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Analysis of Packed-Bed Thermocline Storage Tank Performance by Means of a New Analytical Function

Rocío Bayón[1, a)] and Esther Rojas[1, b)]

[1]*CSSU-Thermal Storage, CIEMAT-PSA, Av. Complutense 40, 28040 Madrid (Spain)*

[a)]Corresponding author: rocio.bayon@ciemat.es
[b)]esther.rojas@ciemat.es

Abstract. In this work we have compared the experimental results from the thermocline tank of STONE installation of CEA with an analytical model which considers an algebraic sigmoid as solution for the energy balance equation. The parameters of this function have been obtained by fitting its mathematical expression to the experimental temperature-time data for different tank heights. From that fitting physical parameters like the velocity of thermocline zone inside the tank and the effective heat capacity of the storage medium could be determined. These values depend on operating conditions like superficial oil velocity and temperature range but they deviate from the theoretical ones due to the specific STONE tank design in which metallic wall shares 10 % tank volume. In this way, considering or not the tank wall in the effective heat capacity leads to 6.6 % error when predicting the thermocline zone position with the model. However, for thermocline tanks with larger volumes the effect of walls is expected to be negligible and hence the analytical model will be able to predict more accurately their performance.

INTRODUCTION

Thermocline tanks either with single or dual media are nowadays within the alternatives with higher cost reduction potential for the storage system in CSP applications. Although thermo-hydraulic performance of thermocline tanks has been widely studied in the literature through different simulation approaches, the availability of experimental data for model validation, especially at the beginning of dynamic processes, is very scarce.

In a previous work [1] we proposed a general analytical model for describing the behavior of thermocline tanks whose implementation in a CSP plant simulation model did not require long computation times [2]. However, this analytical model could not be properly validated due to the lack of experimental data. In order to overcome this limitation, a test campaign was performed at the STONE facility of CEA-Grenoble in collaboration with CIEMAT-PSA supported by both the SFERA-II and the STAGE-STE project, whose preliminary results were presented in a previous SolarPACES Conference [3].

In this work we have analyzed in depth the experimental measurements obtained during that test campaign. For this purpose we have chosen the analytical function that better fitted them and from the fitting we have obtained the thermocline velocity inside the tank and also the effective volumetric heat capacity for the different tests. The results have been discussed in terms of tank configuration and operating conditions. This analysis is the first step for both validating and improving the analytical model previously developed in our institution for predicting thermocline storage tank behavior [1].

EXPERIMENTAL SET-UP AND TEST CONTIDITONS

Experimental data analyzed in this work were obtained at the STONE facility of CEA during a test campaign performed in 2016 [3]. The STONE test-loop consists of a cylindrical thermocline tank and two fluid loops to simulate both charging and discharging processes. The storage tank is made of stainless steel and has 1 m diameter

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(D), 3 m height (H), a domed top and a dished bottom. The tank is filled with a mixture of 3-cm silica gravel and 3-mm silica sand with a global void fraction (ε) of 0.27 and Therminol® 66 [4] is used as heat transfer fluid. Further details and characteristics of the STONE test-loop can be found in Bruch et al. [5].

In this test campaign, charge and discharge processes were performed at different operating conditions in terms of oil superficial velocity (v_m), temperature interval of the oil and initial tank state. The experimental data analyzed in this work correspond to charge and discharge processes starting from the tank at constant temperature, i. e, fully charged or discharged. The specific conditions of each test in terms of oil superficial velocity, mass flow (both mean values) and temperature range (i. e. T_{max} and T_{min}) are recorded in Table 1.

TABLE 1. Operating conditions for the different tests performed at STONE tank and analyzed in the work. v_{TC} values obtained from the fittings of Fig. 2 are recorded as well.

