Thermodynamic Analysis of a Novel Compressed Supercritical Carbon Dioxide Energy Storage System

Jun Huang*, Hui Liu
Shenhua Guohua (Beijing) Electric Power Research Institute CO., LTD, Beijing, 100026, P.R.China
*Corresponding author’s e-mail: huangjun0522@126.com

Abstract. To improve the thermodynamic efficiency of compressed air energy storage system, a novel compressed gas energy storage system using supercritical carbon dioxide was proposed. Energy and exergy analyses were used to study the thermodynamic performance of the novel energy storage system. The thermodynamic analysis results show that the energy storage system based on supercritical CO₂ has a better performance and simpler system configurations compared with CAES (Compressed Air Energy Storage, CAES). The exergy efficiency and round trip efficiency of compressed supercritical CO₂ energy storage system are 57.02% and 73.02%, respectively. And the highest exergy destruction occurs in the recuperator, accounting 36.51% of the total system exergy destruction, followed by the heater (32.33%) and then turbine, low-pressure reservoir etc.

1. Introduction
Now more and more concerns has been put into the security and reliability of power supply, because the renewable energy sources are remarkable and uncontrollable intermittency[1]. Energy storage technologies, especially large-scale energy storage system can effectively regulate the intermittent renewable power production to match customer demand[2].

On a utility scale, CAES (Compressed Air Energy Storage, CAES) is gaining an increasing attention, because it can contribute to integrating fluctuating and intermittent renewable energy sources into electricity grid[3]. Many researches have proved that using CAES can turn fluctuating renewable power (e.g. wind power, solar power, etc) into a flexible power[4-5]. However, lower thermal efficiency (e.g. Huntorf CAES plant efficiency is 42% and AA-CAES efficiency is about 70%(6]) and larger scale of storage cavern (e.g. total volume of Huntorf CAES storage cavern is 310000m³) are the main drawbacks of CAES system.

A lot of innovative compressed gas energy storage systems have been proposed which have a higher efficiency or smaller energy storage cavern volume[7-15]. Although the efficiency can be improved through using the compressed heat or waste heat, some new equipment were added into the new systems that increased the cost and complicated systems.

As we all know, the non-ideal gas CO₂ has been applied to a lot of purposes in different areas, such as the CO₂ Brayton cycle has a higher thermodynamic efficiency than traditional ideal gas Brayton cycle[16-18]. The fluid density and heat capacity of supercritical CO₂ (SCO₂) are higher than ideal gas, while the compressibility factor of SCO₂ is lower than the ideal gas, therefore the SCO₂ storage reservoir can be substantially decreased and turbomachinery designs can be extremely compacted. Considering these advantages, SCO₂ could be used in compressed gas energy storage system, such as
using SCO₂ instead of air in CAES will increase the thermal efficiency and reduce the storage cavern volume[19].

In this paper, exergy analysis is applied to the compressed supercritical CO₂ energy storage using two reservoirs to investigate the exergy destruction among each component and the whole system. The results are expected to help in guiding energy reasonable use and exploring working characteristics of compressed supercritical CO₂ energy storage system.

2. System description

The compressed supercritical CO₂ energy storage system containing two reservoirs, as shown in Fig. 1. The supercritical CO₂ from the last stage turbine train is exhausted into low-pressure cavern to store, whereas the supercritical CO₂ from the last stage compressor train is stored in high-pressure cavern.

The compressed supercritical CO₂ energy storage system works as follows:

1: During off-peak hours, the surplus power is used to drive compressor to compress supercritical CO₂ from low-pressure storage reservoir and stored in high-pressure reservoir.

2: To protect the high-pressure storage reservoir and improve the thermal efficiency, the high temperature SCO₂ came from the last stage compressor train is firstly put into after cooler to absorb the heat from compression process. The waste heat produced during compression process is stored in thermal energy storage.

3: During peak hours, the SCO₂ stored in high-pressure storage reservoir is released and regulated to a certain value by the throttle valve, and absorb the heat exhausted from the compressor and turbine in the pre-heater and recuperator, respectively.

