Parametric Analysis of a Packed Bed Thermal Energy Storage System

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Abstract. Even if the packed bed thermal energy storage concept has been introduced as a promising technology in the concentrated solar power field in the last years, its full deployment in commercial plants presents a clear improvement potential. In order to overcome the under-development of this storage technology, this work attempts to show the great capabilities of packed bed heat storage units after a successful design and operational parametric optimization procedure. The obtained results show that a correct design of this type of facilities together with a successful operation method, allow to increase significantly the storage capacity reaching an overall efficiency higher than 80 %. The design guideline obtained as a result of this work could open new objectives and applications for the packed bed storage technology as it represents a cost-effective and highly performing storage alternative.

INTRODUCTION

In addition to a better exploitation of the available solar resources in concentrated solar power (CSP) plants, thermal energy storage (TES) solutions lead to a more efficient energetic production strategy resulting in a decrease of the final cost of the produced electricity. This optimization of the energetic resources through the deployment of cost-effective TES systems is in agreement with the roadmaps developed by the main energy agencies [1-3] which determine the potential evolution of the CSP production in the next decades. The accomplishment of these objectives implies an important cost reduction [4] of the current CSP technologies, such as the double molten salt tank TES systems [5]. In this regard, solid packed bed arrangements have been proposed in the last years as a technically successful and cost-effective TES system with high thermal performance [6-8]. The increasing number of scientific publications analyzing the mentioned heat storage unit for CSP, industrial waste heat recovery or adiabatic compressed air energy storage (ACAES) applications demonstrates the large capabilities of this concept. This configuration provides a highly efficient TES with a minimized overall cost, as low cost solid materials can be implemented as storage materials in this single tank storage solution. As an example, natural rocks [9], sand [10] or ceramic industrial wastes [11] have been proposed as solid TES material candidates for this application. Although a noticeable research effort has been done on the analysis of the driving mechanisms of this storage system, the commercial implementation maturity of this technology still presents important improvement challenges in real scale applications.

Following the research work performed in our laboratories [12], in this study, the use of steel slag as low cost solid heat storage material is proposed together with air as heat transfer fluid (HTF). This material selection presents a double interest. On one hand, the cost effectiveness of the TES system is guaranteed, given that the solid material is a waste from the steel industry and, the HTF is atmospheric air. On the other hand, these materials permit to remove the current operation temperature range limitation, constrained to the solar salt range between 280 - 565 ºC. The main objective of this work is to provide a complete design guideline of a generic packed bed storage system, showing all the potential optimization gap of this technology. The performed parametric analysis includes a detailed

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study of the impact of the main design parameters such as the tank geometry (cylindrical or conical), aspect ratio, solid particle size and mass flow rate/velocity of the HTF. The impact of all these design parameters on the TES capacity, quality of the released heat and the overall efficiency is discussed. To assess all the mentioned parametric analysis computational fluid dynamic (CFD) calculations have been performed.

MODEL DESCRIPTION

Physical Model

The commercial CFD package, Ansys Fluent®, has been used in this work to model the thermal behavior of an interstitial air flow through a packed-bed storage unit. This fluid flow is modeled by using the continuous porous media approach [13] based on Ergun equation (1) which is valid for a wide range of laminar and turbulent flow conditions [14].

$$\frac{\Delta P}{H} = \frac{150\mu}{d_p^{\ 2}} \frac{(1-\varepsilon)^2}{\varepsilon^3} v_\infty + \frac{1.75\rho_f}{d_p} \frac{(1-\varepsilon)}{\varepsilon^3} v_\infty^{\ 2} \tag{1}$$

Due to the different macro-parameters that will be studied in this analysis and the wide range on which they will be varied, the well-known local thermal equilibrium (LTE) approach to model the heat transfer between the fluid and solid phases cannot always be assumed. Hence, for all the modeled cases, in order to guarantee the accuracy and the reliability of the obtained results, the non-LTE approach will be applied. This method provides a satisfactory and general description of the packed bed thermal behavior, overcoming the local thermal equilibrium approach. Taking this into account, separate energy equations will be solved for the fluid (2) and the solid (3) phases. Both equations are coupled by the solid-fluid heat transfer, represented by the last term on the right side in both equations. This interstitial heat transfer is governed by an overall heat transfer coefficient (h_p), the exchange interface area (A) and the temperature difference between the solid and fluid phases.