Test nº-Charge/Discharge	v_m (m/s) (x10^3)	Mean mass flow (kg/h)	T_{max} (ºC)	T_{min} (ºC)	v_{TC} (m/s) (x10^3)
Test 1-Discharge	1	681	250	100	0.208
Test 2-Charge	1	684	250	100	0.209
Test 2-Discharge	1	681	250	100	0.208
Test 3-Discharge	2	1428	150	85	0.442
Test 4-Charge	1	714	150	85	0.204
Test 4-Discharge	1	711	150	85	0.213
Test 5-Charge	1	714	150	85	0.202
Test 5-Discharge	1	712	150	85	0.213
Test 6-Discharge	2	1427	150	85	0.433
Test 7-Discharge	1.5	1070	150	85	0.322
Test 8-Charge	1.5	1073	150	85	0.309

Temperature inside the storage tank was monitored with time at twelve tank heights (z_c) and different radial positions for both oil and filler (thermocouples were physically inserted in the stones of the filler) [5]. All thermocouples embedded in the storage tank were calibrated in the range 0-350 ºC with an accuracy of ± 2 ºC. After checking radial temperature distribution for oil and filler at each height, we found that differences were below the accuracy range of the thermocouples. Therefore it can be assumed that thermal losses are negligible and hence for each height (z_c), temperature (T) has a unique value which is the same for both oil and solid filler.

ANALYSIS OF THE EXPERIMENTAL RESULTS

Selection of the Analytical Function

The one-dimension differential energy balance equation that describes the behavior of a thermocline storage tank with a single effective storage medium and for which thermal losses can be neglected is [6]:

$$\frac{\partial T}{\partial t} + v_{TC}\frac{\partial T}{\partial z} = \alpha_{eff}\frac{\partial^2 T}{\partial z^2} \tag{1}$$

Where v_{TC} corresponds to the velocity at which thermocline zone moves inside the tank [6] and α_{eff} is the effective diffusivity of the storage medium. These parameters can be expressed in terms of thermos-physical properties as:

$$v_{TC} = \frac{\left(\rho C_p\right)_{liquid}}{\left(\rho C_p\right)_{eff}} v_m \tag{2}$$

$$\alpha_{eff} = \frac{k_{eff}}{\left(\rho C_p\right)_{eff}} \tag{3}$$

Where v_m is the superficial liquid velocity (oil velocity), which can be calculated from the oil mass flow $\dot{m}$ with the oil density and the tank diameter (D):

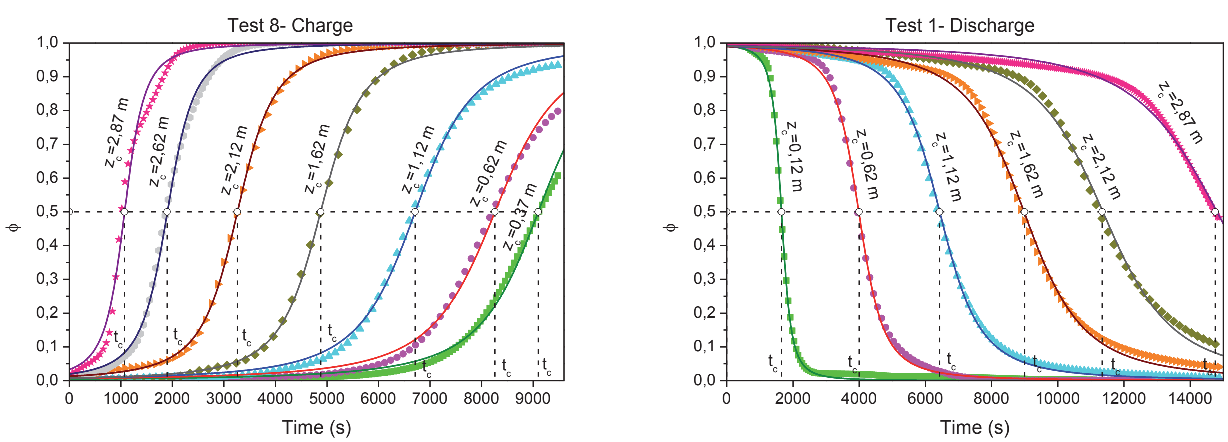
$$v_m = \frac{\dot{m}}{\rho_{liquid} 0.25\pi D^2} \tag{4}$$

