Applied Energy Symposium 2020: Low carbon cities and urban energy systems
October 10-17, 2020
Paper ID: CUE2020-D-020

Proposal and thermodynamic analysis of a novel combined thermal and compressed air energy storage system integrated with an absorption power cycle

Xuqing Yang[1], Zhenzhu Yu[1], Zhan Liu[1*], Xiaohu Yang[2*]

1 College of Electromechanical Engineering, Qingdao University of Science and Technology, Qingdao 266061, PR China

2 School of Human Settlements and Civil Engineering, Xi'an Jiaotong University, Xi'an 710049, PR China
(Corresponding Author: zhanliu168@qust.edu.cn, xiaohuyang@xjtu.edu.cn)

ABSTRACT

Integrating energy storage technology with renewable energy can improve largely the penetration of renewable energy. A novel energy storage system based on compressed air energy storage, electrical heater and Kalina cycle is thus proposed. This work focuses on demonstrating the feasiblity and effectiveness of the hybrid system for achieving the energy cascade utilization through parametric analysis. Results show that within certain ranges, higher discharging pressure and lower electrical heating temperature are beneficial for increasing round trip efficiency of the hybrid system. Decreasing discharging pressure and increasing separator pressure would reach higher power production of the Kalina cycle turbine.

Keywords: compressed air energy storage; Kalina cycle; electrical heater; thermodynamic study

NONMENCLATURE

Abbreviations	
CAES	Compressed air energy storage
PHS	Pumped hydro storage
CH-CAES	Combined heat and CAES
RTE	Round trip efficiency
TES	Thermal energy storage
KC	Kalina cycle
Symbols	
X	Ammonia concentration
h	Specific enthalpy ($J \cdot kg^{-1}$)
λ	Ratio of specific heats
π	Pressure ratio of machine
$\dot{Q}$	Heat transfer rate (W)
s	Specific entropy ($J \cdot kg^{-1} \cdot K^{-1}$)
T	Temperature (K)
$\dot{m}$	Mass flow rate ($kg \cdot s^{-1}$)
$\dot{W}$	Power (W)
$\dot{E}$	Exergy flow rate (W)

1. INTRODUCTION

Compressed air energy storage (CAES) has been considered as one of the two mature and commercially acceptable large-scale energy storage technologies [1]. Nowadays, there have been two commercial diabatic CAES power plants built in the world. One was 290 MW plant commissioned in Huntorf, Germany, 1978, and the other was 110 MW plant commissioned in McIntosh, Alabama, USA, 1991 [2].

In order to solve the problems of greenhouse gas emission and low round trip efficiency in diabatic CAES plant [3]. Adiabatic CAES (A-CAES) system is a new development direction based on diabatic CAES plant [4]. However, CAES energy density cannot be sufficient just rely on the compression unless to increase the size condition of air cavern or employing higher storage pressure. Recently, resistance heating is considered as a promising method to solve the above problem for increasing the system energy capacity without any configuration refactoring.

Considering redundant heat stored in TES by introduction of electrical heater, a thermodynamic cycle is necessary to recovery the residual heat for efficiency improvement. Kalina cycle (KC) which can convert

Selection and peer-review under responsibility of the scientific committee of CUE2020

thermal energy to usable power has attracted more attention. Zhao et al. [5] proposed a CAES-KCS6 system to recovery the heat energy from high temperature exhaust of low pressure turbine. Li et al. [6] made a comparison between the KC-CAES system and ORC-CAES system, and results showed that it was more feasible to choose the KC as the bottoming cycle of CAES system. In order to reduce the loss of heat stored in the thermal energy storage vessel, Li et al. [7] further designed a KC-ACAES system.

In this paper, a novel combined thermal and compressed air energy storage (CH-CAES) system integrated with a Kalina cycle is proposed, denoted as CH-CAES-KC. A Kalina bottoming cycle is employed to recycle the residual heat of thermal oil for efficiency improvement. Parametric analysis is carried out to investigate variation in system performance. The system can achieve the energy cascade utilization and obtain higher system efficiency, making it more competitive in energy storage and supply.

2. SYSTEM DESCRIPTION

Fig. 1 represents the schematic diagram of the CH-CAES-KC system. During charging process, the ambient air is compressed by three compressors, while the compression heat is recovered in the three coolers by thermal oil. Afterwards, the thermal oil is further heated up by electrical heater. During discharging process, the pressurized air passes through the throttle valve 2 to keep a constant pressure. At the same time, the high-temperature thermal oil is divided into three streams which are fed to the corresponding heaters to heat the inlet air of the turbines. Afterwards, high-temperature and high-pressure air expands in the air turbines to generate power. Considering the exhaust of the low-pressure turbine still has a high temperature, a recuperator is employed to recycle the rest heat energy.

