Pilot-scale demonstration of advanced adiabatic compressed air energy storage, Part 1: Plant description and tests wit....
Pilot-scale demonstration of advanced adiabatic compressed air energy storage, Part 1: Plant description and tests with sensible thermal-energy storage

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\textbf{A B S T R A C T}

Experimental and numerical results from the world’s first advanced adiabatic compressed air energy storage (AA-CAES) pilot-scale plant are presented. The plant was built in an unused tunnel with a diameter of 4.9 m in which two concrete plugs delimited a mostly unlined cavern of 120 m length. The sensible thermal-energy storage (TES) with a capacity of 12 MW\textsubscript{h}, was placed inside the cavern. The pilot plant was operated with charging/discharging cycles of various durations, air temperatures of up to 550 °C, and maximum cavern gauge pressures of 7 bar. Higher pressures could not be reached because of leaks that were traced mainly to the concrete plugs. Simulations using a coupled model of the TES and cavern showed good agreement with measurements. Cycle energy efficiencies of the TES were determined to lie between 76% and 90%. The estimated round-trip efficiency of the pilot plant was based on the measured TES performance and estimated performances of the other components, yielding values of 63–74%, which compares favorably with the usually quoted values of 60–75% for prospective AA-CAES plants.

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1. Introduction

The growing share of intermittent renewable-energy sources such as wind and solar requires short- and long-term energy storage to guarantee the power supply. At present, pumped hydroelectric storage (PHS) accounts for more than 99% of worldwide bulk storage capacity, see Luo et al. [1]. The large contribution of PHS is explained by the simplicity of the underlying technology and by the large storage capacities and high efficiencies that can be attained by PHS plants. The construction of additional plants is hampered by high capital costs and restrictive site requirements. These requirements include the need for substantial elevation differences between the two reservoirs, which usually requires the flooding of valleys in mountainous regions, see Venkataramani et al. [2].

Advanced adiabatic compressed air energy storage (AA-CAES) is so far the only alternative to PHS that can compete in terms of capacity and efficiency and has the advantages of lower expected capital costs and less strict site requirements, see Chen et al. [3] and Luo et al. [1]. Because CAES plants do not require elevation differences, they can be built in non-mountainous regions also. Furthermore, CAES in its simpler diabatic form (D-CAES) is the only storage technology other than PHS that has been proven at the utility scale with the plants in Huntorf, Germany (321 MW\textsubscript{e}, since 1978), see Crotogino and Quast [4] and Crotogino et al. [5], and McIntosh, USA (110 MW\textsubscript{e}, since 1991), see Nakhamkin et al. [6] and Daly et al. [7].

The working principle of CAES is straightforward: During periods of low demand, surplus electrical energy is used to drive a compressor operating with ambient air and the compressed air is stored in a cavern. During periods of high demand, electrical energy is generated by expanding the stored air in a turbine. To reduce the required cavern volume for a given storage capacity and to reduce thermal stresses in the cavern wall, the air flowing into the cavern should be at a low temperature. In D-CAES, this is achieved by rejecting the thermal energy generated by the compression. This necessitates that the air flowing out of the cavern during discharging is reheated to prevent ice buildup in the

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turbine during expansion. In the existing D-CAES plants, the reheating is achieved by burning fossil fuels. It is the need to reheat the air prior to expansion in the turbine that is responsible for the relatively low cycle efficiencies of about 45–50% of D-CAES plants. By contrast, in AA-CAES the thermal energy generated by the compression is stored in a thermal-energy storage (TES), which increases cycle efficiencies to about 60–75%, see Budt et al. [8] and Sciacovelli et al. [9].

The D-CAES plants in Huntorf and McIntosh use caverns that were solution-mined from salt deposits. Because suitable salt deposits are not ubiquitous, the use of aquifers and rock caverns has been studied by Allen et al. [10,11], and Bauer et al. [12]. Rock caverns are of particular interest because rock formations are more prevalent than salt deposits and would allow reusing decommissioned tunnels and mines. In addition, rock caverns can be accessed more easily than solution-mined salt caverns, which leads to two advantages. First, in principle CAES plants could be built entirely underground. Second, and perhaps more importantly, the TES can be placed inside the cavern. This simplifies the construction of the TES considerably because its structure does not need to sustain a pressure difference, in contrast to plants in which the TES is placed above ground, see, e.g., Zunft et al. [13]. The main disadvantage of rock caverns compared to salt caverns relates to tightness. Salt caverns are often considered to be virtually airtight because of very low permeability and self-healing nature of rock salt, see, e.g., Bérest and Brouard [14] and Chen et al. [15]. Conversely, for rock caverns, based on the experience with the storage of compressed natural gas, the use of linings or hydodynamic containment is advocated to ensure tightness, see, e.g., Goodall et al. [16], Kovári [17], Lu [18], and Rutqvist et al. [19].