$$\frac{d}{dt}\left(\varepsilon\rho_f E_f\right) + \nabla\cdot\left(\vec{v}\left(\rho_f E_f + p\right)\right) = \nabla\cdot\left(\varepsilon k_f \nabla T_f - \left(\sum_i h_i J_i\right) + \left(\bar{\bar{\tau}}\cdot\vec{v}\right)\right) + S_f^h + h_p A\left(T_s - T_f\right) \tag{2}$$

$$\frac{d}{dt}\left((1-\varepsilon)\rho_s E_s\right) = \nabla\cdot\left((1-\varepsilon)k_s \nabla T_s\right) + S_s^h + h_p A\left(T_f - T_s\right) \tag{3}$$

For the calculation of the interstitial heat transfer coefficient, two different correlations have been used depending on the particle Reynolds number (Re_p), defined as:

$$\mathrm{Re}_p = \frac{v_\infty \rho_f d_p}{\mu} \tag{4}$$

Where v_∞ corresponds to the fluid free velocity in the tank, calculated in the absence of the solid bed, ρ_f is the fluid density, d_p is the particle diameter and, μ the fluid viscosity.

For $\mathrm{Re}_p < 15$, the correlation given by Pfeffer [15] has been used (Eq. (5)).

$$h_p = 1.26\left[\frac{1-(1-\varepsilon)^{5/3}}{2-3(1-\varepsilon)^{1/3}+3(1-\varepsilon)^{5/3}-2(1-\varepsilon)^2}\right]^{1/3}\left(c_f G\right)^{1/3}\left(k_f/d_p\right)^{2/3} \tag{5}$$

On the other hand, for $\mathrm{Re}_p \geq 15$, equation (6) given by *Alazmi* and *Vafai* [16] has been applied.

$$h_p = \frac{k_f}{d_p}\left(2 + 1.1\mathrm{Re}_p^{\ 0.6}\,\mathrm{Pr}^{1/3}\right) \tag{6}$$

The modeling of all the heat transfer phenomena that occur in a packed bed involves complex mechanisms and contributions of different natures (convective, conductive and radiative) depending on the fluid flow regime, packing arrangement and other physical parameters. In order to consider the most representative ones in a simple approach, an effective thermal conductivity of the solid k_s^{eff} and fluid k_f^{eff} materials is introduced in the energy equation instead k_s and k_f. For simplicity, in our modeling approach, isotropic behavior in the axial and radial directions is considered. Quantitatively, the numerical values for this thermal transport parameter has been obtained following the models developed by *Yagi* and *Kunii* [17], *Kunii* and *Smith* [18] and *Wakao* and *Kaguei* [19].

Finally, thermal losses through the outer lateral wall (insulating material) of the storage tank have been introduced in the model. The heat transfer coefficient used for its evaluation is calculated according to the laminar convective heat transfer equations given in ref. [20]. The average value for all the modeled units is h_{loss} = 3.5 W/m^{2}K.

Energy and Efficiency Evaluation

For the energy evaluation in a charge/discharge run of the proposed storage system, equation (7) was used. The subindex "in" is associated to the input fluid of the tank, corresponding to the upper part during the charging process or to the lower part in the case of the discharge. On the other hand, the "out" subindex stands for the output fluid coming out from the storage.

$$E_{charged/discharged} = \int \dot{m} c_f (T_{in} - T_{out}) dt \tag{7}$$

The energy required for the pumping of the fluid in the tank is calculated following equation (8), where the pressure drop (*ΔP*) induced by the packed bed is calculated from the Ergun equation (1).

$$E_{pumping} = \int \frac{\dot{m}}{\rho_f} \Delta P dt \tag{8}$$

Overall, the efficiency (*η*) of a complete cycle (charge + discharge) is calculated according to equation (9). It has to be pointed out, that the energy losses through the lateral wall are included in the charged and discharged energies and, that the pumping energy term accounts for the sum of charge and discharge pumping energies.

$$\eta = \frac{E_{discharged}}{E_{charged} + E_{pumping}} \tag{9}$$

Model Validation

Prior to the calculations, the proposed theoretical model was validated with the experimental results published by *Zanganeh et al.* in ref. [8]. The results of the performed validation are collected in Fig. 1, where a very good agreement can be observed between the experimental results and the data obtained from the model. This results show the validity of the developed modeling strategy, which allows performing precise calculations to determine a general optimization process of packed bed storage units as a function of a wide variety of design parameters.

