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Performance Analysis of Isobaric Compressed CO_2 Energy Storage System with Pumping Compensation

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ABSTRACT

Compared with the compressed air energy storage system, energy storage system with CO_2 as working fluid has the advantages of high energy storage density and compactness. In this paper, a novel isobaric compressed CO_2 energy storage system with a flexible gas holder is proposed. The thermodynamic modeling of the compressed CO_2 energy storage system is established and the impact of some key parameters on system performance is studied. The results show that round-trip efficiency gradually increases from 67.86% to 77.00% as the compressor isentropic efficiency rises from 75% to 95%. As the relative pressure loss rate of the heat exchanger increases from 0.25% to 4.5%, the system round-trip efficiency experiences a decline from 73.12% to 71.49%.

Keywords: compressed CO_2 energy storage, thermodynamic analysis, key parametric analysis, isobaric storage

NONMENCLATURE

Abbreviations	
CO_2	Carbon dioxide
CCES	Compressed carbon dioxide energy storage
LCES	Liquid carbon dioxide energy storage
Symbols	
$\dot{W}$	Power(W)
$\dot{m}$	Mass flow rate (kg/s)
h	Specific enthalpy (J/kg)
$\dot{Q}$	Heat transfer rate (W)
η	Isentropic efficiency (%)
RTE	Round-trip efficiency (%)
$\dot{E}$	Exergy (kW)

1. INTRODUCTION

The escalation of energy demand coupled with the rapid pace of urbanization has led to a worsening of environmental pollution issues. Therefore, the development of a low-carbon economy has become a social consensus [1]. However, renewable energy poses a certain threat to the stable output of electricity due to its instability and fluctuation [2].

Energy storage technology is an effective way to address this issue. Large-scale energy storage technologies suitable for power grids currently encompass pumped hydro storage and compressed gas energy storage [3]. Compressed CO_2 energy storage (CCES) has attracted the attention of researchers from various countries due to its compact system components and high energy conversion efficiency. Liu et al. [4] proposed two kinds of supercritical CO_2 Brayton cycle tower solar thermal power generation systems using compressed CO_2 energy storage. The findings indicated that utilizing transcritical CO_2 as the bottom cycle working fluid resulted in higher efficiency for the combined cycle compared to other organic working fluids. Zhao et al. [5] examined a CO_2 energy storage cycle, configured by three section compression/expansion, two-tank heat storage, artificial tank for storing high pressure CO_2 liquid and gas holder for storing ambient pressure CO_2. The results demonstrated that the round trip efficiency could reach 71 %, with a levelized cost of electricity of 0.1252 $/kWh. Fu et al. [6] proposed a photothermal-assisted liquid carbon dioxide energy storage (LCES) system to increase the inlet temperature of expander. The numerical data showed that incorporating photothermal assistance greatly improved the thermal efficiency of the LCES system. The exergy efficiency rised from 16.00 % to 62.23 %, while the energy storage density jumped from 8.06 kWh/m^3 to 18.59 kWh/m^3. Peng et al. [7] reported a new type of adsorption transcritical compressed CO_2 energy storage system and found that the heat exchange

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efficiency greatly affected the response time of the system during the application. Zhang et al. [8] put forward a novel trans-critical compressed CO_2 energy storage system based on 13X zeolite variable temperature adsorption. The result showed that the release pressure at the critical point caused abrupt changes in system performance.

In this work, an isobaric compressed CO_2 energy storage system with pumping compensation is proposed. Thermodynamic models of system components are established for exergy analysis. Parametric study is carried out such as compressor isentropic efficiency, liquid CO_2 tank pressure and heat exchanger relative pressure loss rate to identify the impacts of key thermodynamic parameters on system performance.

2. SYSTEM DESCRIPTION

The system is specifically provided with all specific details in Fig. 1. The system consists of compressors, turbines, heat exchangers, a CO_2 tank, a flexible gas holder, hot tanks, cold tanks, water turbine, a water tank and water pumps. Specifically, The CO_2 tank is divided into two parts by a seal membrane, with CO_2 inside the seal membrane and water outside. The volume of the seal membrane changes with the filled capacity of CO_2 to maintain the constant pressure. The entire process involves water circulation and CO_2 circulation, both of which are closed-loop systems.

