include rhombohedral and cubic packing, which place the range of attainable porosities between 0.259 and 0.476. Alternatively, disordered packings exhibit a smaller porosity range with most of them falling into the range 0.36–0.40 [18]. The porosity of mixed-size particle beds depends on the volume fraction of the large particles and the relative size of small and large particles. From the theoretical point of view [19], the minimal porosity value in this case could be as low as 0.16 or even 0.14. However, for a real packed-bed formed by non spherical particles the porosity must be experimentally obtained by measuring the free volume between particles for example with the help of a liquid.

In CIEMAT1D1SF model, the porosity is also used for calculating the effective volumetric heat capacity of the storage medium through the equation:

$$(\rho C_p)_{eff}=\varepsilon(\rho C_p)_{liquid}+(1-\varepsilon)(\rho C_p)_{solid} \tag{2}$$

where *solid* refers to the packed bed. For calculating the effective thermal conductivity different equations can be found in the literature [20]. In our simulations we have chosen, as a first approach, the simplest formulation, which also depends on packed-bed porosity:

$$k_{eff}=\varepsilon k_{liquid}+(1-\varepsilon)k_{solid} \tag{3}$$

When the tank contains only liquid, porosity equals 1 and hence $(\rho C_p)_{eff}=(\rho C_p)_{liquid}$ and $k_{eff}=k_{liquid}$.

During charge and discharge processes, thermal losses term can be neglected because the main contribution to temperature variation with time and position is the movement of thermocline zone and hence (Eq. (1)) becomes:

$$(\rho C_p)_{eff}\frac{\partial T}{\partial t}+\varepsilon(\rho C_p)_{liquid}v_{liquid}\frac{\partial T}{\partial z}=k_{eff}\frac{\partial^2 T}{\partial z^2} \tag{4}$$

This equation indicates that the results obtained will be independent on tank diameter, *D*, which means that the model is really one-dimensional and depends only on *z* coordinate. In order to simplify the solving process of (Eq. (4)), all variables have been expressed in dimensionless form by means of the following transformations:

$$\phi=\frac{T}{T_{\max}} \tag{5}$$

$$z^*=\frac{z}{L} \tag{6}$$

$$t^*=\frac{t\alpha_{eff}}{L^2} \tag{7}$$

$$v^*=\frac{\varepsilon(\rho C_p)_{liquid}L v_{liquid}}{k_{eff}}=\frac{(\rho C_p)_{liquid}L v_m}{k_{eff}}=\frac{v_{TC}L}{\alpha_{eff}} \tag{8}$$

where T_{max} is the maximum temperature of the tank (inlet temperature in charge or outlet temperature in discharge), *L* is the total tank height, α_{eff} is the effective thermal diffusivity, v_m is the velocity of the liquid at tank inlet/outlet and v_{TC} is the velocity at which thermocline zone moves. This velocity is constant and has been already introduced by other authors [12]. The expressions for calculating α_{eff}, v_m and v_{TC} are:

$$\alpha_{eff}=\frac{k_{eff}}{(\rho C_p)_{eff}} \tag{9}$$

$$v_m=\varepsilon v_{liquid} \tag{10}$$

$$v_{TC}=\frac{(\rho C_p)_{liquid}\varepsilon v_{liquid}}{(\rho C_p)_{eff}}=\frac{(\rho C_p)_{liquid}v_m}{(\rho C_p)_{eff}} \tag{11}$$

The resulting energy balance equation in dimensionless coordinates is: