Adiabatic compressed air energy storage – a study on dynamic performance with thermal energy storage

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Abstract:

Up to day, 70\% of the gross electricity (~ 20000 TWh) is produced via fossil fuel; to reduce this share and abate CO2 emissions the energy system will face a dramatic change in the near future consisting in a significant introduction of intermittent renewable energy sources. The implementation of this step change will be possible only through efficient and economic grid-scale electrical energy storage. In this paper we study the operation of a 220MWe adiabatic compressed air energy storage (ACAES) plant with a packed bed thermal energy storage. Although in the literature designs of thermal stores for ACAES have been presented, there is a lack of studies addressing how the TES system affects the performance of ACAES plant. This work contribute to fill this gap in the literature by linking in a quantitative manner the TES sizing and ACAES global efficiency. We achieve this goal via an algebraic-differential model that account both dynamic and off-design performance of ACAES plant's components. The results shows that after 30 charge/discharge cycles the thermal store reaches cycling; this enable the A-CAES to achieve a round trip efficiency of 70\%.

Keywords: Energy Storage, Thermal Storage, Compressed Air, Grid scale

2. Introduction

Overcome our current fossil fuels based energy scenario is imperative as CO2 emissions and global warming are already taking their toll on our society and planet Earth [1]. To contain global warming below 2°C carbon dioxide emission must decrease by 90\% by 2050 through an intense penetration of renewable resources which could reach a global share of 65\% according to scenarios forecasted by IEA [2]. This great potential can be untapped only if the intrinsic variability of renewables, such wind and solar energy, is mitigated through energy storage (ES). ES technology provides several functions to facilitate the use of renewables: it enables to capture “wrong time” energy and make it available when needed, it helps to shave and shift load peaks, and it improves reliability of energy systems [3,4].

Alongside with pumped hydroelectricity storage, compressed air energy storage (CAES) is among the few grid-scale energy storage technology with power rating of 100s MW [5,6]. CAES operates in such a way that electrical energy is stored in form of compressed air confined in a natural or artificial reservoir. Then, during periods of high energy demand, stored energy is retrieved by withdrawing high pressure air and expand it through a series of turbines to generate electricity. Traditionally, for example in the Huntorf plant [6,7], before expansion air is heated in a combustion chamber burning conventional fossil fuels. This leads to two drawbacks: CAES is not CO2 free and round trip efficiency is limited to 40-50\% [5,6]. To overcame such disadvantages adiabatic compressed air energy storage (A-CAES) has been proposed. Instead of burning fuel, in A-CAES the heat generated by compression is stored in a Thermal Energy Storage (TES) and then used to heat air from the reservoir before it enters the turbines [6,8]. The vast majority of the studies on A-CAES consider indirect heat exchangers (HEXs) and a separate thermo-fluid to store the heat of
compression. Another A-CAES configuration proposed uses a solid medium, typically natural rocks, to store the heat of compression [6,9]: during A-CAES charging heat is stored by flowing hot air from compressors through a packed bed of rocks; when discharge occurs air from the cavern flows through the packed bed retrieving the heat previously stored and then expands though turbines train to generate electricity. In this study we focus on the dynamic performance of A-CAES plant with an integrated packed bed thermal energy storage system. We developed a model that details the transient features of the thermal store, the cavern, and compression/expansion stages. This allows to link the performance of the components to the performance of the whole A-CAES plant. Such device-to-plant link is crucial: only through integration of TES in the whole A-CAES system is possible to assess the benefit and added value of thermal energy storage. To the authors’ knowledge such link between components performance and A-CAES plant performance has been marginally addressed in the literature. Hence the motivation for this work.

2. System description and mathematical model

Figure 1 presents the adiabatic compressed air energy storage system (A-CAES) studied in this work. Table 1 summarizes the major features of the A-CAES plant. A packed bed thermal energy storage (TES) ensures the “adiabatic” conditions: after HPC compression stage hot air flows through the packed bed and exchanges heat with the gravel contained in the TES. The gravel acts as sensible storage material and captures heat for later purposes. Air leaves the TES system near ambient temperature and enters the cavern at high pressure. A similar plant configuration is also considered by RWE Power in the EU project “ADELE” [10].

