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Performance of a carnot battery system with seawater as cold reservoir

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ABSTRACT

As an energy storage technology, carnot battery stores the electric energy as thermal energy. It has the characteristics of large capacity, long cycle time and independence for fossil fuels. This paper presents a novel transcritical carbon dioxide carnot battery system. Different from previous studies, seawater of different temperature is selected as cold reservoir in this study. The main work is to analyze the thermodynamic performance of this system. The obtained investigation indicates that the maximum value of round-trip efficiency is 54.2% when the compressor and pump outlet pressure is 20MPa and 19 MPa respectively. Furthermore, the decrease in evaporator pinch temperature and cold seawater temperature result in an increase of round-trip efficiency.

Keywords: carnot battery, seawater, thermodynamic analysis, transcritical carbon dioxide

NONMENCLATURE

Abbreviations	
RTE	Round-trip efficiency
PHS	Pumped hydro storage
CAES	Compressed air energy storage
ch	Charge
dis	Discharge
Symbols	
h	Specific enthalpy (J/kg)
η	Isentropic efficiency (%)
$\dot{m}$	Mass flow rate (kg/s)
$\dot{Q}$	Heat transfer rate (W)
$\dot{W}$	Power (W)

1. INTRODUCTION

Renewable energy sources have obtained significant interest owing to fossil energy shortages and environmental pollution. It is estimated that renewable energy sources will account for 29% of the total power generation in 2040[1]. Nevertheless, renewable energy sources have the characteristics of intermittency and stochasticity so that the stability of power grid is threaten. Energy storage technologies with grid have been considered to address these issues [2].

There are several types of massive energy storage technologies such as compressed air energy storage (CAES), pumped hydro storage (PHS), and carnot battery. However, the wide application of CAES and PHS are limited because of high capital cost and requirement for special geographical conditions [3].

Comparison with CAES and PHS, carnot battery has the characteristics of low capital cost and independence on geological conditions [4]. In the charging progress, surplus electricity is stored as sensible and/or latent heat through a heat pump cycle. In the discharging progress, the heat engine cycle is driven by the heat energy stored in the charging progress to work and generate electricity.

In recent years, transcritical CO_2 carnot battery system has gradually received the attention of researchers. Mercangöz et al. [5] proposed a transcritical CO_2 carnot battery system based on the Rankine cycle. In this system, the hot water was used to store the pumped heat, and ice was generated and melted at the cold end of the cycles. Zhao et al. [4] summarized influence of different heat storage materials on the system performance. Tauveron et al. [6] introduced a new concept of Thermo-carnot battery with transcritical CO_2 cycles and ground heat storage. Result show that RTE of this system up to 66%. However, in the above systems, ice or environment is used as the cold source and the stability of the system cannot be guaranteed.

Ocean has great development potential in thermal energy resources. The ocean's surface temperature ranges from 295.15 K to 302.15 K, while at the lower depths, the temperature of seawater can drop below 279.15 K [7]. This temperature range satisfies the requirement of the cold reservoir for the transcritical CO_2 carnot battery.

In this study, a transcritical CO_2 carnot battery system with seawater is presented. Different from the

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previous studies, seawater of different temperature is selected as cold reservoir for the charging and discharging process. The thermal energy of the ocean is harnessed effectively so there is potential to gain better thermodynamic performance. Thermodynamic model is built to analyze the system performance and the impact of several significant parameters on this system is studied.

2. SYSTEM DESCRIPTION

The schematic diagram of the proposed system is shown in Fig. 1. In the charging time, CO_2 absorbs thermal energy from warm seawater before entering the compressor. After storing the heat in the circulating water through the gas cooler, CO_2 is expanded to generate power. In the discharging time, CO_2 is liquefied in the condenser using the cooling volume of cold seawater and then pressurized by the pump. Subsequently, the high pressure CO_2 absorbs thermal energy in the gas heater and produces power in the expander.

It is worth noting that three water tanks are used to store the thermal energy. This is because the heat capacity of supercritical CO_2 will change dramatically for the duration of heat transfer process and the increase of heat transfer temperature difference will bring about the decrease of cycle efficiency in the ideal case [5]. Therefore, a middle-temperature tank is added to achieve higher round-trip efficiency.

