Advanced Adiabatic Compressed Air Energy Storage design and modelling accounting for turbomachinery performance

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Abstract. Energy storage plants are going to become a strategic asset in electric grids. This statement is confirmed looking at the increasing shares of renewables composing the energy portfolio of several nations. Therefore the power demand and production mismatches, caused by the intermittent nature of renewables, must be reconciled. Many energy storage solutions are available but Advanced Adiabatic Compressed Air Energy Storage (AA-CAES) plants have potentials similar to pumped hydro systems (PHS). A physical model was developed in Matlab-Simscape to simulate the dynamics of AA-CAES plants, implementing temperature-dependent air properties, efficiency maps for turbomachinery and realistic power ramps. Furthermore, start-up and shut-down phases and energy consumption during idle periods were accounted for. The model embeds a 1D Fortran code to model the detailed behaviour of a packed-bed TES. The grid-to-grid performance of an AA-CAES plant was determined and the assumptions implemented to take into account real turbomachinery behaviour are presented.

1. Introduction

Advanced Adiabatic Compressed Air Energy Storage systems (AA-CAES) store electric energy by pressurizing air and storing it in underground caverns. Electric energy can be recovered later by expanding the high-pressure air in turbines. AA-CAES technology rose as an improvement of CAES plants, which found their first installations in Huntorf (Germany, 1978) and in McIntosh (United States, 1991). In these plants the thermal energy content of air due to its compression is wasted in both intercoolers and after-coolers, therefore efficiencies are limited to 42% and 54%, respectively [1]. In the AA-CAES concept, the need of fossil fuels was overcome by storing the thermal energy content of the hot air exiting the compressor and recovering it before air expansion. AA-CAES technology competes with Pumped Hydro Storage (PHS) in terms of energy capacity, power and expected round-trip efficiency. The availability of adequate turbomachinery and their reaction time determine if and how a plant can participate in the storage markets and provide grid ancillary services. Start-up times are limited by thermal stresses, which are proportional to the highest temperature in the plant: for high-temperature AA-CAES plants (> 400°C) reaction times are expected to be as flexible as gas-turbines plants (10-15 min) but for PHS plants these are a maximum of 2 minutes [1].
The investigation of how real efficiency maps of compressors and turbines would affect the plant performance, including also turbinomachinery transients, was performed to establish grid-to-grid performance of a realistic AA-CAES plant. MAN Energy Solutions Schweiz AG and ALACAES SA conceived the AA-CAES plant depicted in figure 1. In this layout, the overall pressure ratio is evenly divided between the LP and HP components, allowing for the exploitation of commercially available state-of-the-art turbinomachinery components while balancing the effects of high temperatures. The LP-TES (Thermal Energy Storage) is contained in a dedicated pressurized chamber whereas the HP-TES is placed in the cavern where the high pressure air is stored.

![Figure 1. AA-CAES plant schematics.](image)

### 2. Physical model

During charging, the AA-CAES plant exploits electric energy to compress ambient air and then transfers its thermal energy to the LP-TES. In the HP compressor, air is further compressed and sent to the cavern where it leaves its thermal energy in the HP-TES and is stored at a low temperature. An intercooler is used to prevent excessive temperatures at the HP compressor inlet. During discharging, mechanical power is extracted from the hot air flow passing through the turbines connected to the generator.

The AA-CAES plant model was built in the Matlab-Simscape environment, which solves the differential-algebraic equations constituting the mathematical model of a system that evolves in time [2]. The evolution of the pressure and temperature in the caverns was modeled using conservation of mass and energy. Convective thermal losses at the cavern walls surface were accounted for assuming an average area and a constant heat-transfer coefficient. The plant model embeds a 1D code developed in Fortran to accurately describe the packed-bed TES behaviors and thermocline evolutions, taking into account geometries, insulation layers and rock properties [3].

Part of the modeling efforts were focused on the turbinomachinery modelling, which included the energy required for their transients (start-ups and shut-downs) and the integration of their efficiency maps. The compressor and turbine constitutive equations describe the thermodynamic transformations that air undergoes through the turbinomachinery: in the model, either isentropic or polytropic efficiencies can be specified to characterize compressions and expansions. In the former case, efficiency maps were defined as function of the flow rate $\dot{V}$ and the pressure ratio $\beta$ while in the latter the reduced mass flow rate $\dot{G}_R$ and the pressure ratio $\beta$ were exploited.

