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Title: Cyclic transient behavior of the Joule-Brayton based pumped heat electricity storage: Modeling and analysis

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Keywords: pumped heat electricity storage, pumped thermal electricity storage, Brayton, thermal energy storage, heat storage, energy storage

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Abstract: Pumped heat electricity storage (PHES) has the advantages of a high energy density and high efficiency and is especially suitable for large-scale energy storage. The performance of PHES has attracted much attention which has been studied mostly based on steady thermodynamics, whereas the transient characteristic of the real energy storage process of PHES cannot be presented. In this paper, a transient analysis method for the PHES system coupling dynamics, heat transfer, and thermodynamics is proposed. Judging with the round trip efficiency and the stability of delivery power, the energy storage behavior of a 10 MW/4 h PHES system is studied with argon and helium as the working gas. The influencing factors such as the pressure ratio, polytropic efficiency, particle diameters, structure of thermal energy storage reservoirs are also analyzed. The results obtained indicate that, mainly owing to a small resistance loss, helium with a round-trip efficiency of 56.9% has an overwhelming advantage over argon with an efficiency of 39.3%. Furthermore, the increases in the pressure ratio and isentropic efficiencies improve the energy storage performance considerably. There also exist optimal values of the delivery compression ratio, particle sizes, length-to-diameter ratios of the reservoirs, and discharging durations corresponding to the maximum round-trip efficiency and preferable discharging power stability. The above can provide a basis for the optimal design and operation of the Joule-Brayton based PHES.

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Reply: The authors thank the reviewer's comments.

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1. In the transient simulation of this paper, the author seems to set the pressure ratio of compressor a constant value (7, 10 or 13 as shown in Fig. 10). However, should the pressure ratio keep a constant during the transient process of charging or discharging?

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2. The descriptions of models used in this paper are not clear. For example, the transient models of turbo machines are missing. The relationship between pressure ratio, mass flow rate and shaft speed during the transient process should be presented in the paper. And Eq. (15) and (16) should be explained in detail.

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In section 3.2.1, the following analysis have been added.

"During the charging and discharging process, temperatures and densities of the HR and CR outflow gas transiently vary leading to the variation of volume flow rates and rotation rates in the compressor and the expander. The unsteady variation of the turbo-machines shaft power  $P(t)$  owing to the inertia of rotors can be calculated by equation (5).

Where  $I$  is the moment of inertia of rotor and  $\omega(t)$  is the angular velocity. The angular velocity is proportional to the volume flow rate  $Q(t)$  and inversely proportional to the gas density at the constant mass flow rate, with equation (6)

Where  $\omega_{des}$  and  $Q_{des}$  are the angular velocity and the volume flow rate under the design condition, respectively."

In section 3.3, the following analysis have been added.

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By bringing equation (3), (4) into equation (15), (16), and neglecting the unsteady variation of the turbine machines, the transient specific energy can be calculated using equation (19) for the charging process and equation (20) for the discharging process.

Where  $e_{chr}$  and  $e_{dis}$  are specific energy (J/kg) of shaft work during charging and discharging,  $T_{c,in}$  and  $T_{e,in}$  are the inflow temperatures (K) of the compressor and the expander during charging, and the superscript 'denotes the discharging process."

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$$P(t) = -I \cdot \omega(t) \frac{d\omega(t)}{dt} \quad (5)$$

Where  $I$  is the moment of inertia of rotor and  $\omega(t)$  is the angular velocity. The angular velocity is proportional to the volume flow rate  $Q(t)$  and inversely proportional to the gas density at the constant mass flow rate.

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By bringing equation (3), (4) into equation (15), (16), and neglecting the unsteady variation of the turbine machines, the transient specific energy can be calculated as below:

For the charging process,

$$e_{\text{chr}}(t) = T_{\text{c,in}}(t) \cdot \left( r_c(t)^{\kappa/\eta_c} - 1 \right) - T_{\text{e,in}}(t) \cdot \left( 1 - r_e(t)^{-\kappa\eta_e} \right) \quad (19)$$

For the discharging process,

$$e_{\text{dis}}(t) = T'_{\text{e,in}}(t) \cdot \left( 1 - r'_e(t)^{-\kappa\eta_e} \right) - T'_{\text{c,in}}(t) \cdot \left( r'_c(t)^{\kappa/\eta_c} - 1 \right) \quad (20)$$

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## Abstract

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26 Key words: pumped heat electricity storage, pumped thermal electricity storage, Brayton, thermal  
27 energy storage, heat storage, energy storage

## 28 *1 Introduction*

29 The increase in energy consumption and the demand for decrease in carbon emission have  
30 result in great changes in the global energy structure owing to which the proportion of renewable  
31 energy usage has increased and that of fossil energy has gradually decreased [1]. From 2007 to  
32 2017, the total renewable power capacity of non-hydropower renewables increased more than  
33 six-fold (that of solar energy and wind energy increased 48-fold and six-fold respectively) [1, 2].  
34 In particular in 2017, renewable power accounted for 70% of net additions to the global power  
35 generation capacity and 26.5% of the global electricity production [1, 2]. However, the majority of  
36 renewable energy resources have inherent intermittency and instability characteristics, which  
37 results in the carryover of oscillation and unreliability to the power network. For example, 6%  
38 photovoltaic power and 12% wind power was wasted in China in 2017 [3]. Electrical energy  
39 Storage (EES) that converts electrical energy into another form of energy for storage and converts  
40 it back to electrical energy when required, is considered as one of the most promising solutions for  
41 increasing the penetration depth of renewable energy resources [4, 5]. Moreover, EES is an

42 essential link in the energy supply chain, which provides services such as load leveling, peaking  
43 shaving, power quality improvement, and frequency regulation for the traditional power grid, thus  
44 improving the security and utilization rate of the power grid [6-8].

45 Nowadays, there exist various energy storage technologies and different criteria for their  
46 classification. Based on the form of energy storage in the system, the energy storage technologies  
47 can be mainly categorized into five classes: chemical (hydrogen and synthetic natural gas),  
48 electrical (capacitors and superconducting magnetic), electrochemical (classic batteries and flow  
49 batteries), mechanical (flywheels, adiabatic compressed air, pumped heat electrical storage,  
50 pumped hydro and cryogenic energy storage) and thermal (sensible heat, latent heat and  
51 thermochemical heat) [4, 5]. Each EES technology has a suitable range of applications (e.g.  
52 batteries, compressed air energy storage (CAES), and pumped hydro storage are suitable  
53 candidates for peak shaving; flywheels, super-capacitors and superconducting magnetic energy  
54 storage are suitable candidates for frequency regulation) depending on its advantages, drawbacks,  
55 and scales [4, 9].

56 Among the available storage technologies, only pumped hydro storage (PHS) and CAES  
57 are mature large-scale stand-alone electricity storage technologies that can be used to store power  
58 greater than 100 MW under commercial operation [4, 5, 10]. PHS is the most mature EES  
59 technology having a high capacity, long storage period, high efficiency and relatively low cost per  
60 unit of energy. To date, there are more than 300 facilities with a total power of over 170 GW in  
61 operation, which accounts for approximately 96% of the global energy storage capacity [4, 11].  
62 The Bath County Pumped Storage Station in the USA is the largest PHS power station in the  
63 world which has a generation capacity of 3 GW and a storage capacity of 11 h [12]. CAES is

64 another mature technology that is typically used for large scale energy storage. The operational  
65 CAES units in the world are 290 MW/2 h CAES in Huntorf, Germany with an underground  
66 storage cavern of approximately 310,000 m<sup>3</sup> and 110 MW/26 h CAES in McIntosh, Alabama,  
67 USA, with a cavern of approximately 500,000 m<sup>3</sup> [4, 5, 13]. The main barriers for PHS and  
68 CAES plants are similar, in that their construction requires appropriate geographical conditions for  
69 the huge volume of storage.

70 A category of novel energy storage technologies “pumped heat electricity storage (PHES)”  
71 was proposed, which is also called “pumped thermal electricity storage (PTES)” and  
72 “thermo-electrical energy storage (TEES)”. During the charging process of the energy storage,  
73 heat is pumped from cold reservoirs (CRs) to hot reservoirs (HRs) via a heat pump circle and then  
74 stored; during the discharging process electricity is generated by the stored thermal energy through  
75 the heat-work conversion circle. Owing to the advantages of its high energy density and high  
76 efficiency, PHES has captured the attention of researchers as a promising technology for  
77 large-scale energy storage in recent years [14-31]. The categories of the PHES systems is mainly  
78 based on two types of reversible heat-work conversion circles thus far: The Joule–Brayton cycles  
79 [25-31] and the Rankine cycles [14-24].

80 The Rankine-cycle-based PHES system was first proposed by the ABB Company by the  
81 name of TEES [14, 15]. It mainly includes the transcritical CO<sub>2</sub> Rankine cycle, organic Rankine  
82 cycles (ORCs), and subcritical steam Rankine cycle. Morandin et al. studied a TEES system  
83 based on a transcritical CO<sub>2</sub> Rankine cycle with hot-water thermal storage and ice-cold storage,  
84 and then optimized the system with an achieved round-trip efficiency of 60% on using the pinch  
85 analysis approach [16, 17]. Kim et al. then presented an isothermal TEES system based on the

86 transcritical CO<sub>2</sub> Rankine cycle wherein water was sprayed to cool/heat transcritical CO<sub>2</sub> directly,  
87 and it was found that the expansion work and efficiency were improved via the isothermal  
88 expansion owing to the high efficient heat transfer with the thermal storage tanks [18]. Abar et al.  
89 proposed the use of a PTES and bottoming system based on the transcritical ammonia cycle  
90 connected to a natural-gas peak plant and the obtained result indicates that the stand-alone energy  
91 storage efficiencies is between 51%-66% with a stand-alone bottoming efficiency of 24% [19, 20].  
92 Wang and Zhang proposed and analyzed a PHES based on the transcritical CO<sub>2</sub> heat pump cycle  
93 during charging and the cascaded system of the transcritical CO<sub>2</sub> Rankine cycle and the subcritical  
94 NH<sub>3</sub> Rankine cycle utilizing liquid natural gas cold energy with a round-trip efficiency of up to  
95 139% [21]. Steinmann developed the compressed heat energy storage (CHEST) concept based on  
96 stream Rankine cycles combined with sensible and latent heat storage with an estimated round-trip  
97 efficiency of 70% based on the isentropic efficiencies of 0.9 [22]. A PHES based on the ORC  
98 system with the integration of low-temperature heat was also studied. Jockenhöfer et al. found that  
99 the ORC-CHEST system could provide 1.25 times the net power with a heat resource temperature  
100 of 100°C and a maximum exergetic efficiency of 0.59 [23]. Frate et al. studied a PHES system  
101 comprising of a vapor-compression heat pump integrated with a low-grade heat source for  
102 charging and an ORC system for discharging and found that the achievable round-trip efficiency  
103 was 130% on using R1233zd at the heat source temperature of 110 °C and the isentropic  
104 efficiency was 0.8 [24].

105 Using a single-phase gas as the working fluid, the Joule–Brayton-cycle based PHES  
106 generally consists of cold (low-pressure) thermal energy storage (TES) reservoirs, hot  
107 (high-pressure) TES reservoirs, and compressor–turbine-pairs, wherein the CRs and HRs are

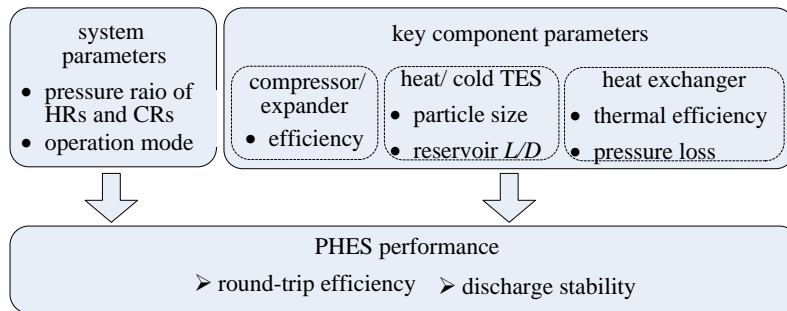


108 generally comprise packed-bed solid thermal energy storage owing to its wide temperature range,  
109 high efficiency, and small pressure loss. Desrues et al. presented a PHES system based on the  
110 Joule–Brayton cycle consisting of two TES reservoirs connected by two compressor-turbine-pairs  
111 and two heat exchangers comprising argon as the working gas and obtained an optimized  
112 round-trip efficiency of 66.7% based on the turbo machines’ polytropic efficiency of 0.9 [25]. Ni  
113 and Caram analyzed the influence of gas and pressure ratios etc. through a simulation and found  
114 the efficiency of the turbomachinery to be the factor limiting the round-trip efficiency [26]. Howes  
115 from the company Isentropic introduced three prototype of PTES and proposed a 2 MW PTES  
116 system with heat and cold thermal storage temperatures of 500 °C and -160 °C having a round-trip  
117 efficiency of up to 72% [27]. White et al. found that the round-trip efficiency and energy storage  
118 density increase with the temperature ratio between the hot and cold TES [28]. McTigue et al.  
119 presented a PTES system based on the Joule–Brayton cycle with a buffer vessel and performed a  
120 theoretical analysis on the PTES system coupled with a packed bed model of the HRs and CRs  
121 [29]. Benato presented a Joule–Brayton PHES system with an electric heater settled after the  
122 compressor in order to maintain the hot–tank temperature during charging, and the performance  
123 and cost evaluation of such a system with different TES materials and different working gases was  
124 analyzed [30,31].

125 There are mainly three categories of TES technologies: sensible heat storage, latent heat  
126 storage, and chemical heat storage [32]. Among the TES technologies, packed bed sensible TES  
127 has been identified as the most suitable technology for the PHES system owing to its advantages  
128 of low cost, small pressure loss, wide applicable temperature range, and large heat transfer surface  
129 area that results in a small temperature difference, etc. [30].

130 The performance of a PHES comprising heat and cold packed-bed reservoirs of different  
131 materials was analyzed in terms of the round-trip efficiency [25, 29, 30], energy density [30, 31],  
132 and costs [30, 31]. However, there still exist defects in the published studies: (1) such a PHES  
133 comprising heat and cold packed-bed reservoirs have strong unsteady characteristics whereas the  
134 majority of the analyses on the PHES were performed using the stable thermodynamics method,  
135 (2) it is not based on continuous cycles, and the initial state of each cycle is strong related to the  
136 state at the end of last cycle for the continuous cycles, (3) it neglects the coupling effect of  
137 dynamics, heat transfer and thermodynamics, (4) it involves the oversimplification of heat  
138 exchangers, and (5) argon or air is used as the working fluid.

139 In this context, we make the first attempt to investigate the cyclic transient behavior of the  
140 Joule–Brayton PHES system. Specifically, on a 10 MW/4 h PHES system, a transient analysis  
141 method for the coupling of the dynamics, heat transfer and thermodynamics of the PHES system  
142 with the components including the compressor, expander, TES reservoirs and heat exchangers is  
143 proposed and solved numerically for multiple continuous cycles. The research presents a more  
144 realistic behavior that is close to the real cyclic operations of the Joule–Brayton PHES, wherein  
145 the working performance including both the round-trip efficiency and power attenuation during  
146 discharging can be obtained. Helium is studied as a monoatomic molecular gas with a high energy  
147 density that can be used as the working gas. This paper is thus focused on the influencing  
148 mechanism of the parameters of the PHES system and the key components that are presented in  
149 figure 1.



150

151 Fig.1. Parameters influencing on PHEs performance

152 In the following, section 2 presents a detailed description of the Joule–Brayton based PHEs  
 153 system, section 3 describes the coupling analysis method of the PHEs system and the components,  
 154 ~~section 4 presents the reliability of the packed beds simulation, section 5 and~~ introduces the  
 155 parameters design of the 10 MW/4 h PHEs system, section ~~6-4~~ presents the results and findings,  
 156 and the last section concludes the paper.

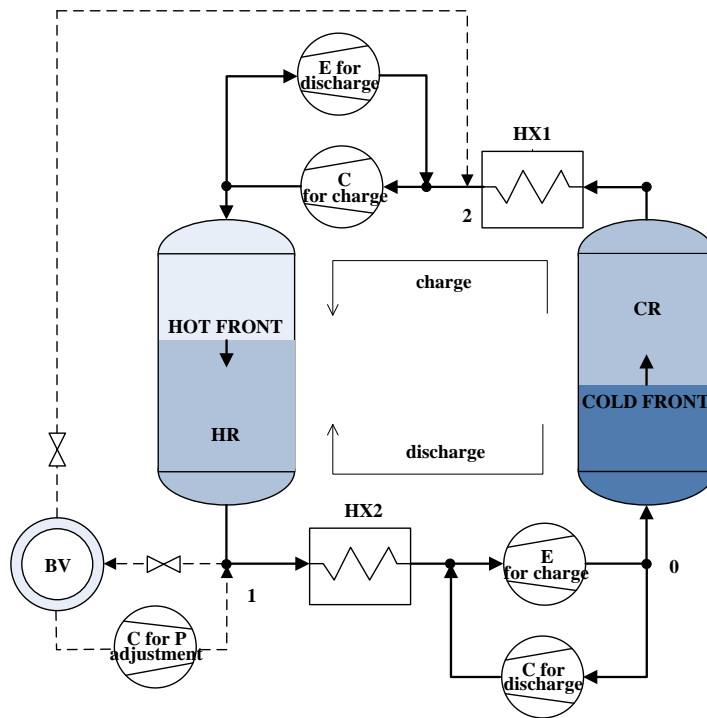
157 *2 Description of Joule–Brayton based PHEs system*

158 Based on the PHEs system proposed by White et al. [28], and McTigue et al. [29], the  
 159 Joule–Brayton PHEs discussed in this paper, as shown in figure 2, mainly consists of a cold  
 160 (low–pressure) TES reservoir, a hot (high–pressure) TES reservoirs, two  
 161 compressor–turbine–pairs(one for charging and the other for discharging) and two heat exchangers.  
 162 The heat exchangers are required to remove surplus heat from the PHEs system and stabilize the  
 163 temperature variation in the packed–bed reservoirs during the charging process. A buffer vessel is  
 164 also required to store/release gas in order to stabilize the system pressure during  
 165 charging/discharging to balance the gas mass changes in the two reservoirs. During the charging

166 and discharging processes, approximately 0.36% of the total flow rate of the gas is required to be  
167 exported to the buffer vessel through position 1 in figure 2 to maintain the system under a constant  
168 pressure. Furthermore, the same amount of gas returns the system through position 2 during the  
169 discharging process. Moreover, a different pressure ratio of the compressor and expander during  
170 the charging and discharging processes can be obtained by adjusting the buffer vessel, valves, and  
171 a pressure adjustment compressor coordinately during the idle period.