Independently on the solving method (numerical or analytical), the solution of Eq. (1) is a set of sigmoid curves that represent the temperature variation as a function of time and tank height [6]. When analytical methods are applied for solving Eq. (1), the solutions normally found in the literature are sigmoid functions used in the field of statistics and based on the complementary error function, which is directly related to the Normal Cumulative Distribution Function (Normal-CDF) [7, 8]. In a previous work [1] we used a logistic-CDF curve as an alternative approximation to the Normal-CDF for describing the thermal behavior of thermocline tanks. In this way we used the logistic-CDF equation for fitting the sigmoidal solution curves of Eq. (1) previously obtained by means of a numerical method [6]. However when analyzing the experimental data of STONE tank we observed that the logistic-CDF did not fit them very well. To our opinion, the selection of the most appropriate analytical function might depend on the specific kind of thermocline storage tank; although we expect that choosing a curve or another is not critical for predicting its behavior. Actually other authors used different sigmoid curves as analytical functions for simulating the performance of thermocline tanks and they all obtained quite good results [9, 10, 11]. In order to check other sigmoid equations that could better fit the experimental data, we previously transformed temperature into dimensionless form (ϕ) [5] by using the following transformation so that temperature interval is [0, 1]:

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

After this transformation we plotted temperature-time data for the different tank heights in order to check various sigmoid equations and find out which one led to the best fitting [12]. Using temperature-time curves for the fitting is more convenient than using temperature-position ones, because the number of data is much larger [3]. After several tries, an algebraic sigmoid [12] was finally chosen as the analytical function that better fitted the experimental data of STONE thermocline tank. Figure 1 displays temperature-time data recorded at different tank heights (z_c) together with the resulting fitting curves for the Test 8-Charge (a) and the Test 1-Discharge (b) (see Table 1. for operating conditions).

As we can see, the selected algebraic sigmoid fits quite well the experimental results obtained from the STONE thermocline tank not only during charge but also during discharge processes under different operating conditions of superficial oil velocity and temperature range.



(a) (b)

FIGURE 1. Dimensionless temperature-time data recorded at different tank heights (z_c) with the corresponding fitting curves using the algebraic sigmoid for Test 8-Charge (a) and Test 1-Discharge (b) (See Table 1 for operating conditions).

The mathematical expression of the algebraic sigmoid used for the experimental data fitting in terms of dimensionless temperature variation with time at a certain tank position (z_c) is:

$$\phi(t,z_c)=\frac{1}{2}\left\{1\pm\frac{(t-t_c)}{\left[S_t+(t-t_c)^2\right]^{1/2}}\right\}\ \text{with}\ \begin{array}{l}+\ \text{in charge}\\ -\ \text{in discharge}\end{array} \tag{6}$$

Whereas the expression for dimensionless temperature variation with tank position at a certain time (t_c) is:

$$\phi(z,t_c)=\frac{1}{2}\left\{1+\frac{(z-z_c)}{\left[S_z+(z-z_c)^2\right]^{1/2}}\right\}\ \text{with}\ \begin{array}{l}z_c=H-v_{TC}t_c\ \text{in charge}\\ z_c=v_{TC}t_c\ \text{in discharge}\end{array} \tag{7}$$

Taking into account that thermocline moves downwards in charge and upwards in discharge. Apart of t_c and z_c, which correspond to the positions of inflection points of Eq. (6) and Eq. (7) respectively, these functions have a unique parameter S_t or S_z, which is related to the sigmoid slope at the inflection point. The relation between these parameters and their dependence on tank operating conditions and characteristics is an issue still under study and will be reported in a future work.