Rock caverns constructed in two former mines were investigated in Japan. Experimental data was presented for a cavern lined with backfilling concrete, reinforced concrete, and a rubber sheet in Ishihata [20]. The cavern had a volume of about 1600 m³ and was located at a depth of about 450 m. Starting from a maximum pressure of about 0.9 MPa, leakage rates of about 0.5% per day were observed and traced to cracks in the concrete lining and the air tightness of the rubber sheet. Experimental data was also reported for an unlined blast-excavated cavern with a volume of about 200 m³ located at a depth of about 1000 m in Nakata et al. [21]. The pressure in the cavern, starting from 0.6 MPa, reduced to ambient pressure within about five hours. The leaks were traced to the piping. Because of the above-mentioned advantages, rock caverns continue to be of interest despite the leaks. This interest is reflected in a growing number of studies of rock caverns for CAES from geological and geomechanical perspectives, see, e.g., Shidahara et al. [22], Kim et al. [23], Perazzelli et al. [24], Perazzelli and Anagnostou [25], and Carranza-Torres et al. [26].

So far, AA-CAES has not been demonstrated at the utility scale. An above-ground pilot plant was tested with five compressor and three expansion stages, see Wang et al. [27]. The compressed air was stored in two steel tanks with a combined volume of 100 m³ and with pressures ranging from 2.5 MPa to 9.5 MPa. Two pressurized water tanks with a volume of 12 m³ were used as a TES. The water was used to cool/heat the air between the compression/expansion stages, respectively. An average round-trip energy efficiency of 22.6% was reported. The low efficiency was ascribed to the unsteady operation of the compressor, to mass flow rates during expansion that were lower than expected, and to heat losses from the TES that were larger than expected.

In this article, experimental data obtained with the world’s first underground AA-CAES pilot plant are presented. The specific objectives of the pilot plant were:

1. Demonstrate the ability of rock caverns to withstand the stresses induced by cyclic charging/discharging of the cavern.

### Nomenclature

**Abbreviations**

AA-CAES  advanced adiabatic compressed air energy storage  
D-CAES  diabatic compressed air energy storage  
LDC  low density concrete  
PHE  pumped hydroelectric storage  
RTD  resistance temperature detector  
TES  thermal-energy storage  
UHPC  ultra high performance concrete

**Greek characters**

η  efficiency  
γ  ratio of specific heat capacities  
κ  exponent in analytical solution of cavern temperature  
π  pressure ratio  
ρ  density  
Ξ  exergy

**Latin characters**

ṁ  mass flow rate  
̅p  gauge pressure  
A  area  
$c_p$  specific heat capacity at constant pressure  
$c_v$  specific heat capacity at constant volume  
E  thermal energy  
e  specific internal energy  
h  heat-transfer coefficient  
$h$  specific enthalpy  
k  thermal conductivity  
p  pressure  
Q  heat flux  
s  specific entropy  
T  temperature  
t  time  
V  volume  
W  work  
x  axial coordinate

**Subscripts**

0  initial/reference state  
amb  ambient  
cav  cavern  
comp  compressor  
el  electric  
est  estimated  
exp  experimental results  
ex  exergetic  
gen  generator  
ins  insulation  
in  inlet  
out  outlet  
sim  simulation results  
w  work  
c  charging  
d  discharging  
p  pre-charging  
s  isentropic  
t  total (stagnation)  
w  wall
2. Evaluate the tightness of a mostly unlined rock cavern: Reducing the area covered by a lining simplifies the construction and reduces the cost.

3. Investigate the performance of a sensible TES using a packed bed of rocks placed inside the cavern.

4. Investigate the performance of a combined sensible/latent TES at the pilot scale: Adding a layer of encapsulated phase-change material on top of the packed bed of rocks decreases the drop in the outflow temperature during discharging, see Hahne et al. [28], Zanganeh et al. [29], and Geissbühler et al. [30].

The pilot plant was built in a mountain tunnel that was used during the excavation of the Gotthard base tunnel near Biasca in southern Switzerland. The tunnel has a length of 3.16 km and a diameter of 4.9 m. Further information on the geological and geotechnical conditions at the plant site as well as information on a preliminary design of the pilot plant were presented in Pedretti et al. [31]. Two experimental campaigns were performed with the pilot plant. The first campaign was performed with a sensible TES and is described in this paper. The second campaign was performed with a combined sensible/latent TES and is described in the companion paper Becattini et al. [32].

2. Plant description

A schematic of the pilot plant is shown in Fig. 1. During charging, hot compressed air enters the cavern through an insulated pipe that directs the air to the top of the TES. By flowing through the thermocline TES, the air is cooled. The cooled air then exits the TES at the bottom and enters the cavern. During discharging, the flow is reversed: cold air from the cavern enters the TES at the bottom, is heated, leaves the TES at the top, and exits the cavern through the insulated tube. As can be seen from Fig. 1, the pilot plant does not include a turbine because none was necessary to meet the plant’s objectives. Therefore, the hot air that exits the cavern is routed through a silencer (not indicated in Fig. 1 for simplicity) and released to the ambient. In the following, the various components of the plant are described in more detail.

