Energy and exergy analysis of adiabatic compressed air energy storage system

Lukasz Szablowski a, *, Piotr Krawczyk a, Krzysztof Badyda a, Sotirios Karellas b, Emmanuel Kakaras b, Wojciech Bujalski a

a Institute of Heat Engineering, Warsaw University of Technology, Warsaw, Poland
b National Technical University of Athens, Athens, Greece

ABSTRACT

The low efficiency of existing CAES systems is due to large energy losses during the air compression process. This could be remedied by building an adiabatic CAES system, where the heat of compression is stored and subsequently used during the expansion process in the turbine.

An energy and exergy analysis of A-CAES is presented in this article. A dynamic mathematical model of an adiabatic CAES system was constructed using Aspen Hysys software.

The volume of the CAES cavern is 310000 m³ and the operation pressure inside the cavern ranges from 43 to 70 bar. Thermal oil was used as the working medium in thermal energy storage system. The temperature in the hot oil tank was 300 °C and in the cold oil tank 80 °C.

Simulations of processes of loading and unloading of compressed air storage were performed and then places where exergy destruction occurs were identified. The biggest exergy destruction occurs in the compressors (276.191 MWh/cycle in total) and turbines (190.394 MWh/cycle in total). Major destruction of exergy was also reported at control valve V2 (103.688 MWh/cycle). Round trip efficiency of the system is 50%.

© 2017 Elsevier Ltd. All rights reserved.

1. Introduction

There are only two methods of energy storage used on a large scale at present: pumped storage and Compressed Air Energy Storage (CAES).

CAES power plants feature a huge volume tank, used to store compressed air. Due to the amount of air required and the resulting financial constraints, only natural reservoirs are currently economically viable: salt caverns, salt mine workings, spaces in aquifers, mines of limestone and other minerals formed in the structure of hard rocks.

CAES technology is relatively well known. Worldwide there are two large plants of this type: Huntorf in Germany (290 MW) and McIntosh in the USA (110 MW). The first one went operational in 1978, the second in 1991.

An overview of the state of knowledge about CAES technology was given in Refs. [1–5].

In article [6] a hybrid system using CAES in combination with Renewable Technologies (e.g. wind turbines) was shown and a possible application of CAES-RT in Poland was presented. A dynamic mathematical model of CAES was given in Refs. [6,7].

In Ref. [8] a simulation and thermodynamic analysis of the Compressed Air Energy Storage-Combined Cycle (CAES-CC) proposed by the authors were performed. The overall efficiency of the CAES-CC system was about 10% higher than the conventional CAES. The reference system in this case was CAES, without regeneration. According to the authors, the heat obtained from the compressor intercoolers when charging the air reservoir can be used to keep the steam part of the system on hot standby.

In Ref. [9] a system consisting of a CAES system and Kalina Cycle to recover waste heat was presented. This combination delivered an increase in efficiency of only 4% compared to the CAES system. The reference system in this case was CAES, with regeneration.

An analysis of the impact of selected parameters (pressure ratio, temperature, mass flow of air) on the performance of an adiabatic CAES system was presented in Ref. [10]. Calculations were

http://dx.doi.org/10.1016/j.energy.2017.07.055
0360-5442/© 2017 Elsevier Ltd. All rights reserved.
Abbreviations

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>B</td>
<td>exergy [kJ]</td>
</tr>
<tr>
<td>B_in</td>
<td>exergy at the inlet [kJ]</td>
</tr>
<tr>
<td>B_in cs</td>
<td>exergy at the inlet of the cold side of the heat exchanger [kJ]</td>
</tr>
<tr>
<td>B_in hs</td>
<td>exergy at the inlet of the hot side of the heat exchanger [kJ]</td>
</tr>
<tr>
<td>B_out</td>
<td>exergy at the outlet [kJ]</td>
</tr>
<tr>
<td>B_out cs</td>
<td>exergy at the outlet of the cold side of the heat exchanger [kJ]</td>
</tr>
<tr>
<td>B_out hs</td>
<td>exergy at the outlet of the hot side of the heat exchanger [kJ]</td>
</tr>
<tr>
<td>Eel g</td>
<td>energy from the generator [kJ]</td>
</tr>
<tr>
<td>Eel c</td>
<td>energy consumed by the compressor [kJ]</td>
</tr>
<tr>
<td>Eel p</td>
<td>energy consumed by the pumps [kJ]</td>
</tr>
<tr>
<td>Eel t</td>
<td>energy from the turbines [kJ]</td>
</tr>
<tr>
<td>h</td>
<td>specific enthalpy of the working medium [kJ/kg]</td>
</tr>
<tr>
<td>h0</td>
<td>specific enthalpy of the working medium in ambient conditions [kJ/kg]</td>
</tr>
<tr>
<td>h1</td>
<td>specific enthalpy of the working medium at the inlet [kJ/kg]</td>
</tr>
<tr>
<td>h2</td>
<td>specific enthalpy of the working medium at the outlet [kJ/kg]</td>
</tr>
<tr>
<td>Qf</td>
<td>chemical energy of fuel (0 in adiabatic systems) [kJ]</td>
</tr>
<tr>
<td>S</td>
<td>entropy [kJ/K]</td>
</tr>
<tr>
<td>S2</td>
<td>specific entropy of the working fluid at outlet [kJ/(kgK)]</td>
</tr>
<tr>
<td>S0</td>
<td>specific entropy of the working fluid in ambient conditions [kJ/(kgK)]</td>
</tr>
<tr>
<td>T</td>
<td>absolute temperature [K]</td>
</tr>
<tr>
<td>T_0</td>
<td>ambient temperature [K]</td>
</tr>
</tbody>
</table>