Calculation of Thermocline Velocity

For each tank position (z_c), curve fitting with Eq. (6) leads to a value of time (t_c) that corresponds to the sigmoid inflection point in which ϕ=0.5 (See Fig. 1 (b)). In this way, if we plot z_c vs. its corresponding t_c we will obtain a linear variation whose slope is the thermocline velocity v_{TC} (see Eq. (1)). These pots were done for charge and discharge tests recorded in Table 1 and all plots lead to very good linear fittings from which the corresponding v_{TC} values could be obtained. Figure 2 displays those plots together with their corresponding linear fitting for all charge (a) and discharge (b) tests and the resulting v_{TC} values have been included in Table 1. The plots of Fig. 2 clearly demonstrate the reproducibility of the tests performed with the STONE thermocline tank since repeated test performed under the same conditions show the same linear fitting and hence the same v_{TC} value.

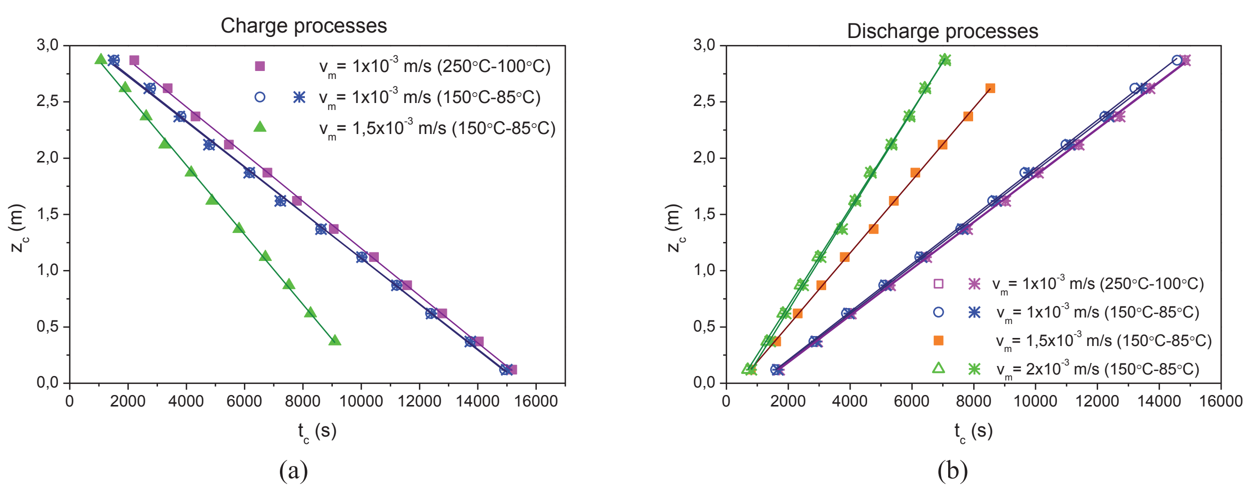


FIGURE 2. v_{TC} calculation from z_c vs. t_c plots for charge (a) and discharge (b) tests performed at different superficial oil velocity and temperature intervals (see Table 1.)

Calculation of Volumetric Heat Capacity

Once v_{TC} value is known, the effective volumetric heat capacity $(\rho C_p)_{eff}$ for each test can be calculated by combining Eq. (2) and Eq. (4):

$$\left(\rho C_p\right)_{eff} = \frac{\dot{m}\left(C_p\right)_{liquid}}{0.25\pi D^2 v_{TC}} \tag{8}$$

Hence it is possible to calculate the experimental values $(\rho Cp)_{eff}$ since all parameters included of Eq, (8) are known and $(C_p)_{liquid}$ can be calculated from the corresponding equation reported in the commercial brochure of Therminol 66 [4]. The temperature used for calculating $(C_p)_{liquid}$ was the mean temperature of each test, which depended on the specific conditions at which the test was performed.

