1 **Thermodynamic analysis of a compressed carbon dioxide energy storage system**
2 **using two saline aquifers at different depths as storage reservoirs**

3 Hui Liu^{1,2}, Qing He¹, Andrea Borgia², Lehua Pan ², Curtis M. Oldenburg^{2*}

4
5 ¹School of Energy Power and Mechanical Engineering,
6 North China Electric Power University,
7 2 Beinong Road, Changping, Beijing 102206
8 China
9 ²Energy Geosciences Division, 74-316C
10 Lawrence Berkeley National Laboratory
11 Berkeley, CA 94720
12 USA

13
14
15 Corresponding author: Curtis M. Oldenburg,
16 Energy Geosciences Division, 74-316C
17 Lawrence Berkeley National Laboratory
18 Berkeley, CA 94720
19 USA
20 Email address: cmoldenburg@lbl.gov
21

22 **Abstract:** Compressed air energy storage (CAES) is one of the leading large-scale
23 energy storage technologies. However, low thermal efficiency and low energy storage
24 density restrict its application. To improve the energy storage density, we propose a
25 two-reservoir compressed CO₂ energy storage system. We present here
26 thermodynamic and parametric analyses of the performance of an idealized
27 two-reservoir CO₂ energy storage system under supercritical and transcritical
28 conditions using a steady-state mathematical model. Results show that the
29 transcritical compressed CO₂ energy storage system has higher round-trip efficiency
30 and exergy efficiency, and larger energy storage density than the supercritical

31 compressed CO₂ energy storage. However, the configuration of supercritical
32 compressed CO₂ energy storage is simpler, and the energy storage densities of the two
33 systems are both higher than that of CAES, which is advantageous in terms of storage
34 volume for a given power rating.

35 **Key words:** Subsurface energy storage; Compressed CO₂ energy storage system;
36 Utilization of CO₂; Two saline aquifers reservoirs; Thermodynamic analysis;
37 Parametric analysis.

38 Nomenclature

H	enthalpy, kJ/kg
S	entropy, kJ/(kg·K)
P	pressure, MPa
$\dot{E}$	exergy, kW
T	temperature, K
T_s	surface temperature, K
W	shaft work, kW
C_p	specific heat capacity at constant pressure, kJ/(kg·K)
G	geothermal gradient, K/km
V	volume, m ³
$\dot{m}$	mass flow rate, kg/s
$\dot{Q}$	heat transfer, W
Z	depth of saline reservoir, m

Abbreviations

A-CAES	adiabatic CAES
AA-CAES	advanced adiabatic CAES
C	compressor
CAES	compressed air energy storage
CCES	compressed CO ₂ energy storage
HE	heater
HS	high pressure reservoir

LS	low pressure reservoir
PM-CAES	porous media CAES
RE	recuperator
SC-CCES	supercritical compressed CO ₂ energy storage
SC-CO ₂	supercritical CO ₂
T	turbine
TC-CCES	transcritical compressed CO ₂ energy storage
TC-CO ₂	transcritical CO ₂

Greek symbols

β_p	pore compressibility, Pa ⁻¹
β_w	change in brine density
η	efficiency
τ	temperature difference, K
ρ	density, kg/m ³

Subscripts

S	isentropic process
Comp	compressor
1	inlet stream
2	outlet stream
T	turbine
NG	nature gas
F	fuel

tot

total

D

destruction

L

loss

39 **1. Introduction**

40 In recent years, renewable energy, particularly wind power and solar photovoltaic (PV)
41 generation has demonstrated robust growth-worldwide motivated by concerns about
42 energy security and climate change due to CO₂ emission levels^[1-2]. But Renewable
43 energy sources (e.g., solar and wind energy) exhibit significant and uncontrollable
44 intermittency during power production. When these renewable energy sources are
45 connected to an electrical grid, they can cause serious safety problems for the grid;
46 hence, it is difficult to deliver power from renewable energy sources that instantly
47 matches electricity demand^[3].

48
49 To solve this dilemma and develop renewable energy sources further, viable energy
50 storage systems (ESS) are required. For example, an efficient ESS can increase the
51 penetration of wind power generation by controlling wind power plant output and
52 storage, in addition to providing ancillary services to the power system^[4-5].

53
54 On a utility scale, compressed air energy storage (CAES) is one of the technologies
55 with the highest economic feasibility with potential to contribute to a flexible energy
56 system with an improved utilization of intermittent renewable energy sources^[1]. The
57 feasibility of using CAES to integrate fluctuating renewable power into the electricity
58 grid has been proven by many researchers^[6-9]. Bosio and Verda^[6] analyzed the
59 thermo-economics of a CAES system integrated into a wind power plant in the

60 framework of the Italian Power Exchange market, which showed that a hydroelectric
61 power plant (HPP)-CAES system was cost-effective in terms of solving local
62 imbalances of the grid. Clearly et al.^[7] evaluated the economic benefits of CAES in
63 mitigating wind curtailment. They showed that both wind curtailment levels and
64 wind-farm total annual generation costs could be decreased. Arabkoohsar et al.^[8-9]
65 simulated and analyzed CAES equipped with a solar heating system. The results
66 showed that CAES could increase the efficiency and reliability of a PV plant.

67
68 However, the main drawbacks of a CAES system include its low thermal efficiency
69 (e.g., Huntorf CAES plant efficiency is 42% and AA-CAES efficiency is about
70 70%^[10]), CO₂ emissions from combustion of natural gas in the recovery system for
71 conventional CAES, the need for high temperature thermal storage and temperature
72 resistant materials for adiabatic CAES (A-CAES). These factors limit further
73 development of CAES. Although large-scale caverns are also required for CAES as it
74 is carried out today, porous media systems such as aquifers and depleted natural gas
75 reservoirs, so-called porous media CAES (PM-CAES) systems, offer much more
76 storage capacity^[11].

77
78 Thermodynamic analyses of CAES systems have been performed to optimize these
79 systems and improve their thermal efficiency. For example, Buffa et al.^[12] conducted
80 an exergy analysis of A-CAES and found that exergy destruction mostly occurred in
81 the compressors and coolers. Proczka et al.^[13] analyzed the effects of pressure and the

82 efficient sizing of pressure vessels on CAES. Zhang et al.^[14-15] analyzed the
83 thermodynamic effects of thermal energy storage (TES) and the air storage chamber
84 model on a CAES system. Jubeh and Najjar et al.^[16] explored the effects of operating
85 variables on A-CAES performance. Najjar and Zamout analyzed the effects of dry
86 regions on the performance of a CAES plant^[17]. The operation, experience, and
87 characteristics of Huntorf CAES were also investigated^[18]. Thermodynamic analyses
88 have shown that, both, decreasing the exhaust temperature and using heat of
89 compression during expansion can significantly improve CAES efficiency.