3. THERMODYNAMIC MODEL

To assess the performance of proposed system, a thermodynamic model is created based on the following assumptions:

(1) The heat leak of thermal storage tanks is not considered.

(2) This system operates under stable conditions.

3.1 compressor, expander and pump

Isentropic efficiencies are introduced for modeling the machines. For a compressor and a pump, the isentropic efficiency can be defined as follows:

$$\eta_c = \frac{h_{c,is,out} - h_{c,in}}{h_{c,out} - h_{c,in}} \tag{1}$$

$$\eta_p = \frac{h_{p,is,out} - h_{p,in}}{h_{p,out} - h_{p,in}} \tag{2}$$

where subscript '*c*' and '*p*' represent compressor and pump, respectively, the subscript 'is' denotes the condition of isentropic process.

With regard to an expander, the isentropic efficiency can be calculated as:

$$\eta_e = \frac{h_{e,in} - h_{e,out}}{h_{e,in} - h_{e,is,out}} \tag{3}$$

The power consumption of compressor and pump are shown as:

$$\dot{W}_{com} = \dot{m}_{CO_2}(h_3 - h_2) \tag{4}$$

$$\dot{W}_{pump} = \dot{m}_{CO_2}(h_8 - h_7) \tag{5}$$

Furthermore, the pressure losses, Δp_{sw}, of the seawater pipes are evaluated as:

$$\Delta p_{sw} = f\frac{L_{sw}}{D_{sw}}\frac{\rho_{sw}V_{sw}^2}{2} \tag{6}$$

where f, L_{sw} and D_{sw} are friction factor, length and inner diameter of the seawater pipes. Moreover, V_{sw} and ρ_{sw} stand for velocity and density of the seawater.

For the pumps delivering the seawater, the required power can be calculated using Eq. (7):

$$\dot{W}_{p,sw} = \frac{\dot{m}_{sw}\Delta p_{sw}}{\rho_{sw}\eta_{p,sw}} \tag{7}$$

where $\dot{m}_{sw}$ is the mass flowrate of seawater in the pipe.

The electricity generated by expanders can be expressed as:

$$\dot{W}_{exp1} = \dot{m}_{CO_2}(h_5 - h_1) \tag{8}$$

$$\dot{W}_{exp2} = \dot{m}_{CO_2}(h_{10} - h_6) \tag{9}$$

3.2 heat exchanger

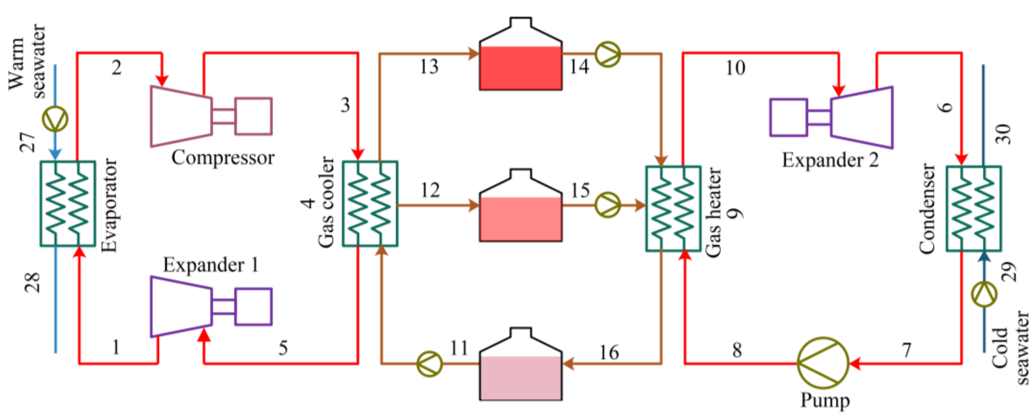

Fig. 1 The schematic diagram of carnot battery with seawater

For heat exchangers, the following equation is given representing their heat balance:

$$\begin{aligned}\dot{Q}_{gc} &= \dot{m}_{CO_2}(h_3 - h_5) \\ &= \dot{m}_{water,21}(h_{22} - h_{21}) + \dot{m}_{water,23}(h_{23} - h_{22})\end{aligned} \tag{10}$$