2.1. Cavern

The cavern is located approximately 800 m from the northeast-erly portal of the tunnel and has an average rock overburden of about 450 m. The cavern is 120 m long and confined by two concrete plugs and steel doors, see Fig. 1. The plugs are 5 m thick and have a double-conical shape to transfer the cavern pressure to the surrounding rock. A picture of one of the plugs and the associated steel door is shown in Fig. 2. The volume and surface area of the cavern are estimated to be about 1942 m³ and 1991 m², respectively. Note that the volume includes the void space inside the TES.

The cavern surface exposed to rock is covered with about 5 cm of shotcrete that had been applied during the construction of the tunnel. With the exception of the cavern surface near the two plugs, no lining is used and therefore the cavern is referred to as “mostly unlined.” Near the two plugs, the shotcrete is lined with an impermeable high-strength glass-fiber membrane, see Fig. 3. The objective of the lining was to reduce the likelihood of air leakages near the plugs by forcing the air to travel a longer path through the rock to reach the ambient.

The plant and the components have been designed for a maximum operating pressure of 33 bar.

2.2. TES

The packed-bed thermocline TES is located inside the cavern and built against the cavern wall as can be seen from Figs. 4 and 5. As indicated in Fig. 5, the TES is charged from top to bottom with hot air from the compressor and discharged from bottom to top with cold air from the cavern. The TES has a height of about 3.1 m and an average length and average width of about 9.9 m and 2.4 m, respectively. From inside to outside, the walls consist of 1 cm of ultra-high-performance concrete (UHPC), 4 cm of low density concrete (LDC), 10 cm of microporous thermal insulation and 25 cm of reinforced concrete. In the wall adjacent to the cavern surface, the thickness of the reinforced concrete is 39 cm. The walls facing the tunnel axis are angled outward to reduce the stresses exerted by the rocks on each other and on the UHPC due to thermal cycling, see Zanganeh et al. [33]. The TES rests on a foundation of reinforced concrete. A steel cover lined with 15 cm microporous thermal insulation is placed on top of the TES.

The TES is filled to a height of about 2.7 m with a packed bed of rocks with a mean diameter of about 2 cm. The packed bed comprises various rock types (mafic rocks, felsic rocks, limestones, sandstones, and quartz-rich conglomerates) obtained from a fluvial deposit near Zurich, Switzerland. Previous studies used
rugs from the same deposit in lab- and pilot-scale storages, see Zanganah et al. [33,34], and Geissbühler et al. [30]. Flow distributors are mounted above and below the packed bed; they were designed and optimized via CFD to ensure uniform air flow conditions. The fully-charged capacity of the TES is 12 MWhb. The TES was over-dimensioned on purpose to protect the cavern and its measurement equipment from high temperatures.

2.3. Compressor train

A high-pressure compressor train (Atlas Copco, ZD 800-1200 and ZD 1200 VSD air-cooled) was used to provide pressurized air at ambient temperature. To simulate adiabatic compression, the air was then heated to 550 °C with an electrical heater.

2.4. Sensors

The gauge pressure in the cavern was measured by two sensors. The air mass flow rate was measured between the compressor and the inlet to the cavern and, therefore, was measured for the air flowing into the TES during charging and for the air flowing out from the TES during discharging. The temperature in the cavern was measured by two resistance temperature detectors (RTDs). Since one RTD was placed near the TES and found to be affected by heat losses, we only report the temperature measured by the other RTD that was placed in the middle of the cavern. The temperature of the rock was measured by two RTDs located about 10 cm inside the rock. The temperatures in the TES were measured by 19 RTDs whose locations are shown by the symbols in Fig. 5. The deformation of the cavern and the displacement and deformation of the concrete plugs was monitored using extensometers, tachymeters, and strain gauges.

3. Numerical model

A numerical model was developed to assist in the design of the experiments carried out with the pilot plant and to simulate larger plants. This model couples a model for the TES with a simple model for the evolution of the air in the cavern.

3.1. TES

The model for the TES is based on a previously developed model that solves one-dimensional (axial) energy equations in a packed bed of circular cross-section and two-dimensional (axial and radial) energy equations in the annular insulation and structure, see Geissbühler et al. [30]. To simulate the pilot plant, the model was adapted in two ways. First, modifications were made to account for the specific TES configuration. The rectangular cross-section is represented in the model by considering transient heat conduction in linear instead of circular geometries for the structure and insulation. Natural convection is assumed on the outer walls of the TES using the correlations of Fujii and Imura [35] and neglecting that one side of the storage abuts the cavern wall.

The second modification relates to the coupling between the TES and the cavern. During charging, the temperature of the air entering the cavern is equal to the temperature of the air exiting the TES. Conversely, during discharging, the temperature of the air entering the TES is approximately equal to the temperature of the air in the cavern. During both charging and discharging, the pressure at the bottom of the TES is assumed to be equal to the cavern pressure. To compute thermal losses from the TES, the ambient temperature is assumed to be the simulated temperature of the air in the cavern.