172 The working principal of the Joule–Brayton based PHES system is that during the charging  
173 process, the working gas driven by the compressor (for charging) goes through the HR, heat  
174 exchanger 2 (HX2), the expander (for charging), the CR and heat exchanger 1 (HX1) in the  
175 indicated direction of charging. During the charging process, the system operates as a heat pump  
176 wherein the heat is extracted from the CR to the HR while consuming electricity, and cold and  
177 heat thermal energy are stored in the CR and HR respectively. During discharging, the system  
178 operates as a heat engine with the working gas flowing along the indicated direction of discharge,  
179 which is opposite to direction of charging, when the heat returns from the HR to the CR in order to  
180 generate electricity.

181



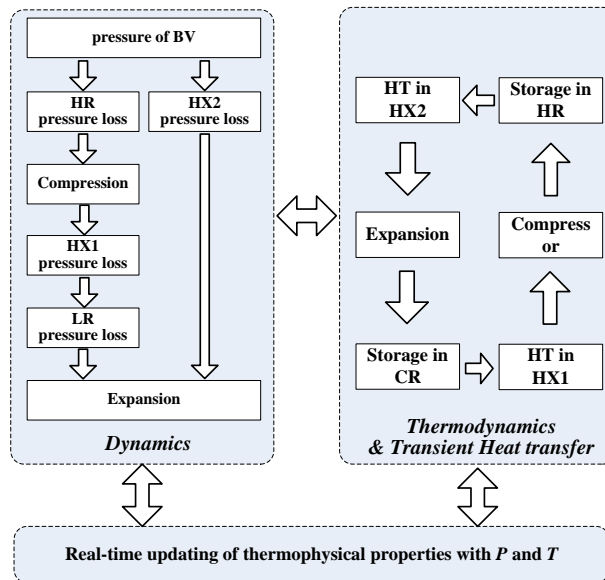
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183 Fig.2. Layout of the PHES system. BV = buffer vessel; C = compressor; E = expander; HX =  
 184 heat exchanger; CR = cold reservoir; HR = hot reservoir.

185 *3 Methodology: coupling analysis of dynamics, transient heat transfer, and thermodynamics*

186 *Dynamics:* In the PHES system, the compressor is the driving component of the gas flow,  
 187 whereas the expander, the cold and hot storage reservoirs and the heat exchangers are the  
 188 components that consume the mechanical energy of the gas during both the processes of charging  
 189 and discharging. During the working process, the temperature profiles and thermophysical  
 190 properties of the gas in the CR and HR are changing with time, thus resulting in a change in the  
 191 pressure loss of the packed bed and leading to a pressure variation of the entire system. The  
 192 pressure at point 1 during charging and at point 2 during discharging are maintained constant by  
 193 the buffer vessel as shown in figure 3. *Heat transfer:* the transient temperature at the outflow of

194 the CR and HR solved using the unsteady mass and energy conservation equations of the packed  
 195 bed. *Thermodynamics*: For a fixed compression ratio of the compressor, the expansion ratio of the  
 196 expander changes with time owing to the variation in the components' pressure loss. Along with  
 197 the transient variation of the temperatures at the inlets and pressure ratios, the power and outflow  
 198 temperatures of the compressor and the expander changes are time-varying. *Thermal properties*:  
 199 The thermal properties of a gas, such as its density, thermal conductivity, and viscosity, have a  
 200 great influence on the system performance. Moreover, the properties of the gas are obtained from  
 201 the National Institute of Standards and Technology (NIST) database and updated in real-time  
 202 during the solution procedure. Therefore, a coupling analysis including dynamics, transient heat  
 203 transfer, thermodynamics and thermal properties is performed to obtain the transient behavior of  
 204 the PHES system as shown in figure 3.



205

206

Fig.3. Coupling analysis of PHES during charging process

207

3.1 Dynamic conservation equation of PHES system

208 In the typically closed PHES system, the compressor provides the driving force of the  
 209 expander and the gas flow in the components including the HR and CR and heat exchangers  
 210 during both the charging and discharging processes. For the PHES system shown in figure 2, if we  
 211 suppose that the total pressure at position 0 is  $P_0$  during the charging and  $p_0'$  during the  
 212 discharging respectively, we obtain:

$$213 \quad (p_0 - \Delta p_{LP} - \Delta p_{HX1}) \beta_c - \Delta p_{HP} - \Delta p_{HX2} - p_0 \beta_e = 0 \quad (1)$$

214 during the charging process and

$$215 \quad p_0' \beta_c' - \Delta p_{HX2}' - \Delta p_{HP}' - (p_0' + \Delta p_{LP}' + \Delta p_{HX1}') \beta_e' = 0 \quad (2)$$

216 | during the discharging process, wherein the superscript ' denotes the discharging process.  $\Delta p$   
 217 indicates the total pressure loss at each component, and  $\beta_c$  and  $\beta_e$  are the compression ratio and  
 218 expansion ratio respectively.

### 219 3.2 Thermodynamics of PHES system

#### 220 3.2.1 Compressor and expander

221 Taking into account the irreversibility loss of turbomachines, the polytropic process of  
 222 compression and expansion occurs with the polytropic efficiencies  $\eta_c$  and  $\eta_e$  respectively. For the  
 223 compressor

$$224 \quad T_{c,out} / T_{c,in} = \beta_c^{\kappa / \eta_c} \quad (3)$$

225 For the expander

$$226 \quad T_{e,out} / T_{e,in} = \beta_e^{-\kappa \eta_e} \quad (4)$$

227 where the parameter  $\kappa$  is defined as  $\kappa = (\gamma - 1) / \gamma$  and  $\gamma$  is the specific heat ratio ( $c_p / c_v$ ) of the gas  
 228 [25, 33].

229 During the charging and discharging process, temperatures and densities of the HR and CR

230 outflow gas transiently vary leading to the variation of volume flow rates and rotation rates in the  
231 compressor and the expander. The unsteady variation of the turbo-machines shaft power  $P(t)$   
232 owing to the inertia of rotors can be calculated by:

$$233 \quad P(t) = -I \cdot \omega(t) \frac{d\omega(t)}{dt} \quad (5)$$

234 Where  $I$  is the moment of inertia of rotor and  $\omega(t)$  is the angular velocity. The angular velocity  
235 is proportional to the volume flow rate  $Q(t)$  and inversely proportional to the gas density at  
236 the constant mass flow rate.

$$237 \quad \omega(t) = \frac{\omega_{des}}{Q_{des}} Q(t) = \frac{\omega_{des} \rho_{des}}{\rho(t)} \quad (6)$$

238 Where  $\omega_{des}$  and  $Q_{des}$  are the angular velocity and the volume flow rate under the design condition,  
239 respectively.

240

### 241 3.2.2 Packed bed heat/cold thermal energy storage reservoirs

242 The domains of the hot and cold thermal energy storage reservoirs are considered as  
243 cylindrical tanks, which include the packed bed of the TES particles and the heat transfer gas  
244 flowing through the void space. On assuming that the flow pattern is a 1D Newtonian plug flow,  
245 neglecting the temperature gradient in the radial direction and neglecting the heat loss through the  
246 well-insulated wall, the governing energy conservation equations of the unsteady two-phase model  
247 of such packed beds is given as follows.

248 For the fluid phase,

$$249 \quad \varphi \frac{\partial \rho_g}{\partial t} + \frac{\partial G}{\partial x} = 0 \quad (57)$$



$$\frac{\partial T_g}{\partial t} + \frac{G}{\rho_g \phi} \frac{\partial T_g}{\partial x} = \frac{h_v}{\rho_g c_{p,g} \phi} (T_s - T_g) \quad (68)$$

For the solid phase,

$$\frac{\partial T_s}{\partial t} = \frac{h_{v,\text{eff}}}{\rho_s c_s (1-\phi)} (T_g - T_s) + \frac{k_{s,\text{eff}}}{\rho_s c_s (1-\phi)} \frac{\partial^2 T}{\partial x^2} \quad (79)$$

where  $h_{v,\text{eff}}$  is the effective volumetric heat transfer coefficient on considering the internal heat conduction resistance in a solid (for a Biot number smaller than 100) having the relationship with the volumetric heat transfer coefficient  $h_v = h_p 6(1-\phi)/d$ . The volumetric heat transfer coefficient of Chandra's equation is used which fits well with the experimental results under both low and high pressures [35, 36]

$$h_{v,\text{eff}} = \begin{cases} h_v & \text{for } Bi \leq 0.1 \\ \frac{1}{\frac{1}{h_v} + \frac{d_p^2}{60k_s(1-\phi)}} & \text{for } 0.1 < Bi \leq 100 \end{cases} \quad (810)$$

$$h_v = 1.45 \frac{Re^{0.7} k_g}{d^2} \quad (119)$$

where the characteristic length for the Biot number is  $d_p/6$  [37].

$$Bi = \frac{h_p d_p}{6k_s} \quad (120)$$

$k_{s,\text{eff}}$  is the effective thermal conductivity for the non-contiguous spherical particles in a dispersion medium given by [38, 39]:

$$\frac{k_s - k_{s,\text{eff}}}{k_s - k_g} \left( \frac{k_{s,\text{eff}}}{k_g} \right)^{\frac{1}{3}} = \phi \quad (131)$$

which is solved by performing iteration.

The dramatic temperature changes dramatically in the packed beds would lead to a change in the volume flow rate and thermoproperty of the gas in the packed bed. In this paper, the packed bed is divided into  $n$  sections along the axis, and the pressure drop across the packed bed and each

269

section are given by the Ergun equation shown as below [34].

270

$$\Delta p(i) = \frac{\Delta L \cdot G^2}{\rho(i) \cdot d} \left( 1.75 \frac{1-\phi}{\phi^3} + 150 \frac{1-\phi}{\phi^3} \frac{\mu(i)}{Gd} \right) \quad (142)$$

271

$$\Delta p = \sum_{i=1}^n \Delta p(i) \quad (153)$$

272

where  $\Delta p$  and  $\Delta p(i)$  are the pressure drop across the packed bed and the pressure drop across

273

the  $i_{th}$  section, respectively, and  $\Delta L$  ( $\Delta L = L/n$ ) is the length of each section.

274

### 3.2.3 Heat exchanger

275

In the PHES system, the heat exchangers play important roles including removing the surplus

276

heat and stabilizing the temperature fluctuations from the HR and CR during the charging process.

277

Water from the cooling towers is usually selected as an efficient cooling media for heat

278

exchangers having a temperature approximately about 2–5° C higher than the ambient temperature.

279

As the heat capacity of the cooling water is greater than that of the gas and on ignoring the

280

influence of the heat exchanger heat capacity, the outflow temperature from the heat exchanger

281

can be obtained as follows.

282

$$T_{g,o}(t) = T_{g,i}(t) - \varepsilon \frac{\dot{m}_g c_{p,g}}{\dot{m}_w c_{p,w}} (T_{g,i}(t) - T_{w,i}) \quad (164)$$

283

where  $\dot{m}$  and  $c_p$  are the mass flow rate and heat capacity ~~of the water and gas~~, and  $\varepsilon$  is the

284

heat exchanger effectiveness.

285

### 3.3 Systemic analyses *of PHES system*

286

~~For the PHES system, In the gas temperature and pressure variation in the PHES system, the~~

287

transient ~~specific shaft work~~energy performed during charging and delivered during discharging.

288

~~with considering the unsteadiness of the compressor and expander, can~~ be obtained using equation

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289 (~~45~~17) and equation (186), respectively.

$$290 \quad e_{\text{chr}}(t) = e_{\text{c,chr}}(t) - e_{\text{e,chr}}(t) + \frac{1}{\dot{m}c_p} (P_e(t) + P_c(t)) \quad (17)$$

$$291 \quad e_{\text{dis}}(t) = e_{\text{e,dis}}(t) - e_{\text{c,dis}}(t) - \frac{1}{\dot{m}c_p} (P_e(t) + P_c(t)) \quad (18)$$

292 As shown in equation (5), the moment of inertia parameters of the compressor and the  
 293 expander are needed for calculating  $P(t)$ , whereas there is no available compressor and the  
 294 expander for the 10MW PTES system. In this study, referring to the compressor and the expander  
 295 in the 10MW Advanced compressed air energy storage, the moment of inertia of compressor and  
 296 the expander rotor is taken 1800 kgm<sup>2</sup> at the rated speed of 1500 rpm, referring to the  
 297 compressor and the expander in the 10MW Advanced Compressed air energy storage [42, 43].  
 298 Among the situations of in this study, the maximum absolute value of angular acceleration of  
 299 the expander rotor and the compressor rotor is 0.0063 rad/s<sup>2</sup> and 0.0026 rad/s<sup>2</sup> respectively, and  
 300 the corresponding  $P_e(t)$  and  $P_c(t)$  is -3.47 kW and 0.36 kW, which are less than ±0.04% of the  
 301 transient shaft power and can be negligible.

302 By bringing equation (3), (4) into equation (15), (16), and neglecting the unsteady variation  
 303 of the turbine machines, the transient specific energy/the transient shaft work can be calculated as  
 304 below:

305 For the charging process,

$$306 \quad e_{\text{chr}}(t) = T_{\text{c,in}}(t) \cdot (r_c(t)^{k/\eta_c} - 1) + T_{\text{e,in}}(t) \cdot (r_e(t)^{-k/\eta_e} - 1)$$

$$307 \quad e_{\text{chr}}(t) = T_{\text{c,in}}(t) \cdot (r_c(t)^{k/\eta_c} - 1) - T_{\text{e,in}}(t) \cdot (1 - r_e(t)^{-k/\eta_e}) \quad (195)$$

308 For the discharging process,

$$309 \quad e_{\text{dis}}(t) = T'_{\text{c,in}}(t) \cdot (1 - r'_c(t)^{k/\eta_c}) + T'_{\text{e,in}}(t) \cdot (1 - r'_e(t)^{-k/\eta_e})$$

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$$e_{\text{dis}}(t) = T'_{\text{e,in}}(t) \cdot \left(1 - r'_e(t)^{-\kappa/\eta_e}\right) - T'_{\text{c,in}}(t) \cdot \left(r'_c(t)^{\kappa/\eta_c} - 1\right) \quad (2046)$$

Where  $e_{\text{chr}}$  and  $e_{\text{dis}}$  are specific energy (J/kg) of shaft work during charging and discharging,  $T_{\text{e,in}}$  and  $T_{\text{c,in}}$  are the inflow temperatures (K) of the compressor and the expander during charging, and the superscript ' denotes the discharging process.

On assuming no mechanical loss, the round-trip coefficient of the PHES system is obtained on using the quotient of the net delivered shaft work during the discharging process and the consumed shaft work during the charging process, as shown in equation (2147)

$$\chi = \frac{\text{net work output}}{\text{net work input}} = \frac{\int_{\text{dis}} \dot{m}_{\text{dis}} c_p e_{\text{dis}}(t) dt}{\int_{\text{chr}} \dot{m}_{\text{chr}} c_p e_{\text{chr}}(t) dt} \quad (4721)$$

where  $\dot{m}$  is the mass flow rate through the compressors and expanders.

The stability of the delivery power is another important factor affecting for the energy storage system. In this paper, the offset ratio of the delivery power is increased to evaluate the stability which is defined as the ratio of the offset range of the delivery power to the maximum value during the delivery period, as presented in equation (2248).

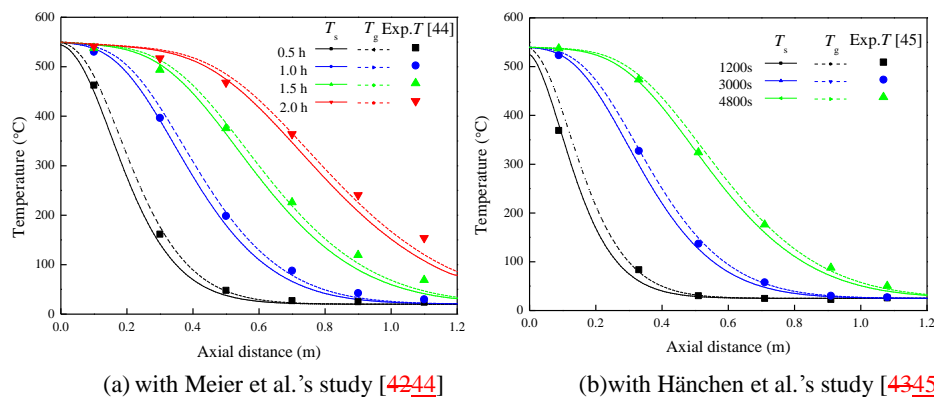
$$\theta = \frac{\text{Max}(e_{\text{dis}}(t)) - \text{Min}(e_{\text{dis}}(t))}{\text{Max}(e_{\text{dis}}(t))} \quad (48)$$

For the PHES system, a smaller offset ratio indicates a more stable delivery power during the discharging process.