90

91 Several novel CAES systems have been proposed that reduce waste heat. A
92 recuperator was utilized to capture heat from the turbine exhaust, which could reduce
93 the fuel consumption of the McIntosh plant by 25%^[19-20]. Safaei and Keith^[17]
94 proposed a distributed CAES (D-CAES) system that placed compressors near heat
95 demand loads to recover the heat generated during the compression stage. Liu^[2]
96 proposed a modified A-CAES system that used a pneumatic motor instead of a low
97 pressure turbine (LT) to reduce the exhaust temperature caused by LT, and the exergy
98 efficiency can be improved by nearly 3% compared with that of the conventional
99 A-CAES system. Guo et al.^[21] proposed a novel A-CAES system in which an ejector
100 was integrated into an A- CAES system to recover pressure reduction losses; energy
101 conversion efficiency could reach 65.36%. Several demonstration A-CAES plants
102 have been built, such as a 1.5 MW A-CAES in China, where initial experimental tests
103 are on-going. An A-CAES technology that uses reversible reciprocating piston

104 machines is being developed by LightSail Energy Ltd. in the U.S. Other new systems
105 include a tri-generation system based on compressed air and thermal energy storage^[22],
106 biomass-fueled CAES, isobaric adiabatic CAES with combined cycle^[23], combined
107 cooling, heating and power system based on small-scale CAES^[24], CAES using a
108 cascade of phase change materials^[25], CAES combined with solar thermal capture^[26],
109 integrating CAES with diesel engine^[27], and compressed carbon dioxide energy
110 storage^[28].

111
112 Although thermal efficiency can be improved by various methods, CAES has low
113 energy density and requires large-scale storage reservoirs^[29]. To overcome these
114 restrictions, several studies have been conducted on novel energy storage technologies.
115 For instance, Kim^[30] proposed a constant-pressure CAES system combined with
116 pumped hydro-storage to reduce the cavern volume. Guo et al.^[31] presented a
117 supercritical compressed air energy storage (SC-CAES). Oldenburg and Pan^[11]
118 modeled a porous media CAES (PM-CAES) system that uses aquifers or depleted
119 natural gas reservoirs for storage. Underwater compressed air energy storage
120 (UWCAES) stores the compressed air under water by using a large elastic bladder^[32].
121 Small scale CAES (SS-CAES) that stores high-pressure air in a tank or an
122 underground pipeline was also proposed^[33]. Each of these novel approaches brings
123 with it additional requirements and limitations.

124

125 As popularly known, CAES is derived from the Brayton cycles, and gases like CO₂

126 that are non-ideal at operating conditions are more efficient in a Brayton cycle^[34].
127
128 Using CO₂ as the working fluid in a compressed gas energy storage system can also
129 achieve better performance than AA-CAES^[35]. At the same time geological CO₂
130 sequestration in deep formations (e.g., saline aquifers, gas and oil reservoirs, and coal
131 beds) is a promising measure for reducing greenhouse gas emissions^[36]. Therefore,
132 the combination of compressed gas energy storage in the deep subsurface and
133 large-scale utilization of CO₂ is both possible and beneficial.
134
135 Although, some research has been conducted on energy power cycle and energy
136 storage systems based on CO₂ and liquid CO₂^[28,35], we are not aware of published
137 analyses of energy storage systems based on transcritical CO₂ (transition from
138 supercritical to gas) or based on supercritical CO₂ throughout the cycle. Therefore, the
139 innovation of this paper resides in the exergy analysis of a closed-loop gas storage
140 system, conceived by two of us (Borgia and Oldenburg in January of 2012), which
141 comprises two reservoirs, in this case in saline aquifers but which could also be in
142 caverns, located at different depths and uses transcritical and supercritical CO₂ as the
143 working fluid. This novel energy storage system can be used in two different energy
144 cycles (e.g., transcritical CO₂ energy storage cycle, and supercritical CO₂ energy
145 storage cycle) according to the physical state of CO₂ in the process. We conducted
146 energy and exergy analyses to understand the thermal properties of the compressed
147 CO₂ energy storage system. In addition, parametric analysis was performed to

investigate the effects of the physical conditions of two saline aquifer reservoir (e.g., energy storage pressure, energy releasing pressure, and pressure of low-pressure reservoir) on system performance.

2. System description

The proposed compressed CO₂ energy storage system using two saline aquifers as storage reservoirs is a closed energy-storage cycle. The first reservoir is a low-pressure reservoir used to store CO₂ exhausted from the turbine, whereas the second reservoir is at higher pressure to store CO₂ from the compressor. This energy storage system, although based on the same principles, can be operated in two different ways according to the state of CO₂, (1) by allowing the CO₂ to transition from supercritical to gaseous conditions in the turbine, which we refer to as the transcritical compressed CO₂ energy storage (TC-CCES) system, and (2) by keeping the CO₂ above the critical pressure throughout the cycle, which we refer to as the supercritical compressed CO₂ energy storage (SC-CCES) system. The schematic and *T-S* diagram of compressed CO₂ energy storage (CCES) is shown in Figs. 1 and 2.

1: During off-peak hours, the working fluid (low-pressure CO₂) stored in a shallow low-pressure reservoir is removed, pressurized, and injected into a deeper high-pressure reservoir using surplus renewable power, such as that from wind or solar.

169 2: For multi-stage compressor, the heat of compression of the CO₂ is absorbed by a
170 cooling fluid which is stored in a TES system, while the CO₂ from the last stage
171 compressor is directly injected into the high-pressure storage reservoir.

172 3: During peak hours, the high-pressure CO₂ is regulated to a certain pressure through
173 the throttle valve, and is transported to the recuperator system to absorb the heat
174 exhausted from the turbine in the TES.

175 4: The heated high-pressure CO₂ is fed into the turbine.

176 5: The high pressure CO₂ expands through the turbine generating shaft work.

177 6: The exhaust CO₂ is stored in the low-pressure reservoir.

178

179 Considering that because of the geothermal gradient the temperature of the
180 saline-aquifer reservoirs increases with depth, the output CO₂ from the compressor
181 stage can be directly injected into the high-pressure reservoir storing, both heat and
182 CO₂ directly into the rock formation. Therefore, the proposed SC-CCES does not
183 need a TES to store the compressed heat generated during compression, implying that
184 the aftercooler is theoretically unnecessary. In our analysis, though, to allow a direct
185 comparison with CAES and highlight the benefits of the two-reservoir CCES system,
186 we retain the heater.

Fig.1. Schematic illustration of CCES using two saline-aquifer reservoirs.

