Analysis of an integrated packed bed thermal energy storage system for heat recovery in compressed air energy storage technology

Iñigo Ortega-Fernández a, Simone A. Zavattoni b, Javier Rodríguez-Asequinolaza a,c, Bruno D’Aguanno d,a, Maurizio C. Barbato b,*

a CIC Energigune, Albert Einstein 48, 01510 Miñano (Álava), Spain
b Department of Innovative Technologies, SUPSI, 6928 Manno, Switzerland
c Departamento de Física Aplicada I, Escuela Técnica Superior de Ingeniería, Universidad del País Vasco, Alameda Urquijo s/n, 48013 Bilbao, Spain
d Present address: Koiné Multimedia, Via Alfredo Catalani 33, 56125 Pisa, Italy

HIGHLIGHTS

- A packed bed TES system is proposed for heat recovery in CAES technology.
- A CFD-based approach has been developed to evaluate the behaviour of the TES unit.
- TES system enhancement and improvement alternatives are also demonstrated.
- TES performance evaluated according to the first and second law of thermodynamics.

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ABSTRACT

Compressed air energy storage (CAES) represents a very attracting option to grid electric energy storage. Although this technology is mature and well established, its overall electricity-to-electricity cycle efficiency is lower with respect to other alternatives such as pumped hydroelectric energy storage. A meager heat management strategy in the CAES technology is among the main reasons of this gap of efficiency. In current CAES plants, during the compression stage, a large amount of thermal energy is produced and wasted. On the other hand, during the electricity generation stage, an extensive heat supply is required, currently provided by burning natural gas. In this work, the coupling of both CAES stages through a thermal energy storage (TES) unit is introduced as an effective solution to achieve a noticeable increase of the overall CAES cycle efficiency. In this frame, the thermal energy produced in the compression stage is stored in a TES unit for its subsequent deployment during the expansion stage, realizing an Adiabatic-CAES plant. The present study addresses the conceptual design of a TES system based on a packed bed of gravel to be integrated in an Adiabatic-CAES plant. With this objective, a complete thermo-fluid dynamics model has been developed, including the implications derived from the TES operating under variable-pressure conditions. The formulation and treatment of the high pressure conditions were found being particularly relevant issues. Finally, the model provided a detailed performance and efficiency analysis of the TES system under charge/discharge cyclic conditions including a realistic operative scenario. Overall, the results show the high potential of integrating this type of TES systems in a CAES plant.

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1. Introduction

Currently, the worldwide installed capacity for electrical energy storage (EES) is dominated by pumped hydroelectric energy storage (PHES). In 2015, with 145 GW installed, PHES represented about 97% of the global EES capacity [1]. The power ratings of the existing PHES plants are in the range of 1 MW up to 3 GW with a cycle efficiency of 70–85% [2]. Despite PHES is a well-known, mature and efficient solution it has also some major limitations such as: applicability limited to suitable locations and relatively low energy density, which translates into a considerable environmental impact.

In the field of large-scale EES, a valid alternative to PHES is represented by compressed-air energy storage (CAES). CAES plants operate on a “decoupled” Brayton cycle. During electric energy...
storage, the air compression operation occurs and electricity is absorbed from the grid to activate a motor-compressor train (see Fig. 1). The thermal energy produced during compression is removed by means of intercoolers and after-coolers and the high-pressure low-temperature air is then stored in a large air reservoir, usually a cavern. When electric energy is requested from the grid, the compressed air is extracted from the reservoir, is flown and heated in a combustion chamber and, at high enthalpy, it is expanded in a gas turbine that drives itself an electric generator. As of today, two industrial-scale CAES plants are successfully in operation: the 321 MW Huntorf plant in Germany, and the 110 MW McIntosh plant located in Alabama, (U.S.). Commissioned at the end of 1978, the Huntorf plant is the world’s first CAES plant, whereas the McIntosh CAES plant, commissioned in 1991, can be considered as a second-generation CAES in which a recuperator is exploited to pre-heat the compressed air before entering the combustion chamber. With this enhancement the electricity-to-electricity cycle efficiencies of McIntosh reached 54% vs. 42% of the Huntorf plant [3].

Since several CAES concepts have been proposed/developed, a general classification can be based upon how thermal energy is managed during air compression/expansion stages [3]. If thermal energy is wasted during compression and provided prior to expansion by burning natural gas in a combustion chamber, the CAES concept is known as diabatic (D-CAES); Huntorf and McIntosh are D-CAES plants. Conversely, if thermal energy produced during compression is stored into a thermal energy storage (TES) system, from which is recovered before expansion, the associated CAES concept is known as adiabatic (A-CAES) [4]. The development of the A-CAES concept was the subject of the research project “ADELE” (2010–2013) [5] with the construction of the world’s first 260 MW prototype expected as outcome of the “ADELE-ING” project (2013–present) [6]. Since in the A-CAES concept there is no need of burning fuel for the air heating process before expansion, the expected round-trip efficiency is in the order of 70% [5]. As a consequence, the TES system becomes a key component for a successful commercial implementation of the A-CAES technology.