4: SCO₂ is heated into a certain value in the heater, and then fed into the turbine to generate electricity.

5: The CO₂ from the last stage turbine is exhausted and stored in the low-pressure storage reservoir.

3. Methodology and modeling

3.1. Exergy analysis

To make a thermodynamic analysis using Second Law of Thermodynamics, the input exergy (fuel exergy) and output exergy (product exergy) concepts are applied to exergy analysis [23]. The exergy balance equation is
\[ \dot{E}_{D,j} = \dot{E}_{F,j} - \dot{E}_{P,j} \]  

where \( \dot{E}_{D,j} \), \( \dot{E}_{F,j} \), \( \dot{E}_{P,j} \) respectively represent the exergy destruction rate, fuel exergy rate and product exergy rate within the \( i \)th component.

For the overall system, the general exergy balance can be presented as following
\[ \dot{E}_F = \dot{E}_P + \sum_{i} \dot{E}_{D,i} + \dot{E}_L \]  

where \( \dot{E}_F \) represents the fuel exergy rate within the overall considered system, \( \dot{E}_p \) represents the product exergy rate within the overall considered system, \( \sum_{i} \dot{E}_{D,i} \) represents the total amount of exergy destruction rate within the overall considered system, and \( \dot{E}_L \) is the exergy loss rate within the overall considered system.

The \( i \)th component exergy efficiency, \( \eta_{e,i} \), is defined as
\[ \eta_{e,i} = \frac{\dot{E}_{P,i}}{\dot{E}_{F,i}} \]  

To compare the exergy destruction of different components within the overall considered system, the \( i \)th component exergy destruction ratio, \( \eta_{D,i} \), and relative exergy destruction, \( \eta_{D,i}^* \), can be respectively calculated by the following equations.
\[ \eta_{D,i} = \frac{\dot{E}_{D,i}}{\dot{E}_{F,i}} \]  
\[ \eta_{D,i}^* = \frac{\dot{E}_{D,i}}{\sum_{i} \dot{E}_{D,i}} \]

The exergy analysis methodology of each component in the compressed supercritical CO\(_2\) energy storage system have been detailed analyzed in Ref.[19],and are shown in Table 1.

Table 1. Exergy analysis model of each component of the compressed supercritical CO\(_2\) energy storage

| Component     | Schematic view | \( \dot{E}_{F,i} \) | \( \dot{E}_{P,i} \) | \( \dot{E}_{D,i} \) |
|---------------|----------------|----------------------|---------------------|---------------------|
| Compressors   |                | \( \dot{E}_{C,C} \)  | \( \dot{E}_{C,C} \)  | \( \dot{E}_{C,C} \)  |
| Storage cavern|                | \( \dot{E}_{1,SC} \) | \( \dot{E}_{2,SC} \) | \( \dot{E}_{1,SC} \) - \( \dot{E}_{2,SC} \) |
| Heater        |                | \( \dot{E}_{Q,H} \)  | \( \dot{E}_{2,H} \)  | \( \dot{E}_{1,H} \)  |
| Heat exchanger|                | \( \dot{E}_{1,HE} \) | \( \dot{E}_{2,HE} \) | \( \dot{E}_{1,HE} \) |
Table 2. Main design parameters of the compressed supercritical CO$_2$ energy storage system

| Name                          | Value |
|-------------------------------|-------|
| Inlet temperature of turbine/°C | 600   |
| Inlet pressure of turbine/bar  | 200   |
| Inlet pressure of the first stage compressor/bar | 75    |
| Outlet pressure of the last stage turbine/bar | 80    |

3.2. Compressed supercritical CO$_2$ energy storage model

3.2.1. Compressor model.
The compressor isentropic efficiency, $\eta_c$, is defined as

$$\eta_c = \frac{\dot{h}_{out,c} - \dot{h}_{in,c}}{\dot{h}_{out,c} - \dot{h}_{in,c}}$$  \hspace{1cm} (6)

where $\dot{h}_{in,c}$ represents the inlet enthalpy of compressor, $\dot{h}_{out,c}^s$ and $\dot{h}_{out,c}$ represent the outlet enthalpy during isentropic compression and real compression process, respectively.