$$\begin{aligned}\dot{Q}_{gh} &= \dot{m}_{CO_2}(h_{10} - h_8) \\ &= \dot{m}_{water,24}(h_{24} - h_{25}) + \dot{m}_{water,26}(h_{25} - h_{26})\end{aligned} \tag{11}$$

$$\dot{Q}_{eva} = \dot{m}_{CO_2}(h_2 - h_1) = \dot{m}_{sw,29}(h_{27} - h_{28}) \tag{12}$$

$$\dot{Q}_{con} = \dot{m}_{CO_2}(h_6 - h_7) = \dot{m}_{sw,29}(h_{30} - h_{29}) \tag{13}$$

where the subscript '*gc*' and '*gh*' denotes the gas cooler and gas heater, 'eva' and 'con' mean evaporator and condenser.

3.3 thermal storage tanks

When the temperature of hot water is lower than 373.15 K, the pressure of stored water tank is obtained as:

$$P_w = 1.2 P_{AMB} \tag{14}$$

Otherwise, the pressure of stored water tank is show as:

$$P_w = 1.2 P_{sat} \tag{15}$$

where P_{AMB} and P_{sat} are ambient pressure and saturated pressure at the temperature of hot water respectively.

Also, a 20% margin of water tank volume is selected to accommodate deep charging and discharging progress and the volume of tank is expressed as:

$$V_{tank} = 1.2 M_w / \rho_w \tag{16}$$

where M_w and ρ_w is the total mass and density of circulating water.

3.4 Performance indicators

Round-trip efficiency (RTE) is defined as the ratio of net work output to net work input:

$$RTE = \frac{\dot{W}_{net,dis} t_{dis}}{\dot{W}_{net,ch} t_{ch}} = \frac{\left(\dot{W}_{\exp 2} - \dot{W}_{pump} - \dot{W}_{p,csw}\right) t_{dis}}{\left(\dot{W}_{com} - \dot{W}_{\exp 1} + \dot{W}_{p,wsw}\right) t_{ch}} \tag{17}$$

The energy storage density (ρ_E) of carnot battery is defined as the available heat storage per unit volume

$$\rho_E = \frac{\dot{W}_{net,ch} t_{dis}}{V_{HT} + V_{MT} + V_{LT}} \tag{18}$$

where the subscript 'HT', 'MT' and 'LT' mean the high-temperature tank, middle-temperature tank and low-temperature tank respectively.

4. RESULTS AND DISCUSSION

In the following sections, the effect of several significant parameters on the system performance is assessed. The main baseline parameters are shown in Table 1.

Table 1

Main baseline parameters of the system

Parameter	Unit	Value
Ambient pressure	MPa	0.1
Ambient temperature	K	298.15
Charging duration	h	8
Discharging duration	h	8
Isentropic efficiency of compressors	%	0.85
Isentropic efficiency of expander 1	%	0.8
Isentropic efficiency of pump	%	0.8
Isentropic efficiency of expander 2	%	0.88
Output power	MW	10