An inter-refrigeration heat exchanger cools air flow before it enters the high pressure stage. This configuration is considered in to prevent excessively high air temperature at the outlet of HPC. Compressors operate for a range of compression ratios because air pressure in the cavern spans the range 46 to 72 bar, which is the typical range adopted for the Huntorf plant and Machintosh plant [6,7]. During discharge process energy is retrieved by withdrawing air from the cavern at high pressure and expand it through the train of turbines. We considered constant inlet pressure mode: as depicted in Fig. 1, a throttling system maintains constant the turbine inlet pressure. Such an operating mode allows to operate the turbine train at constant expansion ratio and near to design conditions – thus at maximum efficiency – for the entire discharge process. Design expansion ratio (Table 1) for HPT and LPT were chosen as the one for existing CAES plants [7].

For the purpose of simulation of A-CAES plant operation we considered n equal cycles of 10 hours charge, 4 hours discharge and 10 hours idle, as shown in Figure 2. Such a figure present the nominal cycle with constant power input during charge and constant power output during discharge. The actual profile of each cycle was determined through the simulations performed as detailed in the Results section. The nominal profile of Fig. 2 was chosen considering the A-CAES plant operating for peak shaving, minute reserve, or compensation of fluctuation in wind power. Such operation modes are typical of existing CAES plants [6,7], and present discharge time of 3-4 hours, as in the case of Fig. 2. The total number n of cycles considered in the study was 30.

Table 1. Major parameters of A-CAES system

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ambient temperature</td>
<td>293.15 K</td>
</tr>
<tr>
<td>Ambient pressure</td>
<td>1.01325 bar</td>
</tr>
<tr>
<td>Expansion train rated power</td>
<td>220 MW</td>
</tr>
<tr>
<td>HP turbine design inlet temperature</td>
<td>905.15 K</td>
</tr>
<tr>
<td>HP turbine design inlet pressure</td>
<td>46 bar</td>
</tr>
<tr>
<td>HP turbine design expansion ratio</td>
<td>4.18</td>
</tr>
<tr>
<td>LP turbine design inlet temperature</td>
<td>655.15 K</td>
</tr>
<tr>
<td>LP turbine design inlet pressure</td>
<td>11 bar</td>
</tr>
<tr>
<td>LP turbine design expansion ratio</td>
<td>11</td>
</tr>
</tbody>
</table>
Turbines design efficiency 88%
Compression train rated power 100 MW
HP compressor design inlet temperature 480.15 K
HP compressor design compression ratio 8.4
LP compressor design compression ratio 8.4
Cavern volume 230 000 m³
Cavern min/max pressure 46/72 bar
Cavern wall heat transfer coefficient $0.02356 + 0.0149|\dot{m}_{\text{in}} - \dot{m}_{\text{out}}|^{0.8}$

Figure 1: Adiabatic compressed air energy storage (A-CAES) plant with sensible thermal energy storage.

Figure 2: Charge and discharge cycle.
2. Mathematical model

This section presents the mathematical models for the components of the A-CAES plant depicted in Fig. 1. Each model is first presented separately along with the underlying assumption adopted in the study. Where not stated explicitly the modelling was performed in Matlab/Simulink 2014 [11].

2.1. Compressors and turbines

Modelling of compressors turbine stages involves mass and energy balance to compute temperature of air exiting each stage and the compression work. Isoentropic air outlet temperature from compressors was computed as:

\[ T_{c_{out}}^{is} = T_{c_{in}}(k \beta_i k-1) \]  

(1)

where \( \beta_i = \beta_{HPC}, \beta_{LPC} \) is the compression ratio of each stage. Actual outlet temperature \( T_{c_{out}} \) was obtained using compressor isoentropic efficiency defined as:

\[ \eta_c = \frac{T_{c_{out}}^{is} - T_{c_{in}}}{T_{c_{out}}^{is} - T_{c_{in}}} \]  

(2)

The power of compressors consumed during charge was evaluate by an energy balance at each compressor neglecting variations in inlet to outlet kinetic energy of air:

\[ W_c = \dot{m}_c(h_{c_{out}} - h_{c_{in}}) \]  

(3)

We included off-design calculations through compressors characteristic maps that quantify compression ratios \( \beta_i \) and isoentropic efficiency \( \eta_i \) in terms of dimensionless flow rate. The characteristic maps were approximated according to [12].