In order to validate the transient equation of the packed beds, the numerical simulations of the TES process of the crushed steatite (magnesium silicate rock) packed beds are performed by solving equations (57)–(134) with the parameters used in reference [442] and [453].

The temperature dependence of the heat capacity of the crushed steatite ( $\text{Mg}_3\text{Si}_4\text{O}_{10}(\text{OH})_2$ ) is

331 taken in to consideration in the simulation [40]. The temperature profiles along the axial distance  
 332 of the packed beds of the simulated and experimental results are shown in figures 4 (a) and 4(b); it  
 333 can be observed that an obvious thermocline occurs during the charging process and the simulated  
 334 profiles fit well with the experimental results which proves the accuracy of the simulation method  
 335 [42, 43].



338 Fig.4. Comparison between the simulation and experimental results of the temperature  
 339 profiles in the packed beds

340 5.3.4 Parameters design of the 10 MW/4 h PHES system

341 In this paper, a Joule–Brayton based PHES system of 10 MW (nominally discharging  
 342 power 10 MW, 4 h charging, and 4 h discharging) was designed and analyzed. The designed  
 343 parameters of the PHES system with either argon or helium as the working gas are shown in Table  
 344 1 wherein the pressure ratio is 10 as in McTigue et al.'s study [29]. It should be noted that the heat  
 345 capacity of helium is almost ten times that of argon, and thus, the mass flow rate of helium is  
 346 approximately only 1/10th that of argon in a PHES system of the same power. Therefore, the  
 347 pressure loss in the heat exchangers and packed-bed reservoirs would be decreased greatly on  
 348 using helium instead of argon.

349 Table 1 Designed parameters of PHES system of 10 MW discharging power

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Working gas	HP Pressure (MPa)	LP Pressure (MPa)	Average $c_{p,g}$ (J/kg/K)	Mass flow rate (kg/s)	Polytropic efficiency	$\epsilon$ of HXs	$\Delta p$ of HP HXs (kPa)	$\Delta p$ of LP HXs (kPa)	Cooling water temperature (K)
Argon	1.05	0.105	525	85.1	0.9	0.9	3	20	300
Helium	1.05	0.105	5193	8.6	0.9	0.9	0.3	2	300

350

351 The designed 10 MW/4 h PHES system consists of an HR and a CR with a packed bed of  
352 basalt particles. The packed-bed TES is unstable and has a larger packed bed volume, which  
353 results in a more stable output temperature but a higher cost and lower energy storage density. In  
354 consideration of the thermal front volume, the designed volumes of the HR and CR are selected to  
355 be twice the minimum design volume obtained using from the energy balance method  
356  $V = 2Q / (\overline{\rho_s c_s} \Delta T)$ . The detailed parameters of the HR and CR are shown in table 2. In this design,  
357 the basalt is chosen as the hot and cold TES material, as it has a good heat capacity and thermal  
358 stability within the temperature range of  $-196^\circ\text{C}$ – $800^\circ\text{C}$ . Based on the TA Q2000 DSC, the heat  
359 capacity of basalt is found to be strongly dependent on the temperature as shown in figure 5, and  
360 the linear fit equation is given in equation (23+7).

361

$$c_p(T) = 0.23 + 0.00201 \cdot T \quad (23+9)$$

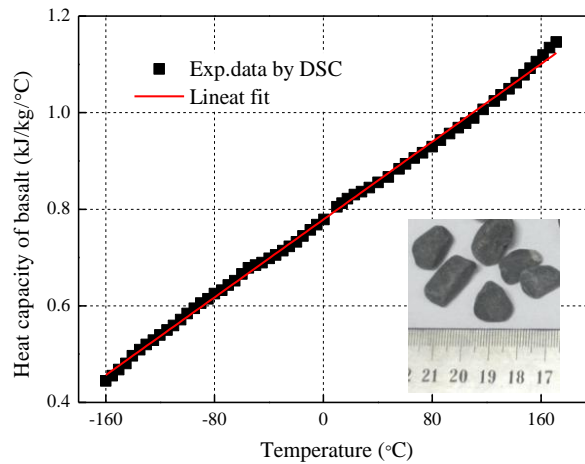


Fig.5. Dependence of heat capacity of basalt with temperature measured using DSC

Table 2 Hot and cold reservoir details for 10 MW/4 h PHES system  
(the total volume is twice the minimum design volume)

Reservoir	Pressure (MPa)	Density of solid material (kg/m <sup>3</sup> )	Porosity	Average $d_p$ (mm)	Total Volume (m <sup>3</sup> )	$L$ (m)	$D$ (m)
Heat	1.05	5175	0.35	30	460	10.96	7.31
Cold	0.105	5175	0.35	30	740	12.86	8.56

#### 5.4.3.5.4.1 Heat exchangers design and analysis

For eliminating surplus heat and stabilizing the temperature variation, two heat exchangers are required for the Joule–Brayton cycle PHES. One heat exchanger is under low pressure and the other is under medium/high pressure, and such heat exchangers are required to be compatible with a wide range of operation conditions, high efficiency and low pressure loss wherein the shell-and-tube heat exchangers are the optimal choices. According to the working conditions of the PHES system, the one shell pass, two tube pass TEMA shell-and-tube heat exchangers were

376 designed for the hot and cold heat exchangers using the  $\epsilon$ - $NTU$  method and an empirical relation  
377 [41], wherein the heat transfer tubes have an outer diameter of 32mm and thickness of 2 mm, and  
378 the working gas passes through the shell side to minimize the pressure loss of the gas side.

379 Figure 6 shows the variation of the heat transfer efficiency and pressure drop of HX1 (low  
380 pressure) and HX2 (high pressure) with the tube number and tube length on using argon and  
381 helium respectively. The heat-transfer tube number ranges from 100 to 1000, and the tube length  
382 ranges from 0.5 m to 10.0 m. It can be found that an increase in the number of tubes would  
383 obviously decrease the pressure loss and improve the efficiency, and an increase in the tube length  
384 would lead to an increase in the efficiency and pressure loss. In order to obtain a high round-trip  
385 efficiency, the PHES system requires heat exchangers with a small pressure loss and high  
386 efficiency which can be obtained by using a large number of long tubes but this amount and length  
387 cannot be increased beyond a certain limit owing to the prohibitive cost.

388 From figure 6, it can be found that for heat exchangers of the same size, the efficiencies are  
389 similar when using argon and helium, ~~whereas but~~ the pressure drop observed when using helium  
390 is only approximately 1/10th the pressure drop observed when using argon owing to the difference  
391 in the mass flow rate. Furthermore, the pressure drop of HX1 under a low pressure is several times  
392 higher than the pressure drop of HX2 under a high pressure because of the high volume flow rate  
393 under the low pressure. From the design of the PHES system, the heat exchangers with an  
394 efficiency of 0.9, the pressure loss of HX1 of 20 kPa and pressure loss of HX2 of 3 kPa on using  
395 argon, and the heat exchangers with an efficiency of 0.9, pressure loss of HX1 of 2 kPa and  
396 pressure loss of HX2 of 0.3 kPa on using helium are achieved and such parameters are selected in  
397 the 10 MW/4 h PHES system.



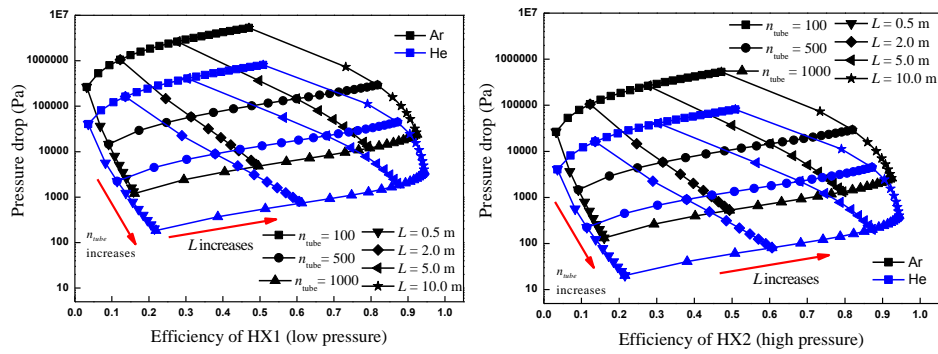


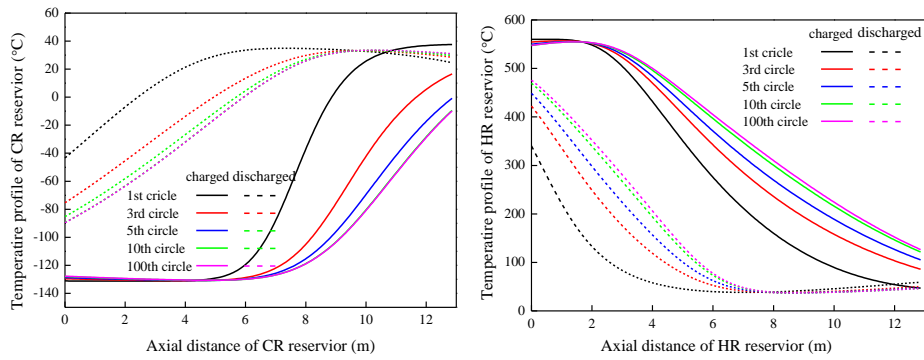
Fig.6. Efficiency versus pressure drop of the shell-and-tube heat exchangers

398  
399  
400

## 6.4 Result and Discussion

### 6.4.1 Cyclic behavior of PHES system

403 Based on the standard parameters in table 1 and 2, and the modeling method described in  
 404 section 3, the working behavior of the PHES system running 100 circles was simulated using  
 405 argon as the working gas; each cycle included 4 h of charging and 4 h of discharging. The axial  
 406 temperature profile of the HR and CR at the end of the charging and discharging processes from  
 407 the 1<sup>st</sup> circle to the 100<sup>th</sup> circle are shown in figures 7(a) and 7(b), respectively. It can be observed  
 408 that, the profiles at the end of the charging and discharging process tend to coincide after several  
 409 cycles. The temperature profiles in the reservoirs can be roughly divided into a stable temperature  
 410 region and a thermocline region wherein the temperature gradient in the thermocline region  
 411 decreases gradually with the cycling.



412

413

Fig.7. Cyclic behaviors of the HR and CR

414

In order to study the cyclic convergence of the PHES system, the factor  $\Delta T_{\text{Max}}(N)$

415

indicates the maximum temperature difference between the adjacent circles at the same axial

416

position and is defined as shown in the equation (1822). As shown in figure 8, the factor

417

$\Delta T_{\text{Max}}(N)$  declines exponential with the circle number where argon has a higher decline rate than

418

helium. After 40 circles, the maximum temperature difference at the same axial position between

419

the adjacent circles is below  $0.1 \text{ } ^\circ\text{C}$  for all the gases and reservoirs which is deemed cyclically

420

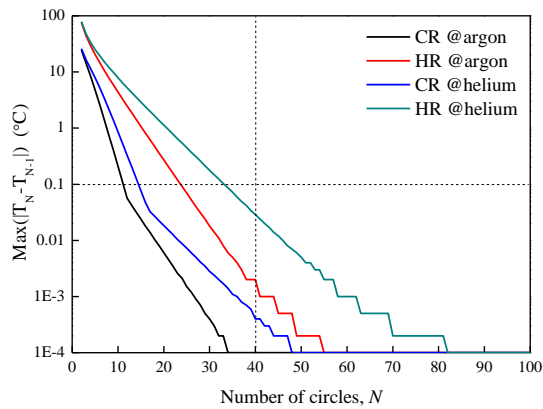
stable. According to this, the following analysis is based on the data of the 40th circles which have

421

achieved the cyclic stable state.

422

$$\Delta T_{\text{Max}}(N) = \text{Max}(|T_{i,N} - T_{i,N-1}|) \quad N=1, 2, 3, \dots \quad (1824)$$



423

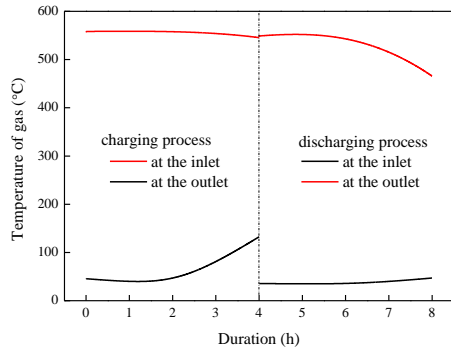
424

Fig.8. Maximum temperature differences between circles versus the number of circles

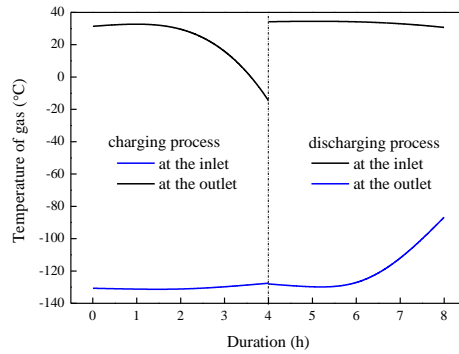
425 Under the cyclic stable state, figures 9(a) and 9(b) show the transient variation of the inflow  
426 and outflow temperatures of the HR and CR during the charging and discharging, respectively,  
427 when using argon as the working gas. This shows that the outflow temperature from the HR  
428 increases continuously after a period of stable state (approximately 1.5 h) during the charging  
429 process and decreases continuously after a period of stable state (approximately 1.5 h) during the  
430 discharging. The outflow temperature from the CR also has a similar unstable behavior but the  
431 temperature variation trend is opposite to that of the HR. Figure 9(c) shows the variation in the  
432 pressure loss of the HR and CR during the charging and discharging processes. It can be found  
433 that the pressure loss of the CR decreases linearly during the charging and increases during the  
434 discharging process, and the opposite phenomenon is observed in the case of the HR. This is  
435 because, during the charging period in the CR, the cold region grows gradually where the volume  
436 flow rate decreases owing to the high density which results in a decrease in the pressure loss, and  
437 during the discharging, the cold region retracts gradually and the pressure loss increases gradually.  
438 For similar reasons, the increase in the hot region in the HR could lead to a higher volume flow  
439 rate, hence increasing the pressure loss during the charging. The expansion ratio increases slightly  
440 during the charging and decreases during the discharging, as shown in figure 8(c), and is mainly  
441 influenced by variations in the pressure loss of the reservoirs. Figure 8(d) shows that the powers of  
442 the PHES compressor, expander and shaft are rather stable during the charging process, and during  
443 the delivery process, the compressor power increases and the expander power decreases gradually,  
444 thus leading to a decrease in shaft power. Based on the parameters listed in tables 1 and 2, the  
445 round-trip efficiency  $\chi$  and the delivery working offset ratio  $\theta$  using argon as the working gas is  
446 39.3% and 71.0%, respectively, and the round-trip efficiency  $\chi$  and delivery working offset ratio  $\theta$

447 using helium is 56.9% and 45.9%, respectively.

448

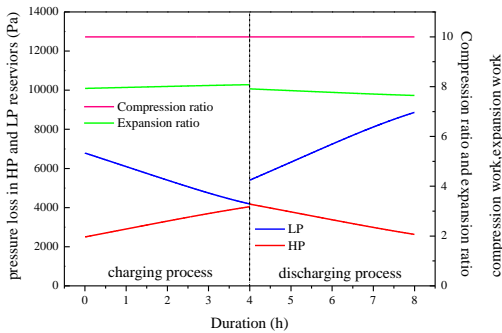


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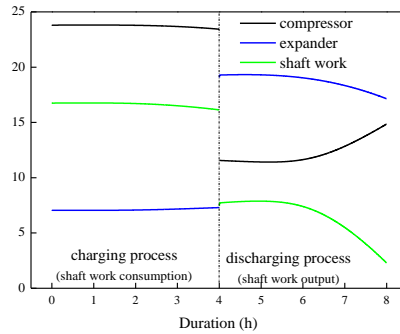
450 (a) inflow and outflow temperature of HP reservoir

(b) inflow and outflow temperature of LP reservoir



451

(c) pressure loss of the HP and LP reservoirs



(d) transient power variation of PHES

Fig.9. Transient behaviors of the HR and CR and PHES system.