In the simulations presented in Section 4, the flow distributors at the top and the bottom of the TES were neglected. The convective heat-transfer coefficient in the packed bed was calculated from the correlation of Alanis et al. [36]. Natural convection on the lateral walls and the cover was modelled using the correlation of Fujii and Imura [35]. At the bottom, conduction
losses through the reinforced concrete were considered based on a mean thickness of 0.6 m. The properties of the rocks were taken from Becattini et al. [37] and the properties of reinforced concrete and microporous insulation are given in Table 1, where the thermal conductivity of the microporous insulation is given by

\[ k_{\text{ins}}(T) = 2.151 \times 10^{-3} + 9.219 \times 10^{-5} T - 1.421 \times 10^{-7} T^2 \]

\[ + 8.33 \times 10^{-11} T^3, \]  
(1)

where the temperature is in K. The material properties for LDC and UHPC are taken from Zanganeh [38]. Air was considered to be an ideal gas and thermo-physical properties were taken from Incropera et al. [39].

3.2. Cavern

The cavern model is based on the equations for the conservation of mass and energy for an open control volume. Assuming that the air in the cavern can be treated as an ideal gas, the conservation equations simplify to

\[ V_{\text{cav}} \frac{d \rho_{\text{cav}}}{dt} + \dot{m} = 0, \]

\[ (2) \]

\[ c_{v}V_{\text{cav}} \frac{d T_{\text{cav}}}{dt} + \dot{m} c_{p} T_{i} = -Q^{\prime} A_{\text{cav}}, \]

\[ (3) \]

where \( V_{\text{cav}} \) and \( A_{\text{cav}} \) are the volume and surface area of the cavern, \( \rho_{\text{cav}} \) is the average density of the air in the cavern, \( \dot{m} \) is the mass flow rate of the air, \( c_{v} \) and \( c_{p} \) are the specific heat capacities of the air, \( T_{\text{cav}} \) is the average temperature of the air in the cavern, \( T_{i} \) is the total temperature of the air flowing into/out of the cavern, and \( Q^{\prime} \) is the wall heat flux. We assume that the wall heat flux can be expressed as

\[ Q^{\prime} = h(T_{\text{cav}} - T_{w}), \]

\[ (4) \]

where \( h \) is a convective heat-transfer coefficient and \( T_{w} \) is the cavern wall temperature. Zaugg [42] appears to have been the first to present the analytical solution of Eqs. (2) and (3) for charging at constant mass flow rate, inflow total temperature, wall temperature, and air properties. In Appendix A, Zaugg’s solution is extended to simultaneous charging and discharging. With the extended solution, the evolution of the density and temperature of the air in the cavern can be described if some of the air escapes through a leak during charging. If the mass flow rate, the total inlet and wall temperatures, or the air properties are not constant, Eqs. (2) and (3) can either be solved numerically or the analytical solution can be applied for small time steps. In our simulations, the cavern wall temperature was approximated by the average of the measured rock temperatures.

4. Results and discussion

Two sets of experiments were conducted with the pilot plant. In the first set, the cavern was charged with compressed air at ambient temperature to assess the tightness of the cavern. In the second set, the cavern was charged with compressed air at about 525–550 °C. In both tests, the deformation of the cavern walls and the plugs was found to be negligible. This is not further discussed as it was not the focus of this paper.

4.1. Ambient-temperature tests

The tests revealed discrepancies between the measured cavern gauge pressure and that predicted by the cavern model using the measured charging mass flow rate. The discrepancy was ascribed to leaks through the plugs and/or the unlined portion of the cavern. In Fig. 6, the measured cavern gauge pressure \( p_{\text{cav}} = p_{\text{cav}} - p_{\text{amb}} \), the measured mass flow rate, and the estimated total leakage mass flow rate \( \dot{m}_{\text{leak,tot}} \) are shown as a function of time. The total leakage mass flow rate was estimated by subtracting the measured mass flow rate from that necessary to reproduce the measured cavern gauge pressure. Once the cavern was no longer being charged, the pressure approached the ambient within about 4 days.

The leakage through the southwesterly plug could be estimated thanks to an anemometer in an antechamber located adjacent to the plug. (The antechamber is not shown in Fig. 1 for simplicity.) Comparing the estimated leakage mass flow rate \( \dot{m}_{\text{leak,plug,w}} \) with the total leakage mass flow rate indicates that at least at high gauge pressures, around 60% of the total leakage can be attributed to the southwesterly plug. Although the leakage rate through the northwesterly plug could not be measured, it is unlikely that the behavior of that plug is considerably different. Therefore it can be concluded that leaks originated mainly from the concrete plugs and that losses through the unlined portion of the cavern could be considered to be small or even negligible. Since the tightness of the concrete plugs depends on their design, and improved designs will be considered for future plants, the results are encouraging concerning the tightness of unlined rock caverns.