In the bottom cycle, the basic solution is first heated by ammonia-poor liquid and is heated again by thermal oil. After two-stage heating, the ammonia-water two-phase mixtures are separated to ammonia-rich vapor and ammonia-poor liquid. The ammonia-rich vapor flows through the KC turbine to produce power. Next, the ammonia-poor liquid at the exit of the recuperator passes the throttle valve 3 and mixes with the exhaust of KC turbine. Finally, the mixture is condensed to saturated liquid by cooling water.

3. THERMODYNAMIC MODEL

The mathematical model of the proposed system was established based on energy and mass conservation laws. To simplify the investigations, the CH-CAES-KC system is assumed with steady state condition and air is assumed as an ideal gas. There are no pressure/heat losses in the heat exchangers, separator and pipelines.

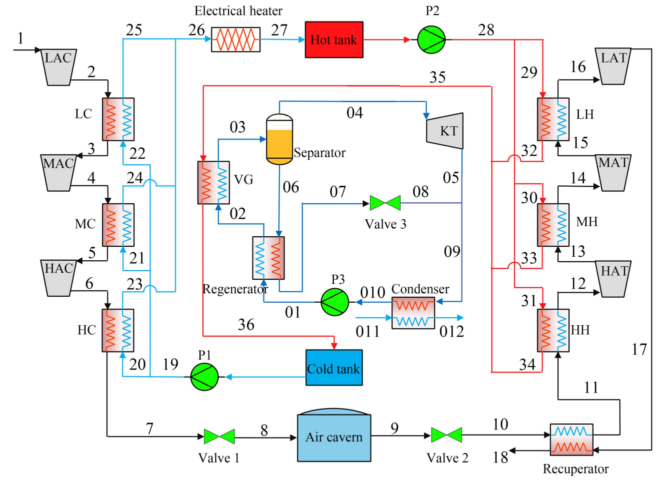

Fig. 1. Layout of the CH-CAES-KC system.

3.1 CH-CAES section

For compressor and turbine, the powers are [8]:

$$\dot{W}_{AC,i} = \dot{m}_{air} c_p T_{in,i} \frac{\pi_{AC}^{\frac{\lambda-1}{\lambda}} - 1}{\eta_{AC}} \tag{1}$$

$$\dot{W}_{AT,i} = \dot{m}_{air} c_p T_{in,i} \left(1 - \pi_{AT}^{\frac{1-\lambda}{\lambda}}\right) \eta_{AT} \tag{2}$$

For coolers and heaters:

$$\dot{m}_{air} c_{p,air} \Delta T_{air} = \dot{m}_{oil} c_{oil} \Delta T_{oil} \tag{3}$$

For the electrical heater:

$$\dot{Q}_{EH} = \dot{m}_{26} c_{oil} (T_{27} - T_{26}) \tag{4}$$

As air passes through the throttle valve, its specific enthalpy remains constant and the temperature of air at cavern outlet is equal to ambient temperature:

$$T_9 = T_0, \quad h_9 = h_{10} \tag{5}$$

3.2 Absorption power cycle section

For the vapor generator [9]:

$$\dot{Q}_{in} = \dot{m}_{35} c_{oil} (T_{35} - T_{36}) = \dot{m}_{02} (h_{03} - h_{02}) \tag{6}$$

For the separator[9]:

$$\dot{m}_{03} = \dot{m}_{04} + \dot{m}_{06} \tag{7}$$

$$\dot{m}_{03} X_{03} = \dot{m}_{04} X_{04} + \dot{m}_{06} X_{06} \tag{8}$$

Power produced by KC turbine can be given as:

$$\dot{W}_{KT} = \dot{m}_{04} (h_{04} - h_{05}) \tag{9}$$

For the regenerator and throttle valve 3:

$$\dot{m}_{06} (h_{06} - h_{07}) = \dot{m}_{02} (h_{02} - h_{01}), \quad h_{07} = h_{08} \tag{10}$$

The mixing process of ammonia-poor liquid and exhaust of the KC turbine:

$$\dot{m}_{05} h_{05} + \dot{m}_{08} h_{08} = \dot{m}_{09} h_{09} \tag{11}$$

For the condenser:

$$\dot{m}_{09}(h_{09}-h_{010})=\dot{m}_{011}(h_{012}-h_{011}) \tag{12}$$

The power consumption of the pump is:

$$W_{P3}=\dot{m}_{01}(h_{01}-h_{010}) \tag{13}$$

Net output power of the Kalina cycle is:

$$\dot{W}_{KC}=\dot{W}_{KT}-\dot{W}_{P3} \tag{14}$$

The exergy at each stream is identified as [10]:

$$\dot{E}_j=\dot{m}_j\left[\left(h_j-h_0\right)-T_0\left(s_j-s_0\right)\right] \tag{15}$$