~~The influencing factors include the compression ratio in the discharging process only and that for the entire processes, the polytropic efficiency of compressors and expanders, the particle diameter of the particles in the reservoirs, the length to diameter ratio of the reservoirs, the efficiency and pressure loss in the heat exchangers and the discharging duration of the PHES system performance are studied using argon and helium as the working gases.~~

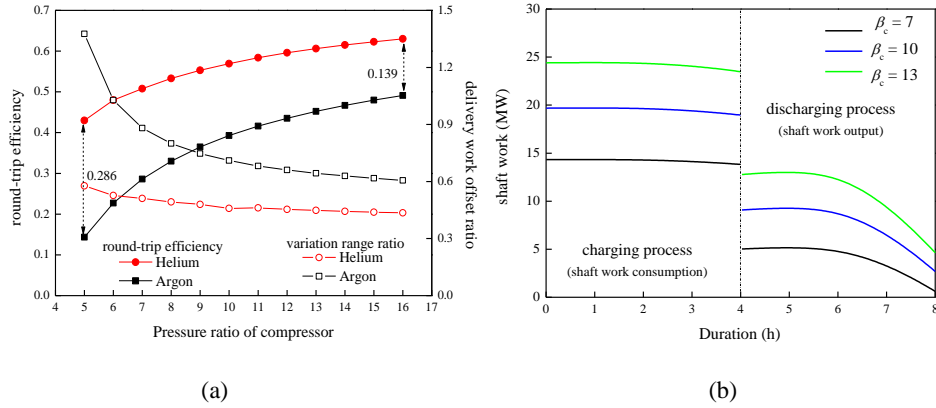
~~6.4.2.1 Effect of compression ratio during charging and discharging~~

The influencing factors include the compression ratio in the discharging process only and that for the entire processes, the polytropic efficiency of compressors and expanders, the particle diameter of the particles in the reservoirs, the length-to-diameter ratio of the reservoirs, the efficiency and pressure loss in the heat exchangers and the discharging duration of the PHES system performance are studied using argon and helium as the working gases.

Figure 10(a) shows the influence of the compression ratio of the compressors ranging from 5 to 16 during both charging and discharging processes on the round-trip efficiency  $\chi$  and the delivery working offset ratio  $\theta$  wherein the other parameters are obtained from in tables 1 and 2. It can be found that the round-trip efficiency increases gradually with the compression ratio  $\beta_c$  from 14.3% at  $\beta_c = 5$  to 49.1% at  $\beta_c = 16$  for argon and from 43.0% at  $\beta_c = 5$  to 63.0% at  $\beta_c = 16$  for helium; the round-trip efficiency of helium is considerably higher than that of argon, with a range of 13.9% to 28.6%. This is mainly because a much smaller pressure loss occurs in the reservoirs and heat exchangers of helium than those of argon, and a greater expansion work can be obtained on using helium. From figure 10(a), it can also be observed that the delivery working offset ratio  $\theta$  decreases with the compression ratio  $\beta_c$ , and the offset ratio  $\theta$  of helium is much lower than that of argon; such a result indicates that the delivery work during the discharging using helium is more stable than that using argon. The transient charging power and delivery power profiles at the compression ratio  $\beta_c$  of 7, 10 and 13 on using argon are shown in figure 10(b). It can be found that both the charging power and discharging power increase with the compressor ratio and an obvious decrease in delivery power occurs during the late discharging period.

Périlhon et al. recommended that the maximum fluid temperature should not exceed 800 °C for a reasonable life of the turbomachines [464]. The maximum temperature of the gas is

482 approximately 750 °C in the PHES system at the compression ratio  $\beta_c$  of 16 for both argon and  
 483 helium, which is within the permitted temperature range.



484  
 485

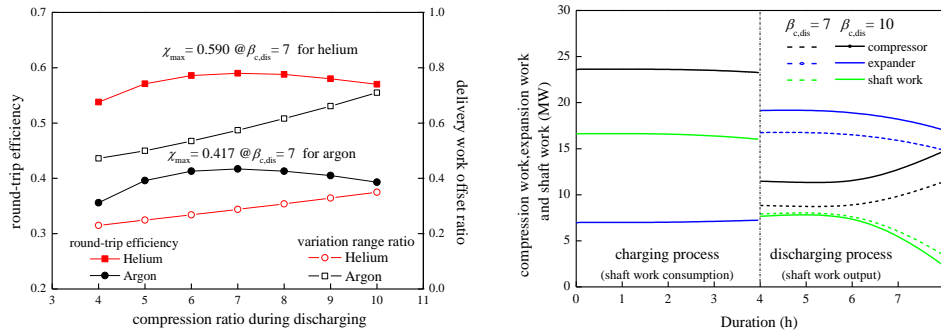
486 Fig.10. Impact of compression ratio during both charging and discharging

487 6.2.4.32 Effect of compressor pressure ratio during discharging

488 Owing to the pressure loss, heat transfer loss and the irreversible loss of the compressor and  
 489 expanders, setting the pressure ratio of the compressor during discharging as the same as that of  
 490 during charging may not be the best choice. After the charging process, the compression ratio of  
 491 the delivery process can be reset by storing some gas in the BV and recharging the system by the  
 492 adjustment compressor during the idle time. At the charging compression ratio of 10 and the other  
 493 parameters listed in tables 1 and 2, figure 11(a) shows the influence of the compression ratio  
 494 ranging from 4 to 10 during the discharging process on the round-trip efficiency  $\chi$  and the delivery  
 495 working offset ratio  $\theta$ . This result indicates that the round-trip efficiency  $\chi$  increased  
 496 first and then decreased with the discharging compress ratio and the maximum round-trip  
 497 efficiency  $\chi$  occurs at the discharging compress ratio of 7 for both argon and helium, the maximum  
 498 round-trip efficiency  $\chi$  obtained using helium is 59.0%, which is considerably higher than that  
 499 obtained using argon: 41.7%. Moreover, it is also indicated from figure 11(a) that the offset ratio  $\theta$

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500 using helium and argon increases gradually with the increase in the discharging compression ratio. As  
 501 shown in figure 11(b), when the charging compression ratio  $\beta_{c,chr}$  is 10, the discharging  
 502 compression power and discharging expansion power at a high pressure ratio of 10 are both higher  
 503 than those at a low pressure ratio of 7. The shaft power at a compression ratio of 10 is lower than  
 504 that at a compression ratio of 7; this is because, the variation amplitude of the compression power  
 505 is greater than that of the expansion power when the discharging compression ratio increases from  
 506 7 to 10.



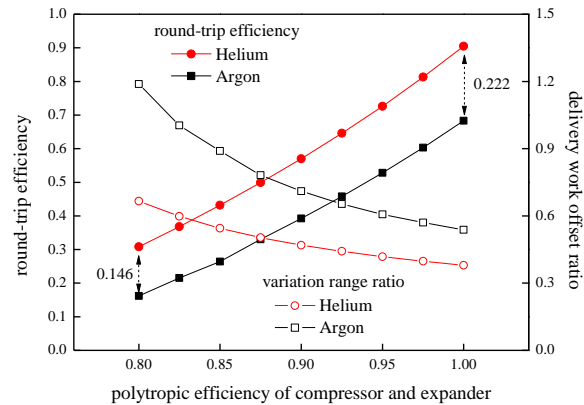
(a) (b)

Fig.11. Impact of compression ratio during discharging (at  $\beta_{c,chr} = 10$ )

6.2.34.4 Effect of polytropic efficiency of both compressors and expanders

511 The plots of the round-trip efficiency  $\chi$  with the polytropic efficiency of both the compressors  
 512 and expanders ranging from 0.8 to 1.0 during both charging and discharging are shown in figure  
 513 12, which the use of argon and helium respectively, and the other parameters are obtained from  
 514 tables 1 and 2. It can be observed that the polytropic efficiency of the compressors and expanders  
 515 have an almost dominant effect on the round-trip efficiency  $\chi$ , such that the round-trip efficiency  
 516 increases from 16.2% at  $\eta = 0.8$  to 68.3% at  $\eta = 1.0$  when using argon, while the round-trip  
 517 efficiency increases from 30.8% at  $\eta = 0.8$  to 90.5% at  $\eta = 1.0$  on using helium. The delivery

518 working offset ratio  $\theta$  in figure 11 shows that the increase in the polytropic efficiency also  
 519 improves the stability of the delivery power.



520

521 Fig.12. Impact of polytropic efficiency of compressor and expander

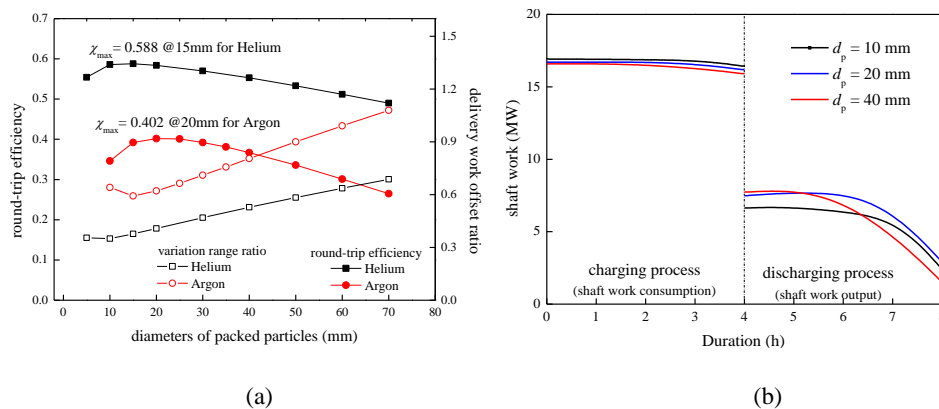
522 6.2.4.5 Effect of TES particles diameter

523 The diameters of the solid TES particles would affect the pressure loss and heat transfer in  
 524 the packed beds and, hence, affect the PHES efficiency. Figure 13(a) shows the influence of the  
 525 particle size in both the HR and CR in the range from 5mm to 70mm on the round-trip efficiency  $\chi$   
 526 and the delivery working offset ratio  $\theta$ . It can be observed that, the round-trip efficiency  $\chi$  first  
 527 increases and then gradually decreases with the particles sizes, the maximum round-trip efficiency  
 528 of 40.2% occurs at  $d_p = 20$  mm for argon and for helium the maximum round-trip efficiency of  
 529 58.8% is obtained at  $d_p = 15$  mm, and such particle sizes always correspond to a small delivery  
 530 working offset ratio  $\theta$ . Such a result is mainly attributed to the joint action of the decrease in the  
 531 pressure loss and increase in the heat transfer temperature difference between the gas and the TES  
 532 materials as the particle size increases. Figure 13(b) shows the transient charging and delivery  
 533 power in the case of particles sizes of 10 mm, 20 mm, and 40 mm using argon. It can be observed  
 534 that large particles result in a relatively small charging power during the charging process; The

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535 discharging power is the lowest at  $d_p = 10\text{mm}$  during the entire discharging process which is  
 536 relatively stable. However, although the discharging power at  $d_p = 40\text{mm}$  is higher than that at  $d_p =$   
 537  $20\text{mm}$  during the first discharging hour, it then declines fast and drops below that at  $d_p = 20\text{ mm}$   
 538 during the following discharging hours. The influence of the particle diameter mainly includes two  
 539 aspects: large particles result in small pressure loss and also large thermal resistance in particles  
 540 and large delivery temperature variation.



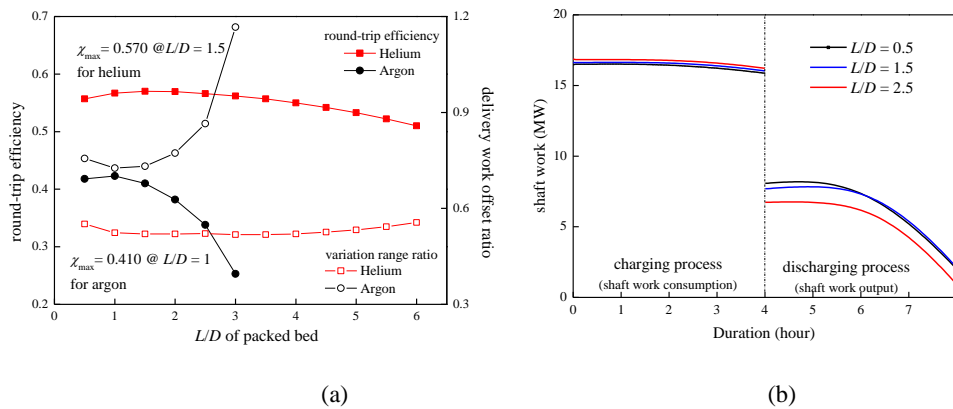
541  
542 (a) (b)  
543 Fig.13. Impact of particle diameter of compressor and expander

544 6.2.54.6 Effect of length-to-diameter ratio of reservoirs

545 As described in section 5, the volume of the designed HR and CR is  $460\text{ m}^3$  and  $740\text{ m}^3$ ,  
 546 respectively, for the 10 MW/4 h PHES system. For the cylindrical reservoirs with a fixed volume,  
 547 the length-to-diameter ratio  $L/D$  of the reservoirs is an important factor that influences the  
 548 pressure loss and heat transfer of the packed beds. Figure 14(a) shows the variation in the  
 549 round-trip efficiency  $\chi$  and the delivery working offset ratio  $\theta$  with the length-to-diameter ratio  
 550  $L/D$  of both the HR and CR, and the ranges of  $L/D$  are 0.5–3 for argon and 0.5–6 for helium. It can  
 551 be observed in figure 14(a) that the influence of  $L/D$  is rather gentle in the case of helium whereas  
 552 it is great in the case of argon. The round-trip efficiency  $\chi$  increases at the beginning and decreases

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553 gradually with the increase in  $L/D$ , and a maximum round-trip efficiency of 41.0% and a  
 554 minimum discharging power offset ratio of 72.6% occurs at  $L/D = 1$  for argon; for helium the  
 555 maximum round-trip efficiency is 57.0% and the minimum discharging power offset ratio of 51.8%  
 556 occurs at  $L/D = 1.5$ . This is because a larger length-to-diameter ratio  $L/D$  would result in a larger  
 557 pressure loss and a relatively smaller proportion of the thermozone region in the packed beds  
 558 simultaneously, which is also a joint effect. Figure 14(b) shows the transient charging and  
 559 discharging power under the conditions of the length-to-diameter ratio  $L/D$  of 0.5, 1.5, and 2.5  
 560 using argon. During the charging process, the larger length-to-diameter ratio  $L/D$  results in  
 561 relatively higher charging power owing to the higher pressure loss; the discharging power is the  
 562 lowest at  $L/D = 2.5$  during the discharging process. However, the discharging power at  $L/D = 0.5$   
 563 is higher than that at  $L/D = 1.5$  during the discharging, and then declines fast and drops below that  
 564 at  $L/D = 1.5$ .



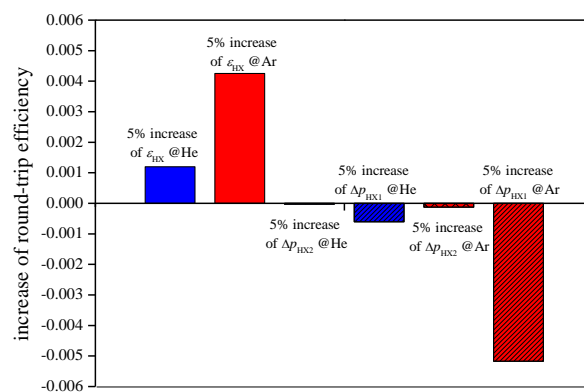
565 (a) 566 (b) 567 Fig.14. Impact of  $L/D$  of packed bed reservoirs

568 6.2.64.7 Effect of efficiency and pressure drop of heat exchangers

569 Figure 15 shows the round-trip efficiency variation of the PHES with a 5% increase in the  
 570 efficiency and pressure drop of the heat exchangers (including HX1 and HX2) based on the

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571 parameters listed in tables 1 and 2. It can be observed that the increase in the heat transfer  
 572 efficiency of the heat exchangers improves the round-trip efficiency whereas the increase in the  
 573 pressure loss decreases the round-trip efficiency; the effect of the heat exchangers efficiency and  
 574 pressure drop on the PHES efficiency using argon is several times higher than that of helium; and  
 575 the influence of the pressure loss of the low pressure heat exchanger (HX1) is more obvious than  
 576 that of the high pressure heat exchanger (HX2).



577

578 Fig.15. Impact of efficiency and pressure drop of heat exchangers

578

579 6.2.74.8 *Effect of discharging duration*

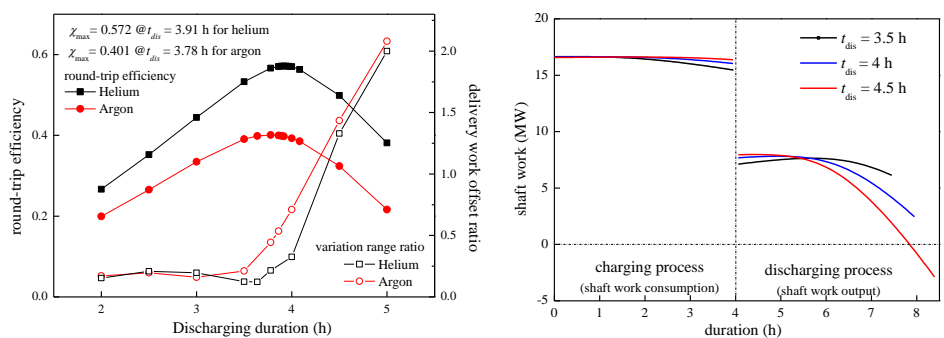
580 In the above analysis, each energy storage circle comprise a charging process of 4 h and a  
 581 discharging process of 4 h; however, an equal discharging and charging duration may not be  
 582 optimal for such a PHES system. Figure 16(a) shows the influence of the discharging time ranging  
 583 from 2 h to 5 h (one circle consists of a 4 h charging process and 2–5 h discharging process) on the  
 584 round-trip efficiency  $\chi$  and the delivery working offset ratio  $\theta$  using argon and helium, respectively.  
 585 From figure 15(a), it can be observed that the round-trip efficiency  $\chi$  increases at first and  
 586 then decreases with the discharging time. The best selection of the discharging duration is a few  
 587 minutes shorter than the charging time such that the maximum round-trip efficiency of 40.1%

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588 occurs at the delivery duration of 3.78 h for argon, and the maximum round-trip efficiency is 57.2%  
 589 at the delivery duration of 3.91 h for helium. The delivery working offset ratio  $\theta$  is relatively low  
 590 (<20%) for a discharging duration less than approximately 3.5 h and then increases sharply.