The experimental values of $(\rho Cp)_{eff}$ were compared with the expected theoretical ones calculated with the typical expression used for estimating the effective volumetric heat capacity of a packed-bed system with a certain porosity, ε:

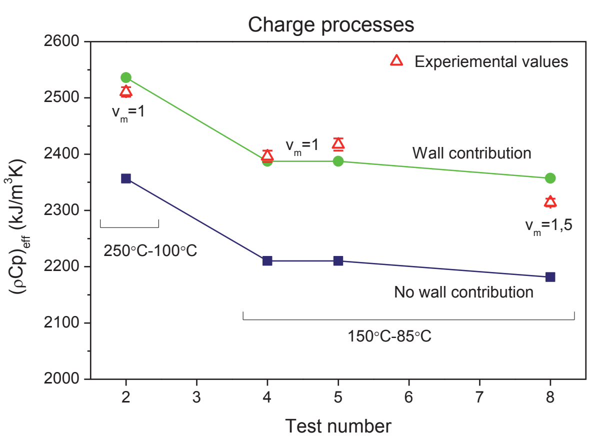
$$\left(\rho C_p\right)_{eff} = \varepsilon\left(\rho C_p\right)_{liquid} + \left(1-\varepsilon\right)\left(\rho C_p\right)_{solid} \tag{9}$$

Again density, ρ, and heat capacity, C_p, for the liquid were calculated with the equations included in the commercial brochure Therminol 66 [4] at the mean temperature of each test, whereas the values taken for the solid filler are those previously reported in ref. [5].

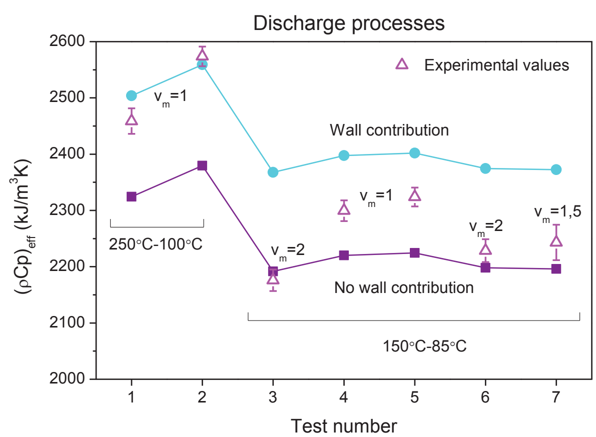
When comparing results from Eq. (8) with those from Eq. (9), we observed that, in general, the former were higher, especially for the case of charge processes. This means that the storage tank presents an additional thermal inertia. Actually Bruch et al. observed that their numerical model better fitted the experimental results of the STONE storage tank when they considered in the calculations the metallic tank wall as an additional mass of storage medium [5]. In a similar way we could consider the contribution of tank wall to the effective volumetric heat capacity by using the following expression where δ accounts for the volume fraction of the metallic wall in the whole tank volume.

$$\left(\rho C_p\right)_{eff} = \left(1-\delta\right)\left[\varepsilon\left(\rho C_p\right)_{liquid} + \left(1-\varepsilon\right)\left(\rho C_p\right)_{solid}\right] + \delta\left(\rho C_p\right)_{wall} \tag{10}$$

In Fig. 3 the experimental values of $(\rho C_p)_{eff}$ obtained for charge (a) and discharge (b) tests have been plotted together with the theoretical ones considering or not the tank wall contribution (Eq. (10) and Eq. (9), respectively). In the experimental values error bars of $\dot{m}$ have been included in order to show that the oscillations of oil mass flow are not the responsible for the deviations of experimental $(\rho C_p)_{eff}$ from the theoretical values given by Eq. (9).



(a)



(b)

FIGURE 3. Comparison of $(\rho C_p)_{eff}$ experimental values with those considering (Eq. (10)) or not (Eq. (9)) the wall contribution for both charge (a) and discharge (b) tests. The corresponding superficial velocities used in each test have been indicated in mm/s.