In the analysis of suitable turbinomachinery, constant speed machines were chosen for both compressors and turbines since variable speed components are more expensive and would require dedicated electronics equipment, such as Variable Frequency Drives. The performance flexibility of turbinomachinery, whenever possible, is guaranteed by systems that include variable valve opening and variable inlet guide vanes.

When dealing with turbinomachinery transients such as start-ups, it is useful to distinguish between two phases:
1. a mechanical start-up phase, during which the turbomachinery speed accelerates from 0 rpm to a nominal speed at off-design process conditions;
2. a process/thermodynamic start-up phase, during which the turbomachinery, at constant rpm, reaches the operating point starting from off-design conditions. The duration of this phase is machine-specific.

To avoid frequent and costly machines starts and stops, the possibility of letting the compressor or turbine train rotate during idle (i.e., when the plant is neither storing nor producing electric energy) was examined. Unfortunately, the energy necessary for a turbomachinery train that rotates synchronously with the grid (3000 rpm for 50 Hz) during an idle phase was not negligible and this procedure was considered not sustainable. The most suitable option for idle phases in an AA-CAES plant was to keep both compressors and expanders warm and in slow rotation with a relatively small power consumption. This option avoids too-long transients, strictly connected to the thermal inertia of the components and of the lubricating oil system, which would also affect the responsiveness of the plant. The modeling of turbomachinery transients considered the energy spent and the time required for typical turbomachine start-ups, without the need of modelling all of the components. To avoid efficiency losses, no throttling valves were foreseen: as a direct consequence, neither the turbine power nor the mass flow rate during the discharging phases of the plant were constant.

3. Plant design and simulation
A full-scale AA-CAES plant based on the layout depicted in figure 1 was tested under regular cycling conditions: after an initial pre-charge 50 identical cycles composed of 5 hours of charge and 5 hours of discharge, including 0.1 hour of idle between them, were simulated. The mechanical efficiencies of the electric motor and generator were set to 0.98. The cavern had a volume of 177’000 m$^3$ and a wall area of 18’150 m$^2$. Both the TESs had a truncated cone shape, a height of 15 m, a top and bottom diameters of 32 m and 24.5 m and they were assumed adiabatic. The TES chamber dimensions were fixed to 34 x 34 x 16 m to accommodate the LP TES. Thermal losses in the cavern and the TES chamber were modelled with a constant convective heat-transfer coefficient of 20 W/m$^2$K.

The power consumed by the plant during the charging phases was set to 140 MW. The TES chamber pressure was controlled regulating the HP compressor during charging phases while it was “sliding” during discharging phases. The outlet temperature of the intercooler before the HP compressor was set to not exceed 20°C.

Figure 2 shows the thermoclines spread after 50 cycles in both TESs, each of which reached a stable operating conditions (i.e. successive thermoclines, after a certain number of cycles, are superimposed). The larger temperature difference at the top of the HP TES was caused by the sliding operation of the HP compressor and by the dynamics of the cavern.

The simulated efficiency of the plant was around 75%, calculated as the ratio of the energy delivered by the generator over the one consumed by the motor, including the energy required for turbomachinery transients and stops. Due to the sliding pressure condition, during a discharge phase the turbine train power output decreased from 114.4 MW to 102.4 MW.

![Figure 2](image-url)
The operating points followed by the turbomachinery during the simulation are reported in Figure 3: for most of turbomachinery operation time (>96%) points fell into normal operating regions of the efficiency maps. The few points outside occurred during the transients as they were accounted for assuming a minimum efficiency of the component.

![Figure 3. Operating points of the turbomachinery during the last cycle superimposed on polytropic efficiency maps. Dashed lines enclose the normal operating region of each machine. Namely: (a) LP compressor, (b) HP compressor, (c) LP turbine, (d) HP turbine.](image)

4. Conclusions

This work presented an AA-CAES numerical model and its application to simulate the dynamic behavior of a specific plant layout. The model includes real turbomachinery efficiency maps, accounts for auxiliary energy consumption (due to transients and stand-by) and can evaluate the plant grid-to-grid performance. The results showed that, during plant operation, the selected turbomachinery worked most of the time within the nominal operating range except for few intervals during their start and stop transients. The calculated 75% (grid-to-grid) plant efficiency confirms that this technology can play a relevant role in future energy storage scenarios. Future modeling efforts will focus on the inclusion of additional plants details and on the consideration of an optimized schedule for the plant operation.

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