According to the First Law of Thermodynamics, the inlet and outlet entropies are equal during ideal compression process, i.e.

$$S_{out,c}^s = S_{out,c}$$  \hspace{1cm} (7)

The outlet enthalpy during the real compression process can be calculated using the following thermodynamic relationship equation

$$\dot{h}_{out,c}^s = f\left(\dot{h}_{out,c}^s, P_{out,c}\right)$$  \hspace{1cm} (8)

$$\dot{h}_{out,c} = \dot{h}_{in,c} + \eta_c(\dot{h}_{out,c}^s - \dot{h}_{in,c})$$  \hspace{1cm} (9)

3.2.2. Heat exchanger model

The thermodynamic properties of supercritical CO$_2$ (e.g. specific heat, density and viscosity, etc) drastically vary with variation of temperature, which will have a huge impact on system thermodynamic performance. To make sure constant properties in each section, dividing the after cooler, pre-heater and recuperator into relatively small sections is essential.

$$\dot{Q}_i = m_{CO_2} C_{p,CO_2,j} \left(\tau_{CO_2,j+1} - \tau_{CO_2,j}\right)$$  \hspace{1cm} (10)

$$\dot{Q}_i = m_{water} C_{p,water,j} \left(\tau_{water,j+1} - \tau_{water,j}\right)$$  \hspace{1cm} (11)

$$m_{water} = \frac{\sum_{j} \dot{Q}_i}{(h_{water,out} - h_{water,in})}$$  \hspace{1cm} (12)
3.2.3. Turbine model

The isentropic expansion process efficiency is defined as

\[ \eta_{\text{ie}} = \frac{h_3 - h_4}{h_3 - h_1} \]  

(13)

According to the First Law of Thermodynamics, the inlet and outlet entropies are equal during isentropic expansion process, i.e.

\[ S_{4s} = S_3 \]  

(14)

The outlet stream enthalpy during the isentropic expansion process can be calculated from the thermodynamic property relationship, \( f(x_1, x_2) \), which is calculated using following equation, which is derived from the equation of state. That is

\[ h_{4s} = f(s_{4s}, p_4) \]  

(15)

The actual outlet enthalpy of supercritical CO\(_2\) from turbine train can be calculated using equations (13) and (15). Therefore, the output work during expansion process can be obtained as following

\[ w_{\text{te}} = h_3 - h_4 \]  

(16)

3.2.4. Throttle valve model

As we all know, the thermodynamic parameters, especially pressure, in the high-pressure and low-pressure reservoirs will change. To make sure that the inlet thermodynamic parameters of compressor and turbine is constant, throttle valves are necessary. During the throttling process the enthalpy is constant, as shown in following equation.

\[ h_1(P_1, T_1) = h_2(P_2, T_2) \]  

(17)

The outlet temperature, \( T_2 \), can be calculated using following equation

\[ T_2 = f(P_2, P_1, T_1) \]  

(18)

4. Results and discussions

We use MATLAB language and CoolProp database to evaluate the performance of compressed supercritical CO\(_2\) energy storage[20]. The compressed supercritical CO\(_2\) energy storage system round trip efficiency and exergy efficiency are 73.02% and 57.02%, respectively. The thermodynamic performance is better than that of AA-CAES and energy storage system based on liquid carbon dioxide (round trip efficiency is 67.22% and 56.64%).