Fig.2. CCES using two saline aquifers reservoirs. a. The schematic of TC-CCES. b. The schematic of SC-CCES. c. T - S diagram of TC-CCES. d. T - S diagram of SC-CCES. C = compressor; RE = recuperator; T = turbine; HE = heater; LS = low pressure reservoir; and HS = high pressure reservoir.

195 **3. Theoretical model**

196 The following assumptions are made to simplify the theoretical model of the
197 compressed CO₂ energy storage cycle:

- 198 1. The pressure drop and heat loss in the pipes, heat exchanger, TES, and recuperator
199 are ignored.
- 200 2. The compressor and the turbine have a given isentropic efficiency.
- 201 3. Changes in kinetic energy and potential energy are negligible relative to the stored
202 energy.
- 203 4. The storage in the two saline aquifers is considered a closed system at hydrostatic
204 pressure.
- 205 5. The mass flow rate of CO₂ is the same in the storage and recovery modes of
206 operation, and for TC-CCES and SC-CCES.

207 **3.1. Compressor model**

208 The isentropic efficiency of compressor η_{comp} is,

$$209 \qquad \eta_{\text{comp}} = \frac{h_{2s} - h_1}{h_2 - h_1}, \qquad (1)$$

210 where h_1 is inlet enthalpy and h_{2s} is outlet enthalpy during isentropic compression, h_2
211 is the real enthalpy during compression.

212 During isentropic compression, the entropies of the initial and final states are the same,
213 i.e.,

$$214 \qquad s_{2s} = s_1, \qquad (2)$$

215 The corresponding enthalpy of the outlet stream at the end of isentropic compression
 216 can be calculated from the property relationship, f , which is derived from the equation
 217 of state,

$$218 \quad h_{2s} = f(s_{2s}, p_2), \quad (3)$$

219 The actual enthalpy of the pressurized CO₂ at the outlet of the compressor can be
 220 calculated using the definition of compression efficiency. Hence, the power consumed,
 221 w_{comp} , is,

$$222 \quad w_{\text{comp}} = h_2 - h_1, \quad (4)$$

223 **3.2. Heat exchanger model**

224 The properties, such as density, specific heat, and viscosity are observed to undergo
 225 drastic variations within a very narrow range of temperature if the CO₂ is under
 226 supercritical pressure, which will have a great effect on system performance.
 227 Therefore, it is essential to divide the heat exchanging process into adequately small
 228 sections, such that property variations in each section are so small that constant
 229 properties can be assumed ^[37].

230
 231 The inner cooler, pre-cooler, and recuperator function together as the heat exchanger.
 232 For the inner and pre-cooler, we assumed that the upper terminal temperature
 233 difference $\Delta\tau_{\text{upper}}$, and inner and pre-cooler temperatures are both constant. To obtain
 234 the amount of compressed heat, the overall temperature change for CO₂ is divided
 235 into N equal differences $\Delta\tau$. The specific heat at constant pressure at each

intermediate state is determined from the known pressure and temperature. The heat transfer for each step i and mass flow rate of water are calculated from the following equations:

$$\dot{Q}_i = \dot{m}_{\text{CO}_2} C_{\text{P,CO}_2,i} (\tau_{\text{CO}_2,i+1} - \tau_{\text{CO}_2,i}), \quad (5)$$

$$\dot{Q}_i = \dot{m}_{\text{water}} C_{\text{P,water},i} (\tau_{\text{water},i+1} - \tau_{\text{water},i}), \quad (6)$$

$$\dot{m}_{\text{water}} = \frac{\sum_i^N \dot{Q}_i}{(h_{\text{water,out}} - h_{\text{water,in}})}, \quad (7)$$

For the recuperator, we assumed that lower terminal temperature difference, $\Delta\tau_{\text{lower}}$, is constant. Also the enthalpy change for hot streams is divided into N equal differences Δh . The cooling working fluid temperatures of preheater and recuperator at each intermediate state can be determined using the CoolProp database from the known enthalpy and pressure.

3.3. Aquifer CO₂ storage model

To inject CO₂ into a saline aquifer reservoir, the CO₂ pressure has to be at least as high as the initial groundwater pressure in the reservoir. In the present study, the CO₂ pressures in the low-pressure reservoir and high-pressure reservoir are assumed to exhibit hydrostatic variation with depth at a constant pressure gradient.

The geothermal gradient (underground temperature increases with depth) will control the temperature of the reservoir. Using values for the surface temperature and geothermal gradient, the underground temperature as a function of depth can be

256 determined by,

$$257 \quad T = T_s + Gz, \quad (8)$$

258 where T_s is the surface temperature; G is the geothermal gradient; z is the depth.

259

260 We assume the saline aquifer is a closed-storage formations^[36]. To estimate aquifer

261 storage reservoir volume, V_s , the following equation is used,

$$262 \quad V_s = \frac{M_{CO_2}}{(\beta_p + \beta_w) \rho_{CO_2} \Delta P}, \quad (9)$$

263 where M_{CO_2} and ρ_{CO_2} are respectively the mass and density of CO_2 at reservoir

264 conditions; Δp is the pressure buildup from beginning to the end of injection; β_p is the

265 pore compressibility; β_w is the change in brine density.

266 **3.4. Turbine model**

267 The calculation method for the actual expansion is the same as that for compression.

268 The isentropic efficiency of the turbine can be calculated using,

$$269 \quad \eta_t = \frac{h_3 - h_{4s}}{h_3 - h_4}, \quad (10)$$

270

The entropies of the initial and final states are the same during isentropic expansion,

271

i.e.,

$$272 \quad S_{4s} = S_3, \quad (11)$$

273

The corresponding enthalpy of the outlet stream at the end of isentropic expansion can

274

be calculated from the property relationship, f , which is derived from the equation of

275

state. That is,

$$276 \quad h_{4s} = f(s_{4s}, p_4), \quad (12)$$

277 The actual enthalpy of the pressurized CO_2 at the outlet of the compressor can be

278 calculated using the definition of expansion efficiency. Thus, expansion work can be
279 calculated using,

$$280 \quad w_T = h_3 - h_4, \quad (13)$$

281 **4. Performance`criteria**

282 To analyze the performance of compressed CO₂ energy storage using two saline
283 aquifers as reservoirs, exergy efficiency, exergy destruction, round-trip efficiency, and
284 energy storage density are introduced as the performance criteria of the overall system
285 and the main components^[2,38].