Greek symbols:

- \( \eta_{cycle} \) round trip efficiency of a cycle \([-\] \)

performed with the tacit assumption of the authors that the system operates at constant pressure.

In Ref. [11] a thermodynamic analysis of an adiabatic CAES system with packed bed regenerators was presented. In this case, the heat of compression was stored in a solid material, which achieved simulated efficiency in the range of 70.5%–71.1%. The maximum packed bed temperature was 713 K (for a 2-stage system).

A completely different approach to the adiabatic CAES system was presented in Ref. [12], where a low temperature CAES concept was developed. Roundtrip efficiency was achieved in the range of 52%–60% and the maximum temperature in the thermal energy storage was 200 °C. According to the authors of [12], this solution uses less expensive system elements and achieves a start-up time of less than 5 min.

Low temperature CAES systems were studied in Refs. [13–15]. In these cases, water was used as the heat storage medium.

Article [16] contains analysis of off-design work of a CAES system integrated with a hybrid power plant and connected to a wind power plant. Operation of the system on the Italian Power Exchange market for one year was simulated. The costs analysis was performed through a thermo-economic approach.

In Ref. [17] a concept was introduced of a small CAES cooperating with photovoltaic panels and working for the needs of a radio base station. This system was integrated with storage of the heat which was generated during compression of air and consumed during expansion of this air in the turbine. An additional function of this solution was the production of cold, as the air left the turbine at a temperature of just 3 °C. Despite the small scale of this CAES system, storage efficiency was 57%.

[18] examines a hybrid DG system combined with Compressed Air Energy Storage (CAES) and Thermal Energy Storage (TES).

Descriptions of other energy storage systems can be found in previous works of the authors [19,20].

Exergy analyses of different configurations of CAES systems formed the subject of several articles.

An exergy analysis of a Combined Cooling, Heating and Power system integrated with conventional CAES and wind turbine was presented in Ref. [21], while exergy analysis of constant-pressure CAES combined with pumped hydro storage was dealt with in Ref. [22]. This type of analysis relating to a micro gas turbine, CAES and solar dish integrated system was the topic of article [23].

Exergy analysis of CAES combined with thermal energy storage operating on hot water for district heating needs was a subject of [24]. Exergy analysis of CAES-Combined Cycle was described in Ref. [8]. Exergy analysis of an underwater CAES system was presented in Ref. [25].

Exergy analysis of adiabatic CAES systems was the subject of articles [15] and [26]. [15] analyzes an A-CAES using water as the working medium in thermal energy storage (TES), while in the case of [26] TES was made of phase change materials (PCM).

The novelty of this study is that it features an exergy analysis of an adiabatic compressed air energy storage system which uses thermal oil as the working medium in a thermal energy storage system.

The aims of the study were to investigate the computational efficiency of the system and to identify the main places of exergy destruction.

Within this work a dynamic model of adiabatic CAES system was built. Cycles of loading and unloading of this system have been performed. System parameters at different time intervals have been archived. Then the system performance and exergy destruction on its individual parts were calculated.

2. Methods and materials

Commercial software was used for the purposes of the calculations shown below [27,28]. Linear and nonlinear differential equations describe the dynamic calculations of physical phenomena and are presented in Refs. [27–29]. The Peng-Robinson equation of state underpins the model of the working medium [30].