591 Figure 16(b) shows the transient shaft power during the charging and discharging with the  
 592 discharging duration of 3.5 h, 4 h and 4.5 h using argon. It can be observed that for the PHES  
 593 system having a 3.5 h discharging duration has the most stable delivery power, and the obvious  
 594 decline of the delivery power at the later stage of the discharging process can be observed with a  
 595 longer discharging duration. Figure 16(c) shows the axial temperature profile of the hot TES  
 596 reservoir at the end of the charging and discharging processes for the discharging durations of 3.5  
 597 h, 4 h and 4.5 h. It also shows that more exergy with a high temperature is stored in the hot TES  
 598 reservoir in the PHES system in the case of the discharging duration of 3.5 h, and a relatively  
 599 stable delivery thermal energy profile can be obtained during the discharging process, but it has  
 600 the drawback of relatively unstable charging power, which can be reduced through the heat  
 601 exchangers.

602

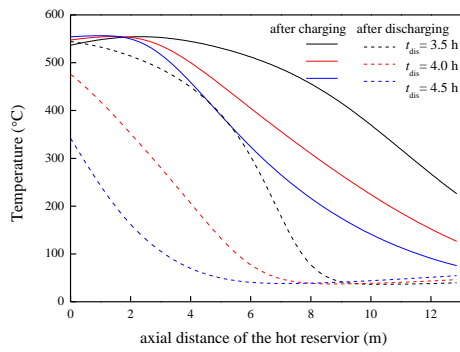


603

604

(a)

(b)



(c)

Fig.16. Impact of the discharging duration on the PHES behavior

## 7.5 Conclusions

In this paper, the use of the transient analysis method on the Joule–Brayton based PHES system is proposed for the coupling dynamics, thermodynamics and heat transfer process. The cyclic transient behavior of the 10 MW/4 h Joule–Brayton PHES system is studied using argon and helium as the working gases. Based on the round-trip efficiency and the variation range ratio of the delivery power, the mechanisms influencing PHES system and components parameters on the PHES system performance are further discussed. From the result of the analysis, the following conclusions can be obtained:

1. The delivery power clearly declines during the discharging process mainly owing to the thermal energy reduction from the packed bed TES reservoirs.
2. The gas resistance loss through the TES reservoirs and heat exchangers has a great influence on the system performance. In addition, helium, with small resistance losses, has an overwhelming advantage over argon for application in the PHES. The round-trip efficiency  $\chi$  of helium is 56.9%, which is much higher than 39.3%, which is obtained on using argon under the

622 design conditions. The PHES system using helium can also provide more stable electricity with  
623 the delivery power offset ratio of 45.9% than that using argon with a delivery power offset ratio of  
624 71.0%.

625 3. The increase in the pressure ratio and isentropic efficiencies would lead to an obviously  
626 improvement in the round-trip efficiency and delivery stability. Furthermore, an appropriate  
627 discharging compression ratio that is less than the charging compression ratio will aid in  
628 improving the round-trip efficiency. For the 10 MW/4 h PHES system, the optimum round-trip  
629 efficiency is obtained at the discharging compression ratio of 7 when the charging compression  
630 ratio is 10.

631 4. For the TES reservoirs, there exists optimal selections of particle sizes, ratios of length  
632 to-diameter, and discharging durations corresponding to the maximum round-trip efficiency and  
633 preferable discharging power stability; this is mainly owing to the joint effects of the pressure loss,  
634 heat transfer and thermodynamics.

635 Further research is required for improving the improvement of the round-trip efficiency and  
636 discharging power stability and decreasing the costs, which will be the subject of the authors'  
637 future research.

638

#### 639 **Conflict of Interest**

640 The authors declare no conflict of interest.

641

#### 642 **Acknowledgements**

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744

745 **Nomenclature**

746 *Abbreviations*

BOT	Bottoming system	747
BV	Buffer vessel	748
CAES	Compressed air energy storage	749
CHEST	Compressed heat energy storage	750
CR	Cold Reservoir	751
DSC	differential scanning calorimetry	752
EES	Electrical energy storage	753
HP	High pressure	754
HR	Hot reservoir	755
HX	Heat exchanger	756
LNG	Liquefied natural gas	757
LP	Low pressure	758
NIST	National Institute of Standards and Technology	
ORC	Organic Rankine cycle	761
PHS	Pumped hydro storage	762
PHES	Pumped heat electricity storage	763
PTES	Pumped thermal electricity storage	764
TEES	Thermo-electrical energy storage	765
TEMA	Tubular Exchanger Manufacturers Association	766 767
TES	Thermal energy storage	768 769

770 *Symbols*

<i>Bi</i>	Biot number
<i>C</i>	Specific heat capacity, J K <sup>-1</sup> kg <sup>-1</sup>
<i>d</i>	Diameter of particles, m
<i>D</i>	Diameter of packed bed reservoir, m
<i>e</i>	Specific energy, J kg <sup>-1</sup>
<i>G</i>	Mass flow rate, kg s <sup>-1</sup>
<i>h</i>	Volumetric heat transfer coefficient, W m <sup>-3</sup> K <sup>-1</sup>

$i$	Number $i$	771
$I$	<u>Moment of inertia, <math>\text{kg m}^2</math></u>	
$K$	Thermal conductivity, $\text{W m}^{-1} \text{K}^{-1}$	
$L$	Length scale of packed bed, m	
$m$	Mass of gas, kg	
$n$	Number	
$N$	Number of circles	
$P$	<u>Power, W</u>	
$Q$	<u>Volume flow rate, <math>\text{m}^3 \text{s}^{-1}</math></u>	
$Re$	Reynolds number	
$t$	Time, s	
$T$	Temperature, K	
$\beta$	Compression/expansion ratio of compressor/expander	
$\gamma$	Adiabatic exponent of gas	
$\varepsilon$	Efficiency of heat exchanger	
$\eta$	Polytropic efficiency of compressor/expander	
$\theta$	Offset ratio of delivery power	
$\kappa$	Parameter, $(\gamma-1)/\gamma$	
$\mu$	Dynamic viscosity, Pa s	
$\rho$	Density, $\text{kg m}^{-3}$	
$\Phi$	Porosity of packed bed	
$X$	Round-trip efficiency	
$\omega$	<u>Angular velocity, <math>\text{rad s}^{-1}</math></u>	
<i>Subscript</i>		
0	Point 0	
1	Point 1	
c	Compressor	
chr	Charge	
<u>des</u>	<u>Design</u>	
dis	Discharge	
e	Expander	
eff	Effective	
g	Gas	
HP	High pressure	
HX1	Heat exchanger 1	
HX2	Heat exchanger 2	
$i$	Number $i$	
in	At the inlet	
LP	Low pressure	
p	Particle	
s	Solid	
w	Water	

# Cyclic transient behavior of the Joule–Brayton based pumped heat electricity storage: Modeling and analysis

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## *Abstract*

Pumped heat electricity storage (PHES) has the advantages of a high energy density and high efficiency and is especially suitable for large-scale energy storage. The performance of PHES has attracted much attention which has been studied mostly based on steady thermodynamics, whereas the transient characteristic of the real energy storage process of PHES cannot be presented. In this paper, a transient analysis method for the PHES system coupling dynamics, heat transfer, and thermodynamics is proposed. Judging with the round trip efficiency and the stability of delivery power, the energy storage behavior of a 10 MW/4 h PHES system is studied with argon and helium as the working gas. The influencing factors such as the pressure ratio, polytropic efficiency, particle diameters, structure of thermal energy storage reservoirs are also analyzed. The results obtained indicate that, mainly owing to a small resistance loss, helium with a round-trip efficiency of 56.9% has an overwhelming advantage over argon with an efficiency of 39.3%. Furthermore,

1 21 the increases in the pressure ratio and isentropic efficiencies improve the energy storage  
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3 22 performance considerably. There also exist optimal values of the delivery compression ratio,  
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6 23 particle sizes, length-to-diameter ratios of the reservoirs, and discharging durations corresponding  
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9 24 to the maximum round-trip efficiency and preferable discharging power stability. The above  
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12 25 can provide a basis for the optimal design and operation of the Joule–Brayton based PHES.

13  
14 26 Key words: pumped heat electricity storage, pumped thermal electricity storage, Brayton, thermal  
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17 27 energy storage, heat storage, energy storage  
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## 21 28 *1 Introduction*

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24 29 The increase in energy consumption and the demand for decrease in carbon emission have  
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27 30 result in great changes in the global energy structure owing to which the proportion of renewable  
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30 31 energy usage has increased and that of fossil energy has gradually decreased [1]. From 2007 to  
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33 32 2017, the total renewable power capacity of non-hydropower renewables increased more than  
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36 33 six-fold (that of solar energy and wind energy increased 48-fold and six-fold respectively) [1, 2].  
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39 34 In particular in 2017, renewable power accounted for 70% of net additions to the global power  
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42 35 generation capacity and 26.5% of the global electricity production [1, 2]. However, the majority of  
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45 36 renewable energy resources have inherent intermittency and instability characteristics, which  
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48 37 results in the carryover of oscillation and unreliability to the power network. For example, 6%  
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51 38 photovoltaic power and 12% wind power was wasted in China in 2017 [3]. Electrical energy  
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54 39 Storage (EES) that converts electrical energy into another form of energy for storage and converts  
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57 40 it back to electrical energy when required, is considered as one of the most promising solutions for  
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60 41 increasing the penetration depth of renewable energy resources [4, 5]. Moreover, EES is an  
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1 42 essential link in the energy supply chain, which provides services such as load leveling, peaking  
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3 43 shaving, power quality improvement, and frequency regulation for the traditional power grid, thus  
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6 44 improving the security and utilization rate of the power grid [6-8].  
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9 45 Nowadays, there exist various energy storage technologies and different criteria for their  
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11 46 classification. Based on the form of energy storage in the system, the energy storage technologies  
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13 47 can be mainly categorized into five classes: chemical (hydrogen and synthetic natural gas),  
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15 48 electrical (capacitors and superconducting magnetic), electrochemical (classic batteries and flow  
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17 49 batteries), mechanical (flywheels, adiabatic compressed air, pumped heat electrical storage,  
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20 50 pumped hydro and cryogenic energy storage) and thermal (sensible heat, latent heat and  
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22 51 thermochemical heat) [4, 5]. Each EES technology has a suitable range of applications (e.g.  
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25 52 batteries, compressed air energy storage (CAES), and pumped hydro storage are suitable  
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27 53 candidates for peak shaving; flywheels, super-capacitors and superconducting magnetic energy  
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30 54 storage are suitable candidates for frequency regulation) depending on its advantages, drawbacks,  
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33 55 and scales [4, 9].  
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39 56 Among the available storage technologies, only pumped hydro storage (PHS) and CAES  
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41 57 are mature large-scale stand-alone electricity storage technologies that can be used to store power  
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43 58 greater than 100 MW under commercial operation [4, 5, 10]. PHS is the most mature EES  
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45 59 technology having a high capacity, long storage period, high efficiency and relatively low cost per  
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48 60 unit of energy. To date, there are more than 300 facilities with a total power of over 170 GW in  
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51 61 operation, which accounts for approximately 96% of the global energy storage capacity [4, 11].  
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54 62 The Bath County Pumped Storage Station in the USA is the largest PHS power station in the  
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57 63 world which has a generation capacity of 3 GW and a storage capacity of 11 h [12]. CAES is  
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1 64 another mature technology that is typically used for large scale energy storage. The operational  
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3 65 CAES units in the world are 290 MW/2 h CAES in Huntorf, Germany with an underground  
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6 66 storage cavern of approximately 310,000 m<sup>3</sup> and 110 MW/26 h CAES in McIntosh, Alabama,  
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9 67 USA, with a cavern of approximately 500,000 m<sup>3</sup> [4, 5, 13]. The main barriers for PHS and  
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12 68 CAES plants are similar, in that their construction requires appropriate geographical conditions for  
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15 69 the huge volume of storage.

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17 70 A category of novel energy storage technologies “pumped heat electricity storage (PHES)”  
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20 71 was proposed, which is also called “pumped thermal electricity storage (PTES)” and  
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23 72 “thermo-electrical energy storage (TEES)”. During the charging process of the energy storage,  
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26 73 heat is pumped from cold reservoirs (CRs) to hot reservoirs (HRs) via a heat pump circle and then  
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29 74 stored; during the discharging process electricity is generated by the stored thermal energy through  
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32 75 the heat-work conversion circle. Owing to the advantages of its high energy density and high  
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35 76 efficiency, PHES has captured the attention of researchers as a promising technology for  
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38 77 large-scale energy storage in recent years [14-31]. The categories of the PHES systems is mainly  
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41 78 based on two types of reversible heat-work conversion circles thus far: The Joule–Brayton cycles  
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44 79 [25-31] and the Rankine cycles [14-24].

45 80 The Rankine-cycle-based PHES system was first proposed by the ABB Company by the  
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48 81 name of TEES [14, 15]. It mainly includes the transcritical CO<sub>2</sub> Rankine cycle, organic Rankine  
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51 82 cycles (ORCs), and subcritical steam Rankine cycle. Morandin et al. studied a TEES system  
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54 83 based on a transcritical CO<sub>2</sub> Rankine cycle with hot-water thermal storage and ice-cold storage,  
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57 84 and then optimized the system with an achieved round-trip efficiency of 60% on using the pinch  
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60 85 analysis approach [16, 17]. Kim et al. then presented an isothermal TEES system based on the

1 86 transcritical CO<sub>2</sub> Rankine cycle wherein water was sprayed to cool/heat transcritical CO<sub>2</sub> directly,  
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3 87 and it was found that the expansion work and efficiency were improved via the isothermal  
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6 88 expansion owing to the high efficient heat transfer with the thermal storage tanks [18]. Abarr et al.  
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9 89 proposed the use of a PTES and bottoming system based on the transcritical ammonia cycle  
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11  
12 90 connected to a natural-gas peak plant and the obtained result indicates that the stand-alone energy  
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15 91 storage efficiencies is between 51%-66% with a stand-alone bottoming efficiency of 24% [19, 20].  
16  
17 92 Wang and Zhang proposed and analyzed a PHES based on the transcritical CO<sub>2</sub> heat pump cycle  
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20 93 during charging and the cascaded system of the transcritical CO<sub>2</sub> Rankine cycle and the subcritical  
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22  
23 94 NH<sub>3</sub> Rankine cycle utilizing liquid natural gas cold energy with a round-trip efficiency of up to  
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26 95 139% [21]. Steinmann developed the compressed heat energy storage (CHEST) concept based on  
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29 96 stream Rankine cycles combined with sensible and latent heat storage with an estimated round-trip  
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32 97 efficiency of 70% based on the isentropic efficiencies of 0.9 [22]. A PHES based on the ORC  
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35 98 system with the integration of low-temperature heat was also studied. Jockenhöfer et al. found that  
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38 99 the ORC-CHEST system could provide 1.25 times the net power with a heat resource temperature  
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41 100 of 100°C and a maximum exergetic efficiency of 0.59 [23]. Frate et al. studied a PHES system  
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44 101 comprising of a vapor-compression heat pump integrated with a low-grade heat source for  
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47 102 charging and an ORC system for discharging and found that the achievable round-trip efficiency  
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50 103 was 130% on using R1233zd at the heat source temperature of 110 °C and the isentropic  
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53 104 efficiency was 0.8 [24].

54 105 Using a single-phase gas as the working fluid, the Joule–Brayton-cycle based PHES  
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56 106 generally consists of cold (low-pressure) thermal energy storage (TES) reservoirs, hot  
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59 107 (high-pressure) TES reservoirs, and compressor–turbine-pairs, wherein the CRs and HRs are

1 108 generally comprise packed-bed solid thermal energy storage owing to its wide temperature range,  
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3 109 high efficiency, and small pressure loss. Desrues et al. presented a PHES system based on the  
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6 110 Joule–Brayton cycle consisting of two TES reservoirs connected by two compressor-turbine-pairs  
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9 111 and two heat exchangers comprising argon as the working gas and obtained an optimized  
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11 112 round-trip efficiency of 66.7% based on the turbo machines’ polytropic efficiency of 0.9 [25]. Ni  
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14 113 and Caram analyzed the influence of gas and pressure ratios etc. through a simulation and found  
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17 114 the efficiency of the turbomachinery to be the factor limiting the round-trip efficiency [26]. Howes  
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20 115 from the company Isentropic introduced three prototype of PTES and proposed a 2 MW PTES  
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22  
23 116 system with heat and cold thermal storage temperatures of 500 °C and -160 °C having a round-trip  
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26 117 efficiency of up to 72% [27]. White et al. found that the round-trip efficiency and energy storage  
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29 118 density increase with the temperature ratio between the hot and cold TES [28]. McTigue et al.  
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31 119 presented a PTES system based on the Joule–Brayton cycle with a buffer vessel and performed a  
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34 120 theoretical analysis on the PTES system coupled with a packed bed model of the HRs and CRs  
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37 121 [29]. Benato presented a Joule–Brayton PHES system with an electric heater settled after the  
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40 122 compressor in order to maintain the hot–tank temperature during charging, and the performance  
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43 123 and cost evaluation of such a system with different TES materials and different working gases was  
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45 124 analyzed [30,31].