During the charging process, the gaseous CO_2 stored in the flexible gas holder is compressed by compressors to a supercritical state, the supercritical CO_2 exhausted from compressor is cooled by the cooling fluid to a liquid state while the heat produced in the compression process is stored in hot tanks, and the liquid CO_2 is injected into the seal membrane inside CO_2 tank, the expanded seal membrane squeezes water out of the CO_2 tank, which then enters the water turbine for operation, and is subsequently stored in the water tank.

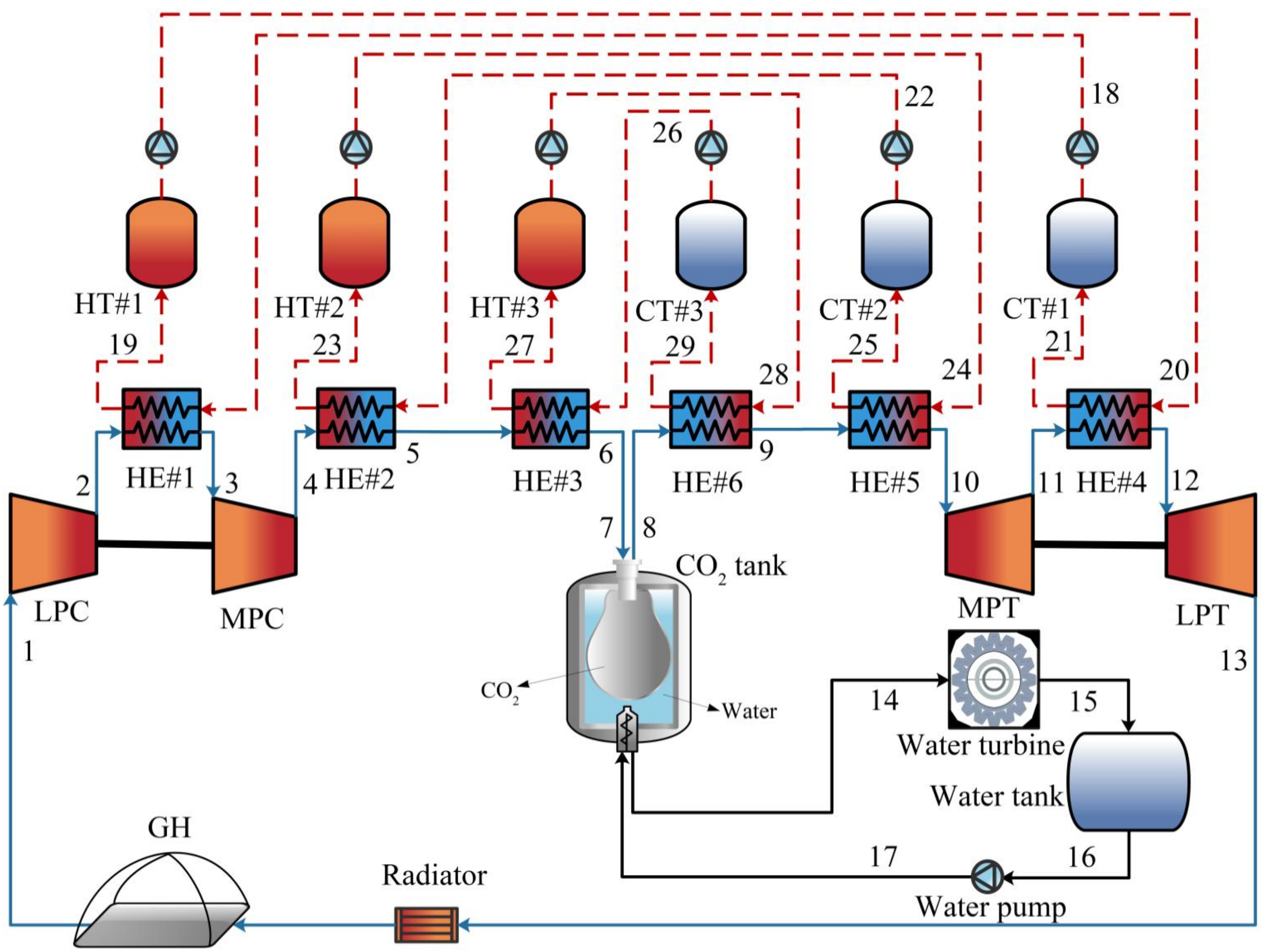

Fig. 1 Specific details of the proposed CCES

During the discharging process, the water in the water tank is pumped into the CO_2 tank through a water pump, squeezing out the CO_2 inside the seal membrane, causing the volume of the seal membrane to gradually decrease. CO_2 is heated by the heat exchangers to a gaseous state, and then fed into the turbines, high-temperature CO_2 generates power through turbines expansion. Finally, CO_2 enters the radiator to release excess heat into the environment before re-entering the flexible gas holder for storage.