Table 3. Thermodynamic data for the material streams of compressed supercritical CO\(_2\) energy storage

| Streams | Material stream | T/°C | P/bar | h/kJ/kg | s/kJ/(kg·K) | e/kJ/kg |
|---------|----------------|------|-------|---------|-------------|--------|
| 1       | CO\(_2\)       | 31.92| 75.00 | 347.56  | 1.48        | 228.79 |
| 2       | CO\(_2\)       | 91.37| 250.00| 384.87  | 1.50        | 261.36 |
| 3       | CO\(_2\)       | 40.00| 250.00| 272.34  | 1.17        | 251.23 |
| 4       | CO\(_2\)       | 33.41| 200.00| 262.08  | 1.15        | 245.51 |
| 5       | CO\(_2\)       | 76.68| 200.00| 367.15  | 1.47        | 252.00 |
| 6       | CO\(_2\)       | 396.59| 200.00| 846.18  | 2.49        | 418.16 |
| 7       | CO\(_2\)       | 600.00| 200.00| 1097.20 | 2.81        | 568.41 |
| 8       | CO\(_2\)       | 490.21| 80.00 | 973.01  | 2.84        | 436.65 |
| 9       | CO\(_2\)       | 81.68| 80.00 | 493.98  | 1.93        | 236.31 |
| 10      | Water          | 35   | 10.00 | 146.90  | 0.50        | 0.71   |
| 11      | Water          | 86.37| 10.00 | 361.87  | 1.15        | 17.07  |
| 12      | Water          | 86.37| 10.00 | 361.87  | 1.15        | 17.07  |
The exergy analysis results of compressed supercritical CO₂ energy storage system are shown in Fig. 2 and Table 4, respectively. The results shown in Fig. 2 and Table 4 reveal that the largest exergy destruction is happen in the recuperator, \( \eta_{D,RE} = 36.58\% \), the second one is heater, \( \eta_{D,HE} = 32.39\% \). From Fig. 2 we can know that the exergy destruction of low-pressure reservoir and high-pressure reservoir are 8.05\% and 6.12\%, respectively, but both of them have a high exergy efficiency (\( \eta_{e,LS} = 0.97 \), \( \eta_{e,HS} = 0.98 \)). The overall exergy destruction consists of the exergy destruction within each component, while the exergy loss is from the two reservoirs since it is not further used. The overall considered system exergy efficiency is 57.02\%, and as a consequence, 42.98\% of the total input exergy is destroyed. Nearly one third of overall exergy of fuel is destroyed in heater and recuperator (\( \eta_{D,HE} = 0.14 \), \( \eta_{D,RE} = 0.16 \)), which also explains that the more attention should be paid to heater and recuperator, as shown in Table 4.

### Table 4. Results of the conventional exergy analyses.

| Component | \( \dot{E}_{P,i} / \text{kJ/h} \) | \( \dot{E}_{P,i} / \text{kJ/h} \) | \( \dot{E}_{D,i} / \text{kJ/h} \) | \( \eta_{e,i} \) | \( \eta_{D,i} \) |
|-----------|-------------------------------|-------------------------------|-------------------------------|----------------|----------------|
| C1        | 134316                        | 117252                        | 17064                         | 0.87           | 0.22           |
| AC        | 36468                         | 30816                         | 5652                          | 0.85           | 0.01           |
| PH        | 30132                         | 23400                         | 6732                          | 0.78           | 0.01           |
| HE        | 649836                        | 540864                        | 108972                        | 0.83           | 0.14           |
| T1        | 474336                        | 447084                        | 27252                         | 0.94           | 0.03           |
| RE        | 721224                        | 598176                        | 123058                        | 0.83           | 0.16           |
| LS        | 850716                        | 823644                        | 27072                         | 0.97           | 0.03           |
| HS        | 904428                        | 883836                        | 20592                         | 0.98           | 0.03           |
| Total     | 784152                        | 447084                        | 337068                        | 0.57           | 0.43           |

![Figure 2. Exergy destruction ratio of compressed supercritical CO₂ energy storage system.](image)

### 5. Conclusion

A compressed gas energy storage using supercritical CO₂ as working fluid has been analyzed in this paper. An exergy analyses of the proposed system is carried out. The main conclusions are summarized as follows:

1. The compressed supercritical CO₂ energy storage system has a better round-trip efficiency (73.02\%) than that of AA-CAES and energy storage system based on liquid carbon dioxide (67.22\% and 56.64\%)


(2) It is obtained from the exergy analyses that the highest exergy destruction occurs in the recuperator, accounting 36.51% of the total system exergy destruction, followed by the heater (32.33%) and then turbine, low-pressure reservoir etc.