Round trip efficiency (RTE) is defined as:

$$RTE=\frac{\left(\sum\dot{W}_{dch}\right)\cdot T_{dch}}{\left(\sum\dot{W}_{ch}+\dot{Q}_{EH}\right)\cdot T_{ch}} \tag{16}$$

Moreover, the thermal efficiency (η_{th}) and exergy efficiency (η_{exg}) of the Kalina cycle can be expressed as:

$$\eta_{th}=\frac{\dot{W}_{KC}\cdot T_{dch}}{\dot{Q}_{in}\cdot T_{ch}},\quad \eta_{exg}=\frac{\dot{W}_{KC}\cdot T_{dch}}{\left(\dot{E}_{35}-\dot{E}_{36}\right)\cdot T_{ch}} \tag{17}$$

4. RESULTS AND DISCUSSION

The charging/discharging periods of the CAES are both set to 6 hours and the power generation of air turbines is fixed at 10 MW. Default operating parameters are listed in Table 1.

Table 1
Default conditions of the CH-CAES-KC system.

Parameter	Unit	Value
Ambient pressure	MPa	0.1013
Ambient temperature	K	293.15
Charge pressure	MPa	7.2
Discharge pressure	MPa	4.6
The electrical heating temperature	K	573.15
Isentropic efficiency of air compressor	-	0.84
Isentropic efficiency of air turbine	-	0.88
Isentropic efficiency of pump	-	0.70
Ammonia mass fraction	-	0.60
Separation temperature	K	383.15
Separation pressure	MPa	1.6

4.1 Effect of discharging pressures

Fig. 2 shows the variation of the RTE of both the CH-CAES and CH-CAES-KC systems, thermal efficiency, exergy efficiency, power production of the KC turbine and power consumption of the pump with altering the discharging pressure. As can be seen, the increasing discharging pressure improves the RTE of both the CH-CAES and CH-CAES-KC systems. On the other hand, the CH-CAES-KC system is more efficient than the system without KC with a 1.68-2.04% enhancement. Moreover, it can be seen that the power production of the KC turbine and power consumption of the pump are decreased gradually while the thermal efficiency and exergy efficiency of the Kalina cycle changed little.

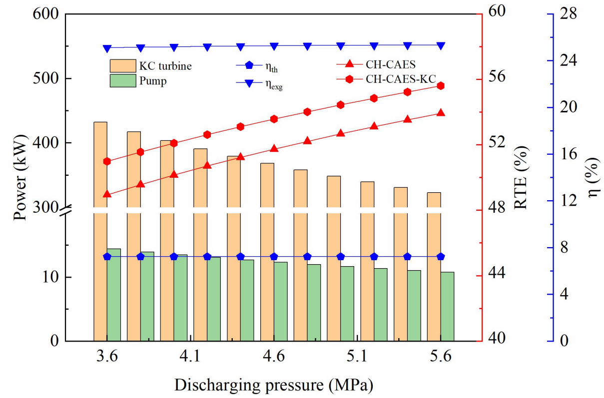

Fig. 2. Effect of discharging pressure on the system performance.

4.2 Effect of electrical heating temperature

As the electrical heating temperature increases, the RTE of both systems show a decreasing trend and the power production of the KC turbine and power consumption of the pump show an increasing trend, see Fig. 3. With the electrical heating temperature varying from 270 K to 340 K, the RTE of CH-CAES-KC system is 1.44-2.27% higher than that of the CH-CAES system. The thermal efficiency of the KC remains unchanged and the exergy efficiency of that increases first and then decreases but changes little.

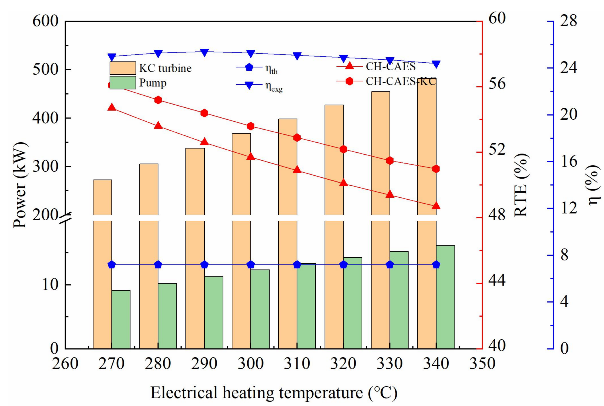

Fig. 3. Effect of the electrical heating temperature on the system performance.

4.3 Effect of separation temperature

Fig. 4 gives the effect of separation temperature on the system performance. With the increase in separation temperature, the RTE of both systems are always unchanged. Moreover, the output power generated by KC turbine and the power consumed by pump all decrease. So it's easy to obtain by calculation that the net power produced by KC increases first and then decreases in general, but this changes reflected in the thermal efficiency and the exergy efficiency of the KC varies a small scale.