High pressure and low pressure turbines were were modelled through mass and energy balance following the same approach adopted for the compressors. An improved Flugel formula [12] was used to describe the off-design performance of turbines:

\[ \frac{\dot{m}}{\dot{m}_{\infty}} = \alpha \frac{T_{r_{0,in}}}{T_{r_{out}}} \sqrt{\frac{\pi_i^2 - 1}{\pi_{r_{0}}^2 - 1}} \]  

(4)

2.2. Compressed air reservoir and packed bed thermal energy storage

We employed a dynamic model to simulate the transient behaviour of temperature of air within the cavern. The model consists in two ordinary differential equations that stem from energy balance and mass balance equations for the air in the cavern:

\[ \frac{dT_r}{dt} = \frac{1}{m_r} \left[ (1 - \frac{1}{k}) (\dot{m}_{in}T_{in} - \dot{m}_{out}T_r) + h_wA_w(T_w - T_r) \right] \]  

(5)

\[ \frac{dm_r}{dt} = \dot{m}_{in} - \dot{m}_{out} \]  

(6)

In equation (5) the first of the right hand side term accounts for energy transfer due to injection/withdraw of air form the cavern under the assumption that air leaves the cavern at the cavern’s air temperature. The second term quantifies the heat transfer between air and cavern’s walls.
We adopted a non-equilibrium model to study heat transfer within the packed bed thermal energy storage [13,14,15]; it consists in a set of two energy balance equations, the first one for the air (subscript \(a\)) in the TES while the second one for the solid filler material (subscript \(s\)):

\[
\rho_a c_{p,a} \frac{\partial T_a}{\partial t} + \rho_a c_{p,a} A T_a = k_a \frac{\partial^2 T_a}{\partial x^2} - h_v(T_a - T_s) - U_w(T_a - T_0) + \left(1 - \epsilon\right) \rho_s c_{p,s} \left(\frac{\partial T_s}{\partial t} = k_s \frac{\partial^2 T_s}{\partial x^2} + h_v(T_s - T_a)\right)
\]

(7)

(8)

Table 2 summarizes the major parameters for the model of the packed bed thermal energy storage system. The diameter \(D\) and height \(H\) of the TES system were obtained through a preliminary design on the basis of data in Tables 1 together with charge/discharge cycle of Fig. 2. Such data allows to estimate heat to be stored and thus the geometrical dimensions of the TES system.

**Table 2. Input parameters for TES model**

<table>
<thead>
<tr>
<th>Property</th>
<th>Formulation</th>
</tr>
</thead>
<tbody>
<tr>
<td>(\rho_s) (kg/m(^3))</td>
<td>2911</td>
</tr>
<tr>
<td>(c_{p,s}) (J/kg K)</td>
<td>(A(B + C T + B^2))</td>
</tr>
<tr>
<td>(k_s) (W/m K)</td>
<td>Zehner-Bauer–Schlunder model</td>
</tr>
<tr>
<td>(d_p) (m)</td>
<td>0.02</td>
</tr>
<tr>
<td>(H) (m)</td>
<td>22</td>
</tr>
<tr>
<td>(D) (m)</td>
<td>20</td>
</tr>
</tbody>
</table>

2.3. Performance indicators

Round trip efficiency and thermal storage efficiency were used to assess the performance of the whole A-CAES plant and the thermal energy storage system. The round trip efficiency for each charge/discharge cycle was calculated as:

\[
\eta_{cycle} = \frac{E_{out}}{E_{in}} = \frac{\int_0^{\Delta t_c} W_i \, dt}{\int_0^{\Delta t_d} W_c \, dt}
\]

(9)

The time integration is performed over each charging period \((\Delta t_c)\) and discharging period \((\Delta t_d)\).

The performance of the TES system was assessed through the thermal storage efficiency defined as follows:

\[
\eta_{th} = \frac{\int_0^{\Delta t_c} m_i (h_4 - h_0) \, dt}{\int_0^{\Delta t_c} m_i (h_1 - h_0) \, dt}
\]

(10)

Where \(h_4\) is the specific enthalpy of air at the outlet of the TES system during discharging while \(h_1\) is the specific enthalpy of air the inlet of TES system during charging.