Acknowledgments
This work was supported by National Key R&D Program of China (2018YFB0604203).

References
[1] Venkataramani G, Parankusam P, Ramalingam V, et al. A review on compressed air energy storage—A pathway for smart grid and polygeneration[J]. Renewable and Sustainable Energy Reviews, 2016, 62: 895-907.
[2] Briola S, Di Marco P, Gabbielli R, et al. A novel mathematical model for the performance assessment of diabatic compressed air energy storage systems including the turbomachinery characteristic curves[J]. Applied Energy, 2016, 178: 758-772.
[3] Swider D J. Compressed air energy storage in an electricity system with significant wind power generation[J]. IEEE transactions on energy conversion, 2007.
[4] Paul D, Ramteen S. The value of compressed air energy storage with wind in transmission-constrained electric power systems[J]. Energy policy, 2009, 37(8): 3149-3158.
[5] Sayyad N, Afshin N G, Majid M, et. al. Optimal bidding and offering strategies of merchant compressed air energy storage in deregulated electricity market using robust optimization approach[J]. Energy, 2018, 142: 250-257.
[6] Ivan C, Claudio A C, Kankar B. Compressed air energy storage system modeling for power system studies[J]. IEEE transactions on power systems, 2019.
[7] An integrated design for hybrid combined cooling, heating and power system with compressed air energy storage[J]. Appl Energy, 2018, 210: 1151-1166.
[8] Nielsen L, Leithner R. Dynamic Simulation of an Innovative Compressed Air Energy Storage Plant-Detailed Modelling of the Storage Cavern[J]. WSEAS Trans Power Sys 2009; 4(8): 253-263.
[9] Qing H, Hui L, Yinping H, et. al. Thermodynamic analysis of a novel supercritical compressed carbon dioxide energy storage system through advanced exergy analysis[J]. Renewable energy, 2018, 127: 835-849.
[10] Sammy H, Mohammad J, Peggy I P, et. al. Thermodynamic analysis of a high temperature hybrid compressed air energy storage (HTH-CAES) system[J]. Renewable energy, 2018, 115:1043-1054.
[11] Amin M, Mehdi, M. Exergy analysis and optimization of an integrated micro gas turbine, compressed air energy storage and solar dish collector process[J]. Journal of cleaner production, 2016, 139: 372-383.
[12] Andrew P, Seamus D G. Underwater compressed air energy storage[J]. Storing energy, 2016:135-154.
[13] Akinyele D O, Rayudu R K. Review of energy storage technologies for sustainable power networks[J]. Sustainable energy technologies and assessments, 2014, 8:74-91.
[14] Bharath K, Seamus G, Andrew P. Compressed air energy storage with liquid air capacity extension[J]. Applied energy, 2015, 157: 152-164.
[15] Hui L, Qing H, Andrea B, et al. Thermodynamic analysis of a compressed carbon dioxide energy storage system using two saline aquifers at different depths as storage reservoirs[J]. Energy conversion and management. 2016, 127: 149-159.
[16] Kim Y M, Favrat D. Energy and exergy analysis of a micro-compressed air energy storage and air cycle heating and cooling system[J]. Energy, 2010, 35(1): 213-220.
[17] Dostal V, Driscoll M J, Hejzlar P. A supercritical carbon dioxide cycle for next generation nuclear reactors[J]. Massachusetts Institute of Technology. Dept. of Nuclear Engineering, Cambridge, MA, Paper No. MIT-ANP-TR-100, 2004.
[18] Nguyen T V, Voldsund M, Elmegaard B, et al. On the definition of exergy efficiencies for petroleum systems: Application to offshore oil and gas processing[J]. Energy, 2014, 73: 264-281.
[19] Hui L, Qing H, Saeed S B. Thermodynamic analysis of a compressed air energy storage system through advanced exergetic analysis[J]. Journal of Renewable and Sustainable Energy, 2016, 8(3):03401.
[20] CoolProp integration in MATLAB. <https://www.mathworks.com/matlabcentral/answers/182384-coolprop-integration-in-matlab>