The measurements from all the ambient-temperature tests were used to derive a correlation for the total leakage mass flow rate,

\[ \dot{m}_{\text{leak,tot}} = -2.746 \times 10^{-7} \dot{p}_{\text{cav}} + 2.989 \times 10^{-3} \dot{p}_{\text{cav}}^2 \]

\[ - 6.210 \times 10^{-1} \dot{p}_{\text{cav}}^3, \]

\[ (5) \]

where \( \dot{m}_{\text{leak,tot}} \) is in kg/s and \( \dot{p}_{\text{cav}} \) in bars. The fit was used in the simulation of the high-temperature pressure tests to account for the leakage.

4.2. High-temperature tests

Two test runs were conducted, each consisting of up to five days of operation. Run 1 and 2 consisted of six and seven charging/
discharging cycles, respectively. A so-called pre-charging was performed before the first charging of each run to approach steady cycling behavior of the TES more quickly, see Zanganeh [38] and the comments below. The durations of the pre-charging, charging, and discharging phases of each cycle are given in Table 2. The table also lists the figures in which the mass flow rates, the charging temperature, and the cavern gauge pressure are shown. In general, cycles of equal charging and discharging times were performed, except for cycles B1, B2, and B3 of run 1, which were performed after a re-charge phase in a constant gauge pressure range between 3.5 and 4 bar.

Figs. 7 and 8 show the measured charging temperature, the measured mass flow rates, and the measured and simulated cavern gauge pressures during the pre-charging phases of the two runs, respectively. In these and the following figures, light blue, white, and gray backgrounds indicate pre-/re-charging, charging, and discharging phases, respectively. The charging temperature was measured inside the tube just above the distributor, as indicated by the filled black circle in Fig. 5. During both pre-charging phases, the plant was charged at the maximum mass flow rate, which increased with increasing back pressure from the cavern. The maximum mass flow rate was limited to protect measurement equipment and the steel cover.

As can be seen from Fig. 7, during the pre-charging phase of run 1 the plant was charged with cold compressed air for the first five hours. Afterwards, the charging temperature was rapidly increased and kept constant at \(T_{\text{c,in}} \approx 525 \, ^\circ\text{C}\). During the pre-charging phase of run 2, the inlet temperature was ramped slowly from 20 \(^\circ\text{C}\) to about 550 \(^\circ\text{C}\) and then kept constant. The temporal evolution of the inflow temperature results in a spatial temperature distribution in the TES due to the flow. The ramp, which was designed with the numerical model, therefore leads to a shallower thermocline than if the inflow temperature were immediately set to its maximum of 550 \(^\circ\text{C}\). Because the shallower thermocline more closely resembles the thermocline at quasi-steady conditions, pre-charging allows quasi-steady cycling conditions of the TES to be reached more quickly. There was a short interruption during pre-charging where the plant was switched to discharge mode. In both runs, the cavern gauge pressure increased to about 6 bar.

We make three observations about the agreement between the measured and simulated cavern gauge pressures. First, the agreement is seen to be fairly good overall, with better agreement at lower gauge pressures. Second, the agreement worsens from run 1 to run 2. These two observations may indicate that the leakage rate changes with time. Finally, it should be noted that perfect agreement between the measured and simulated cavern gauge pressures can be obtained by extracting for each run a separate correlation for the total leakage mass flow rate. We restrained from doing so because this causes the comparison with experimental data to degenerate into a curve-fitting exercise.

### 4.2.1. Cavern

Figs. 9 and 10 show the measured and simulated temperatures at the top of the TES, the measured mass flow rates, and the measured and simulated cavern gauge pressure during the cycles of run 1 and run 2, respectively. The charging temperature for all cycles was \(T_{\text{c,in}} \approx 550 \, ^\circ\text{C}\).

<table>
<thead>
<tr>
<th>Table 2</th>
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<tr>
<td>Operating conditions of the high-temperature test runs.</td>
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<table>
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<tr>
<th>Run</th>
<th>Cycle</th>
<th>(\Delta t_{\text{in}}) [h]</th>
<th>(\Delta t_{\text{c}}) [h]</th>
<th>(\Delta t_{\text{d}}) [h]</th>
<th>(m_{\text{in}}, T_{\text{c,in}}, P_{\text{c,in}})</th>
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<tbody>
<tr>
<td>1</td>
<td>A1</td>
<td>40:10</td>
<td>2:45</td>
<td>2:45</td>
<td>Figs. 7 and 9</td>
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<td></td>
<td>A2</td>
<td>2:45</td>
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<td>Fig. 9</td>
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<td></td>
<td>A3</td>
<td>4:59</td>
<td>4:00</td>
<td>1:07</td>
<td>Fig. 9</td>
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<td></td>
<td>B1</td>
<td>3:22</td>
<td>3:34</td>
<td>1:05</td>
<td>Fig. 9</td>
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<tr>
<td></td>
<td>B2</td>
<td>2:45</td>
<td>2:45</td>
<td>2:45</td>
<td>Fig. 9</td>
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<tr>
<td></td>
<td>B3</td>
<td>4:07</td>
<td>4:07</td>
<td>4:07</td>
<td>Fig. 10</td>
</tr>
</tbody>
</table>