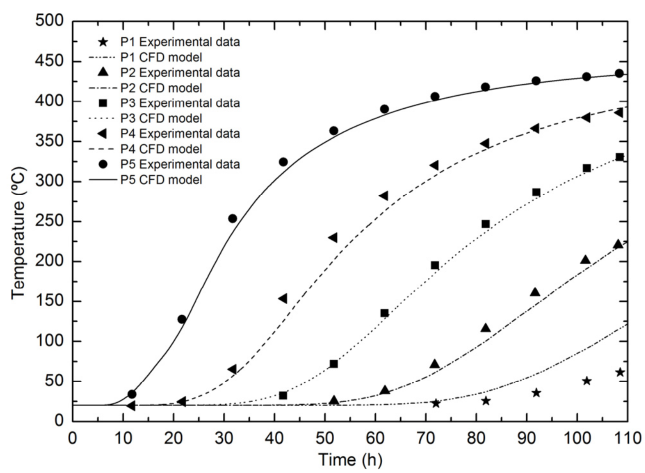


FIGURE 1. Validation of the CFD model used in this work with the experimental results published in ref. [8]. Lines correspond to the data obtained from the model and the points with the experimental measurements.

SYSTEM DESCRIPTION AND METHODOLOGY

To guarantee the maximum generality of the obtained results, a benchmark case study was pre-selected, with a full compatibility to a new generation CSP production frame. As a reference, the volume of the TES unit was fixed to 3 m^3, corresponding to a maximum storage capacity of around 1 MWh_t. The selected HTF was air and the TES material is the aforementioned steel slag [12]. The operation temperature was fixed in the 700 ºC - 20 ºC range, oriented to the next generation CSP electricity production. A 25 cm insulation layer was considered in all the modeled units. As an example, the thermophysical properties of the mentioned materials at the average operational temperature (360 ºC) are included in Table 1 (in the model all thermophysical properties are considered temperature dependant).

TABLE 1. Air, steel slag and insulating material thermo physical properties at the average temperature (360 ºC) considered in this work.

Property	Air	Steel Slag	Insulation
Density (kg/m^3)	0.555	3430	128
Heat capacity (J/kg.K)	1056	877	1250
Thermal conductivity (W/m.K)	0.047	1.47	0.08
Viscosity (kg/m.s)	3.140E-5	-	-

In the operation of the modeled TES unit, a full charge/discharge criterion was fixed. In this regards, the system is assumed to be completely charged when the outlet temperature is 100 ºC higher (temperature tolerance) than the cold fluid (20 ºC). On the other hand, in the discharge stage, all the released energy is considered useful for the power block while the fluid outlet temperature does not drop below 600 ºC (hot fluid temperature minus the temperature tolerance). This selection ensures a good quality of the heat for the steam turbine electric production above 600 ºC. At the same time, during the charge, this selection does not introduce an "over-heated" fluid flow, which implies an energy loss together with an operation difficulty to manage appropriately the output fluid.

The parametric study performed in this work included the analysis of two different tank geometries, cylindrical and conical. The first one has extensively been used in the TES field as well as in many other industrial applications. On the other hand, the conical tank has recently been introduced by some authors [8] aiming the reduction of the mechanical stresses generated between the TES solid media and the container with the thermal cycling. In addition to these geometries, two orientations of the conical shape have been analyzed: the first one with the upper part of the tank larger than the bottom (henceforth "regular") and, the second one with the larger diameter on the bottom (henceforth "inverse").

For all the investigated geometries, the influence of the tank aspect ratio on the stored energy and on a charge and discharge cycle efficiency has been analysed. In the case of the cylindrical tank, the aspect ratio is defined as the tank length (H) divided by the tank diameter (D). Meanwhile, in the conical shape, an additional degree of freedom is present since the diameter of the storage tank depends on the height. In this work, the total volume of the tank together with the conicity angle are fixed (see Table 2) leaving the tank length as the only free parameter. Furthermore, the TES material particle diameter and the fluid mass flow rate (velocity) influence on these parameters will be studied.