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47 125 There are mainly three categories of TES technologies: sensible heat storage, latent heat  
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50 126 storage, and chemical heat storage [32]. Among the TES technologies, packed bed sensible TES  
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53 127 has been identified as the most suitable technology for the PHES system owing to its advantages  
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56 128 of low cost, small pressure loss, wide applicable temperature range, and large heat transfer surface  
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59 129 area that results in a small temperature difference, etc. [30].



1 130 The performance of a PHES comprising heat and cold packed-bed reservoirs of different  
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3 131 materials was analyzed in terms of the round-trip efficiency [25, 29, 30], energy density [30, 31],  
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6 132 and costs [30, 31]. However, there still exist defects in the published studies: (1) such a PHES  
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9 133 comprising heat and cold packed-bed reservoirs have strong unsteady characteristics whereas the  
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12 134 majority of the analyses on the PHES were performed using the stable thermodynamics method,  
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15 135 (2) it is not based on continuous cycles, and the initial state of each cycle is strong related to the  
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18 136 state at the end of last cycle for the continuous cycles, (3) it neglects the coupling effect of  
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21 137 dynamics, heat transfer and thermodynamics, (4) it involves the oversimplification of heat  
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24 138 exchangers, and (5) argon or air is used as the working fluid.

25 139 In this context, we make the first attempt to investigate the cyclic transient behavior of the  
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28 140 Joule–Brayton PHES system. Specifically, on a 10 MW/4 h PHES system, a transient analysis  
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31 141 method for the coupling of the dynamics, heat transfer and thermodynamics of the PHES system  
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34 142 with the components including the compressor, expander, TES reservoirs and heat exchangers is  
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37 143 proposed and solved numerically for multiple continuous cycles. The research presents a more  
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40 144 realistic behavior that is close to the real cyclic operations of the Joule–Brayton PHES, wherein  
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43 145 the working performance including both the round-trip efficiency and power attenuation during  
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46 146 discharging can be obtained. Helium is studied as a monoatomic molecular gas with a high energy  
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49 147 density that can be used as the working gas. This paper is thus focused on the influencing  
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52 148 mechanism of the parameters of the PHES system and the key components that are presented in  
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55 149 figure 1.  
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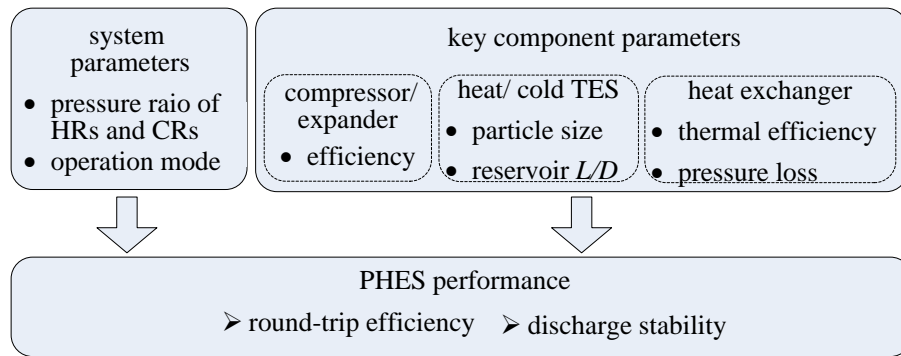


Fig. 1. Parameters influencing on PHEs performance

In the following, section 2 presents a detailed description of the Joule–Brayton based PHEs system, section 3 describes the coupling analysis method of the PHEs system and the components, and introduces the parameters design of the 10 MW/4 h PHEs system, section 4 presents the results and findings, and the last section concludes the paper.

## 2 Description of Joule–Brayton based PHEs system

Based on the PHEs system proposed by White et al. [28], and McTigue et al. [29], the Joule–Brayton PHEs discussed in this paper, as shown in figure 2, mainly consists of a cold (low–pressure) TES reservoir, a hot (high–pressure) TES reservoir, two compressor–turbine–pairs (one for charging and the other for discharging) and two heat exchangers. The heat exchangers are required to remove surplus heat from the PHEs system and stabilize the temperature variation in the packed–bed reservoirs during the charging process. A buffer vessel is also required to store/release gas in order to stabilize the system pressure during charging/discharging to balance the gas mass changes in the two reservoirs. During the charging and discharging processes, approximately 0.36% of the total flow rate of the gas is required to be

1 166 exported to the buffer vessel through position 1 in figure 2 to maintain the system under a constant  
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3 167 pressure. Furthermore, the same amount of gas returns the system through position 2 during the  
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6 168 discharging process. Moreover, a different pressure ratio of the compressor and expander during  
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9 169 the charging and discharging processes can be obtained by adjusting the buffer vessel, valves, and  
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12 170 a pressure adjustment compressor coordinately during the idle period.

13  
14 171 The working principal of the Joule–Brayton based PHES system is that during the charging  
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17 172 process, the working gas driven by the compressor (for charging) goes through the HR, heat  
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20 173 exchanger 2 (HX2), the expander (for charging), the CR and heat exchanger 1 (HX1) in the  
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23 174 indicated direction of charging. During the charging process, the system operates as a heat pump  
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26 175 wherein the heat is extracted from the CR to the HR while consuming electricity, and cold and  
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29 176 heat thermal energy are stored in the CR and HR respectively. During discharging, the system  
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32 177 operates as a heat engine with the working gas flowing along the indicated direction of discharge,  
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35 178 which is opposite to direction of charging, when the heat returns from the HR to the CR in order to  
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37 179 generate electricity.

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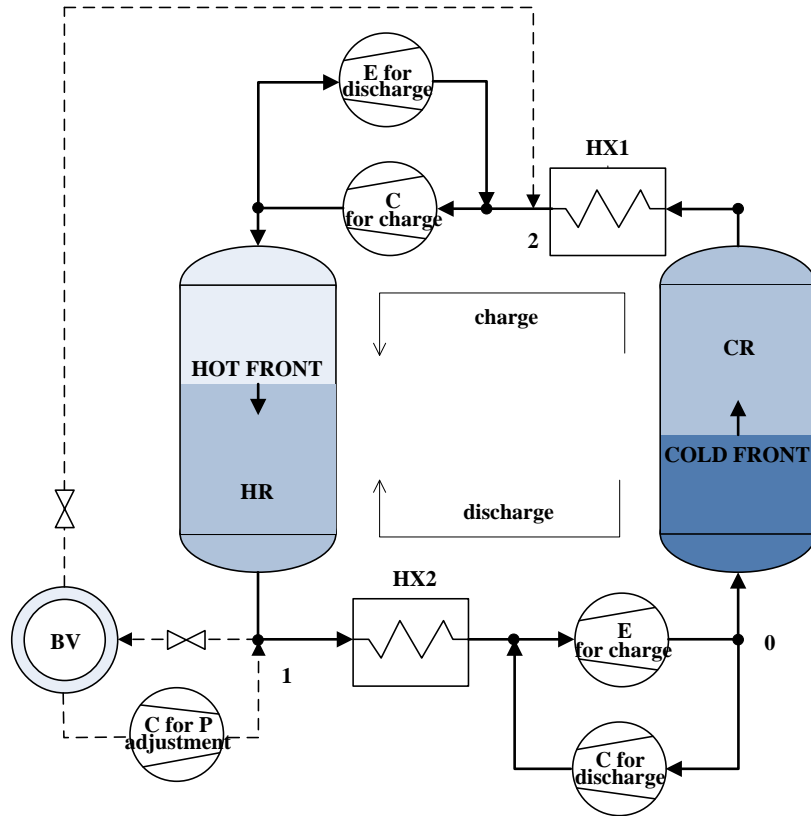


Fig.2. Layout of the PHES system. BV = buffer vessel; C = compressor; E = expander; HX = heat exchanger; CR = cold reservoir; HR = hot reservoir.

### 3 Methodology: coupling analysis of dynamics, transient heat transfer, and thermodynamics

*Dynamics:* In the PHES system, the compressor is the driving component of the gas flow, whereas the expander, the cold and hot storage reservoirs and the heat exchangers are the components that consume the mechanical energy of the gas during both the processes of charging and discharging. During the working process, the temperature profiles and thermophysical properties of the gas in the CR and HR are changing with time, thus resulting in a change in the pressure loss of the packed bed and leading to a pressure variation of the entire system. The pressure at point 1 during charging and at point 2 during discharging are maintained constant by the buffer vessel as shown in figure 3. *Heat transfer:* the transient temperature at the outflow of

193 the CR and HR solved using the unsteady mass and energy conservation equations of the packed  
 194 bed. *Thermodynamics*: For a fixed compression ratio of the compressor, the expansion ratio of the  
 195 expander changes with time owing to the variation in the components' pressure loss. Along with  
 196 the transient variation of the temperatures at the inlets and pressure ratios, the power and outflow  
 197 temperatures of the compressor and the expander changes are time-varying. *Thermal properties*:  
 198 The thermal properties of a gas, such as its density, thermal conductivity, and viscosity, have a  
 199 great influence on the system performance. Moreover, the properties of the gas are obtained from  
 200 the National Institute of Standards and Technology (NIST) database and updated in real-time  
 201 during the solution procedure. Therefore, a coupling analysis including dynamics, transient heat  
 202 transfer, thermodynamics and thermal properties is performed to obtain the transient behavior of  
 203 the PHES system as shown in figure 3.

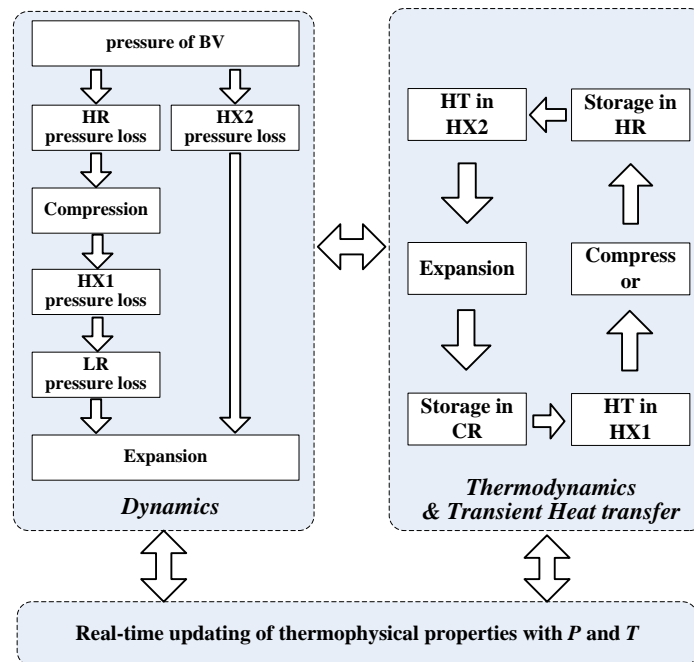


Fig.3. Coupling analysis of PHES during charging process

### 3.1 Dynamic conservation equation of PHES system

207 In the typically closed PHES system, the compressor provides the driving force of the  
 208 expander and the gas flow in the components including the HR and CR and heat exchangers  
 209 during both the charging and discharging processes. For the PHES system shown in figure 2, if we  
 210 suppose that the total pressure at position 0 is  $P_0$  during the charging and  $p_0'$  during the  
 211 discharging respectively, we obtain:

$$(p_0 - \Delta p_{LP} - \Delta p_{HX1}) \beta_c - \Delta p_{HP} - \Delta p_{HX2} - p_0 \beta_e = 0 \quad (1)$$

during the charging process and

$$p_0' \beta_c' - \Delta p_{HX2}' - \Delta p_{HP}' - p_0' \beta_e' = 0 \quad (2)$$

during the discharging process, wherein the superscript ' denotes the discharging process.  $\Delta p$   
 indicates the total pressure loss at each component, and  $\beta_c$  and  $\beta_e$  are the compression ratio and  
 expansion ratio respectively.

### 3.2 Thermodynamics of PHES system

#### 3.2.1 Compressor and expander

Taking into account the irreversibility loss of turbomachines, the polytropic process of  
 compression and expansion occurs with the polytropic efficiencies  $\eta_c$  and  $\eta_e$  respectively. For the  
 compressor

$$T_{c,out} / T_{c,in} = \beta_c^{\kappa / \eta_c} \quad (3)$$

For the expander

$$T_{e,out} / T_{e,in} = \beta_e^{-\kappa \eta_e} \quad (4)$$

where the parameter  $\kappa$  is defined as  $\kappa = (\gamma - 1) / \gamma$  and  $\gamma$  is the specific heat ratio ( $c_p / c_v$ ) of the gas  
 [25, 33].

During the charging and discharging process, temperatures and densities of the HR and CR

1 229 outflow gas transiently vary leading to the variation of volume flow rates and rotation rates in the  
 2  
 3 230 compressor and the expander. The unsteady variation of the turbo-machines shaft power  $P(t)$   
 4  
 5  
 6 231 owing to the inertia of rotors can be calculated by:

$$9 \quad 232 \quad P(t) = -I \cdot \omega(t) \frac{d\omega(t)}{dt} \quad (5)$$

11  
 12 233 Where  $I$  is the moment of inertia of rotor and  $\omega(t)$  is the angular velocity. The angular velocity  
 13  
 14 234 is proportional to the volume flow rate  $Q(t)$  and inversely proportional to the gas density at  
 15  
 16 235 the constant mass flow rate.

$$17 \quad 236 \quad \omega(t) = \frac{\omega_{des}}{Q_{des}} Q(t) = \frac{\omega_{des} \rho_{des}}{\rho(t)} \quad (6)$$

18  
 19 237 Where  $\omega_{des}$  and  $Q_{des}$  are the angular velocity and the volume flow rate under the design condition,  
 20  
 21 238 respectively.

### 22 239 *3.2.2 Packed bed heat/cold thermal energy storage reservoirs*

23  
 24  
 25 240 The domains of the hot and cold thermal energy storage reservoirs are considered as  
 26  
 27 241 cylindrical tanks, which include the packed bed of the TES particles and the heat transfer gas  
 28  
 29 242 flowing through the void space. On assuming that the flow pattern is a 1D Newtonian plug flow,  
 30  
 31 243 neglecting the temperature gradient in the radial direction and neglecting the heat loss through the  
 32  
 33 244 well-insulated wall, the governing energy conservation equations of the unsteady two-phase model  
 34  
 35 245 of such packed beds is given as follows.

36 246 For the fluid phase,

$$37 \quad 247 \quad \varphi \frac{\partial \rho_g}{\partial t} + \frac{\partial G}{\partial x} = 0 \quad (7)$$

$$\frac{\partial T_g}{\partial t} + \frac{G}{\rho_g \varphi} \frac{\partial T_g}{\partial x} = \frac{h_v}{\rho_g c_{p,g} \varphi} (T_s - T_g) \quad (8)$$

For the solid phase,

$$\frac{\partial T_s}{\partial t} = \frac{h_{v,\text{eff}}}{\rho_s c_s (1-\varphi)} (T_g - T_s) + \frac{k_{s,\text{eff}}}{\rho_s c_s (1-\varphi)} \frac{\partial^2 T}{\partial x^2} \quad (9)$$

where  $h_{v,\text{eff}}$  is the effective volumetric heat transfer coefficient on considering the internal heat conduction resistance in a solid (for a Biot number smaller than 100) having the relationship with the volumetric heat transfer coefficient  $h_v = h_p 6(1-\varphi)/d$ . The volumetric heat transfer coefficient of Chandra's equation is used which fits well with the experimental results under both low and high pressures [35, 36]

$$h_{v,\text{eff}} = \begin{cases} h_v & \text{for } Bi \leq 0.1 \\ \frac{1}{\frac{1}{h_v} + \frac{d_p^2}{60k_s(1-\varphi)}} & \text{for } 0.1 < Bi \leq 100 \end{cases} \quad (10)$$

$$h_v = 1.45 \frac{Re^{0.7} k_g}{d^2} \quad (11)$$

where the characteristic length for the Biot number is  $d_p/6$  [37].

$$Bi = \frac{h_p d_p}{6k_s} \quad (12)$$

$k_{s,\text{eff}}$  is the effective thermal conductivity for the non-contiguous spherical particles in a dispersion medium given by [38, 39]:

$$\frac{k_s - k_{s,\text{eff}}}{k_s - k_g} \left( \frac{k_{s,\text{eff}}}{k_g} \right)^{\frac{1}{3}} = \varphi \quad (13)$$

which is solved by performing iteration.

The dramatic temperature changes dramatically in the packed beds would lead to a change in the volume flow rate and thermoproperty of the gas in the packed bed. In this paper, the packed bed is divided into  $n$  sections along the axis, and the pressure drop across the packed bed and each



267

section are given by the Ergun equation shown as below [34].

$$\Delta p(i) = \frac{\Delta L \cdot G^2}{\rho(i) \cdot d} \left( 1.75 \frac{1-\phi}{\phi^3} + 150 \frac{1-\phi}{\phi^3} \frac{\mu(i)}{Gd} \right) \quad (14)$$

$$\Delta p = \sum_{i=1}^n \Delta p(i) \quad (15)$$

270

where  $\Delta p$  and  $\Delta p(i)$  are the pressure drop across the packed bed and the pressure drop across

271

the  $i_{th}$  section, respectively, and  $\Delta L$  ( $\Delta L = L/n$ ) is the length of each section.

### 272 3.2.3 Heat exchanger

273 In the PHES system, the heat exchangers play important roles including removing the surplus  
274 heat and stabilizing the temperature fluctuations from the HR and CR during the charging process.

275 Water from the cooling towers is usually selected as an efficient cooling media for heat  
276 exchangers having a temperature approximately about 2–5° C higher than the ambient temperature.