TES systems can store thermal energy in the form of sensible heat, latent heat [7] or by thermochemical reactions [8]. The large majority of the high-temperature TES systems nowadays in operation in concentrating solar power (CSP) applications [9] or industrial process heat recovery [10], store sensible heat with a two-tank storage configuration [11]. However, and by considering its high-efficiency, affordability and simplicity, the single-tank or...
thermocline storage technology represents a valuable alternative [12–14].

This work reports a detailed performance analysis of a packed bed TES system exploited to store the thermal energy produced during the compression stage of an existing CAES plant, i.e. charging operation, to be then reused to increase the enthalpy of high-pressure air prior to be expanded in the CAES plant turbine during electric energy production phase.

For this purpose, initially a single-tank solution was studied for continuous and complete charge/discharge cycles. Hereinafter, assuming this first simulation as a reference case, different TES enhancement alternatives were analysed. Among them, the impact of the tank volume size, the number of TES tanks used, and the overall heat management strategy were studied. The results of this part of the study were used to define the multiple benefits of the proposed storage technology. A pre-charge stage was analysed to possibly enhance the TES cyclic performance. Finally, the performance of the packed bed TES under partial charge/discharge operations, due to a realistic CAES daily operational scheme, was also evaluated for a month of continuous operation.

2. A-CAES plant description and TES tank characteristics

2.1. CAES plant operational parameters

As a real plant of reference for this study, the Huntorf CAES plant was chosen. It is the world’s first and longest-operating CAES facility and is located near Huntorf in Northern Germany. This plant is designed for a peak power generation of 2 h at full load with an air mass flow rate of 417 kg/s and a power of 290 MW. The full charge of the two caverns, acting as high pressure air reservoirs, occurs in 8 h with an air mass flow rate of 108 kg/s. Plant components are designed to operate with a pressure range that goes from 22 bars up to 76 bars. However, the maximum plant performance is reached when operating between 46 bars and 66 bars. Before being injected into the caverns, compressed air is cooled below 50 °C with the twofold aim of maximizing the air reservoirs storage capacity and to avoid environmental issues. The thermal energy generated during compression is then wasted. In this work, we discuss the possible recovery of this large amount of thermal energy by means of a packed bed TES system.

As a summary, in Table 1 the main parameters of the Huntorf plant, which are considered within this work, are collected.

2.2. From CAES to A-CAES: TES system configuration

The TES solution proposed in this work, to move from a CAES to an A-CAES plant is a single-tank packed bed configuration in which natural rocks are exploited as low-cost heat storage material (HSM) and air is the heat transfer fluid (HTF) (see Fig. 1).

In a single-tank packed bed TES, different mechanisms governing the heat transfer between the fluid and solid materials, and also through the fluid and solid themselves, leading to the formation of a temperature stratification zone, so-called thermocline. This region presents segregated temperature gradients and physically separates the hot region located on the top of the tank and the cold region which stays at the tank bottom. During TES charge process, the high-temperature HTF is introduced from the top; it exchanges thermal energy passing throughout the solid HSM leaving then the storage at lower temperature from the bottom. On the contrary, during the discharge process, the HTF flow direction is reversed: the high-pressure low-temperature HTF enters from the TES bottom, it passes throughout the solid HSM, recovering the thermal energy stored, leaving then the storage from the top. This high enthalpy HTF is then exploited to feed the turbine of the power block.

According to the operational parameters of the reference CAES plant (Table 1), preliminary calculations showed that a TES capacity of about 450 MWhth is required to store the thermal energy produced during the nominal 8 h of charge. A secure design up-scaling factor of 1.5 was introduced in order to account for any potential performance losses. Overall, introducing the influence of the pebble packing, considered as a close random packing, a total storage tank volume of around 3500 m^3 was obtained (see Table 2).

On the basis of previous experiences at pilot scale [13], a truncated cone shape of the TES tank was selected (see Fig. 1). The main benefits of this design are connected to the lateral wall inclination, which favours the movement of the particles inside the tank during thermal expansion. This effect promotes a better structural force distribution, decreasing both, particle-particle and particle-tank thermo-mechanical stresses reducing, as a consequence, the thermal ratcheting [15–17].