Exergy destruction could be described using the Gouy-Stodola equation [31,32]:

\[
\delta B = T_0 \sum \Delta S
\] (1)

The outflow specific exergy loss for the pneumatic engine is [31]:

\[
\delta b_{out\ flow} = h_2 - h_0 - T_0(s_2 - s_0)
\] (2)

And the outflow exergy loss \( \delta b_{out\ flow} \) is obtained by multiplying outflow specific exergy loss \( \delta b_{out\ flow} \) by mass of working fluid.
Exergy destruction in the elements of modelled system is described below:

Compressors:

\[ \delta B = B_{\text{out}} - B_{\text{in}} - E_{\text{el c}} \]  

(3)

Turbines:

\[ \delta B = B_{\text{out}} - B_{\text{in}} + E_{\text{el t}} \]  

(4)

Heat exchangers:

\[ \delta B = B_{\text{out}} + B_{\text{out}} - B_{\text{in}} + B_{\text{in}} - B_{\text{in}} \]  

(5)

Cavern, tanks, valves:

\[ \delta B = B_{\text{out}} - B_{\text{in}} \]  

(6)

Pumps:

\[ \delta B = B_{\text{out}} - B_{\text{in}} - E_{\text{el p}} \]  

(7)

Mixer:

\[ \delta B = \sum B_{\text{out}} - B_{\text{in}} \]  

(8)

Round trip efficiency of the cycle is obtained through the following formula [6,7]:

\[ \eta_{\text{cycle}} = \frac{E_{\text{el g}}}{E_{\text{el c}} + Q_f} \]  

(14)

3. Description of model

The model was built using Aspen HYSYS software in dynamics mode. Air parameters assumed according to ISO conditions, i.e. temperature of 15 °C, pressure 1.013 bar. Time step was set at 0.05 s. Air composition is shown in Table 1. The energy storage system is charged during the valleys of load of the power system and discharged at peaks. Therminol 55 oil was used as the working medium in the heat store. It is a synthetic oil used in heat storage systems [33] and its operating range is −28 °C to 315 °C [34].

A schematic diagram of the modelled Adiabatic CAES system is shown in Fig. 1. The installation analysed consists of the two-stage compressor with the intercooler and two heat exchangers (thermal oil/air), the underground storage with the relatively large volume (salt cavern), three-stage gas turbine with three heat exchangers (thermal oil/air), two oil tanks (for hot and cold oil), two oil coolers and nine pumps. The air is taken from the environment and pumped into the cavern during the storage charging, powered by electricity. In this process, air is cooled in the two heat exchangers and in intercooler. Excess heat of air is transported to thermal oil, which is directed to hot oil tank. In the last stage of charging air is cooled by water (in the intercooler). When the storage is discharging, after the pressure pre-reduction the air is transported to the turbine. Air is heated before each part of the turbine in three heat exchangers powered by hot oil (from hot oil tank). After the expansion process, the exhaust gases are discharged into the environment.

The polytropic efficiency of the rotating machines (turbines and compressors) was set at 75%. The volume of the cavern is 310,000 m³, while the pressure inside it during normal operation ranges from 43 to 70 bar. The air temperature before the cavern is 50 °C. Air pressure drop at exchangers HEX1, HEX2, IC1, HEX3, HEX4 and HEX5 in the nominal operating point are respectively 0.1, 0.25, 0.25, 0.1, 0.1 and 0.1 bar (they change during change of flow pressure drops). Cooling water at inlets (of IC1, OC1 and OC2) has a temperature of 15 °C and pressure of 2 bar, whereas at the outlet – 75 °C and 1 bar. The oil pressure drop on each exchanger was 1 bar (also on oil coolers OC1 and OC2). Each of the heat exchangers (HEX1, HEX2, IC1, HEX3, HEX4, HEX5, IC1, OC1 and OC2) is computationally divided into 10 intervals. The temperature in the hot oil tank was 300 °C, and pumps P1 and P2 were controlled to obtain a constant oil temperature after HEX1 and HEX2 (to avoid exergy destruction due to mixing streams of different temperatures). The volume of the hot oil tank was 15,627.9 m³. The temperature in the cold oil tank was 80 °C and its volume was 12,685.6 m³. The task of oil coolers OC1 and OC2 was to remove heat which could not be consumed during discharging of A-CAES (the main reason for this was throttling on valve V2). Heat losses in

---

**Table 1**

Mass composition of air.