3. Results

Simulations provide results for each component depicted in Fig. 1 for 30 consecutive charge/discharge cycles. A subset of these results are shown in Fig. 3 to clarify the plant operation; detailed results for plant components are presented in the following subsections. Fig. 3a plots the power input/output for the A-CAES plant. Both compressor train and expansion train operate
around the corresponding rated power (100 MW/220MW); the variations in compression power and expansion power during charge stage and discharge stage are due to off-design operating conditions which will be detailed in sections 4.2 and 4.3. Fig. 8b shows the state of charge for the thermal store and the compressed air reservoir. The stored thermal energy and the air pressure show a similar time pattern, since both follow charge/discharge cycles; 940 MWh, are stored in the TES on average while TES efficiency, as defined by Eq. (10), is 93%. Average roundtrip efficiency is 74%. Cavern pressure spans the range 48-71 bar with variation also during idle stage: after each discharge process cavern pressure increases from ~ 48 bar to about 50 bar due to heat transfer from cavern walls to compressed air, the latter being relatively cold because of the withdraw process during discharge. During idle after each charge pressure in the cavern slightly drops before the next discharge. In this case heat flows from the compressed air to the cavern wall cooling the mass of air in the cavern and thus leading to a reduction of pressure.

Figure 4 displays the round trip efficiency and thermal storage efficiency over 30 cycles. Both efficiencies reach a stable value after an initial increase during the first operating cycles. The key quantity here is the efficiency of the thermal storage system: as soon as TES starts to operate in an effective way the overall performance significantly improves, which shows how relevant is to carefully integrate thermal storage with the remaining part of the system. Maximum TES efficiency occurs when static cycling operating conditions are established in the thermal storage as explained in detail in the next section.

![Figure 3: A-CAES plant performance between 5th and 10th operation cycle. a) Compression train power during reservoir charge and turbine power output during discharge. b) Thermal energy stored in the sensible TES (left axis) and reservoir pressure variation (right axis) due to injection/withdraw of compressed air.](image-url)
Figure 4: Efficiency of A-CAES plant. a) Round trip efficiency according to Eq. (9); b) Efficiency of the thermal energy storage system (Eq. 10).

3.1. Thermal energy storage (TES) system

Figure 5 presents the temperature profile within the TES system and how the temperature profile varies from cycle to cycle. Figure 5a shows temperature after charge (t = 16h within each cycle). Two key features should be noticed: the position of the thermal front and how the temperature evolves, after a sufficient number of cycles, toward a cycling stationary profile. After cycle 1 the temperature shows a thermal front around $x = 11$ that extends for about 10% of the TES length. The ideal operation of the TES system would preserve the thermal front as sharp as possible from cycle to cycle, while each charge/discharge would consist in such sharp front travelling back and forth from $x = 0$ to $x = H$. Thermal degradation of the front [15] prevents a practical implementation of the ideal TES operation, in fact after 5 cycles thermal front broadens up to 50% of TES length. Therefore, the thermal store actually operates very similarly to a regenerator: air is gradually cooled during charge, while it is gradually heated – from TES inlet to TES outlet – during discharge. Such an operation mode leads to stationary cycling operating conditions, where two stationary temperature profiles that occur after charge and discharge (see cycle 30 in Fig. 5). Stationary profile slightly decreases from $x = 0$ to $x = 10$ m due to increase in air outlet temperature from HP compressor during charge. During discharge air withdrawn from the cavern is slightly above ambient temperature; this causes the hump at $x = 15$ m illustrated in Fig. 5b.

The time evolution of air at the outlet of TES system – corresponding $T_4$ in Fig. 1 – is presented in Fig 6. In the course of the last stage of discharge the outlet temperature drops of about 15%, as the degraded thermal front exits the thermal store. A more marked drop occurs during the first cycles because stationary temperature profile is not established yet in the TES system. Air outlet temperature from the packed bed storage coincides with the HP turbine inlet temperature; thus, any variation of $T_4$ from design point detrims the performance and efficiency of the expansion train, as explained in Sect. 3.2.
3.2. Compression train and expansion train

The performance of low pressure compressor (LPC) and high pressure compressor (HPC) significantly depart from nominal condition, as both compressors operate in off-design during charging. In fact, the pressure of air in the cavern constantly increases, bringing also an increase in the total pressure ratio experienced by the compressor train. Figure 7 lucidly summarizes how the compression train operates during charge. At the beginning of charging the LPC performs the majority of the compression work as LP compression ratio is ~ 16% larger than the HP one. As pressure in the cavern rises, both $\beta_{\text{LPC}}$ and $\beta_{\text{HPC}}$ increment up to the corresponding design values, which is achieved only at the end of charge. As charging starts $\beta_{\text{LPC}} = 7.6$ and $\beta_{\text{LPC}} = 6.5$, thus LPC compression ratio and HPC compression ratio are respectively 10% and 22% lower that the design value. As a result minimum isoentropic efficiency of compressors occurs at the begin of each charge as shown in Fig. 7a. The compression power shows a maximum around $t = 2h$ that can be
explained from the behaviour of mass flow rate and compression ratios. The combination of a decreasing trend for $\dot{m}_c$ with an increasing trend for $\beta_{LPC}$, $\beta_{HPC}$ brings a maximum in compression power $W_c$ calculated with Eq. 3. Mass flow rate monotonically decreases during charge because the operating point of compressors shifts from low compression ratios, so high mass flow rate $\dot{G}_c$, to high compression ratio and lower mass flow rate. On the other hand, compression ratio monotonically increases during charge as cavern pressure rises.