Considering the dimensions of the Huntorf plant, a suitable single-tank TES should have a very large volume. Therefore, the possible division of the total storage volume in different tanks operated in parallel was also considered. Three different arrangements were proposed: single (1-tank), double (2-tank) and quadruple (4-tank) storage tanks solutions. Table 2 collects dimensions together with fluid flow rates associated to each of these TES systems configurations.

Tank structures are assumed to be made of low density concrete with the conical walls having 0.8 m of thickness. To minimize thermal losses, a “sandwich” of two insulating materials is rolled all around the conical walls; the inner layer of 0.24 m thickness is a commercial micro-porous insulating material, namely Microtherm®, followed by a 0.4 m thick layer of Foamglas®.

3. Model description

3.1. Momentum equation

The interstitial fluid movement through the packed bed was modelled using the commercial computational fluid dynamics (CFD) software ANSYS-Fluent v.16.2. In particular, the porous
media formulation was applied. The porous media model considers
the domain as a continuum adding a source term ($S_i$) to the
momentum equation for describing the interference of the packed
bed on the fluid flow:
\[
\frac{d}{dt}(\rho_f \overline{v}) + \nabla \cdot (\rho_f \overline{v} \overline{v}) = -\nabla p + \nabla \cdot (\overline{\tau}) + \rho_f \overline{g} + S_i
\]  
(1)
where $\rho_f$ is the fluid density, $\overline{v}$ the fluid velocity, $p$ the static
pressure, $\overline{\tau}$ the stress tensor and $g$ the gravitational acceleration
vector.

The source term added to Eq. (1), generates a pressure gradient
proportional to the fluid velocity. In the case of homogeneous and
isotropic porous media, as the packed bed subject of this study, the
source term is given by the following equation: m:
\[
S_i = -\frac{\mu}{\sigma} \overline{v} + C_2 \frac{1}{2} \rho_f |\overline{v}| \overline{v}
\]  
(2)
where $\sigma$ is the permeability of the medium and $C_2$ the inertial
resistance factor.

3.1.1. Evaluation of the pressure drop

For the calculation of both the permeability ($\sigma$) and the inertial
resistance coefficient ($C_2$), the Ergun’s semi-empirical correlation
was used [18]:
\[
\frac{\Delta p}{L} = \frac{150 \mu}{d_p^2} \left(1 - \frac{e^3}{e^2}\right) v_e^2 + \frac{1.75 \rho_f}{d_p} \left(1 - \frac{e^3}{e^2}\right) v_e^3
\]  
(3)

This approach was chosen due to its validity for both, a wide
range of Reynolds number and different porous media conformation
in terms of particles arrangement.

Comparing the Ergun Eq. (3) with Eq. (2), the permeability coeffi-
cient might be calculated as:
\[
\sigma = \frac{d_p^2 e^3}{150} \left(1 - \frac{e^3}{e^2}\right)
\]  
(4)
and the inertial resistance coefficient as:
\[
C_2 = \frac{3.5 \left(1 - \frac{e^3}{e^2}\right)}{d_p}
\]  
(5)

In this work, the dependency of $\sigma$ and $C_2$ coefficients on the
packed bed void fraction was explicitly considered (see Section 3.3).
This sub-model was coded in “C” language and coupled as “user-
defined function” (UDF) with the CFD solver.

3.2. Energy equation

The heat transfer modelling in packed bed systems can be
addressed by means of different simulation strategies [19]. Among
them, the continuous media approach implemented in this work
offers a good compromise between the calculation accuracy and
computational effort. This method considers the packed bed as a
continuous porous media without including local effects derived
from the individual particles shape and packing.

For the heat exchange between air and rocks, the local thermal
equilibrium (LTE) model was applied. In this approach, solid and
fluid phases have identical volume averaged temperatures. The
LTE model is valid provided that the solid thermal conductivity is
much larger than that of the fluid [20,21]. As a consequence, the
model requires just one energy equation [22]:
\[
\frac{d}{dt}(\varepsilon \rho_f e_f) + (1 - \varepsilon)(\rho_f e_f) + \nabla \cdot (\overline{\rho} (\rho_f e_f + p)) = \nabla \cdot (k_{eff} \nabla T) + S_i
\]  
(6)

3.2.1. Evaluation of the effective packed bed thermal transport

The modelling of the heat transfer phenomena at high temper-
atures in packed beds involves complicated mechanisms and contribu-
tions (conductive, convective and radiative) [23] that are not
directly accounted for in the energy conservation equation (Eq.
(6)). In order to include all of them in a single term, typically a
modified effective thermal conductivity ($k_{eff}$) is applied. In this
work the inclusion of all the representative heat transport contri-
butions in the packed bed system is performed according to the
semi-empirical models developed by Yagi & Kunii [23], Kunii &
Smith [24] and Wakao & Kaguedi [25].