3. SYSTEM MODELING

To evaluate the system performance, the thermodynamic model is established firstly.

3.1 Compressor model

The power of compressors is expressed as [9]:

$$\dot{W}_{LC} = \dot{m}_1 (h_2 - h_1) \tag{1}$$

$$\dot{W}_{HC} = \dot{m}_3 (h_4 - h_3) \tag{2}$$

Isentropic efficiency (η_C) is defined as [9]:

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

3.2 Turbine model

The power generated by the turbine is expressed as [10]:

$$\dot{W}_{LT} = \dot{m}_{12} (h_{12} - h_{13}) \tag{4}$$

$$\dot{W}_{HT} = \dot{m}_{10} (h_{10} - h_{11}) \tag{5}$$

Isentropic efficiency (η_T) is defined as [10]:

$$\eta_T = \frac{h_{in} - h_{out}}{h_{in} - h_{out,is}} \tag{6}$$

3.3 Heat exchanger model

The energy balance equation is expressed as [10]:

$$\dot{Q} = \sum_{i=1}^{N} \dot{Q}_i \tag{7}$$

$$\dot{Q} = \dot{m}_{CO_2} (h_{CO_2,i+1} - h_{CO_2,i}) = \dot{m}_{water} (h_{water,i+1} - h_{water,i}) \tag{8}$$

3.4 Exergy model

Considering that exergy destruction is an important criteria to evaluate the performance of the system. According to the definition, the exergy destruction of each component is [11]

$$\dot{E}_d = \sum \dot{E}_{in} - \sum \dot{E}_{out} \tag{9}$$

3.5 Performance evaluation criteria

Round-trip efficiency (*RTE*) can be defined as [12]:

$$RTE = \frac{\dot{W}_{discharge} t_{discharge}}{\dot{W}_{charge} t_{charge}} \tag{10}$$

Energy density (ρ_E) can be used to determine the storage volume of the system [12]:

$$\rho_E = \frac{\dot{W}_{discharge} t_{discharge}}{V_{tank}} \tag{11}$$

4. RESULTS AND DISCUSSION

The specific design parameters of the system are shown in Table 1.

Table 1

Preliminary design parameters.

Parameter	Unit	Value
Ambient temperature	°C	20
Ambient pressure	MPa	0.1
Compressor isentropic efficiency	%	85
Turbine isentropic efficiency	%	92
Heat exchanger relative pressure loss rate	%	2
Output power	MW	100
Charge duration	h	7
Discharge duration	h	7

4.1 The influence of compressor isentropic efficiency

The influences of compressor isentropic efficiency on *RTE*, energy density and component powers are illustrated in Fig. 2. As shown in Fig. 2(a), *RTE* increases with the increasing of the compressor isentropic efficiency, while the energy density decreases with the increasing of the compressor isentropic efficiency. As indicated in Fig. 2(b), the compressor power has a more sensitive impact on the system *RTE*. Reducing the compressor power leads to an increase in the system *RTE*. Therefore, considering the efficiency of the CCES system, it is recommended to improve the compressor isentropic efficiency.

The influences of compressor isentropic efficiency on exergy destruction of each module are illustrated in Fig. 3. The exergy destruction of the compressor module or the heater module decreases with the rising of the compressor isentropic efficiency, while exergy destruction of the expander module decreases with its increase. The exergy destruction of the cooler module or the water stabilization module remains consistently stable.