277 As the heat capacity of the cooling water is greater than that of the gas and on ignoring the  
278 influence of the heat exchanger heat capacity, the outflow temperature from the heat exchanger  
279 can be obtained as follows.

$$T_{g,o}(t) = T_{g,i}(t) - \varepsilon \frac{\dot{m}_g c_{p,g}}{\dot{m}_w c_{p,w}} (T_{g,i}(t) - T_{w,i}) \quad (16)$$

281 where  $\dot{m}$  and  $c_p$  are the mass flow rate and heat capacity, and  $\varepsilon$  is the heat exchanger  
282 effectiveness.

### 283 3.3 Systemic analyses of PHES system

284 For the PHES system, the transient specific energy performed during charging and delivered  
285 during discharging, with considering the unsteadiness of the compressor and expander, can be  
286 obtained using equation (17) and equation (18), respectively.

$$e_{\text{chr}}(t) = e_{\text{c,chr}}(t) - e_{\text{e,chr}}(t) + \frac{1}{\dot{m}c_p} (P_e(t) + P_c(t)) \quad (17)$$

$$e_{\text{dis}}(t) = e_{\text{e,dis}}(t) - e_{\text{c,dis}}(t) - \frac{1}{\dot{m}c_p} (P_e(t) + P_c(t)) \quad (18)$$

As shown in equation (5), the moment of inertia of the compressor and the expander are needed for calculating  $P(t)$ , whereas there is no available compressor and expander for the 10MW PTES system. In this study, referring to the compressor and the expander in the 10MW Advanced compressed air energy storage, the moment of inertia of compressor and the expander rotor is taken 1800 kgm<sup>2</sup> at the rated speed of 1500 rpm [42, 43]. Under the situations in this study, the maximum absolute value of angular acceleration of the expander rotor and the compressor rotor is 0.0063 rad/s<sup>2</sup> and 0.0026 rad/s<sup>2</sup> respectively, and the corresponding  $P_e(t)$  and  $P_c(t)$  is -3.47 kW and 0.36 kW, which are less than  $\pm 0.04\%$  of the transient shaft power and can be neglected.

By bringing equation (3), (4) into equation (15), (16), and neglecting the unsteady variation of the turbine machines, the transient specific energy can be calculated as below:

For the charging process,

$$e_{\text{chr}}(t) = T_{\text{c,in}}(t) \cdot (r_c(t)^{\kappa/\eta_c} - 1) - T_{\text{e,in}}(t) \cdot (1 - r_e(t)^{-\kappa\eta_c}) \quad (19)$$

For the discharging process,

$$e_{\text{dis}}(t) = T_{\text{e,in}}'(t) \cdot (1 - r_e'(t)^{-\kappa\eta_c}) - T_{\text{c,in}}'(t) \cdot (r_c'(t)^{\kappa/\eta_c} - 1) \quad (20)$$

Where  $e_{\text{chr}}$  and  $e_{\text{dis}}$  are specific energy (J/kg) of shaft work during charging and discharging,  $T_{\text{c,in}}$  and  $T_{\text{e,in}}$  are the inflow temperatures (K) of the compressor and the expander during charging, and the superscript ' denotes the discharging process.

308 On assuming no mechanical loss, the round-trip coefficient of the PHES system is obtained  
 309 on using the quotient of the net delivered shaft work during the discharging process and the  
 310 consumed shaft work during the charging process, as shown in equation (21)

$$\chi = \frac{\text{net work output}}{\text{net work input}} = \frac{\int_{\text{dis}} \dot{m}_{\text{dis}} c_p e_{\text{dis}}(t) dt}{\int_{\text{chr}} \dot{m}_{\text{chr}} c_p e_{\text{chr}}(t) dt} \quad (21)$$

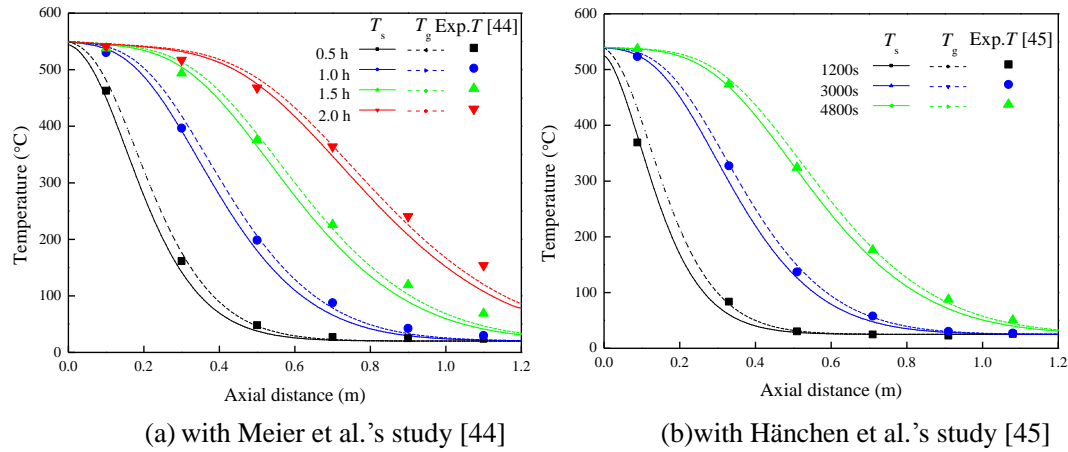
312 where  $\dot{m}$  is the mass flow rate through the compressors and expanders.

313 The stability of the delivery power is another important factor affecting for the energy  
 314 storage system. In this paper, the offset ratio of the delivery power is increased to evaluate the  
 315 stability which is defined as the ratio of the offset range of the delivery power to the maximum  
 316 value during the delivery period, as presented in equation (22).

$$\theta = \frac{\text{Max}(e_{\text{dis}}(t)) - \text{Min}(e_{\text{dis}}(t))}{\text{Max}(e_{\text{dis}}(t))} \quad (22)$$

318 For the PHES system, a smaller offset ratio indicates a more stable delivery power  
 319 during the discharging process.

320 In order to validate the transient equation of the packed beds, the numerical simulations of the  
 321 TES process of the crushed steatite (magnesium silicate rock) packed beds are performed by  
 322 solving equations (7)–(13) with the parameters used in reference [44] and [45].  
 323 The temperature dependence of the heat capacity of the crushed steatite ( $\text{Mg}_3\text{Si}_4\text{O}_{10}(\text{OH})_2$ ) is  
 324 taken in to consideration in the simulation [40]. The temperature profiles along the axial distance  
 325 of the packed beds of the simulated and experimental results are shown in figures 4 (a) and 4(b); it  
 326 can be observed that an obvious thermocline occurs during the charging process and the simulated  
 327 profiles fit well with the experimental results which proves the accuracy of the simulation method  
 328 [42, 43].



329  
330 (a) with Meier et al.'s study [44] (b) with Hänchen et al.'s study [45]  
331 Fig.4. Comparison between the simulation and experimental results of the temperature  
332 profiles in the packed beds

### 333 3.4 Parameters design of the 10 MW/4 h PHES system

334 In this paper, a Joule–Brayton based PHES system of 10 MW (nominally discharging  
335 power 10 MW, 4 h charging, and 4 h discharging) was designed and analyzed. The designed  
336 parameters of the PHES system with either argon or helium as the working gas are shown in Table  
337 1 wherein the pressure ratio is 10 as in McTigue et al.'s study [29]. It should be noted that the heat  
338 capacity of helium is almost ten times that of argon, and thus, the mass flow rate of helium is  
339 approximately only 1/10th that of argon in a PHES system of the same power. Therefore, the  
340 pressure loss in the heat exchangers and packed-bed reservoirs would be decreased greatly on  
341 using helium instead of argon.

342 Table 1 Designed parameters of PHES system of 10 MW discharging power

Working gas	HP Pressure (MPa)	LP Pressure (MPa)	Average $c_{p,g}$ (J/kg/K)	Mass flow rate (kg/s)	Polytropic efficiency	$\varepsilon$ of HXs	$\Delta p$ of HP HXs (kPa)	$\Delta p$ of LP HXs (kPa)	Cooling water temperature (K)
Argon	1.05	0.105	525	85.1	0.9	0.9	3	20	300
Helium	1.05	0.105	5193	8.6	0.9	0.9	0.3	2	300

343  
344 The designed 10 MW/4 h PHES system consists of an HR and a CR with a packed bed of

345 basalt particles. The packed-bed TES is unstable and has a larger packed bed volume, which  
 346 results in a more stable output temperature but a higher cost and lower energy storage density. In  
 347 consideration of the thermal front volume, the designed volumes of the HR and CR are selected to  
 348 be twice the minimum design volume obtained using from the energy balance method  
 349  $V = 2Q / (\overline{\rho_s c_s} \Delta T)$ . The detailed parameters of the HR and CR are shown in table 2. In this design,  
 350 the basalt is chosen as the hot and cold TES material, as it has a good heat capacity and thermal  
 351 stability within the temperature range of -196° C–800° C. Based on the TA Q2000 DSC, the heat  
 352 capacity of basalt is found to be strongly dependent on the temperature as shown in figure 5, and  
 353 the linear fit equation is given in equation (23).

$$c_p(T) = 0.23 + 0.00201 \cdot T \quad (23)$$

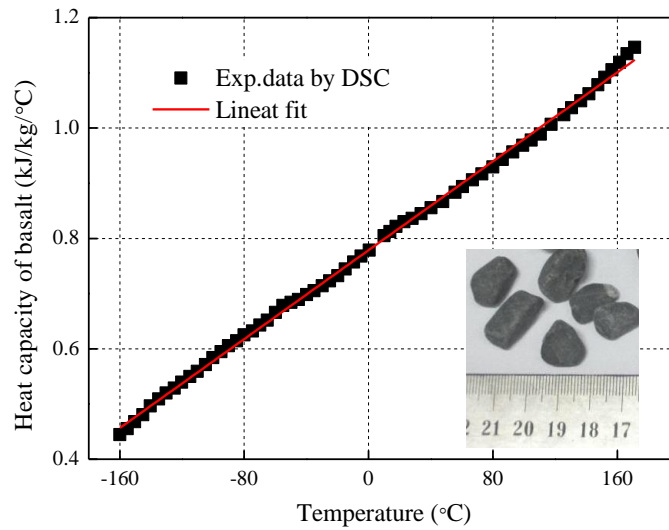


Fig.5. Dependence of heat capacity of basalt with temperature measured using DSC

Table 2 Hot and cold reservoir details for 10 MW/4 h PHES system  
(the total volume is twice the minimum design volume)

Reservoir	Pressure (MPa)	Density of solid material (kg/m <sup>3</sup> )	Porosity	Average $d_p$ (mm)	Total Volume (m <sup>3</sup> )	$L$ (m)	$D$ (m)

Heat	1.05	5175	0.35	30	460	10.96	7.31
Cold	0.105	5175	0.35	30	740	12.86	8.56

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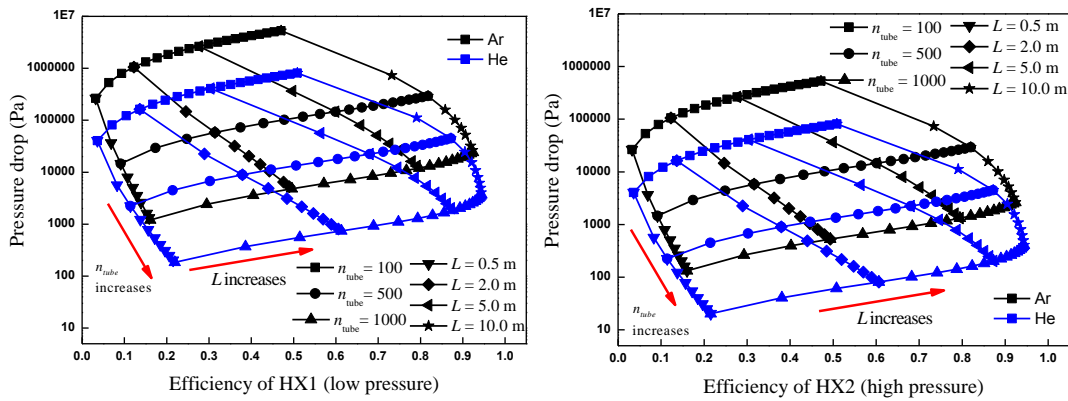
### 362 3.4.1 Heat exchangers design

363 For eliminating surplus heat and stabilizing the temperature variation, two heat exchangers  
364 are required for the Joule–Brayton cycle PHES. One heat exchanger is under low pressure and the  
365 other is under medium/high pressure, and such heat exchangers are required to be compatible with  
366 a wide range of operation conditions, high efficiency and low pressure loss wherein the  
367 shell-and-tube heat exchangers are the optimal choices. According to the working conditions of  
368 the PHES system, the one shell pass, two tube pass TEMA shell-and-tube heat exchangers were  
369 designed for the hot and cold heat exchangers using the  $\varepsilon$ - $NTU$  method and an empirical relation  
370 [41], wherein the heat transfer tubes have an outer diameter of 32mm and thickness of 2 mm, and  
371 the working gas passes through the shell side to minimize the pressure loss of the gas side.

372 Figure 6 shows the variation of the heat transfer efficiency and pressure drop of HX1 (low  
373 pressure) and HX2 (high pressure) with the tube number and tube length on using argon and  
374 helium respectively. The heat-transfer tube number ranges from 100 to 1000, and the tube length  
375 ranges from 0.5 m to 10.0 m. It can be found that an increase in the number of tubes would  
376 obviously decrease the pressure loss and improve the efficiency, and an increase in the tube length  
377 would lead to an increase in the efficiency and pressure loss. In order to obtain a high round–trip  
378 efficiency, the PHES system requires heat exchangers with a small pressure loss and high  
379 efficiency which can be obtained by using a large number of long tubes but this amount and length  
380 cannot be increased beyond a certain limit owing to the prohibitive cost.

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381 From figure 6, it can be found that for heat exchangers of the same size, the efficiencies are  
 382 similar when using argon and helium, but the pressure drop observed when using helium is only  
 383 approximately 1/10th the pressure drop observed when using argon owing to the difference in the  
 384 mass flow rate. Furthermore, the pressure drop of HX1 under a low pressure is several times  
 385 higher than the pressure drop of HX2 under a high pressure because of the high volume flow rate  
 386 under the low pressure. From the design of the PHES system, the heat exchangers with an  
 387 efficiency of 0.9, the pressure loss of HX1 of 20 kPa and pressure loss of HX2 of 3 kPa on using  
 388 argon, and the heat exchangers with an efficiency of 0.9, pressure loss of HX1 of 2 kPa and  
 389 pressure loss of HX2 of 0.3 kPa on using helium are achieved and such parameters are selected in  
 390 the 10 MW/4 h PHES system.



391  
392 Fig.6. Efficiency versus pressure drop of the shell-and-tube heat exchangers  
393

394 *4 Result and Discussion*

395 *4.1 Cyclic behavior of PHES system*

396 Based on the standard parameters in table 1 and 2, and the modeling method described in  
 397 section 3, the working behavior of the PHES system running 100 circles was simulated using  
 398 argon as the working gas; each cycle included 4 h of charging and 4 h of discharging. The axial

399 temperature profile of the HR and CR at the end of the charging and discharging processes from  
 400 the 1<sup>st</sup> circle to the 100<sup>th</sup> circle are shown in figures 7(a) and 7(b), respectively. It can be observed  
 401 that, the profiles at the end of the charging and discharging process tend to coincide after several  
 402 cycles. The temperature profiles in the reservoirs can be roughly divided into a stable temperature  
 403 region and a thermocline region wherein the temperature gradient in the thermocline region  
 404 decreases gradually with the cycling.