The dependency on the material properties and on the packed
bed void fraction was introduced in the CFD model by means of a
purpose-built UDF.

3.3. Packed bed void fraction

In this work the void fraction of the packed bed is considered
variable from a value of 0.37 at the top of the vessel to 0.345 at
the bottom. This variation is modelled with a second order poly-
nomial [13,26]. In the radial direction, instead, as the bottom and the
top tank diameters (see Table 2) are much larger than the average
particles diameter (3 cm), the variation of the void fraction can be
considered negligible. In fact, under these conditions, preferential
paths of the HTF close to the wall, the so called wall-effect or chan-
nelling, can be neglected because they do not affect the system
performance.

4. Solution and data treatment

4.1. Materials properties

The HTF considered in this study is air with thermo-physical
properties assumed as a function of temperature and pressure
(see Fig. 2). The air density was evaluated with the Peng-
Robinson equation of state [27]. Viscosity and thermal conductivity
values show a negligible dependence on pressure. Consequently,
these properties were considered as only dependent on tempera-
ture (see Fig. 2b and d). The air specific heat at different pressures
(46 bars, 56 bars and 66 bars) shows a noticeable spread at tempera-
tures below 200 °C, vanishing at higher temperatures. In order to
simplify the modelling strategy without losing the required accu-
racily, an average specific heat, corresponding to 56 bars, was then
considered.

Fig. 3 and Table 3 summarize the thermo-physical properties of
the TES solid HSM, i.e., rocks, together with the materials used for
the tank structure and for the insulation composite layer (see the r.
h.s. of Fig. 4). In the case of concrete and insulating materials, all
the thermal properties are assumed constant except made for
Microtherm® specific heat that shows a noticeable temperature
dependency.

4.2. Computational domain and boundary conditions

In the definition of the computational domain, the axial symme-
try of the TES was exploited and, hence, a two-dimensional
axisymmetric mesh was considered (see Fig. 4). In all the calcula-
tions, the computational domain was exclusively limited to the
packed bed region and a plug-flow was considered in the inlet sec-
tion of the tank.

For both the charge and the discharge processes, the values of
HTF mass flow rate, inlet temperature and duration are reported in
Table 1. For the modelling of the thermal losses through the lat-
eral wall an effective convective heat transfer coefficient was con-
sidered. The numerical calculation of this parameter was obtained
by means of a mixed convection/conduction heat transfer approach in a multi-layered wall [28]. Its quantification is done through the Newton’s law of cooling.

The relatively low velocities of the fluid throughout the packed bed (the Mach number is always well below 0.3), allow for an “incompressible flow” solution approach.
In the description of the fluid flow regime, the $Re_r$ criterion was used [29]:

$$Re_r = \frac{v_{sup} r_0 \rho_l}{\mu}$$  (7)

Following this criterion, the transition from laminar to turbulent flow regime occurs when $Re_r$ is of order $10^2$. Taking into account that, under the most unfavorable operating conditions $Re_r$ is below $10^2$, the flow throughout the packed bed was assumed to be laminar.

All CFD simulations were time accurate. At $t = 0$ s, the TES tank was assumed at ambient temperature 20 °C and at an initial pressure of 46 bar. In addition, a time step of 2.5 s was considered for both charge and discharge processes. This time step was selected according to an optimized compromise between computational time and accuracy.

4.3. Numerical details

The numerical solution of the above-mentioned equations was accomplished using the “pressure based” solver implemented in the CFD model [30]. In this approach, the pressure is derived from the continuity and the momentum equations so that the velocity field, corrected by the pressure, satisfies the continuity. For the coupling of the pressure and velocity equations, the Pressure-Implicit with Splitting of Operators (PISO) algorithm was selected [31].

The Green-Gauss node-based algorithm was used for the spatial discretization due to its higher accuracy in comparison to cell-based models. In addition, the pressure staggering option (PRESTO!) scheme [31] was applied for the pressure spatial discretization and the momentum and energy equations were solved with a second order accurate numerical scheme (Second Order Upwind).

Convergence criteria were strengthened in order to ensure the good accuracy of the calculation. Namely, for the continuity, radial and axial velocities residuals an accuracy of $10^{-5}$ was fixed while, in the case of the energy equation, the residual threshold was set to $10^{-8}$.

Mesh sensitivity and optimization analyses were performed to determine the best calculation economy together with the maximization of the accuracy. After implementing different alternatives, a uniform mesh of quadrilateral cells was selected. The resulting number of cells, for each of the TES configurations described in Section 2.2, is reported in Table 4 along with the computational CPU time. All calculations were performed on a Linux based Workstation equipped with 32 cores Intel® Xeon® and 64 GB of DDR4 RAM.