As a summary, in Table 2 is presented the main operational parameters of the proposed demonstration scale TES unit together with the range on which the analyzed parameters will be varied.

TABLE 2. Main operational and geometrical parameters of the modeled TES units.

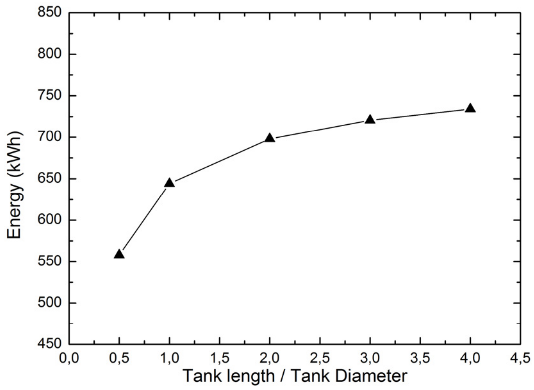
Parameter	Value
Cold fluid temperature (ºC)	20
Hot fluid temperature (ºC)	700
Temperature tolerance (ºC)	100
Fluid mass flow rate (kg/s)	From 0.01 to 1
Slag particle diameter (cm)	From 0.5 to 5
Tank volume (m^3)	3
Insulation thickness (m)	0.25
Cylindrical tank aspect ratio (H/D)	From 0.5 to 4
Conical tank height (m)	From 1 to 3
Conical tank angle (º)	10

RESULTS

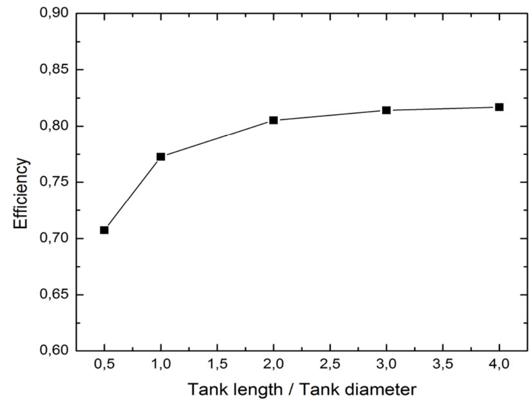
Influence of the Cylindrical Storage Tank Aspect Ratio

Based on the above-mentioned parametric analysis, in Fig. 2 it is presented the behaviour of the cylindrical tank as a function of the aspect ratio, maintaining fixed the air mass flow rate (0.3 kg/s) and the slag particle diameter (1 cm). Figure 2a displays the impact of the aspect ratio on the stored energy (released) during the first discharge run. The influence on the overall cycle efficiency is shown in Fig. 2b. From this figure, it can be observed that higher tank aspect ratio values promote larger energy released within the established temperature tolerance. Comparing the

0.5 and 4 aspect ratio values, an improvement of around 30 % on the total stored energy can be achieved. However, it can also be observed that aspect ratios higher than 2 do not show a noticeable improvement, obtaining only a slight enhancement. Looking at Fig. 2b, a similar behaviour is observed for the cycle efficiency where a saturated value around 80 % is reached for storage aspect ratios above 2. It must be noted that this efficiency value is associated to the first charge/discharge cycle of the storage. This value is subsequently increased with further thermal cycling operation of the TES, not addressed in this work, up to values around 95 % [21]. A detailed analysis of the long-term thermal cycling of the storage and its corresponding impact on the stored energy and overall efficiency is shown in [21]. As a result coming out from this analysis, due to the mechanical complexity of constructing high length tanks with reduced cross section (high aspect ratio), a cylindrical tank with an aspect ratio around two is identified as the most promising configuration.



(a)



(b)

FIGURE 2. Discharged energy (a) and efficiency (b) in the first charging cycle depending on the aspect ratio of the tank.