4.2 The influence of liquid CO_2 tank pressures

The influences of liquid CO_2 tank pressure on *RTE*, energy density and component powers are expressed in

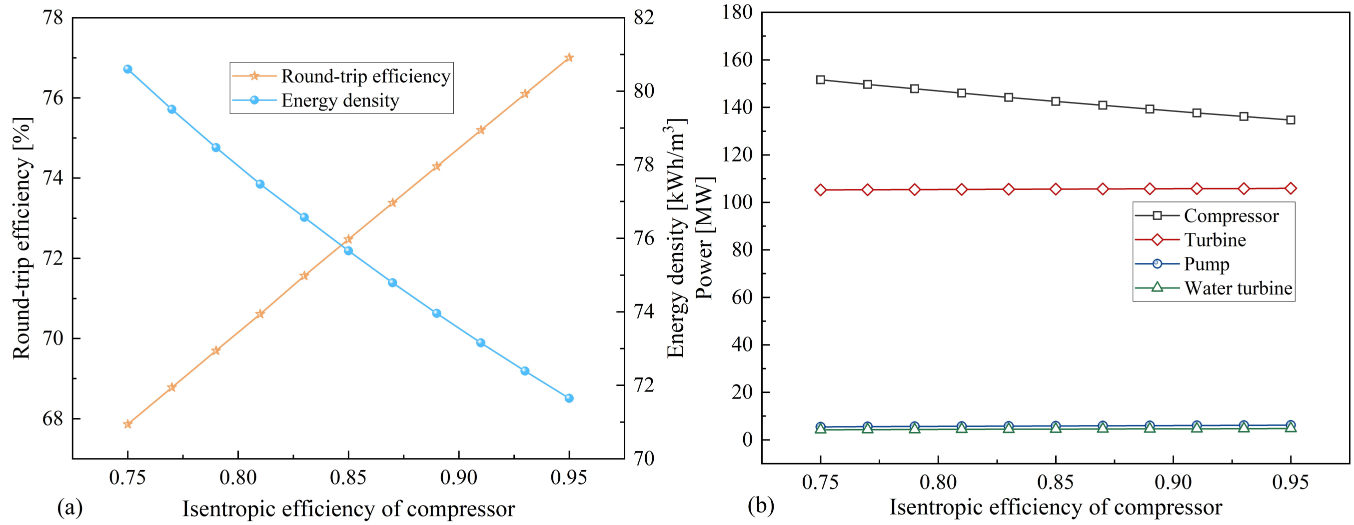

Fig. 2 The influences of compressor isentropic efficiency on RTE, energy density and component powers

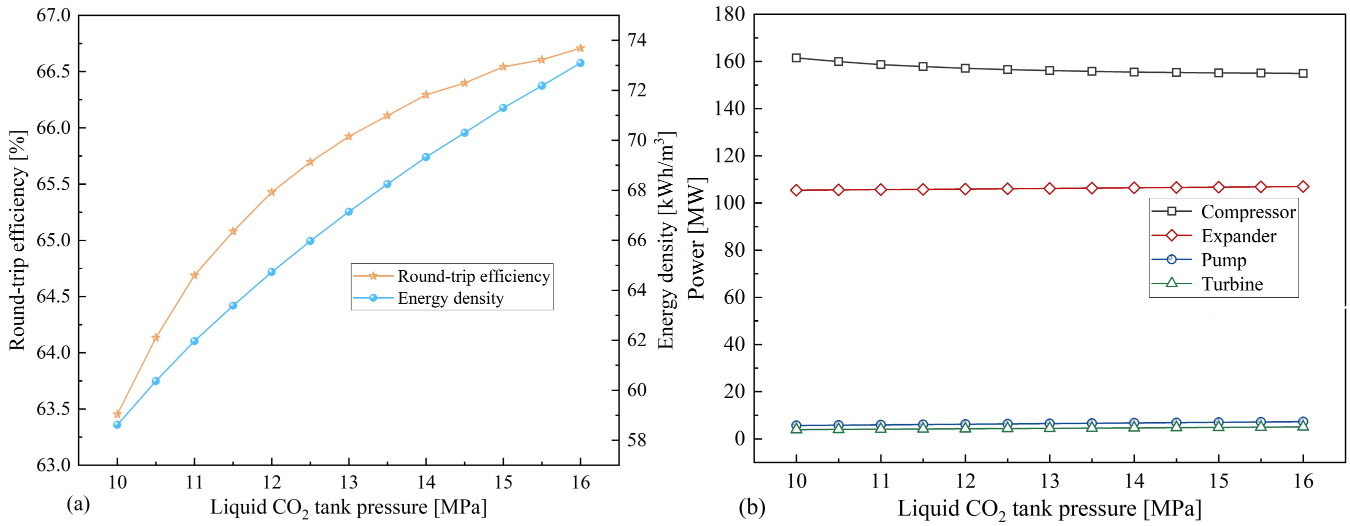

Fig. 4 The influences of liquid CO_2 tank pressure on RTE, energy density and component powers