4.4. Model validation

In order to demonstrate the accuracy of the described model, the experimental results published by Zanganeh et al. [13] were used. The mentioned experimental setup consists of a 6.5 MWh truncated-cone shape packed bed TES unit filled with natural rocks and operated with air as HTF.

Model validation against these experimental data is shown by Fig. 5, which reports the temperature evolution with time in four measurement positions located along the TES height (T1 to T4, from bottom to top).

Fig. 4. Schematic of the computational domain with boundary conditions implemented for charge (l.h.s.) and discharge (r.h.s.) operations respectively.

Table 3
Solid materials density.

<table>
<thead>
<tr>
<th>Material</th>
<th>Density (kg/m³)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Rocks</td>
<td>2635</td>
</tr>
<tr>
<td>Microtherm®</td>
<td>250</td>
</tr>
<tr>
<td>Foamglas®</td>
<td>140</td>
</tr>
<tr>
<td>Concrete</td>
<td>900</td>
</tr>
</tbody>
</table>

Table 4
Mesh elements and required computational times.

<table>
<thead>
<tr>
<th>System</th>
<th>Mesh elements</th>
<th>Charge</th>
<th>Discharge</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Computational time (h)</td>
<td>Iterations per $t_{seg}$</td>
</tr>
<tr>
<td>1-tank</td>
<td>115304</td>
<td>80</td>
<td>32</td>
</tr>
<tr>
<td>2-tank</td>
<td>65012</td>
<td>54</td>
<td>31</td>
</tr>
<tr>
<td>4-tank</td>
<td>49296</td>
<td>34</td>
<td>32</td>
</tr>
</tbody>
</table>
and discharging ($\Delta H_d$) can be calculated by means of the following expressions:

$$\Delta H_c = \int_0^{t_f} (m_c c_p (T_{1,c} - T_{amb})) \, dt$$  \hspace{1cm} (8)

$$\Delta H_d = \int_0^{t_f} (m_d c_p (T_{1,d} - T_{2,d})) \, dt$$  \hspace{1cm} (9)

where $T_{1,c}$ and $T_{amb}$ correspond to the nominal charging temperature (500 °C) and to the ambient temperature (20 °C) respectively; $T_{1,d}$ represents the temperature of the released HTF from the TES tank during the discharge operation and $T_{2,d}$ the nominal HTF outflow temperature during discharging (20 °C).

It has to be mentioned that thermal losses through lateral wall and the effects due to thermocline (temperature stratification) region formation inside the solid media are implicitly included in $\Delta H$ balances.

Therefore, the thermal efficiency of a complete cycle (charge + discharge) can be calculated as:

$$\eta_t = \frac{\Delta H_d}{\Delta H_c}$$  \hspace{1cm} (10)

4.5.2. Second law efficiency

The fluid entropy in the charge and discharge operations is evaluated from the following equations:

$$\Delta S_c = \int_0^{t_f} m_c \left( c_p \ln \frac{T_{1,c}}{T_{amb}} - R \ln \frac{p_{1,c}}{p_{2,c}} \right) \, dt$$  \hspace{1cm} (11)

$$\Delta S_d = \int_0^{t_f} m_d \left( c_p \ln \frac{T_{1,d}}{T_{2,d}} - R \ln \frac{p_{1,d}}{p_{2,d}} \right) \, dt$$  \hspace{1cm} (12)

where $p_1$ and $p_2$ correspond to the pressure in the upper and lower part of the tank, respectively.

4.5.3. Exergy

To discuss the quality and usability of the stored energy the following exergetic ($\Delta B$) criterion is selected:

$$\Delta B = \Delta H - T_{amb} \Delta S$$  \hspace{1cm} (13)

Considering a complete charge/discharge cycle, the process exergy efficiency can be calculated from:

$$\eta_B = \frac{\Delta B_d}{\Delta B_c}$$

5. Results and discussion

5.1. Reference case: single-tank packed bed TES

In this section the thermal performance of a single-tank TES, subjected to a total of 20 consecutive charge/discharge cycles, is evaluated. In Fig. 6 the resulting HTF outflow temperature, during charging (l.h.s) and discharging (r.h.s), is presented. Fig. 6a clearly shows that the HTF outflow temperature remains constant at 20 °C throughout the first 8 h of charging. However, from the second cycle this temperature starts increasing gradually reaching a stable thermal behaviour after 14–15 cycles. Under these conditions, the air outflow temperature starts to increase after 5 h reaching a temperature of about 150 °C at the end of charge. It should be noted that one of the operational requirements of the CAES plant is that the compressed air has to be stored at temperatures not higher than 50 °C. As a consequence, these results show the need of an
air cooler to be installed between the packed bed TES system and the cavern.