286 **4.1. Energy analysis**

287 *4.1.1. Round-trip efficiency*

288 To facilitate comparisons of the novel energy storage system to other electrical
289 storage devices, the round-trip efficiency of energy storage is defined as^[38-39],

$$290 \quad \eta_{RT,1} = \frac{E_T}{E_C + \eta_{NG} E_F}, \quad (14)$$

291 where E_T represents the electricity output; E_C represents the electricity input; $\eta_{NG} E_F$
292 represents the amount of electricity that could have been made from the natural gas
293 input E_F , if that fuel had been used to make electricity in a stand-alone power plant at
294 efficiency η_{NG} instead of to fire an energy storage unit, $\eta_{NG} = 47.6\%$ ^[38].

295 4.1.2. Energy storage density

296 Determining the amount of electrical energy that can be produced per unit volume of
 297 storage capacity (E_{GEN}/V_s) is essential to evaluate the geological requirements for
 298 compressed gas energy storage. The energy produced per unit volume for compressed
 299 CO₂ energy storage with two saline reservoirs is,

$$300 \quad E_{\text{GEN}}/V_s = \frac{w_l (\beta_p + \beta_w) (\rho_{1,\text{CO}_2} \Delta P_1 + \rho_{2,\text{CO}_2} \Delta P_2)}{2}, \quad (15)$$

301 4.2. Exergy analysis

302 The general exergy balance for the overall system is^[40-41],

$$303 \quad \dot{E}_{\text{F,tot}} = \dot{E}_p + \sum_k \dot{E}_{\text{D},k} + \dot{E}_L, \quad (16)$$

304 where $\dot{E}_{\text{F,tot}}$, $\dot{E}_p$, $\sum_k \dot{E}_{\text{D},k}$, and $\dot{E}_L$ represent the total amount rate of fuel exergy, product
 305 exergy, exergy destruction, and exergy loss associated with the overall considered
 306 system, respectively.

307

308 The general exergy balance of the k^{th} component of the overall system can be
 309 expressed as follow:

$$310 \quad \dot{E}_{\text{D},k} = \dot{E}_{\text{F},k} - \dot{E}_{\text{P},k}, \quad (17)$$

311 where $\dot{E}_{\text{D},k}$, $\dot{E}_{\text{F},k}$, and $\dot{E}_{\text{P},k}$ represent the exergy destruction rate, the fuel exergy,
 312 and the product exergy rate within the k^{th} component, respectively. Exergy efficiency
 313 is defined as,

$$314 \quad \eta_{\text{ex}} = \frac{E_{\text{F,tot}} - \sum E_{\text{D},k} - E_L}{E_{\text{F,tot}}} = \frac{E_p}{E_{\text{F,tot}}}, \quad (18)$$

Exergy destruction in each component of the system can be calculated using the equations listed in Table 1. To compare the exergy destruction of dissimilar components, the exergy destruction ratio of the k^{th} component is defined as

$$y_{D,k}^* = \frac{\dot{E}_{D,k}}{\dot{E}_{D,tot}}, \quad (19)$$

Table 1. Exergy calculation in each component of the novel system.

Component	Schematic view	$\dot{E}_{F,k}$	$\dot{E}_{P,k}$	$\dot{E}_{D,k}$
Compressors		W	$\dot{E}_{2,C} - \dot{E}_{1,C}$	$\dot{E}_{F,C} - \dot{E}_{P,C}$
Storage cavern		$\dot{E}_{1,SC}$	$\dot{E}_{2,SC}$	$\dot{E}_{F,SC} - \dot{E}_{P,SC}$
Heater		$\dot{E}_{Q,H}$	$\dot{E}_{2,H} - \dot{E}_{1,H}$	$\dot{E}_{F,H} - \dot{E}_{P,H}$
Heat exchanger		$\dot{E}_{3,HE} - \dot{E}_{4,HE}$	$\dot{E}_{2,HE} - \dot{E}_{1,HE}$	$\dot{E}_{F,HE} - \dot{E}_{P,HE}$
Turbine		$\dot{E}_{1,T} - \dot{E}_{2,T}$	W	$\dot{E}_{F,T} - \dot{E}_{P,T}$

5. Results and discussions

The simulations for the parametric analysis of the compressed transcritical and supercritical CO₂ energy storage are carried out using a MATLABTM with the thermodynamic properties of the working fluid calculated through CoolProp^[42]. The design parameters and detailed conditions for the simulation and analysis of the

325 compressed gas energy storage are summarized in Table 2.

326 **5.1. Thermodynamic analysis**

327 The detailed results of the analysis of compressed CO₂ energy storage using two
328 saline aquifers as reservoirs are presented in Tables 3 and 4, which show the
329 thermodynamic parameters at each node of the two storage systems, respectively. A
330 summary of the results of the energy and exergy analyses are provided in Table 5. The
331 round-trip efficiency and exergy efficiency of compressed transcritical CO₂ energy
332 storage is better than those of the compressed supercritical CO₂ energy storage.
333 Moreover, the compressed transcritical CO₂ energy storage has a higher energy
334 density (E_{GEN}/V_S) than the compressed supercritical CO₂ energy storage. In addition,
335 the energy densities (E_{GEN}/V_S) of the two systems can reach 497.68 kWh/m³
336 (transcritical CO₂) and 255.20 kWh/m³ (supercritical CO₂), which are better compared
337 with that for CAES ($E_{GEN}/V_S = 2\text{-}20 \text{ kWh/m}^3$)^[38] because more energy can be stored
338 in a given reservoir. The reason that the energy storage density using transcritical CO₂
339 is much larger than that of supercritical CO₂ comes from the fact that the output work
340 of compressed transcritical CO₂ energy storage is more than two times larger than that
341 of compressed supercritical CO₂ energy storage. According to Eq. (15), the energy
342 storage density is mainly a function of w_t under the same reservoir conditions. The w_t
343 of transcritical compressed CO₂ energy storage and supercritical compressed CO₂
344 energy storage are 254.82 kW and 123.58 kW, respectively. Therefore, the energy
345 storage density using transcritical CO₂ is about two times as large as that of

346 supercritical CO₂ under a pressure of 40 MPa.

347

348 Fig. 3 shows the exergy destruction ratio of the different components of the
349 compressed CO₂ energy storage under simulation conditions. For the compressed
350 transcritical CO₂ energy storage, 54.37% of the irreversibility takes place in the heater,
351 11.98% in the low pressure reservoir, 10.93% in the compressor, 9.78% in the turbine,
352 9.52% in the recuperator, 3.42% in the high pressure reservoir. However, for the
353 compressed supercritical CO₂ energy storage, the largest exergy destruction is
354 contributed by high pressure reservoir, which exceeds 33.37% of the total exergy
355 destruction of the system. Secondly the heater brings larger exergy destruction, which
356 accounts for 20.98% of total exergy destruction.