| 2    | A1    | 42:00          | 2:45            | 2:45            | Figs. 8 and 10                    |
|      | A2    | 2:45           | 2:45            | 2:45            | Fig. 10                          |
|      | B1    | 1:22           | 1:22            | 1:22            | Fig. 10                          |
|      | B2    | 1:22           | 1:22            | 1:22            | Fig. 10                          |
|      | C1    | 4:07           | 4:07            | 4:07            | Fig. 10                          |
|      | C2    | 4:07           | 4:07            | 4:07            | Fig. 10                          |
|      | C3    | 4:07           | 4:07            | 4:07            | Fig. 10                          |
The measured and simulated temperatures coincide during the charging and re-charging phases because the former were used as boundary conditions in the simulations. The simulations were carried out for two values of the bypass flow fraction in the TES, 0% and 15%. The agreement between the measured and simulated outflow temperatures during discharging is better for the bypass flow fraction of 15%. Reasons for bypassing in the TES are the increased void fraction in the packed bed near the walls and air losses in the steel cover due to deformation of steel components. The deformation of the steel cover resulted from the pressure difference between the inside of the steel cover and the cavern because of the pressure drop across the packed bed. Considering the size of the TES, the wall bypassing is assumed to be small. During run 2, the agreement between the simulated and measured temperatures for the bypass flow fraction of 15% is good during the first discharging phase. In the subsequent discharging phases, the agreement becomes worse, which is thought to be due to increasing air losses from the piping and the steel cover.

The measured and simulated cavern gauge pressures agree quite well. The overall decrease in the gauge pressures is a consequence of the above-mentioned air leaks. It is interesting to note that the relative errors between the measured and simulated absolute cavern pressures remains approximately constant. The relative errors at the beginning and at the end of the first run are 5.4% and 4.5%, and 10.3% and 13.0% for the second run, respectively.

This means that the difference between the pressures during cycling is determined mostly by the difference at the end of the pre-charging phase.

In Figs. 11 and 12, the measured and simulated cavern temperatures and the measured rock temperature are presented. The simulations were performed with heat-transfer coefficients of $h = 5 \text{ W/m}^2\text{K}$ and $10 \text{ W/m}^2\text{K}$. For the former, good agreement was obtained between the simulated and measured cavern temperatures with maximum temperature differences of around 1°C for both runs. The value of $h = 5 \text{ W/m}^2\text{K}$ is somewhat smaller than previously used values for modeling of the salt cavern of the Huntorf plant, see Kushnir et al. [43] and Xia et al. [44]. This is attributed to the different cavern shapes and materials as well as the location of the temperature measurement (inside the rock and therefore affected by its thermophysical properties).

4.2.2. TES

The measured temperatures and the simulated thermoclines are shown in Figs. 13 and 14 at four times during the pre-charging phases. The symbols indicate the measurement locations in Fig. 5 and the solid and dashed lines represent the averages of the simulated air and rock temperatures with 0% bypass and 15% bypass fractions, respectively.

Although the TES is relatively large and wall effects are therefore not expected to be significant, measurements at equal axial positions nevertheless differ by up to 75°C. These differences may be attributed to the following causes. First, the steep thermoclines mean that even small differences in axial placement of the RTDs can result in relatively large temperature differences. For example, the average thermocline slopes extracted from the simulations are approximately 1227°C/m and 545°C/m after 10 and 40 h of run 1, respectively. Assuming that the vertical inaccuracy in placing the RTDs is 5 cm, the corresponding differences in the temperature would be 61°C and 27°C, respectively. The second cause might be non-uniform flow distribution at the top of the packed bed, perhaps partially due to the distributor and partially due to the leaks from the steel cover. This explanation appears plausible given that the differences between the measured temperatures are somewhat larger near the top of the packed bed. Finally, measurement errors are expected to contribute to the differences. The maximum error of the RTD measurement at 550°C is ±3.1°C.

The agreement between the measured temperatures and the simulated thermoclines is considered to be good, given that the latter mostly fall in the ranges of the former at the various axial locations. For the larger bypass fraction of 15%, the agreement is slightly better at later times and further down in the TES.

Figs. 15 and 16 present the measured and simulated temperatures as a function of time at three axial positions in the TES. The experimental data is an average of all measurements at the same axial position and the simulated data is an average of the air and rock temperatures assuming a bypass fraction of 15%. The agreement between the measured and simulated temperatures improves with time for run 1, whereas it becomes worse for run 2.

4.2.3. TES and plant efficiencies

The cycle energy efficiency of the TES is defined as

$$\eta_{\text{TES}} = \frac{E_{\text{th,d}}}{E_{\text{th,c}}},$$

where $E_{\text{th,d}}$ and $E_{\text{th,c}}$ are the thermal energies provided to and recovered from the TES,

$$E_{\text{th,d}} = \int_{0}^{\Delta t} \dot{m}_d(t) \left[ e(T_{d,\text{out}}(t)) - e(T_{d,\text{in}}(t)) \right] \, dt,$$
computed from the indicated measured time-dependent temperatures.