Regarding the discharge operation, Fig. 6b shows that during the first run, the HTF outflow temperature drops down to 380 °C after 120 min of discharging. This result reveals that the HTF outflow temperature does not fulfill the operational requirements of the power block, i.e. air temperature must stay above 480 °C during the 2 h of discharging. However, the final HTF outflow temperature gradually increases with cycles leading to the achievement of the required performance after seven cycles.

The physical cycling behaviour observed in this system is related to two different phenomena, which are characteristic of packed bed TES. On one hand, the thermal performance is essentially driven by the axial temperature stratification due to buoyancy. On the other hand, it is associated to the different initial conditions obtained for each charge/discharge cycle. In fact, the cyclic performance variation occurs due to the amount of energy left inside the packed bed TES after each discharge. As a result, the thermocline region is gradually moved from the top to the bottom part of the TES leading to its partial ejection, as showed by the large increase of air outflow temperature during charging. In any case, after an enough number of charge/discharge cycles, a stable thermal stratification into the packed bed is attained.

Fig. 7 shows the axial temperature profile once the 1st, 5th, 10th, 15th and 20th charge (continuous lines) and discharge (dashed lines) processes are completed. In this figure, the 0 and 1 values of the abscissae correspond to the upper and lower parts of the packed bed respectively. The continuous lines show how, with the charge operations, the thermocline spreads and moves further towards the bottom part of the packed bed. In the discharge process a similar behaviour is observed.

From Fig. 7 a shape variation of the thermocline region can be observed. Comparing the temperature profiles of the same cycle once the system is charged and discharged, it is observed that the slope of the temperature gradient is noticeably different. This slope is much higher in the discharge profiles than in the charge ones, what indicates a thinner thermocline region and hence, better thermal stratification during discharging than the one during charging. These variations are mainly related to the different fluid velocities; in fact, the discharge mass flow rate is four times the one of the charge. Furthermore, the conical shape of the investigated TES tank is also playing a role: this geometry leads to a non-uniform axial distribution of the solid material mass.

### 5.2. Effect of implementing a multiple tanks TES configuration

In this section, the 1-tank TES system was considered as a reference case and compared to a multiple tanks TES solution. The latter consists in splitting the required storage physical volume in several tanks of smaller size. In this study, 2-tank and 4-tank configurations were investigated to determine the viability and the technical impact of the proposed splitting. In all the presented multi-tank cases, the tanks are operated in parallel (see Fig. 8 for the proposed tank arrangements and Table 2 for the associated dimensions).

In Fig. 9 the HTF outflow temperature, during charging and discharging, for the two multiple tanks configurations is presented and compared with the reference 1-tank arrangement. For the sake of clarity, only the HTF outflow temperature as a function of time corresponding to the 1st, 5th, 10th, 15th and 20th cycles are included. In Fig. 9, the dashed-dotted lines correspond to the...
1-tank configuration, the dashed curves to the 2-tank system and the continuous lines to the 4-tank system.

As it can be observed, overall, the performance of the 2-tank and 4-tank configurations is very similar to the 1-tank system. Although the air outflow temperature stays below 75 °C for the first 5 charge processes, a gradual HTF outflow temperature increase can be seen with further cycling. A maximum outflow temperature of 150 °C was obtained after the 20th cycle together with a stable thermal stratification into the packed bed. On the other hand, during discharge processes, the final air outflow temperature of the first discharge drops 140 °C below the nominal charging temperature (500 °C) and only after almost 10 cycles it matches the requirements of the power block throughout the whole discharge process.

Fig. 10 shows the transient evolution of the energy and exergy efficiencies of the three TES configurations considered. The efficiency values increase gradually with cycles up to the stable cyclic performance where a maximum efficiency around 96% is reached. The slightly lower values obtained in the first cycles can be explained with the internal energy variation of the solid materials constituting the TES tank, i.e. concrete and insulating materials together with the exergy deterioration due to the formation of the thermocline.

The small difference observed between the absolute values of the energy and exergy efficiencies is indicative of the low exergy destruction in the operation of the TES. This small difference is related to the fact that, the energy balance is exclusively limited to the energy introduced/stored in the TES tanks. This excludes from the balance any other energy consuming system needed for the proper operation of the proposed technology, such as fluid distributors, piping, pumps and any other ancillary apparatuses.

Even if the 1-tank configuration showed a slightly better behaviour in comparison to the 2-tank and 4-tank arrangements (see Figs. 9 and 10), the differences are very small to allow discarding any of the proposed systems only due to its energetic or thermal performance. Therefore, construction and mechanical aspects should be also considered when looking for the best design solution. Taking these elements into account, the reduction of the tank size could present important benefits on the minimization of usual construction/mechanical problems when dealing with packed bed TES systems. Among them, the ratcheting, the container mechanical stress management and the homogenization of the fluid flow through the storage tank cross section. When these factors are all taken into account, the 4-tank configuration becomes the most interesting solution and, as a consequence, it was selected for further analysis within this work.