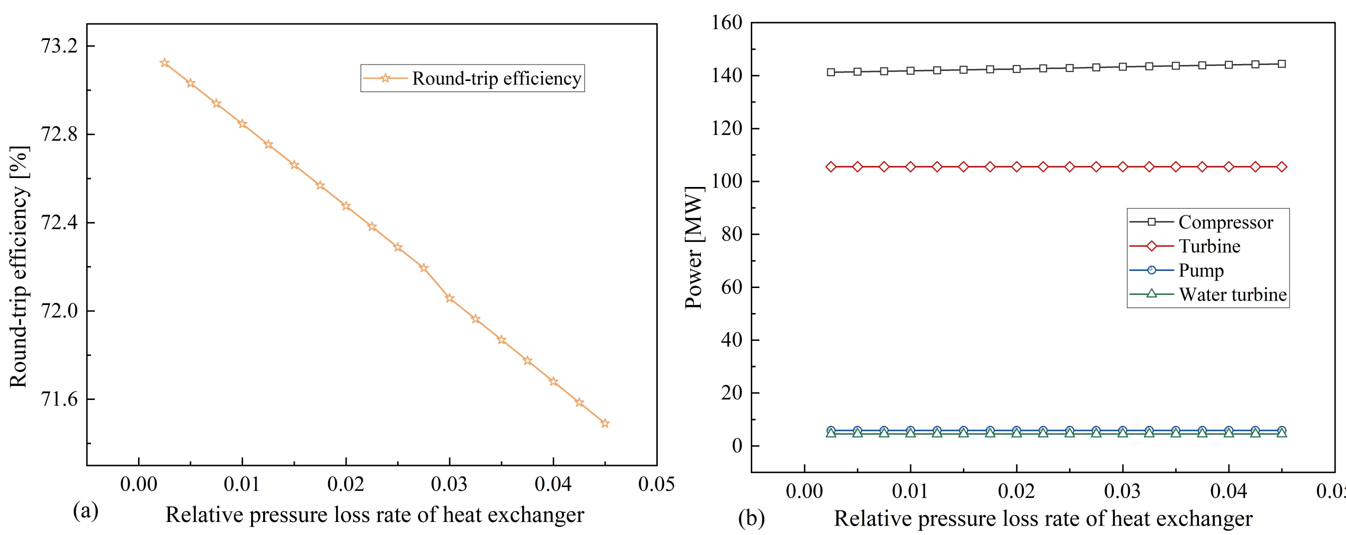

Fig. 6 The influences of heat exchanger relative pressure loss rate on RTE and component powers

Fig. 4. As shown in Fig. 4(a), it can be seen that the increase in *RTE* and energy density is caused by the increase in liquid CO_2 tank pressure. Fig. 4(b) shows the power of the compressor slightly decreases with the increasing in liquid CO_2 tank pressure. Conversely, the power of the expander, pump, or turbine slightly increases with the increasing in liquid CO_2 tank pressure.

The effects of liquid CO_2 tank pressure on exergy destruction of each module are explained in Fig. 5. Except for the water stabilization module, the exergy destruction from other modules decreases with the increase in liquid CO_2 tank pressure, but the main exergy destruction comes from the compressor module and the expander module.

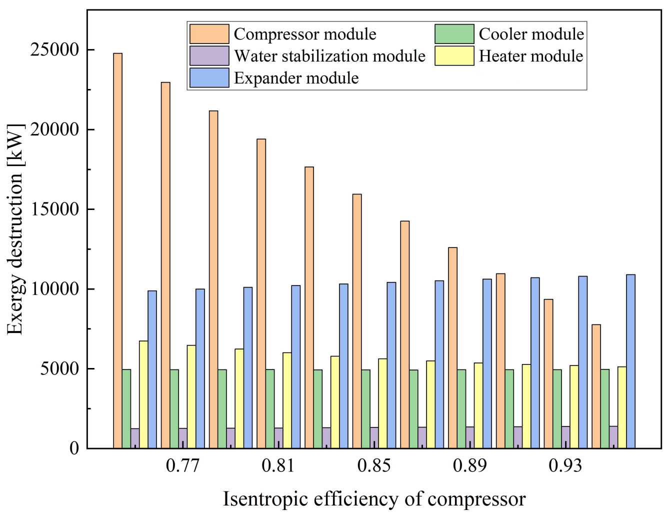

Fig. 3 The influences of compressor isentropic efficiency on exergy destruction of each module