Figure 7: Compression train performance during charge. a) Compression power and isoentropic efficiency of high pressure and low pressure compressors. b) Compression ratio and air outlet temperature for high and low pressure compressors. Compressor train operates under off-design conditions except at the end of the charging process.

The expansion train operates under constant expansion ratio – due to the throttling valve (Fig. 1) – but with variable inlet temperature of air coming from the thermal energy storage system (Fig. 6). As a results departure from design condition are limited in comparison with the compression train, as illustrated in Fig. 8. Both high pressure turbine (HPT) and low pressure turbine (LPT) perform at design isoentropic efficiency for the entire discharge process. Power output drops of about 5% during discharge due to a combined effects of variation in inlet temperature (Fig. 8) and turbine mass flow rate. Although turbine mass flow rate increases during discharge the drop in inlet temperature dominates the behaviour of turbine power, resulting in a reduction of power output from the A-CAES plant. This shows how important is to conceive and operate the thermal storage system in an optimal way, as TES performance reverberate onto the global performance of the plant. Variation in turbine mass flow rate is also caused by reduction of air inlet temperature: according to the Flügel formula (Eq. 4) at constant expansion ratio we have $\dot{m}_t \propto 1/\sqrt{T_{inlet}}$. 

\begin{align*}
\dot{m}_t & \propto 1/\sqrt{T_{inlet}}.
\end{align*}
Figure 8: Expansion train performance during discharge. a) Turbine power and isoentropic efficiency of high pressure and low pressure turbine. b) Expansion ratio and air outlet temperature for high and low pressure turbine. Expansion train operates near design conditions for most of discharge process because of constant inlet pressure.

3.2. Compression train and expansion train

The operation of CAES systems for peak shaving, minute reserve, or compensation of fluctuation in wind power likely involves partial load operation during discharging [7]. The model we developed allows to study A-CAES performance for partial load operating cycle. We considered the cycle of Fig. 16 to show how partial load conditions may detriment A-CAES performance. In the view of peak shaving operation we considered a discharge cycle that last four hours (as in case of Fig. 2) but with three load rates. This mimics operating condition that may realistically occurs, as presented in [7]. Power output is regulated adjusting inlet pressure for HP turbine by throttling air flow from the cavern. Round trip efficiency reduces to 64% due to smaller power output while TES is marginally affected by partial-load operation which causes variation of air flow through the TES as detailed below. As inlet pressure varies with load HP and LP expansion ratios adjust accordingly (Fig 10). Maximum relative variation of $\pi_{HPT}$ is nearly 40% causing non-negligible changes in the corresponding isoentropic efficiency. Outlet temperature from turbine stages (Fig. 10b) adapts following variation in expansion ratios. Outlet temperature drops toward the end of discharging cycle since temperature of air from TES reduces, as previously illustrated for Fig. 6.
Conclusions

In this paper we developed for the first time a fully dynamic and off-design performance model of an A-CAES plant with a packed bed thermal energy storage (TES) system. This was possible by integrating together algebraic and differential sub-models detailing the transient features of the thermal storage, the cavern, and the compression/expansion stages, which is a novelty proposed in this work.

Both design and off-design charging/discharging cycles were studied. The results indicate that under nominal charging/discharging a round trip efficiency exceeding 70% can be achieved when TES efficiency rises above 90%. The link between device performance with plant performance was elucidated. In fact we can conclude that: i) maximum round trip efficiency occurs when cycling stationary temperature profiles establishes in the packed bed TES; ii) A-CAES performance detrims toward the end of each discharging cycle due to degradation of the thermal front within the thermal store; iii) reduction of air outlet temperature from TES system cause turbine to operate in off-design conditions leading to an increase of flow rate; iv) compressors operate under strong off-design conditions which also affect temperature profile in thermal storage system.

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References