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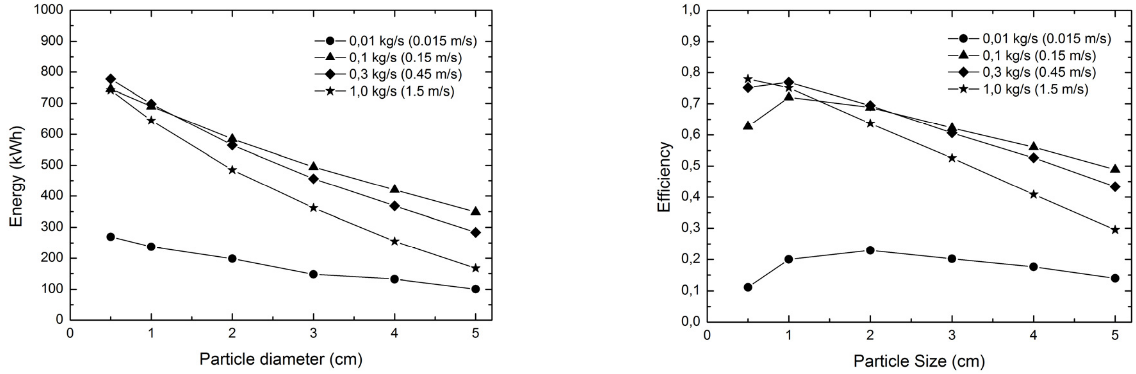
Influence of the Fluid Flow Rate and Solid Particle Diameter

In the study of the heat transfer fluid velocity and solid particle diameter, a tank with an aspect ratio of two is considered. The results of the parametric analysis are collected in Fig. 3. In particular, in Fig. 3a, the energy released during the first discharge run of the system is presented for six different particle diameters and four mass flow rates. In addition, in Fig. 3b the first cycle efficiency calculated following equation (9) for all the analysed cases is presented.

Attending to the influence of the particle diameter on both, the stored energy and efficiency, it can be observed that for the four mass flow rates, the higher is the particle diameter, the smaller is the stored energy and the efficiency. Furthermore, it has to be pointed out that for pebble diameters larger than 3 cm, the storage capacity of the system is reduced to values lower than 500 kWh which indicates that the volume associated to the temperature stratification zone is more than half of the tank. In these cases, the storage capacity reduction connected to the mentioned design parameters represents an important limitation, which establishes a maximum diameter value of the particles. On the other hand, on Fig. 3b it can be observed an important decrease of the efficiency for particle diameters lower than 1 cm caused by a noticeable increase of the pressure drop induced by the packed bed. As a consequence of the previous considerations, in general, the best efficiencies and thermal performances are obtained for particle diameters around 1 cm.

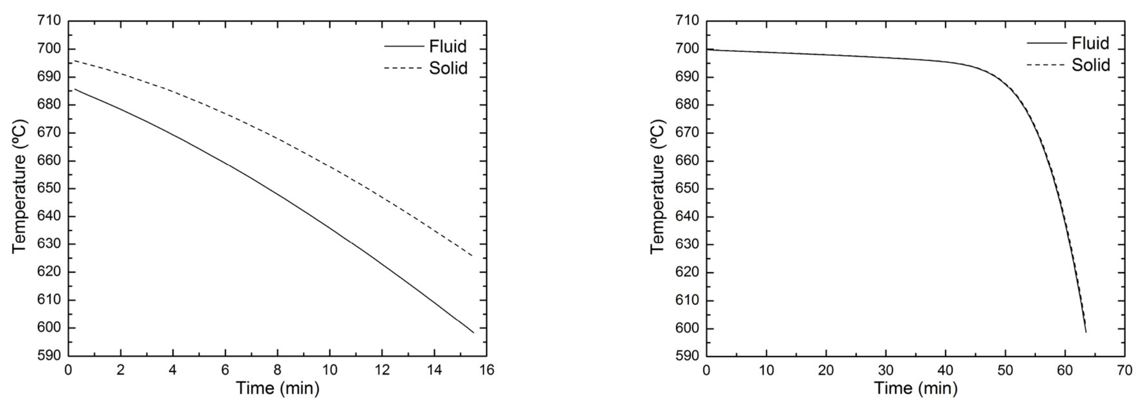
The velocity of the heat transfer fluid through the packed bed influence was analysed by varying the flow from a very laminar (0.01 kg/s) to turbulent (1 kg/s) conditions. The obtained results, presented on Fig. 3a, show a strong impact of this parameter on the performance of the TES unit. As it can be observed in Fig. 3, the storage capacity and efficiency is drastically decreased for the lowest mass flow rate. On the other hand, a maximized storage capacity is achieved for mass flow rates between 0.1 and 0.3 kg/s depending on the particle size. Values above 0.3 kg/s again show a decreasing storage capacity, as depicted in Figure 3a. The reasons for this behaviour are related to the heat transfer mechanisms governing the packed bed thermal transport. In the case of very low fluid velocities, the conduction and radiation between particles control the formation of the thermocline region. Under these conditions, the temperature stratification is mainly caused by the thermal conductivity of the solid media in the axial direction. On the other hand, high fluid velocities favour convective mechanisms because of higher turbulence in the fluid. Depending, among others, on the particle size and the thermo-physical properties of the TES material, the solid media is not able to absorb or release the heat from/to the HTF, leading to the formation of a larger thermocline region on the fluid.