4.1 Effect of the temperature of cold seawater

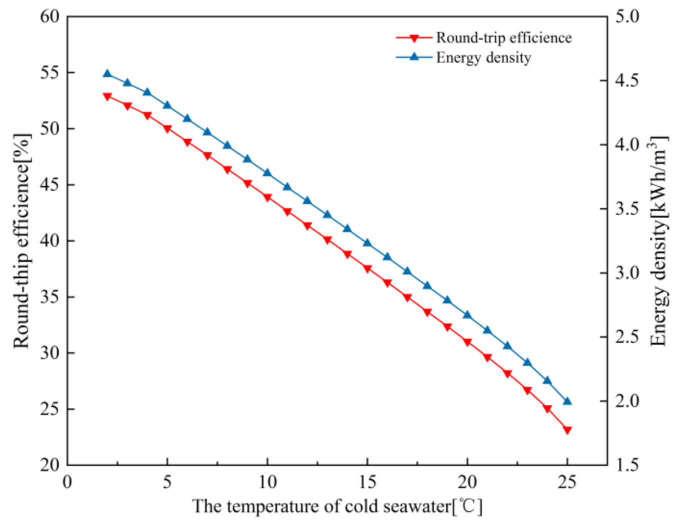

Fig. 2 Effects of the temperature of cold seawater on RTE and energy density

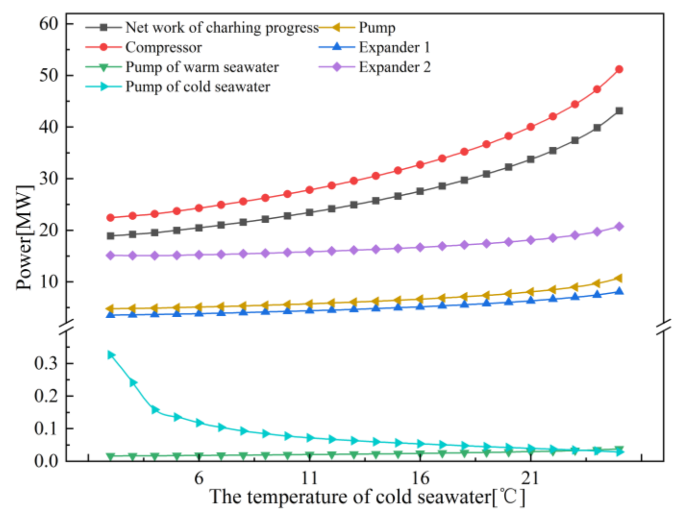

Fig. 3 Variations of power with temperature of cold seawater

Fig. 2 and Fig. 3 illustrate the variations of the power, RTE and energy density. It can be seen that rising temperature of cold seawater gives a decrease in the RTE and energy density. Furthermore, it is obviously that seawater pumps have less impact on system performance due to lower power consumption. With regard to the turbomachinery for transporting CO_2, the power increases monotonically with the temperature of cold seawater and the change sensitivity of the compressor is greater than that of the expander, which result the rise of the net power output.

4.2 Effect of the pinch temperature of evaporator

Fig. 4 represents the variations of RTE and energy density. It is well found that the pinch temperature of evaporator has different impact on RTE and energy density. Specifically, energy density increases monotonically along with the pinch temperature of evaporator while RTE has the opposite trend. Fig.5 illustrates power of the turbomachinery. It is depicted that the network of charging progress increases monotonically with the pinch temperature.