The cycle exergy efficiency of the TES is defined as

$$\eta_{\text{ex,TES}} = \frac{\mathcal{E}_{\text{h, dis}, \text{TES}}}{\mathcal{E}_{\text{h, in}, \text{TES}}}$$

where $\mathcal{E}_{\text{h, dis}, \text{TES}}$ and $\mathcal{E}_{\text{h, in}, \text{TES}}$ are the thermal exergies provided to and recovered from the TES,

$$\mathcal{E}_{\text{h, dis}, \text{TES}} = \int_0^{\Delta t_d} m_c(t) \left[ h(T_{\text{dis}, c}(t)) - h(T_{\text{in}, c}(t)) \right] dt,$$

$$\mathcal{E}_{\text{h, in}, \text{TES}} = \int_0^{\Delta t_c} m_c(t) \left[ h(T_{\text{in}, c}(t)) - h(T_{\text{dis}, c}(t)) \right] dt,$$

where $h(T)$ and $s(T)$ are the specific enthalpy and exergy of the air. The reference state was taken as $T_0 = 15{\degree}C$.

The cycle efficiency of AA-CAES plants is usually defined as

$$\eta_{\text{plant}} = \frac{W_{\text{el, gen}}}{W_{\text{el, mot}}}$$

where $W_{\text{el, gen}}$ and $W_{\text{el, mot}}$ are the electricity produced by the generator and the electricity consumed by the motor, respectively. To apply this definition to the pilot plant, we assume (1) that the motor drove an adiabatic compressor instead of an isothermal compressor and an electric heater and (2) that the pilot plant was equipped with a turbine and a generator. Then

$$W_{\text{el, gen}} = \eta_{\text{gen}} \int_0^{\Delta t_d} \dot{m}_c(t) c_{p, \text{air}} T_{\text{dis}, c}(t) \eta_{\text{h, turb}} \left[ 1 - \left( \frac{p_{\text{amb}}}{p_{\text{cav}}(t)} \right)^{\frac{1}{\gamma}} \right] dt,$$

$$W_{\text{el, mot}} = \frac{1}{\eta_{\text{mot}}} \int_0^{\Delta t_c} \dot{m}_c(t) c_{p, \text{air}} T_{\text{in}, c}(t) \left[ \left( \frac{p_{\text{cav}}(t)}{p_{\text{amb}}} \right)^{\frac{1}{\gamma}} - 1 \right] dt,$$

where $\eta_{\text{h, turb}} = 0.9$ and $\eta_{\text{comp}}, \gamma = 0.85$ are representative isentropic turbine and compressor efficiencies, respectively. $\eta_{\text{gen}} = 0.97$ are representative generator and motor efficiencies, the ambient pressure $p_{\text{amb}}$ was assumed to be 1 bar, and $\gamma = 1.4$ is the ratio of the specific heats of air. The specific heat capacity at constant pressure of the air was assumed to be constant at $c_{p, \text{air}} = 1041$ J/kg K, which corresponds to its value at the average temperature between $T_{\text{amb}} = 15{\degree}C$ and the typical measured TES inlet temperature of $T_{\text{cavern}} \approx 550{\degree}C$.

As indicated in Eqs. (13) and (14), we assume that the compressor and turbine operate between ambient and time-dependent cavern pressures and that the temperature of the air flowing into the turbine is equal to the measured time-dependent temperature of the air flowing out of the TES. The time-dependent cavern pressure was not taken from the experiments because of the leaks. Instead, the model presented in Section 3.2 was used to simulate an airtight cavern with the measured mass flow rates, a constant rock temperature of $15{\degree}C$, and a wall heat-transfer coefficient of $h = 5$ W/m$^2$K.

In estimating the plant efficiency, the pre-charging operation was modified from that used in the experiments so that the cavern pressure is consistent with the typical measured TES inlet temperature of $T_{\text{cavern}} \approx 550{\degree}C$. Using

$$\frac{p_{\text{comp, out}}}{p_{\text{amb}}} = \left( \frac{T_{\text{comp, out}}}{T_{\text{amb}}} \right)^{\frac{1}{\gamma}},$$

Fig. 10. Measured and simulated temperature at the top of the TES, measured mass flow rates, and measured cavern gauge pressure during cycles for run 2 of high-temperature tests (excluding pre-charging phase shown in Fig. 8).

Fig. 11. Measured and simulated cavern temperatures and measured rock temperature for run 1 of the high-temperature tests.

$E_{\text{h, c}} = \int_0^{\Delta t_c} \dot{m}_c(t) \left[ e(T_{\text{in}, c}(t)) - e(T_{\text{dis}, c}(t)) \right] dt,$

where $\Delta t_d$ and $\Delta t_c$ are the discharging and charging durations, respectively, and $e(T)$ is the specific internal energy of the air.
Fig. 12. Measured and simulated cavern temperatures and measured rock temperature for run 2 of the high-temperature tests. See the legend in Fig. 11 for an explanation of the lines and symbols.