357 Table 2. Parameters setting in the compressed CO₂ energy storage with two reservoirs.

Parameter	Value	Unit
Ambient temperature	308.00	K
Pressure drop in throttle valve in compression process	0.50	MPa
Inlet temperature	313.00	K
Outlet pressure of compressor train	25.00	MPa
Temperature of cooling water	308.00	K
Pressure drop in throttle valve in expansion process	5.00	MPa
Outlet temperature of heater	873.00	K
Outlet pressure of turbine train in supercritical cycle	8.00	MPa
Outlet pressure of turbine train in transcritical cycle	2.00	MPa
Isentropic efficiency of compressor	0.85	/
Isentropic efficiency of turbine	0.87	/
SCO ₂ Depth of low pressure reservoir	760.00	M
SCO ₂ Depth of high pressure reservoir	3000.00	M
TCO ₂ Depth of low pressure reservoir	200.00	M
TCO ₂ Depth of high pressure reservoir	3000.00	M

Table 3. Thermodynamic data for the material streams of the compressed transcritical CO₂ energy storage.

Streams	Material stream	$T(K)$	$p(MPa)$	$h(kJ/kg)$	$s(kJ/kg \cdot K)$	$e(kJ/kg)$
1	CO ₂	308.05	1.50	501.50	2.23	153.86
2	CO ₂	405.37	4.49	580.00	2.26	223.17
3	CO ₂	313.00	4.49	473.80	1.96	209.00
4	CO ₂	413.74	13.43	541.34	1.98	268.86
5	CO ₂	313.00	13.43	289.61	1.27	237.68
6	CO ₂	345.89	40.18	328.34	1.28	271.07
7	CO ₂	382.16	20.00	446.52	1.69	267.18
8	CO ₂	587.11	20.00	744.79	2.33	392.83
9	CO ₂	540.60	12.62	704.10	2.34	279.87
10	CO ₂	873.00	12.62	1101.81	2.91	505.91
11	CO ₂	669.75	2.00	873.09	2.96	345.20
12	CO ₂	387.16	2.00	574.82	2.39	174.06

Table 4. Thermodynamic data for the material streams of the compressed supercritical CO₂ energy storage.

Streams	Material stream	$T(K)$	$p(MPa)$	$h(kJ/kg)$	$s(kJ/kg \cdot K)$	$e(kJ/kg)$
1	CO ₂	307.83	7.40	400.71	1.66	228.07
2	CO ₂	432.61	40.18	484.96	1.69	302.86
3	CO ₂	382.16	20.00	446.52	1.69	267.18
4	CO ₂	699.02	20.00	882.22	2.54	437.04
5	CO ₂	873.00	20.00	1097.20	2.81	568.41
6	CO ₂	763.71	8.04	973.57	2.84	437.31
7	CO ₂	387.16	8.04	537.87	2.05	243.42

Table 5. Results of the material streams of the transcritical and supercritical compressed CO₂ energy storage.

Term	Unit	Value of TC-CCES	Value of SC-CCES
C1 power	kW/kg	78.37	83.81
C2 power	kW/kg	67.46	-
C3 power	kW/kg	38.56	-
T1 power	kW/kg	103.62	123.58
T2 power	kW/kg	151.20	-
Thermal energy input	kJ/s	457.70	240.73
Round-trip efficiency	%	63.35	62.28
Exergy efficiency	%	53.02	51.56
E_{GEN}/V_S	kWh/m ³	497.68	255.20

Fig. 3. The exergy destruction ratio of main components. a. transcritical compressed CO₂ energy storage. b. supercritical compressed CO₂ energy storage. C = compressor; RE = recuperator; T = turbine; HE = heater; LS = low pressure reservoir; and HS = high pressure reservoir.

5.2. Sensitivity analysis of key thermodynamic parameters

Based on above analysis, the main advantage of CCES using two saline aquifers reservoirs is the high energy-storage density. Energy storage pressure (outlet pressure of last-stage compressor), release pressure (inlet pressure of first stage-turbine), and pressure of the low-pressure reservoir (outlet pressure of last-stage turbine) are primary and significant parameters that influence the energy storage density. Therefore, conducting a parametric analysis to understand the effects of these various parameters on the performance of the storage system is essential, and parameters range of variation are shown in Table 6.

380 Table 6. The parameters of the compressed CO₂ energy storage system.

Parameters	Range of variation
Energy storage pressure /MPa	40-56
Energy releasing pressure /MPa	10-26
Pressure of low pressure reservoir /MPa	2-10

381 *5.2.1. Effects of the energy storage pressure*

382 Fig. 4 illustrates the effect of the energy storage pressure on round-trip efficiency and
383 exergy efficiency. The round-trip efficiency and exergy efficiency of the TC-CCES
384 and SC-CCES increase along with the increase in the energy storage pressure.
385 Moreover, the round-trip efficiency and exergy efficiency of SC-CCES are higher
386 than those of TC-CCES when the energy storage pressure is higher than 44 MPa. And
387 the round-trip efficiency is even higher than the exergy efficiency. The reason is that
388 for the round-trip efficiency, the input energy is electricity which includes the
389 consumption electricity of compressors and the conversion terms, as shown in Eq.
390 (14). However, for the exergy efficiency, the input exergy consists of two terms:
391 electricity exergy and thermal exergy, as shown in Eq. (18). According to the second
392 law of thermodynamics, the thermal exergy calculated by the input thermal energy
393 multiplied by the Carnot efficiency. The Carnot efficiency will be larger than the η_{NG} ,
394 when the other exergy destructions happened in power plant are considered. Therefore,
395 the round-trip efficiency is higher than the exergy efficiency.

Fig. 4. Effect of energy storage pressure on round-trip efficiency and exergy efficiency.

396
397
398
399

400 The effect of energy storage pressure on energy storage density is shown in Fig. 5. An
401 increase in the energy storage pressure increases net energy output during discharge,
402 which can increase energy storage density. According to Eq. (15), energy storage
403 density is determined by the work output during expansion and the volume of the
404 two-saline aquifers reservoirs. A high energy storage pressure will reduce the required
405 volume of the high-pressure reservoir, whereas the change in the work output is
406 opposite to that of required volume. Therefore, energy storage density will increase
407 along with the increase in the energy storage pressure for TC-CCES and SC-CCES.

Fig. 5. Effect of the energy storage pressure on the energy storage density.