<table>
<thead>
<tr>
<th>Component</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Oxygen</td>
<td>%wt</td>
<td>23.052</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>%wt</td>
<td>74.99</td>
</tr>
<tr>
<td>Carbon dioxide</td>
<td>%wt</td>
<td>0.046</td>
</tr>
<tr>
<td>Argon</td>
<td>%wt</td>
<td>1.276</td>
</tr>
<tr>
<td>Water</td>
<td>%wt</td>
<td>0.636</td>
</tr>
</tbody>
</table>

---

![Fig. 1. Schematic diagram of the modelled Adiabatic CAES power plant.](image-url)
the hot oil tank have been omitted because far more of it (than could potentially be lost through the wall of the tank) remained unused during the unloading of the CAES system. The compressor was divided into 2 parts. LPC needs 33 MW and HPC 27 MW of power during charging (so that the air temperature after both parts was similar). The turbine was divided into 3 parts. The rated power of HPT, IPT and LPT was respectively: 46.5 MW, 52 MW and 62.5 MW. The pressure after each was set so as to maximize the power of the whole turbine. The air temperature before each part of the turbine was 270°C. Valve V2 is controlled in order to maintain a constant flow (417 kg/s) during discharging.

Most of the system parameters (volume of cavern, pressure ranges inside the cavern, air flows during charging and discharging) were adopted on the basis of [35].

4. Results

The results of the simulation are shown in Figs. 2–6. These Figures can be divided into: dynamic charts (Figs. 2–4), energy consumption/production (Fig. 5) and exergy destruction (Fig. 6).

4.1. Description of the processes of charging and discharging

Changes of pressure inside the cavern and injected air mass flow during charging are shown in Fig. 2. Mass flow of the air released from the cavern and V2 valve openness during discharging are shown in Fig. 3. Changes of pressure inside the cavern during discharging are shown in Fig. 4.

During charging, non-linearity is visible in the mass flow (Fig. 2).
As regards pressure, non-linearity would be visible at lower values of this parameter (if the cavern were loaded from a starting point of atmospheric pressure). The same conclusion can be formulated for pressure during discharging of the system (Fig. 4).

As mentioned above, while emptying caverns (Fig. 3) the control system maintains mass flow at a constant level by adjusting the degree of opening of valve V2. The curve showing this parameter is highly nonlinear.

4.2. Performance of the system

Energy consumption and production by chosen elements of the system are shown in Fig. 5. The energy required for the pump is clearly orders of magnitude lower than the energy requirements of compressors or energy from turbines. In total, the pumps consume 1,811 MWh/cycle. Energy consumption of the compressors is 1915 MWh/cycle. The overall energy generated in turbine part of
the system is 952.373 MWh/cycle. Momentary power consumption of compressors was 60 MW (the same as in the previous studies [6,7,19]) and momentary power of whole turbine - 161 MW. With the same air flow, a gas turbine equipped with combustion chambers can achieve power of 320 MW [6,7,19].

Round trip efficiency of the system was 50%. Without recovery of heat of compression, similar system could have efficiency of 43.5% (based on previous studies [6,7,19]). This means that the use of recovery of heat of compression increases the efficiency of the system, but at the same time significantly reduces its power.

4.3. Exergy destruction

Exergy destruction for chosen elements of A-CAES is shown in Fig. 6.

The biggest exergy destruction occurs in compressors (276.191 MWh/cycle in total) and turbines (190.394 MWh/cycle in total). Major destruction of exergy was also reported in control valve V2 (throttling to 43 bar), heat exchanger HEX1 and oil cooler OC1. Exergy destruction in oil cooler OC2 was quite small due to the small oil flow through it (16.842 kg/s). For comparison, the oil flow through OC1 was over 313 kg/s (which resulted in major exergy destruction). Exergy destruction in the pumps is negligible compared to the other elements.

Total exergy destruction was about 440 MWh/cycle during the charging process and about 451 MWh/cycle during the discharging process.

5. Conclusions

In the described adiabatic CAES system heat generated during the compression of air is stored in the hot oil tank. Due to throttling on valve V2 not all of the stored heat could be used during the discharging process. Therefore, oil coolers (OC1 and OC2) had to be used. This excess heat could be used for example in ORC, which would increase the efficiency of the entire system. This process could be performed in a continuous manner via pump P8, thereby avoiding unnecessary repeated start-up and shutdown of ORC. Round trip efficiency of the modelled A-CAES power plant was 50%. The heat delivered to cooling water through IC1, OC1 and OC2 could be used for district heating. This would increase the overall efficiency of the system.

The use of recovery of heat of compression increases the efficiency of the system, but at the same time significantly reduces its power output.

Efficiency could be improved and exergy destruction reduced if the compression and expansion process was divided into more steps, more heat exchangers were inserted (supplied by oil from thermal energy storage) between the steps, and throttling was eliminated upstream of the turbine.

Total exergy destruction during the charging process was almost equal to the total exergy destruction during the discharging process of the modelled CAES system.

Acknowledgments

This work was supported by the European Union in the framework of the European Social Fund (POKL 04.01.01-00-061/10) through the “Didactic Development Program of the Faculty of Power and Aeronautical Engineering of the Warsaw University of Technology”.

References