(a) (b)

FIGURE 3. Particle diameter and mass flow rate influence on: (a) the released energy during the first discharge of the TES unit; (b) the first cycle efficiency calculated following equation (9).

Aligned with these hypotheses, in Fig. 4 it is included the separate fluid and solid temperatures on the outlet of the storage tank (upper part) during the discharge stage of the first cycle. Both figures correspond to a mass flow rate of 1 kg/s; in the left (Fig. 4a) it is presented the case with a particle diameter of 5 cm whereas, in the right (Fig. 4b) the 1 cm diameter case is shown. The former results indicate a temperature difference of around 10 ºC during all the discharge process between the fluid and solid phases. In this case, the solid temperature is higher than the one of the fluid. This behavior is explained by the impossibility of the solid bed to transfer all the heat to the fluid, flowing at high velocity values. Because of the large particle diameter, the contact area is reduced, leading to a poorer heat transfer between both phases. On the contrary, in the case of lower particle diameters, equal to 0.5 cm as shown in Figure 4b, there is no temperature difference between both phases. In this case, the small particle diameter increases the contact area promoting a more effective heat transfer, able to release all the heat content of the solid. Under this condition, the fluid velocity and particle size can be assumed as optimized. Comparing the two cases on Fig. 4 with its stored/released energies on Fig. 3a, it can be seen that an increase of almost four times on the stored energy can be achieved with the optimized system. Overall, these two parameters, fluid velocity and particle size, have revealed to be of paramount importance in order to ensure an adequate thermal performance of the TES unit as well as to minimize the temperature stratification leading to a maximization of the tank volume exploitation.



(a) (b)

FIGURE 4. Solid and fluid temperatures in the upper part of the tank during the discharging stage (fluid outlet). (a) Case with a mass flow rate of 1 kg/m^3 and particle diameter of 5 cm (a) and 0.5 cm (b).

Influence of the Storage Tank Geometry

Once the solid particle size, the mass flow rate and the tank aspect ratio of the cylindrical geometry have been analysed and optimized, on Fig. 5 the results of the parametric analysis performed for the conical tank aspect ratio and its two orientations are presented. Considering the observed behaviour of the cylindrical tank, in this case, the fluid mass flow rate was fixed in 0.3 kg/s in order to operate in the near-to-optimum velocity condition. For the same reasons, in order to optimize the thermal behaviour of the proposed storage system, the slag particle size was fixed in 1 cm. Fig. 5a corresponds to the released energy during the first discharge and, Fig. 5b to the first cycle efficiency.

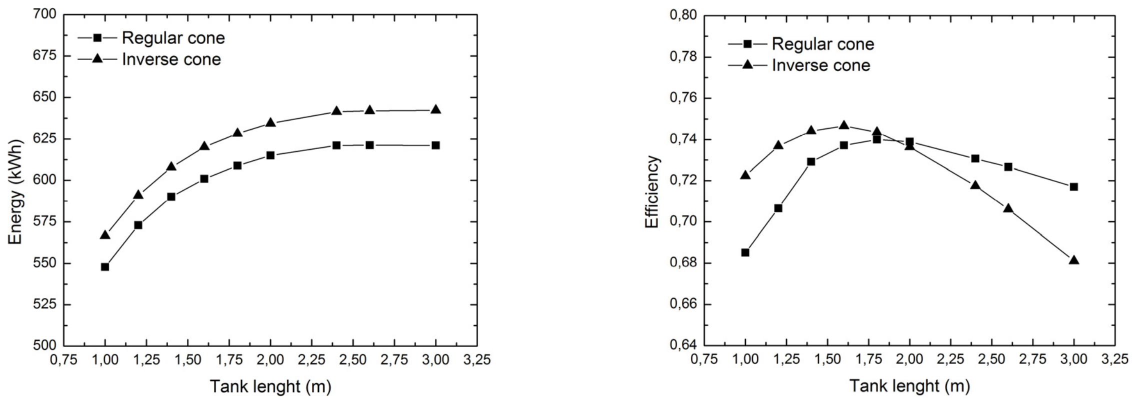


FIGURE 5. Conical tank length an orientation on: (a) the released energy during the first discharge of the TES unit; (b) the first cycle efficiency calculated as equation (9).