Fig. 6 shows the effect of the energy storage pressure on the exergy destruction rate of the main components in TC-CCES and SC-CCES, respectively. The results of the exergy analysis indicate that exergy destruction is mainly contributed by HE and RE in TC-CCES, and the largest exergy destruction is contributed by HS, HE and RE in SC-CCES. Moreover, the rate of RE exergy destruction decreases with the increase in the energy storage pressure, whereas the change in the HE exergy destruction rate is opposite to that of RE in the two energy storage systems. The outlet pressure of the turbine is constant; hence, the outlet temperature of the turbine will decrease with the increase in the energy storage pressure, which will result in a smaller temperature difference in RE. A low outlet temperature of the turbine will lead to a low inlet temperature of HE; hence, the temperature difference will be increased. Therefore, more heat is needed to heat the CO₂ in the heater. The exergy destruction of RE is

mainly contributed by the temperature difference between hot and cold working fluids. The exergy destruction of HE is not only contributed by the temperature difference, but also by the amount of heat put into the heater. Consequently, the exergy destruction rate of RE will decrease with the increase in the energy storage pressure, whereas that of HE will exhibit the opposite trend. The exergy destruction rate of C, T, LS, HS is caused by the pressure difference, the pressure difference is higher, the exergy destruction is larger. Therefore, the exergy destruction rate will increase with the increase in energy storage pressure when the outlet pressure of turbine is constant.

Fig. 6. Effect of energy storage pressure on exergy destruction rate. a. transcritical compressed CO₂ energy storage. b. supercritical compressed CO₂ energy storage. C = compressor; RE = recuperator; T = turbine; HE = heater; LS = low pressure reservoir; and HS = high pressure reservoir.

5.2.2. Effects of energy releasing pressure

The effect of the energy releasing pressure on round-trip efficiency, exergy efficiency, and energy storage density is illustrated in Figs. 7 and 9. The round-trip efficiency and exergy efficiency of TC-CCES and SC-CCES both increase with the increase in energy-release pressure. However, the variation in energy release pressure has a larger effect on the round-trip efficiency and exergy efficiency of SC-CCES. Moreover, the

performance of SC-CCES is better than that of TC-CCES when energy releasing pressure is larger than 21 MPa.

Fig. 7 shows that transcritical CO₂ is better under a releasing pressure of ~20-22 MPa, but SC-CCES becomes better under a high pressure. The reason is that the CO₂ enthalpy will decrease with the increase in pressure at the same temperature. Moreover, the enthalpy difference between different pressures also gradually decreases along with the increase in pressure (cf. Fig. 8).

Fig. 7. Effect of energy releasing pressure on round-trip efficiency and exergy efficiency.

Fig. 8. The h - T curves for CO₂.

In Fig. 7, the outlet pressure of turbine is constant, so that the input work is constant in the compression process, the input thermal energy and output work will change with the variations of releasing pressure. For supercritical CO₂, less thermal energy is needed and more output work is produced with the increase in releasing pressure, and the increasing rate of output work is also higher than that of transcritical CO₂. When the releasing pressure is lower, the supercritical CO₂ need more heat energy and produce less output work compared with transcritical CO₂. Therefore, transcritical CO₂ is better when the releasing pressure is low, but supercritical CO₂ becomes better at high pressure.

Fig. 9 shows the effect of energy releasing pressure on energy storage density. As shown in the preceding analysis, energy storage density is determined by the work

467 output of the turbine train, as well as the outlet pressure of the turbine train and the
 468 compressor train. The increase in energy release pressure will increase the work
 469 output of the turbine train, and the pressure difference in the high-pressure reservoir
 470 will be reduced which results in smaller volume requirement for the high-pressure
 471 reservoir; hence the energy storage densities of TC-CCES and SC-CCES both
 472 increase with increase in energy release pressure.

473
 474 Fig. 9. Effect of energy releasing pressure on energy storage density.

475
 476 The effect of energy releasing pressure on the exergy destruction rate of the main
 477 components is shown in Fig. 10. For TC-CCES and SC-CCES, a greater portion of
 478 exergy destruction occurred in He and RE, and they also have the similar trend with
 479 the variations in energy releasing pressure. The exergy destruction rate of RE in
 480 TC-CCES and SC-CCES decreases with the increase in the energy releasing pressure,
 481 whereas that of HE exhibits the opposite trend.

483 From Fig. 10, we can find that the significant differences between TC-CCES and
484 SC-CCES are that (1) the transcritical exergy destruction rate is larger than that of
485 supercritical CO₂ except in the HS component, and (2) the highest exergy destruction
486 rate happens in different components.

487

488 The lower exergy destruction rate of supercritical CO₂ is caused by the fact that for
489 supercritical CO₂, both the input energy and output work are lower than that of
490 transcritical CO₂, so that the exergy destruction rate of supercritical CO₂ is lower.

491

492 The reason that the highest exergy destruction rate happened in different components
493 in transcritical versus supercritical CO₂ is that the injection temperature of the
494 high-pressure reservoir is much different between the transcritical CO₂ and
495 supercritical CO₂. As shown in Fig. 2, and in Tables 3 and 4, we can find that the
496 injection pressure of the high-pressure reservoir is equal for the transcritical CO₂ and
497 supercritical CO₂, but the injection temperature of supercritical CO₂ is higher than
498 that of transcritical CO₂. So there is a higher exergy destruction rate in the HS in the
499 compressed supercritical CO₂ energy storage system.

Fig. 10. Effect of energy release pressure on exergy destruction rate. a. Transcritical compressed CO₂ energy storage. b. Supercritical compressed CO₂ energy storage. C = compressor; RE = recuperator; T = turbine; HE = heater; LS = low pressure reservoir; and HS = high pressure reservoir.

5.2.3. Effects of pressure of low pressure reservoir

Fig. 11 shows the effect of the pressure of the low-pressure reservoir on the round-trip efficiency and exergy efficiency of TC-CCES and SC-CCES. It indicates that the pressure of the low-pressure reservoir exerts a larger influence on round-trip efficiency and exergy efficiency for TC-CCES and SC-CCES. The round-trip efficiency and exergy efficiency will firstly decrease, and then increase with increase in pressure of the low-pressure reservoir. This phenomenon is caused by the properties of CO₂. As shown in Fig. 12, both input work of the compressor and output work of the turbine decrease along with increase in pressure of the low-pressure reservoir, but the reduction of output work is greater than the decrease of input work of the compressor. Thus the system performance decreases when the storage system runs at transcritical state. Moreover, when the system runs at supercritical state, the compressibility factor of supercritical CO₂ ranges from 0.2-0.5, which means that less

518 compression work can be consumed in the compression process^[43]. Therefore, the
519 reduction of input work will dramatically decrease compared with that at transcritical
520 state. Fig. 13 shows the variations in energy storage density with the increase in
521 pressure of low pressure reservoir. It illustrates that the energy storage density
522 decreases with the increase in pressure of low pressure reservoir. Fig. 14 shows the
523 effect of pressure of low pressure reservoir on exergy destruction rate, which indicates
524 that the exergy destruction rates of HE decrease along with the increase in pressure of
525 the low-pressure reservoir. And when the pressure of the low pressure reservoir is
526 higher than 8 MPa, there is a larger decrease in exergy destruction rate of each
527 component except HS.