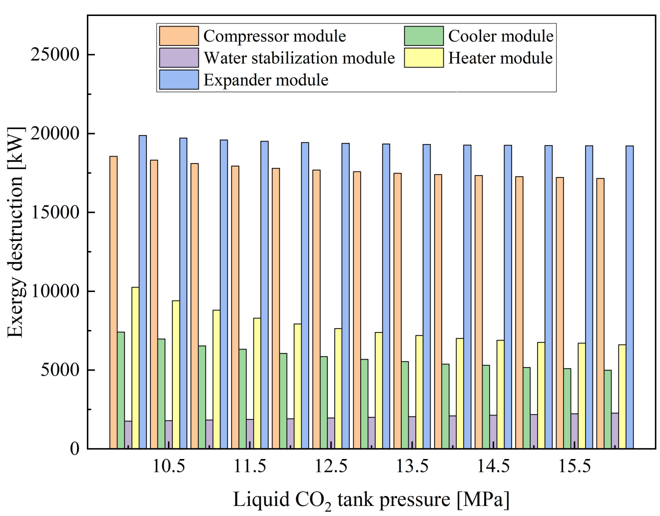

Fig. 5 The influences of liquid CO_2 tank pressure on exergy destruction of each module

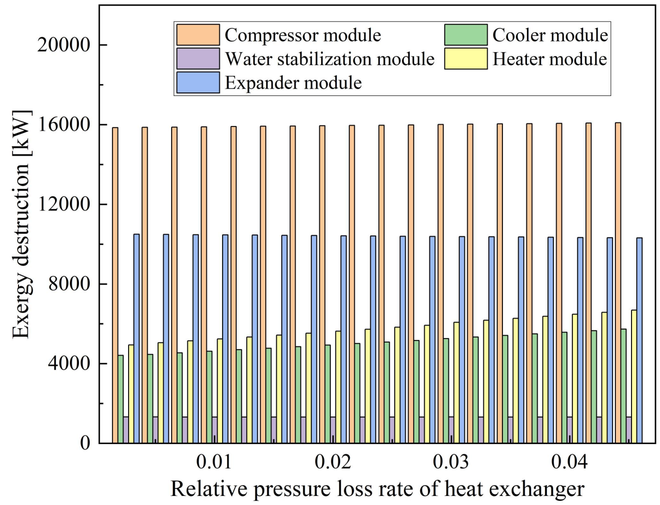

Fig. 7 The influences of heat exchanger relative pressure loss rate on exergy destruction of each module

4.3 The influence of heat exchanger relative pressure loss rate

Fig. 6 exhibits the influences of heat exchanger relative pressure loss rate on *RTE* and component powers. Fig. 6(a) casts a further view that *RTE* drops fast with the increasing heat exchanger relative pressure loss rate, it is mainly due to the increase in compressor power, as indicated in Fig. 6(b).

Fig. 7 exhibits the influences of heat exchanger relative pressure loss rate on exergy destruction of each module. Exergy destruction of the expander module will be reduced as the heat exchanger relative pressure loss rate is improved, exergy destruction of the compressor module, cooler module or heater module increases monotonously with the increasing heat exchanger relative pressure loss rate, and the exergy destruction of water stabilization module almost keeps constant.

5. CONCLUSIONS

The influence of some key parameters on the cycle performance is studied. Results demonstrate that the proposed system round-trip efficiency gradually increases from 67.86% to 77.00% as the compressor isentropic efficiency rises from 75% to 95%. Enhancements to round-trip efficiency and energy density may be achieved through elevation of the liquid CO_2 tank pressure. In addition, when the heat exchanger relative pressure loss rate increases from 0.25% to 4.5%, the round-trip efficiency of the system decreases from 73.12% to 71.49%.

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DECLARATION OF INTEREST STATEMENT

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper. All authors read and approved the final manuscript.

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