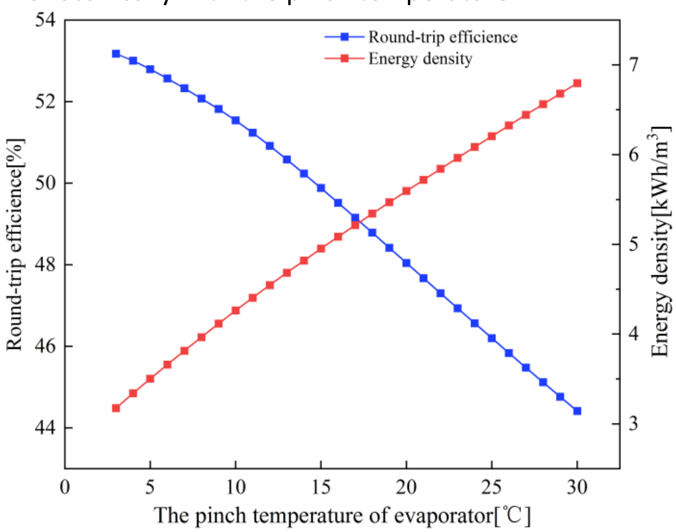

Fig. 4 Effect of the pinch temperature of evaporator on RTE and energy density

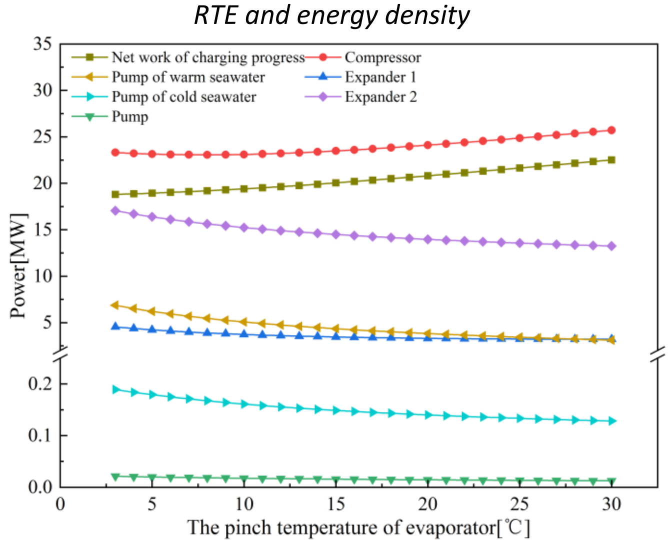

Fig. 5 Variations of power with pinch temperature of evaporator

4.3 Effect of compressor and pump outlet pressure

Fig .6 and Fig .7 reveal the effect of compressor and pump outlet pressure. It is observed that, for a constant compressor outlet pressure, the maximum value of RTE and energy density occurs when the pump outlet pressure is slightly lower than the compressor outlet pressure. The maximum value of RTE and energy density are 54.2% and 7.34 *kWh/m³* respectively.

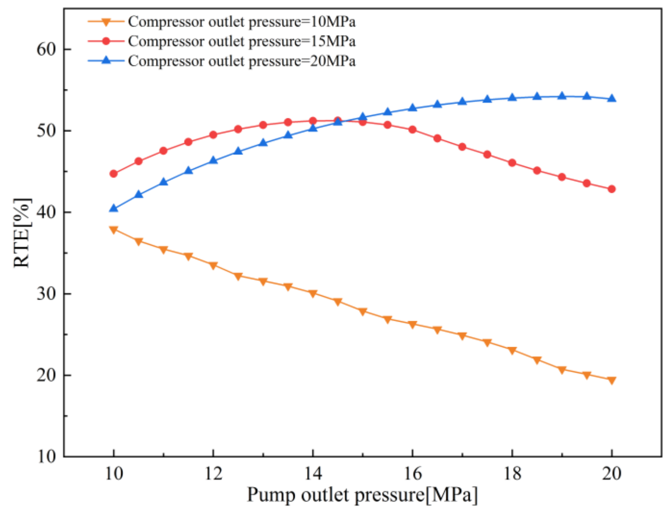

Fig. 6 Variations of RTE with compressor and pump outlet pressure

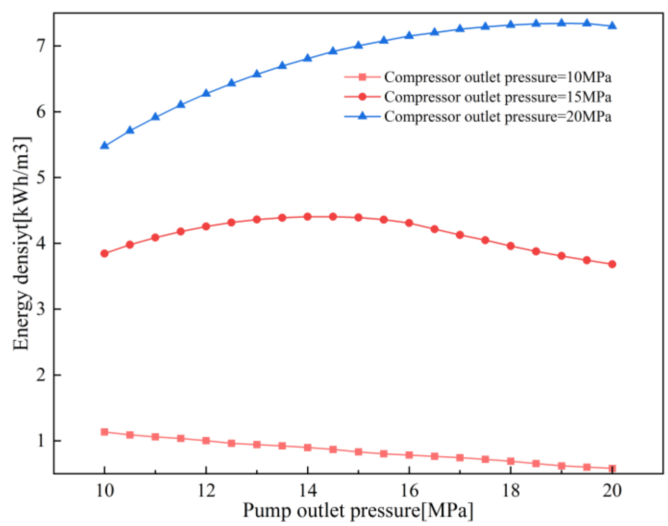

Fig. 7 Variations of energy density with compressor and pump outlet pressure

5. CONCLUSIONS

In this study, a transcritical CO_2 carnot battery system with seawater is discussed. The effects of several key parameters are evaluated. The following results are obtained: the maximum RTE of the system is 54.2% when the compressor and the pump outlet pressure are 20*MPa* and 19*MPa* respectively. The RTE and energy density show a decreasing trend as the temperature of cold seawater. RTE and energy density show opposite trends with increasing pinch temperature of evaporator.

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NONE

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