528
529 The phenomenon we have observed is that performance of SC-CCES can be
530 explained by the properties of supercritical CO₂ and configurations of the energy
531 storage system. Because of the larger mass density and configurations of SC-CCES
532 relative to transcritical CO₂, the input work in compression and the temperature
533 difference in the heater can be reduced. Moreover, the output work in expansion will
534 be increased due to the higher inlet temperature and pressure of the turbine and the
535 higher exhaust temperature caused by a lower pressure ratio will reduce the amount of
536 heat in the heater. Therefore the round-trip efficiency, exergy efficiency, and energy
537 storage density will increase, whereas the exergy destruction rate of heater decreases.

538
539 For Figs. 11-14, the pressure range 6.5-8 MPa shows a different mechanism, which is

540 caused by the thermodynamic properties of CO₂. For CO₂, the critical point is
 541 7.39 MPa and 304.25 K. the thermodynamic properties of CO₂ and supercritical CO₂
 542 are very different. Especially around the critical point, the thermodynamic properties
 543 of CO₂ will greatly change with changes in P and T ^[43]. Therefore, the compression
 544 work can be substantially decreased, as shown in Fig.12.

545
 546 Fig. 11. Effect of pressure of low pressure reservoir on round-trip efficiency and
 547 exergy efficiency.

Fig. 12. Effect of pressure of low pressure reservoir on power input and output.

Fig. 13. Effect of pressure of low pressure reservoir on energy storage density.

553
 554 Fig. 14. Effect of pressure of low pressure reservoir on exergy destruction rate.
 555 C = compressor; RE = recuperator; T = turbine; HE = heater; LS = low pressure
 556 reservoir; and HS = high pressure reservoir.

557 6. Conclusions

558 A thermodynamic analysis is carried out of a system consisting of transcritical and
 559 supercritical compressed CO₂ energy storage using two saline aquifers as reservoirs,
 560 including energy analysis, exergy analysis, and parametric analysis. The main
 561 conclusions are summarized as follows:

562

- 563 (1) According to the energy and exergy analysis, the proposed compressed CO₂
 564 energy storage system with two saline aquifers as reservoirs has a larger energy
 565 storage density (497.68 kWh/m³ for transcritical CO₂ and 255.20 kWh/m³ for
 566 supercritical CO₂ compared to CAES (2-20kWh/m³) and an acceptable round-trip

567 efficiency (63.35% for transcritical CO₂ and 62.28% for supercritical CO₂) and exergy
568 efficiency (53.02% for transcritical CO₂, and 51.56% for supercritical CO₂) compared
569 with conventional CAES (81.7%^[38]).

570

571 (2) The use of compressed transcritical CO₂ for energy storage results in a higher
572 round-trip efficiency, exergy efficiency and energy storage density, but the
573 configuration of SC-CCES is simpler compared with TC-CCES.

574

575 (3) The energy release pressure has a positive effect on the three indicators including
576 round-trip efficiency, exergy efficiency, and energy storage density for TC-CCES and
577 SC-CCES. The performance of the two energy storage systems will become better
578 with the increase in energy release pressure.

579

580 (4) The pressure of the low-pressure reservoir has a large effect on round-trip
581 efficiency and exergy efficiency, especially when the pressure is below 8 MPa.
582 Specifically, for TC-CCES, the round-trip efficiency and exergy efficiency
583 dramatically decrease, whereas the energy storage density will decrease along with
584 the increase in pressure of low pressure reservoir.

585 We note that for the TC-CCES, the low-pressure reservoir needs to be much shallower
586 and larger than the supercritical CO₂ energy storage reservoir. Therefore it may pose
587 larger potential environmental impacts, (for instance to potable groundwater), than the
588 second system. Also, the hazard of induced seismicity and triggered seismicity may

589 exist in both cyclic injection processes. These issues and others such as, how to
590 reduce brine production to a minimum,, or the hydro-geo-mechanical limitations of
591 the saline aquifer reservoirs, and the optimal cycling time given that transient pressure
592 gradients will exist around the injection and production wells^[11] are key questions
593 outside the scope of this study that should be addressed in the future.

594

595 **Acknowledgements**

596 The paper is supported by the National Nature Science Fund of China (No.51276059),
597 Fundamental Research Funds for the Central Universities of China (No. 2014XS27)
598 and China Scholarship Council (No.1510200035). Additional support came from
599 Lawrence Berkeley National Laboratory supported by the U.S. Department of Energy
600 under Contract No. DE-AC02-05CH11231.

601 **References**

- [1] Lund H, Salgi G. The role of compressed air energy storage (CAES) in future sustainable energy systems. *Energy Convers Manage* 2009; 50(5): 1172-1179.
- [2] Liu J L, Wang J H. A comparative research of two adiabatic compressed air energy storage systems. *Energy Convers Manage* 2016;108: 566-578.

- [3] Carrasco J M, Franquelo L G, Bialasiewicz J T, Galvan E, Portillo Guisado R C, Prats M A M, et al. Power-electronic systems for the grid integration of renewable energy sources: A survey. *IEEE Trans Ind Electron* 2006; 53: 1002-1016.
- [4] Díaz-González F, Sumper A, Gomis-Bellmunt O, Villafafila-Robles R. A review of energy storage technologies for wind power applications. *Renew Sustain Energy Rev* 2012; 16(4): 2154-2171.
- [5] Mahlia T M I, Saktisahdan T J, Jannifar A, Hasan M H, Matseelar H S C. A review of available methods and development on energy storage; technology update. *Renew Sustain Energy Rev* 2014; 33: 532-545.
- [6] De Bosio F, Verda V. Thermoeconomic analysis of a Compressed Air Energy Storage (CAES) system integrated with a wind power plant in the framework of the IPEX Market. *Appl Energy* 2015; 152: 173-182.
- [7] Cleary B, Duffy A, O'connor A, Conlon M, Fthenakis V. Assessing the economic benefits of compressed air energy storage for mitigating wind curtailment. *IEEE Trans Sustain Energy* 2015; 6(3): 1021-1028.
- [8] Arabkoohsar A, Machado L, Koury R N N. Operation analysis of a photovoltaic plant integrated with a compressed air energy storage system and a city gate station. *Energy* 2016; 98: 78-91.
- [9] Arabkoohsar A, Machado L, Farzaneh-Gord M, Koury R N N. The first and second law analysis of a grid connected photovoltaic plant equipped with a compressed air energy storage unit. *Energy* 2015; 87: 520-539.
- [10] Long-term energy storage with compressed air storages.
<<http://ees-magazine.com/long-term-energy-storage-with-compressed-air-storages/>>