5.3. Effect of an initial TES pre-charge

The enhancement of the thermal behaviour during the transient stage of the TES is addressed in this section. With this purpose, the quality of the heat released during the discharge operation of the 4-tank configuration was analysed in detail under continuous thermal cycling.
A possible solution to boost the TES performance from the first cycle is to perform a preliminary partial charge of the system before the regular 8 h charge. According to the terminology used in Ref. [32], this procedure is referred to as “pre-charge” in this work. During the TES pre-charge in an A-CAES plant, the high-pressure air leaving the TES is released into the atmosphere instead of being stored into the high-pressure air reservoir, in order not to increase the cavern pressure during the pre-charge procedure.

As an example, in this section, the impact of a pre-charge stage of 6 h using pressurized air at 46 bars and 500 °C is analysed in detail. In Fig. 11, the transient HTF outflow temperature during charging is presented for both the TES system with (Fig. 11a) and without (Fig. 11b) pre-charge respectively. From Fig. 11a, it can be observed that during the pre-charge stage no high-temperature fluid is ejected from the TES unit. It is after two hours of the 1st charge, that the air outflow temperature increases reaching a maximum value of around 415 °C at the end of the process. During consecutive charge processes, the final HTF outflow temperature gradually decreases and it stabilizes around 150 °C after 12–13 cycles.

In the case of the TES system without pre-charge (Fig. 11b), an opposite trend is observed. In this case, the air outflow temperature during the first charge does not experience a noticeable increment. With the cycles, it starts increasing and a stable cyclic behaviour, with a maximum air outflow temperature of 150 °C, is reached after 17–18 cycles. It can be pointed out that when a stable thermal stratification into the packed bed is achieved, both TES systems, with and without pre-charge, show similar HTF outflow temperature behaviour as a function of the charge time.

Fig. 12 represents the air outflow temperature, as a function of time, during the discharge processes in the case of TES system with (Fig. 12a) and without (Fig. 12b) pre-charge respectively. When looking at the first three discharge processes in Fig. 12a, an almost constant HTF outflow temperature is obtained. However, due to the temperature stratification evolution with cycles, from the fourth cycle, a slightly larger temperature drop is obtained after
Anyway, after 6–7 cycles a stable thermal stratification into the packed bed is obtained. Under these conditions the HTF outflow temperature does not decrease more than 10 °C with respect to the nominal charging temperature.

On the other hand, the TES system without pre-charge shows a different thermal behaviour (Fig. 12b). In this case, during the first discharge, the temperature decreases down to 140 °C at the end of the process. This behaviour improves with cycles until, after 17–18 cycles, the TES reaches a stable thermal stratification. In these conditions, the maximum HTF outflow temperature decrease, at the end of discharging, is about 10 °C.

Comparing the charge performance in both operational approaches, it can be concluded that performing a pre-charge stage has a strong impact on the quality of the released heat during the first cycles. A very important consequence of the pre-charge is the possibility of using the stored energy in the power block (air at temperature higher than 480 °C) from the very beginning of the TES thermal cycling. Without pre-charge, the released air would satisfy the power block requirements only after eleven cycles.

In addition to the air outflow temperature variation, the impact of the pre-charge on the energetic and exergetic efficiencies was quantified. Fig. 13 shows the efficiency values with and without the initial pre-charge. It has to be pointed out that, for this part of the study, the energy and exergy associated to the pre-charge process "per se" were not considered; therefore, the data represent exclusively the efficiencies of the TES cyclic behaviour. It must also be noted that, even though the amount of energy involved in the pre-charge could be relevant, this value remains negligible if compared with the total energy introduced/released into/from the TES in the subsequent cyclic operations. As shown in Fig. 13, the pre-charged system leads to constant efficiency values after 2–3 charge/discharge cycles. Under stable cyclic conditions, a maximum efficiency value of 96% was obtained for both energy and exergy.

![Fig. 12. Air outflow temperature during discharge processes: (a) system with 6 h of pre-charge and, (b) system without pre-charge.](image-url)

![Fig. 13. Comparison of the standard (red line) and pre-charged (blue line) TES system energetic (a) and exergetic (b) cycle efficiency. (For interpretation of the references to colour in this figure legend, the reader is referred to the web version of this article.)](image-url)
On the contrary, in the TES system without pre-charge, the transient efficiencies show a lower value. This trend is asymptotically extended with the cycling up to a steady cycling performance is achieved and a maximum efficiency value of around 96% is obtained.