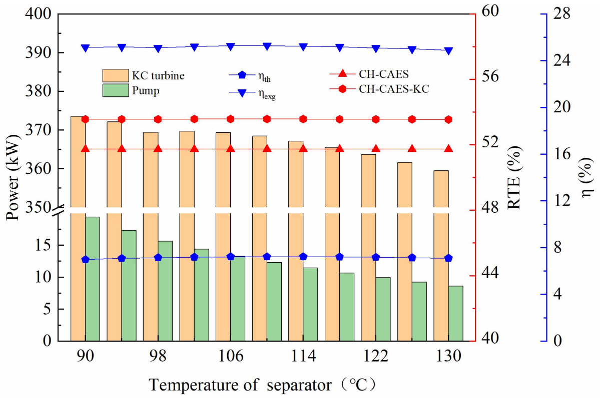

Fig. 4. Effect of the separation temperature on the system performance.

4.4 Effect of separation pressure

Fig. 5 exhibits the effect of separation pressure on the system performance. It can be observed that the exergy efficiency of the KC ranges from 16.53% to 27.61% and the thermal efficiency is raised from 3.83% to 9.78% with the increase of separation pressure.

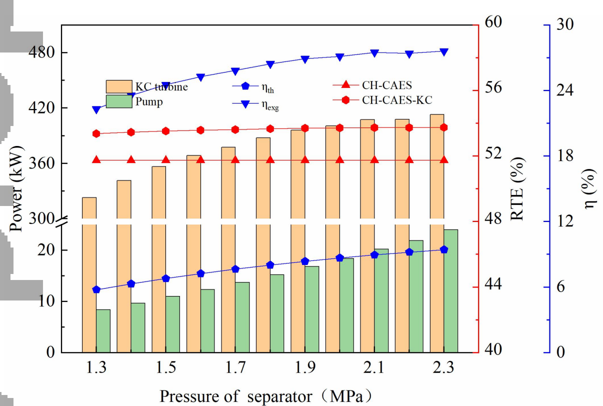

Fig. 5. Effect of the separation pressure on the system performance.

5. CONCLUSIONS

This paper proposes a novel hybrid energy storage system (CH-CAES-KC). The significant conclusions are: within certain ranges, higher discharging pressure and lower electrical heating temperature are beneficial for increasing RTE of the hybrid system while separation temperature and separation pressure have little influence on RTE of the hybrid system; increasing separation pressure and electrical heating temperature, decreasing discharging pressure and separation temperature would reach higher power production of the KC turbine and power consumption of the pump; In addition to the separation temperature and pressure, other parameters have little influence on the thermal efficiency and exergy efficiency of the absorption power cycle.

ACKNOWLEDGEMENT

NONE.

REFERENCE

[1] Evans A, Strezov V, Evans TJ. Assessment of utility energy storage options for increased renewable energy penetration. Renewable and Sustainable Energy Reviews 2012; 16: 4141-7.

[2] Liu Z, Cao F, Guo JZ, Liu J, Zhai HY, Duan ZY. Performance analysis of a novel combined cooling, heating and power system based on carbon dioxide energy storage. Energy Conversion and Management 2019; 188: 151-61.

[3] Budt M, Wolf D, Span R, Yan J. A review on compressed air energy storage: Basic principles, past milestones and recent developments. Applied Energy 2016; 170: 250-68.

[4] Zunft S, Jakiel C, et al. Adiabatic Compressed Air Energy Storage for the Grid Integration of Wind Power. International Workshop on Large-scale Integration of Wind Power & Transmission Networks for Offshore Windfarms 2007; 5:296-306.

[5] Zhao P, Wang JF, Dai YP. Thermodynamic analysis of an integrated energy system based on compressed air energy storage (CAES) system and Kalina cycle. Energy Conversion and Management 2015; 98: 161-172.

[6] Li RX, Wang HR, Yao ER. Thermo-economic comparison and parametric optimizations among two compressed air energy storage system based on Kalina cycle and ORC. Energies 2016; 10(1): 15.

[7] Li RX, Wang HR, Tu SK. Thermo-economic analysis and optimization of adiabatic compressed air energy storage (A-CAES) system coupled with a Kalina cycle. Energy Technology 2018; 6(6): 1011-25.

[8] Wei J, Zhou Y, Sun Y, Zhang W, An BL, Wang JJ. Thermodynamic analysis of a novel hybrid wind-solar-compressed air energy storage system. Energy Conversion and Management 2017; 142: 176-187.

[9] Yuan H, Mei N, Zhou P. Performance analysis of an absorption power cycle for ocean thermal energy conversion. Energy Conversion and Management 2014; 87: 199-207.

[10] Liu Z, Liu B, Guo JZ, Xin X, Yang XH. Conventional and advanced exergy analysis of a novel transcritical compressed carbon dioxide energy storage system. Energy Conversion and Management 2019; 19：111807.