In charge processes (Figure 3(a)) it is quite clear that the experimental values of effective heat capacity are very close to the theoretical ones calculated by taking into account the tank wall contribution. Hence it seems that the thermal inertia of the tank wall can be included in the thermocline model as an additional mass of material that must be heated up, which implies that, in charge, the tank requires more energy supply than expected. However, for the case of discharge processes (Figure 3(b)), the experimental values of $(\rho C_p)_{eff}$ are close to the theoretical ones considering the tank wall only for the tests performed at high temperature range (250 ºC-100 ºC) with ΔT=150 ºC. The tests carried out at the range 150 ºC-85 ºC with ΔT=65 ºC show lower $(\rho C_p)_{eff}$ values so that in the tests performed at the highest superficial velocity (2 mm/s), the experimental heat capacity is almost the same as the theoretical value calculated without taking into account the tank wall contribution.

The reason for this behavior can be that the thermal inertia of tank wall becomes more important when the tests are carried out at high temperature ranges and low superficial velocities. In this way, when tank is operated at 250 ºC-100 ºC range, the effect of tank wall inertia is higher in both charge and discharge processes because the thermal gradient between hot and cold oil is larger (ΔT=150 ºC) whereas for the test performed at low temperature range (150 ºC-85 ºC) with a smaller thermal gradient (ΔT=65 ºC), the influence of wall thermal inertia is lower especially in discharge processes. This effect is more clearly observed in the tests carried out at high oil superficial velocity (2 mm/s), for which experimental $(\rho C_p)_{\mathrm{eff}}$ values are close to the theoretical ones calculated without considering the tank wall.

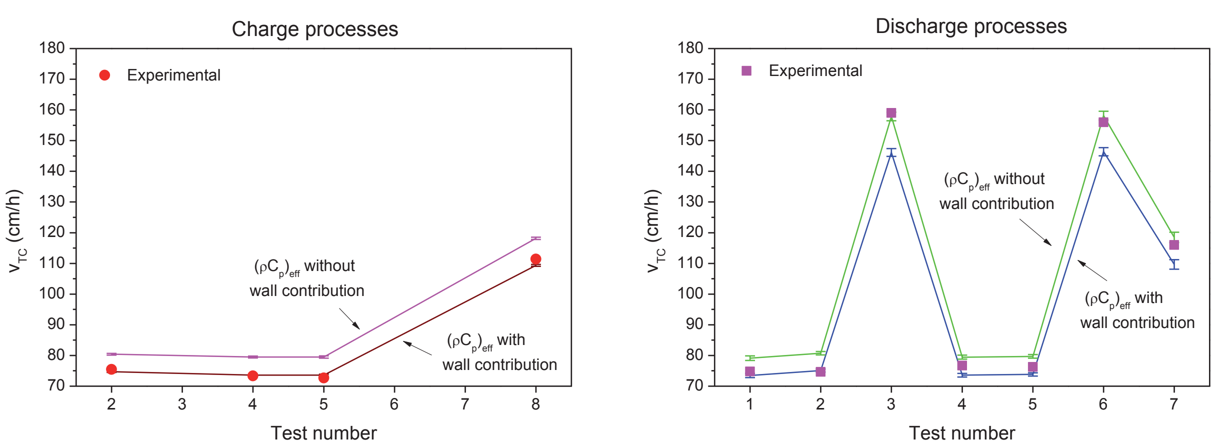
On the other hand, thermal losses could also have some influence in the $(\rho C_p)_{eff}$ values obtained experimentally and somehow explain the differences observed between charge and discharge processes. In discharge, when the tank is cooled down, thermal losses could help the process reducing $(\rho C_p)_{eff}$, whereas in charge, thermal losses have to be overcome and this could make the tank heating more difficult leading to higher $(\rho C_p)_{eff}$ values. In our model we did not considered thermal losses, since no mayor differences was found between outer and inner temperature gauges. However, Bruch et al. [5] did consider them in their model and still had to take into account the wall effect in the storage capacity in order to obtain a good fitting of the experimental results. This means that thermal losses cannot be the only responsible for the deviation of $(\rho C_p)_{eff}$ from the theoretical values.