On Fig. 5a, it can be observed that for all the modeled aspect ratios, the stored energy is always higher for the inverse orientation (around 3.5 %) than the regular one. On the other hand, comparing the maximum storage capacity of all the considered conical storage tanks (642 kWh) with the storage capacities of the cylindrical tanks (Fig. 2a), it can be observed that in all the cases, the conical geometry has a worse thermal performance. The only exception is a cylindrical tank with an aspect ratio of 0.5 which is in the range of the conical tanks.

Looking to the efficiency graph, Fig. 5b, it can be seen that in both conical orientations there is a maximum performance design corresponding to a tank length of 1.8 m and to a 1.6 m for the regular and inverse orientations respectively. This results permit to identify the tank length, or the aspect ratio, as one of the main design parameters for a convenient system optimization.

Overall, when comparing the studied cylindrical and conical geometries, a higher thermal performance is found for the cylindrical case, which is able to store more energy. Taking into account the results presented on Fig. 2 and Fig. 5, it can be concluded that in order to have an optimized packed bed thermal energy storage system from the point of view of the thermal performance, a cylindrical geometry should be selected. On the other hand, the use of a conical geometry would only be justified from the point of view of its mechanical performance, as the appropriate design of the conical lateral tank walls permit to reduce the internal stress distribution.

CONCLUSIONS

In this work, a parametric analysis of the main parameters governing a packed bed thermal energy storage based on air as heat transfer fluid and steel slag as heat storage solid material has been performed by means of CFD calculations. In particular, the effect of the tank aspect ratio, heat transfer fluid velocity, the solid particle size and the storage tank geometry (cylindrical or conical), on the storage capacity and overall thermal efficiency in a complete charge and discharge cycle has been analysed.

The analysis of the cylindrical tank revealed that, taking into account the thermal performance of the system together with the construction constrains, the most promising aspect ratio value for the studied parametric set should be around two.

The consequent study on the influence of the mass flow rate and particle diameter revealed that, particle diameters around 1 cm are for all the considered mass flow rates the most efficient. This particle size balances successfully the pressure drop induced by the packed bed together with a good heat exchange area, which leads to a maximized thermal capacity. On the other hand, the mass flow rate investigation showed that a satisfactory balance between the heat transfer mechanisms and the fluid flow rate must be considered in order to obtain a successful heat management on the storage. This optimization procedure shows that intermediate fluid mass flow rate values are more appropriate (between 0.1-0.3 kg/s) to avoid a noticeable temperature difference between the solid and the fluid phases. Under these conditions, the packed bed storage offers a local thermal equilibrium condition, which promotes a higher thermal performance. If the design conditions do not allow this operation, derived from very low or very high fluid flow velocities, a thermocline spread will be obtained, reducing the thermal performance and the efficiency of the TES system. These results show the importance of a correct balance between the fluid velocity and particle size in order to equilibrate the convective heat transfer mechanisms with the conductive ones.

The analysis of the conical tank aspect ratio and orientation revealed that, the storage capacity and efficiency is lower than the value obtained for the cylindrical geometry. These results permit to conclude that a conical packed bed heat storage unit can only be justified from the point of view of its better mechanical performance, derived from a more effective management of the internal thermal and mechanical stress distribution.

Overall, the obtained results show that a correctly optimized design of a packed bed thermal energy storage unit allow to increase considerable the storage capacity reaching an overall efficiency during the first charge and discharge cycle around 80 %. The design of an optimized packed bed TES by means of maximizing its storage capacity also implies the minimization of the required solid material. This storage material amount reduction, maintaining similar storage capacities also promotes a higher efficiency of the TES system. The design guideline obtained as a result of this work could open new objectives and applications for the packed bed storage technology as it represents a cost effective and highly performing storage alternative.

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