On the basis of the results presented in this section, it can be concluded that performing a TES pre-charge at start-up is highly recommended [32]. In particular, this operation presents two clear benefits: (i) the air outflow temperature obtained during the complete discharge fits the requirements of the power block from the first cycle; (ii) the transient behaviour of the TES system is noticeably shortened.

5.4. Effect of daily cycling in the TES performance

So far, in this work, in order to obtain a clear understanding of the proposed storage system, continuous and complete charge/discharge operations were considered. However, in a real CAES plant, the operating conditions are often far from this ideal behaviour. For this reason, in this section the charge/discharge of the plant is programmed following a realistic CAES plant load scenario. In particular, during the hours with lower electricity price (valley hours) the charge process is performed. The discharging is instead matched with the main energy consumption peaks (i.e., morning, mid-day and evening), corresponding also, in a simplified energy price approach, to the higher electricity price periods. This particular storage strategy is depicted in Fig. 14. Starting from a complete charged initial condition at the beginning of the day (6:00), two partial charge (7:30–12:30 and 13:50–20:00) and partial discharge (6:00–7:30 and 12:30–13:50) processes are programmed. Then, the stored energy is released in a full discharge from 20:00 to 22:00, followed by a complete charge process from 22:00 until next morning at 6:00. It must be noted that, exception made for the charge/discharge duration, the rest of the operational parameters are unchanged (see Table 1). A 6 h TES pre-charge was assumed before the beginning of the cyclic operations.

To draw conclusions about the long-term cyclic behaviour of the TES under realistic load operation, a complete month (31 days) was modelled.

The HTF outflow temperatures as a function of time are presented in Fig. 15. Looking at the curves related to the night charge processes (Fig. 15a), it can be seen that, during the first day, the
The TES thermal behaviour during charge/discharge cycles, allows to conclude that this system works properly not only for continuous and complete charge and discharge operations, but also for partial charge and discharge cycling conditions. Furthermore, the TES performance, during charging and discharging, are only slightly dependent on the particular operation strategy.

6. Summary and conclusions

The performance of a packed bed TES system integrated in a CAES plant was analysed in detail. This proposed solution is known as adiabatic-CAES (A-CAES). In an A-CAES plant, the TES system allows to store the thermal energy recovered from the air compression stage and to, use it to increase the enthalpy of the high-pressure air prior of its expansion in the power block. With this enhancement, the electricity-to-electricity cycle efficiency of an A-CAES plant is higher than that of a CAES plant. Theoretically, it can arrive up to 70%.

To carry out this analysis, a CFD model was developed to evaluate the performance of the packed bed TES systems under investigation. This model, based on a porous media approach, includes the laminar fluid flow governing equations. All the heat transfer mechanisms (conduction, convection and radiation), occurring into the packed bed, are accounted for with a lumped parameters model and an effective thermal conductivity. Furthermore, being the TES operated under variable pressure conditions, the model also accounts for pressure dependency of all relevant variables.

Considering the novelty of the A-CAES technology, which represents a clear progress on the state of the art CAES systems, different potential implementation scenarios were discussed. For all the cases, the operating conditions of the Huntorf CAES plant were selected as reference.

The implementation of a single or a multiple tanks TES configurations was analysed. The results show that splitting the single tank in smaller TES units, operating in parallel, does not present any remarkable implication on the storage performance. On the basis of the results, some important considerations can be drawn:

(i) a certain number of consecutive charge/discharge cycles must be performed before the stored energy can be considered useful to feed the power block in the discharge process. This number of consecutive cycles depends on the initial conditions of the TES system at start-up.

(ii) The HTF outflow temperature during charging resulted to be lower than the maximum threshold value (50°C) in the first cycles. However, it sensibly increases during consecutive cycles indicating the need of a further cooling of the HTF before entering the cavern.

To enhance the TES performance, a partial charge of the TES, the so called pre-charge, was proposed and evaluated. The main effects of including a pre-charge can be summarized as:

(i) the HTF outflow temperature fits the requirements of the power block during the complete discharge from the very first cycle.

(ii) Although the TES can be effectively exploited from the first cycle, it still needs a certain number of consecutive cycles before achieving a stable thermal stratification into the packed bed.

The work was completed with the application of the proposed TES solution under partial charge/discharge conditions. A realistic daily cyclic scenario was implemented and the TES performance was evaluated covering one month of continuous operation. As a result, the reliability, high efficiency and stability of the proposed TES system were pointed out. In fact, it was found that, after a short initial transient, a stable thermal stratification into the packed bed can be obtained. The TES system operates at very high efficiency with the final HTF outflow temperature during discharging suitable for feeding the power block. Therefore, these results allow confirming that a packed bed TES system can be successfully integrated in a real CAES plant leading to a substantial increase of its overall efficiency.

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