- [11] Oldenburg C M, Pan L. Porous Media Compressed-Air Energy Storage (PM-CAES): Theory and Simulation of the Coupled Wellbore–Reservoir System. *Transport Porous Med* 2013; 97(2): 201-221.
- [12] Buffa F, Kemble S, Manfrida G, Milazzo A. Exergy and exergoeconomic model of a ground-based CAES plant for peak-load energy production[J]. *Energies* 2013; 6(2): 1050-1067.
- [13] Proczka J J, Muralidharan K, Villela D, Simmons J H, Frantziskonis G. Guidelines for the pressure and efficient sizing of pressure vessels for compressed air energy storage. *Energy Convers Manage* 2013; 65: 597-605.
- [14] Zhang Y, Yang K, Li X, Xu J. The thermodynamic effect of thermal energy storage on compressed air energy storage system. *Renew Energy* 2013; 50: 227-235.
- [15] Zhang Y, Yang K, Li X, Xu J. The thermodynamic effect of air storage chamber model on Advanced Adiabatic Compressed Air Energy Storage System. *Renew Energy* 2013; 57: 469-478.
- [16] Jubeh N M, Najjar Y S H. Green solution for power generation by adoption of adiabatic CAES system. *Energy Convers Manage* 2012; 44: 85-89.
- [17] Najjar Y S H, Zaamout M S. Performance analysis of compressed air energy storage (CAES) plant for dry regions. *Energy Convers Manage* 1998; 39(15): 1503-1511.
- [18] Crotagino F, Mohmeyer K U, Scharf R. Huntorf CAES: More than 20 years of successful operation, Orlando, Florida, USA; 2001.
- [19] Safaei H, Keith D W. Compressed air energy storage with waste heat export: An Alberta case study. *Energy Convers Manage* 2014; 78: 114-124.

- [20] Luo X, Wang J, Krupke C, Wang Y, Sheng Y, Li J, et al. Modelling study, efficiency analysis and optimisation of large-scale Adiabatic Compressed Air Energy Storage systems with low-temperature thermal storage. *Appl Energy* 2016; 162: 589-600.
- [21] Guo Z, Deng G, Fan Y, Chen G. Performance optimization of adiabatic compressed air energy storage with ejector technology. *Appl Therm Eng* 2016; 94: 193-197.
- [22] Li Y, Wang X, Li D, Ding Y. A trigeneration system based on compressed air and thermal energy storage. *Appl Energy* 2012; 99: 316-323.
- [23] 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.
- [24] Yao E, Wang H, Wang L, Xi G, Marechal F. Thermo-economic optimization of a combined cooling, heating and power system based on small-scale compressed air energy storage. *Energy Convers Manage* 2016; 118: 377-386.
- [25] Tessier M J, Floros M C, Bouzidi L, Narine S S. Exergy analysis of an adiabatic compressed air energy storage system using a cascade of phase change materials. *Energy* 2016; 106: 528-534.
- [26] Simpson M C, Garvey S D, Pimm A, Kantharaj B, Cardenas B, Garvey J E. Integrating solar thermal capture with compressed air energy storage, London, UK; 2016.
- [27] Li Y, Sciacovelli A, Peng X, Radcliffe J, Ding Y. Integrating compressed air energy storage with a diesel engine for electricity generation in isolated areas. *Appl Energy* 2016; 171: 26-36.

- [28] Zhang Y, Yang K, Hong H, Zhong X, Xu J. Thermodynamic analysis of a novel energy storage system with carbon dioxide as working fluid. *Renew Energy* 2016; 99: 682-697.
- [29] Chen H, Cong T N, Yang W, Tan C, Li Y, Ding Y. Progress in electrical energy storage system: A critical review. *Prog Nat Sci* 2009; 19(3): 291-312.
- [30] Kim Y M, Shin D G, Favrat D. Operating characteristics of constant-pressure compressed air energy storage (CAES) system combined with pumped hydro storage based on energy and exergy analysis. *Energy* 2011; 36(10): 6220-6233.
- [31] Guo H, Xu Y, Chen H, Zhou X. Thermodynamic characteristics of a novel supercritical compressed air energy storage system. *Energy Convers Manage* 2016; 115: 167-177.
- [32] Pimm A J, Garvey S D, de Jong M. Design and testing of energy bags for underwater compressed air energy storage. *Energy* 2014; 66: 496-508.
- [33] Jannelli E, Minutillo M, Lavadera A L, Falcucci G. A small-scale CAES (compressed air energy storage) system for stand-alone renewable energy power plant for a radio base station: A sizing-design methodology. *Energy* 2014; 78: 313-322.
- [34] Dostal V. A supercritical carbon dioxide cycle for next generation nuclear reactors. PhD dissertation, Massachusetts Institute of Technology. These N MIT-ANP-TR-100; 2004.
- [35] Wang M, Zhao P, Wu Y, Dai Y. Performance analysis of a novel energy storage system based on liquid carbon dioxide. *Appl Therm Eng* 2015; 91: 812-823.
- [36] Zhou Q, Birkholzer J T, Tsang C F, Rutqvist J. A method for quick assessment of CO₂ storage capacity in closed and semi-closed saline formations. *Int J Greenh Gas Con* 2008; 2(4): 626-639.

- [37] Cayer E, Galanis N, Desilets M, Nesreddine H, Roy P. Analysis of a carbon dioxide transcritical power cycle using a low temperature source. *Appl Energy* 2009; 86(7): 1055-1063.
- [38] Succar, Samir, and Robert H. Williams. Compressed air energy storage: theory, resources, and applications for wind power. Report no. 8, Princeton environmental institute; 2008.
- [39] Kim Y M, Lee J H, Kim S J, Favrat D. Potential and evolution of compressed air energy storage: energy and exergy analyses. *Entropy* 2012; 14(8): 1501-1521.
- [40] Liu H, He Q, Saeed S B. Thermodynamic analysis of a compressed air energy storage system through advanced exergetic analysis. *J Renew Sustain Energy* 2016; 8(3): 034101.
- [41] Wang L, Yang Y, Morosuk T, Tsatsaronis G. Advanced thermodynamic analysis and evaluation of a supercritical power plant. *Energies* 2012; 5(6): 1850-1863.
- [42] Welcome to CoolProp. <<http://www.coolprop.org/>>
- [43] Ahn Y, Bae S J, Kim M, Cho S K, Baik S, Lee J I, et al. Review of supercritical CO₂ power cycle technology and current status of research and development. *Nuc Eng Technol* 2015; 47(6): 647-661.