Fig. 13. Measured temperatures and simulated thermoclines during pre-charging for run 1 of high-temperature tests. The symbols correspond to the measurement locations in Fig. 5. The simulated temperature is the average of the air and rock temperatures.

Fig. 14. Measured temperatures and simulated thermoclines during pre-charging for run 2 of high-temperature tests. The symbols correspond to the measurement locations in Fig. 5. The simulated temperature is the average of the air and rock temperatures.

\[ \eta_{\text{plant,\ max}} = \eta_{\text{gen}} \eta_{\text{net}} \eta_{\text{turb}} \left(1 - \frac{1 - \eta_{\text{comp}}}{\left(p_{\text{comp,\ out}} / p_{\text{amb}}\right)^{\frac{\gamma}{\gamma - 1}}} \right) \]  

which was derived assuming constant and equal charging and discharging durations and mass flow rates. For the assumed isentropic efficiencies and the pressure ratio of 27.52, we obtain \( \eta_{\text{plant,\ max}} = 79.8\% \), indicating that thermal losses cause reductions in the efficiency of 6.0–17.1%.

5. Summary, conclusions, and further work

A pilot-scale AA-CAES plant was built in an unused tunnel. The cavern of 120 m length and 4.9 m diameter was confined by two concrete plugs and was mostly unlined. The TES consisting of a packed bed of rocks was placed inside the cavern to avoid pressure forces acting on its structure and had a maximum capacity of 12 MWth. The pilot plant did not include a turbine.

Tests were carried out with charging temperatures of 550 °C, cavern gauge pressures between 0–7 bar, and various pre-charging, charging and discharging durations. The main conclusions from these tests are:

1. The deformation of the cavern induced by cyclic charging/discharging was found to be negligible.
2. The cavern suffered from leaks. The leaks were traced mostly to the concrete plugs. The unlined portion of the cavern was determined to have small or even negligible leaks. With an empirical correlation for the leakage mass flow rate as a function of the cavern gauge pressure, good agreement between simulated and measured cavern gauge pressures was obtained.
3. The TES performed well with energy efficiencies between 77% and 91% and exergy efficiencies between 72% and 89%. Good agreement was found between simulated and measured thermoclines, with discrepancies attributed partly to the deformation of the steel cover. The concrete structure, the insulation, and the rocks withstood the thermal cycling at high pressures.
4. Plant efficiencies were estimated using the measured mass flow rates and TES inflow/outflow temperatures and assuming isentropic compressor and turbine efficiencies as well as an airtight cavern. The estimated efficiencies were between 63% and 74%, consistent with the usually quoted values of AA-CAES plant efficiencies of 60–75%.

Further work will first focus on addressing the leaks from the concrete plugs. This work will evaluate proven techniques used for the storage of compressed gas in unlined caverns, see, e.g., see Goodall et al. [16], Kovári [17], and Lu [18]. Hydrodynamic containment appears to be a particularly attractive option.
Attention will then shift toward the construction of a larger-scale pilot plant that includes a turbine.

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Appendix A.

Presented is the analytical solution for the cavern temperature at diabatic and simultaneous charge and discharge conditions with constant inlet and wall temperatures as well as constant mass flow rates. Mass conservation yields

\[
\frac{p_{\text{cav}}(t)}{p_{\text{cav},0}} = 1 + \frac{(\dot{m}_c - \dot{m}_d)t}{m_0},
\]

where \( p_{\text{cav},0} \) and \( m_0 \) are the initial air density and mass inside the cavern. Energy conservation gives

\[
\frac{T_{\text{cav}}(t)}{T_{\text{cav},0}} = \left( 1 + \frac{(\dot{m}_c - \dot{m}_d)t}{m_0} \right)^\kappa - \left[ 1 + \frac{(\dot{m}_c - \dot{m}_d)t}{m_0} \right] - \frac{\dot{m}_c}{\dot{m}_c} \frac{T_{T_{\text{cav}}} - T_{T_{\text{cav},0}}}{m_c + (\gamma - 1)(\rho_d + \dot{h}_{\text{cav}})/(\gamma c_v)}
\]

where \( T_{\text{cav},0} \) is the initial cavern temperature, \( T_{T_{\text{cav}}} \) is the total inlet temperature to the cavern during charging and \( \kappa \) is given by

\[
\kappa = -\left( \frac{m_c}{m_c - m_d} + (\gamma - 1) \frac{m_d}{m_c - m_d} + \frac{\dot{h}_{\text{cav}}}{\gamma c_v} \right)
\]

The cavern pressure \( p_{\text{cav}}(t) \) follows from the equation of state of an ideal gas.

Appendix A. Supplementary data

Supplementary data associated with this article can be found, in the online version, at https://doi.org/10.1016/j.est.2018.02.004.

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