Bruch et al. also discussed that the need of including the wall contribution in the whole storage mass comes from the specific tank design, mainly because the ratio between metallic wall and packed bed volume is quite high (δ=10%). To their opinion, it is expected that for a tank with a larger volume, the wall effect in terms of thermal inertia would be much lower and could be disregarded [5].

The fitting results can also be expressed in terms of thermocline velocity (v_{TC}) so that we can represent the experimental values together with the values calculated from an expression similar to the one in Eq. (8) using Eq.(10) or Eq. (9) for calculating $(\rho C_p)_{eff}$ with or without tank wall contribution.

$$v_{TC} = \frac{\dot{m}\left(C_p\right)_{liquid}}{0.25\pi D^2\left(\rho C_p\right)_{eff}} \tag{11}$$

The resulting plots are displayed in Fig. 4 in which the error bars associated to $\dot{m}$ have been included in the calculated values of v_{TC}. In this case velocity has been expressed in cm/h in order to see more clearly the order or magnitude.



(a) (b)

FIGURE 4. Comparison of v_{TC} experimental values with the theoretical ones considering or not the wall contribution in $(\rho C_p)_{eff}$ calculation for both charge (a) and discharge (b) tests.

In this case we observe the same trend as in the heat capacity values so that in charge experimental velocities are similar to the theoretical values considering the tank wall whereas in discharge the velocities are closer to the values calculated without considering the walls specially for the tests performed at higher velocity (Test 3 and Test 6).

However in this case we can see more clearly the order of magnitude for the velocity difference including or not the tank wall contribution in the calculation of $(\rho C_p)_{eff}$. This difference is within the interval 5-10 cm/h which means that for a process of full charge or discharge the error in thermocline position at the end of the process is about 20 cm. Therefore, if we take into account that the tank is 3 m height, the error in predicting the thermocline position considering or not the tank wall effect in the calculation of v_{TC} is around 6.6 %. This value is expected to be reduced in larger storage tanks because in such cases the effect of the walls would be negligible.

CONCLUSIONS

In this work the experimental results of the thermocline storage tank STONE have been analyzed by using an analytical model based on an algebraic sigmoid function. The model allows obtaining the velocity at which thermocline zone moves inside the tank, v_{TC}, by simply fitting the sigmoid function to the temperature values recorded with time at different tank heights. This analysis has proved the high reproducibility of the tests performed and hence the reliability of the STONE installation. From the velocity of thermocline zone, the experimental value of effective volumetric heat capacity, $(\rho C_p)_{eff}$, can be calculated for each test. It has been obtained that experimental values of $(\rho C_p)_{eff}$ depended on operating conditions, i. e. oil temperature range and superficial velocity, and that the differences observed can be associated to the thermal inertia of the tank wall. In this way the error in predicting the thermocline position considering or not the tank wall effect is around 6.6 %. This result is a consequence of the specific STONE tank design because the ratio between metallic wall and packed bed volume is quite high (δ=10%). However it is very likely that for the case of a tank with larger volume, the wall effect would be much lower and hence could be neglected. As a main conclusion we can say that we have proved that the behavior of a real thermocline tank can be predicted by using an analytical sigmoid function, which has been the first step for validating the analytical model previously developed in our institution. Next step will be to establish the dependence of function parameters S_t and S_z on both tank characteristics and operating conditions in order to obtain a general model that is able to predict all thermocline tank situations and states. The kind of sigmoid to be considered in the model might depend on the specific kind of thermocline tank but we do not expect that the function selection strongly affects the prediction of the overall tank performance.

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