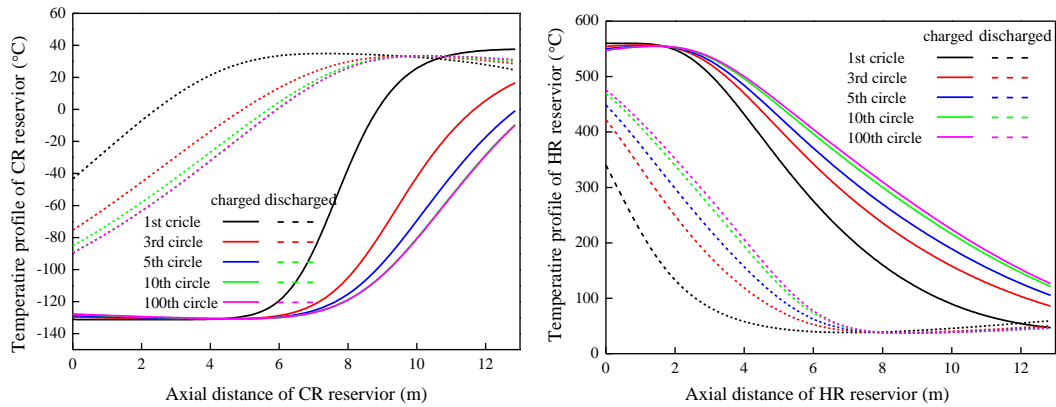
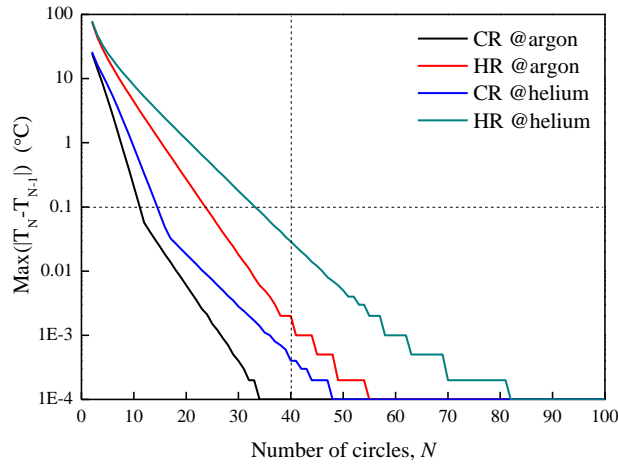


Fig.7. Cyclic behaviors of the HR and CR

407 In order to study the cyclic convergence of the PHES system, the factor  $\Delta T_{Max}(N)$   
 408 indicates the maximum temperature difference between the adjacent circles at the same axial  
 409 position and is defined as shown in the equation (22). As shown in figure 8, the factor  $\Delta T_{Max}(N)$   
 410 declines exponential with the circle number where argon has a higher decline rate than helium.  
 411 After 40 circles, the maximum temperature difference at the same axial position between the  
 412 adjacent circles is below 0.1 °C for all the gases and reservoirs which is deemed cyclically stable.  
 413 According to this, the following analysis is based on the data of the 40th circles which have  
 414 achieved the cyclic stable state.

$$\Delta T_{Max}(N) = Ma \left( \frac{1}{N} T_i, \bar{N} T \right) \quad N=1, 2, 3, \dots \quad (24)$$

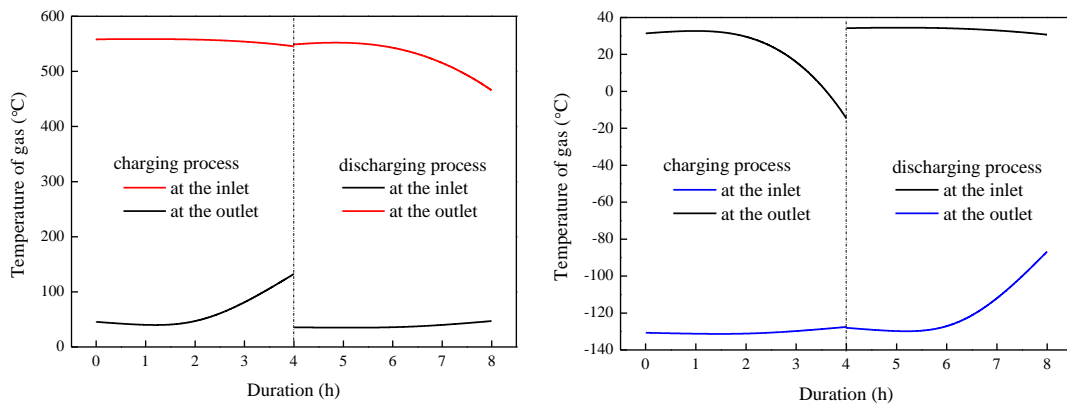




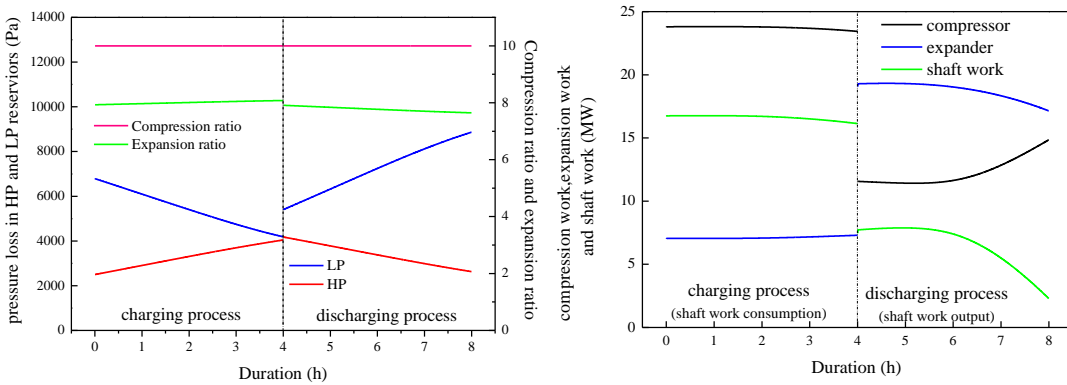
416  
417 Fig.8. Maximum temperature differences between circles versus the number of circles

418 Under the cyclic stable state, figures 9(a) and 9(b) show the transient variation of the inflow  
419 and outflow temperatures of the HR and CR during the charging and discharging, respectively,  
420 when using argon as the working gas. This shows that the outflow temperature from the HR  
421 increases continuously after a period of stable state (approximately 1.5 h) during the charging  
422 process and decreases continuously after a period of stable state (approximately 1.5 h) during the  
423 discharging. The outflow temperature from the CR also has a similar unstable behavior but the  
424 temperature variation trend is opposite to that of the HR. Figure 9(c) shows the variation in the  
425 pressure loss of the HR and CR during the charging and discharging processes. It can be found  
426 that the pressure loss of the CR decreases linearly during the charging and increases during the  
427 discharging process, and the opposite phenomenon is observed in the case of the HR. This is  
428 because, during the charging period in the CR, the cold region grows gradually where the volume  
429 flow rate decreases owing to the high density which results in a decrease in the pressure loss, and  
430 during the discharging, the cold region retracts gradually and the pressure loss increases gradually.  
431 For similar reasons, the increase in the hot region in the HR could lead to a higher volume flow  
432 rate, hence increasing the pressure loss during the charging. The expansion ratio increases slightly

433 during the charging and decreases during the discharging, as shown in figure 8(c), and is mainly  
 434 influenced by variations in the pressure loss of the reservoirs. Figure 8(d) shows that the powers of  
 435 the PHES compressor, expander and shaft are rather stable during the charging process, and during  
 436 the delivery process, the compressor power increases and the expander power decreases gradually,  
 437 thus leading to a decrease in shaft power. Based on the parameters listed in tables 1 and 2, the  
 438 round-trip efficiency  $\chi$  and the delivery working offset ratio  $\theta$  using argon as the working gas is  
 439 39.3% and 71.0%, respectively, and the round-trip efficiency  $\chi$  and delivery working offset ratio  $\theta$   
 440 using helium is 56.9% and 45.9%, respectively.



442 (a) inflow and outflow temperature of HP reservoir (b) inflow and outflow temperature of LP reservoir



444 (c) pressure loss of the HP and LP reservoirs (d) transient power variation of PHES

446 Fig.9. Transient behaviors of the HR and CR and PHES system.

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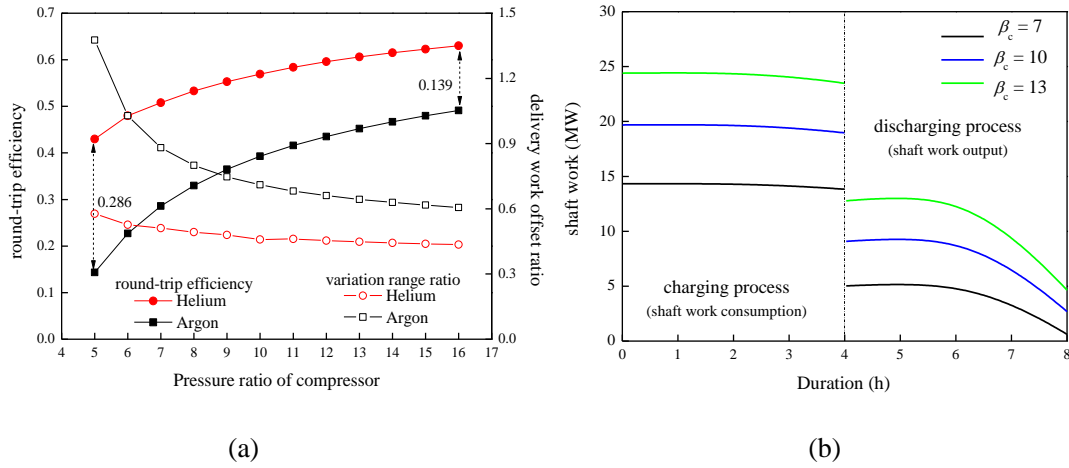
#### 447 4.2 Effect of compression ratio during charging and discharging

448 The influencing factors include the compression ratio in the discharging process only and that  
449 for the entire processes, the polytropic efficiency of compressors and expanders, the particle  
450 diameter of the particles in the reservoirs, the length-to-diameter ratio of the reservoirs, the  
451 efficiency and pressure loss in the heat exchangers and the discharging duration of the PHES  
452 system performance are studied using argon and helium as the working gases.

453 Figure 10(a) shows the influence of the compression ratio of the compressors ranging from 5  
454 to 16 during both charging and discharging processes on the round-trip efficiency  $\chi$  and the  
455 delivery working offset ratio  $\theta$  wherein the other parameters are obtained from in tables 1 and 2. It  
456 can be found that the round-trip efficiency increases gradually with the compression ratio  $\beta_c$  from  
457 14.3% at  $\beta_c = 5$  to 49.1% at  $\beta_c = 16$  for argon and from 43.0% at  $\beta_c = 5$  to 63.0% at  $\beta_c = 16$  for  
458 helium; the round-trip efficiency of helium is considerably higher than that of argon, with a range  
459 of 13.9% to 28.6%. This is mainly because a much smaller pressure loss occurs in the reservoirs  
460 and heat exchangers of helium than those of argon, and a greater expansion work can be obtained  
461 on using helium. From figure 10(a), it can also be observed that the delivery working offset ratio  $\theta$   
462 decreases with the compression ratio  $\beta_c$ , and the offset ratio  $\theta$  of helium is much lower than that of  
463 argon; such a result indicates that the delivery work during the discharging using helium is more  
464 stable than that using argon. The transient charging power and delivery power profiles at the  
465 compression ratio  $\beta_c$  of 7, 10 and 13 on using argon are shown in figure 10(b). It can be found that  
466 both the charging power and discharging power increase with the compressor ratio and an obvious  
467 decrease in delivery power occurs during the late discharging period.

468 P erilhon et al. recommended that the maximum fluid temperature should not exceed 800  C

469 for a reasonable life of the turbomachines [46]. The maximum temperature of the gas is  
 470 approximately 750 °C in the PHES system at the compression ratio  $\beta_c$  of 16 for both argon and  
 471 helium, which is within the permitted temperature range.



472  
473  
474 Fig.10. Impact of compression ratio during both charging and discharging

475 *4.3 Effect of compressor pressure ratio during discharging*

476 Owing to the pressure loss, heat transfer loss and the irreversible loss of the compressor and  
 477 expanders, setting the pressure ratio of the compressor during discharging as the same as that of  
 478 during charging may not be the best choice. After the charging process, the compression ratio of  
 479 the delivery process can be reset by storing some gas in the BV and recharging the system by the  
 480 adjustment compressor during the idle time. At the charging compression ratio of 10 and the other  
 481 parameters listed in tables 1 and 2, figure 11(a) shows the influence of the compression ratio  
 482 ranging from 4 to 10 during the discharging process on the round-trip efficiency  $\chi$  and the delivery  
 483 working offset ratio  $\theta$ . This result indicates that the round-trip efficiency  $\chi$  increased  
 484 first and then decreased with the discharging compress ratio and the maximum round-trip  
 485 efficiency  $\chi$  occurs at the discharging compress ratio of 7 for both argon and helium, the maximum  
 486 round-trip efficiency  $\chi$  obtained using helium is 59.0%, which is considerably higher than that

487 obtained using argon: 41.7%. Moreover, it is also indicated from figure 11(a) that the offset ratio  $\theta$   
 488 using helium and argon increases gradually with the increase in the discharging compress ratio. As  
 489 shown in figure 11(b), when the charging compression ratio  $\beta_{c,chr}$  is 10, the discharging  
 490 compression power and discharging expansion power at a high pressure ratio of 10 are both higher  
 491 than those at a low pressure ratio of 7. The shaft power at a compression ratio of 10 is lower than  
 492 that at a compression ratio of 7; this is because, the variation amplitude of the compression power  
 493 is greater than that of the expansion power when the discharging compression ratio increases from  
 494 7 to 10.

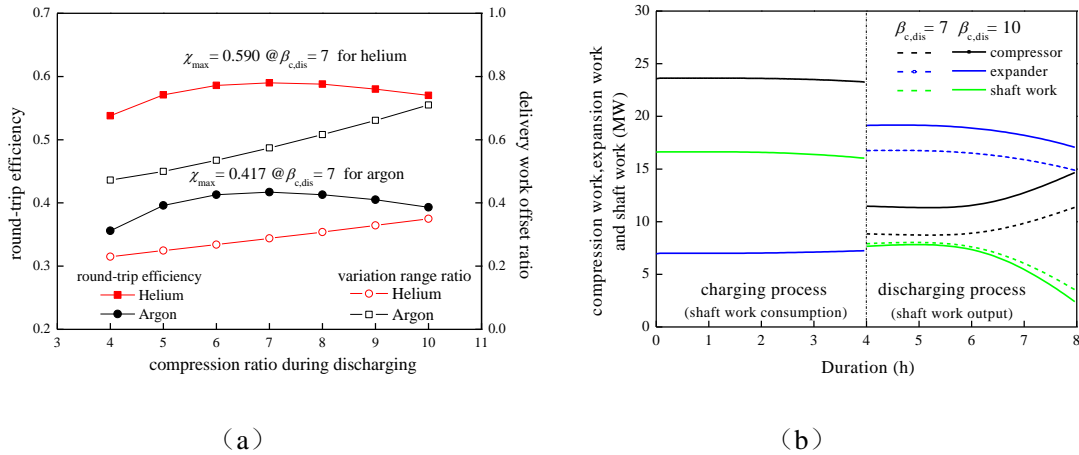


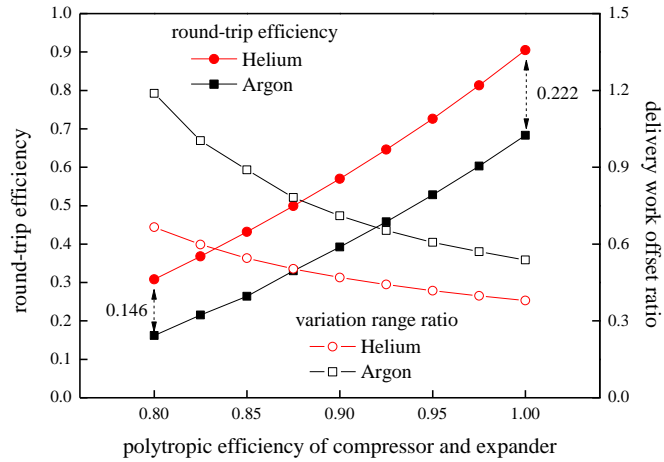
Fig.11. Impact of compression ratio during discharging (at  $\beta_{c,chr} = 10$ )

#### 4.4 Effect of polytropic efficiency of both compressors and expanders

499 The plots of the round-trip efficiency  $\chi$  with the polytropic efficiency of both the compressors  
 500 and expanders ranging from 0.8 to 1.0 during both charging and discharging are shown in figure  
 501 12, which the use of argon and helium respectively, and the other parameters are obtained from  
 502 tables 1 and 2. It can be observed that the polytropic efficiency of the compressors and expanders  
 503 have an almost dominant effect on the round-trip efficiency  $\chi$ , such that the round-trip efficiency  
 504 increases from 16.2% at  $\eta = 0.8$  to 68.3% at  $\eta = 1.0$  when using argon, while the round-trip

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505 efficiency increases from 30.8% at  $\eta = 0.8$  to 90.5% at  $\eta = 1.0$  on using helium. The delivery  
 506 working offset ratio  $\theta$  in figure 11 shows that the increase in the polytropic efficiency also  
 507 improves the stability of the delivery power.



508

509 Fig.12. Impact of polytropic efficiency of compressor and expander

510 *4.5 Effect of TES particles diameter*

511 The diameters of the solid TES particles would affect the pressure loss and heat transfer in  
 512 the packed beds and, hence, affect the PHES efficiency. Figure 13(a) shows the influence of the  
 513 particle size in both the HR and CR in the range from 5mm to 70mm on the round-trip efficiency  $\chi$   
 514 and the delivery working offset ratio  $\theta$ . It can be observed that, the round-trip efficiency  $\chi$  first  
 515 increases and then gradually decreases with the particles sizes, the maximum round-trip efficiency  
 516 of 40.2% occurs at  $d_p = 20$  mm for argon and for helium the maximum round-trip efficiency of  
 517 58.8% is obtained at  $d_p = 15$  mm, and such particle sizes always correspond to a small delivery  
 518 working offset ratio  $\theta$ . Such a result is mainly attributed to the joint action of the decrease in the  
 519 pressure loss and increase in the heat transfer temperature difference between the gas and the TES  
 520 materials as the particle size increases. Figure 13(b) shows the transient charging and delivery  
 521 power in the case of particles sizes of 10 mm, 20 mm, and 40 mm using argon. It can be observed

522 that large particles result in a relatively small charging power during the charging process; The  
 523 discharging power is the lowest at  $d_p = 10\text{mm}$  during the entire discharging process which is  
 524 relatively stable. However, although the discharging power at  $d_p = 40\text{mm}$  is higher than that at  $d_p =$   
 525  $20\text{mm}$  during the first discharging hour, it then declines fast and drops below that at  $d_p = 20\text{ mm}$   
 526 during the following discharging hours. The influence of the particle diameter mainly includes two  
 527 aspects: large particles result in small pressure loss and also large thermal resistance in particles  
 528 and large delivery temperature variation.

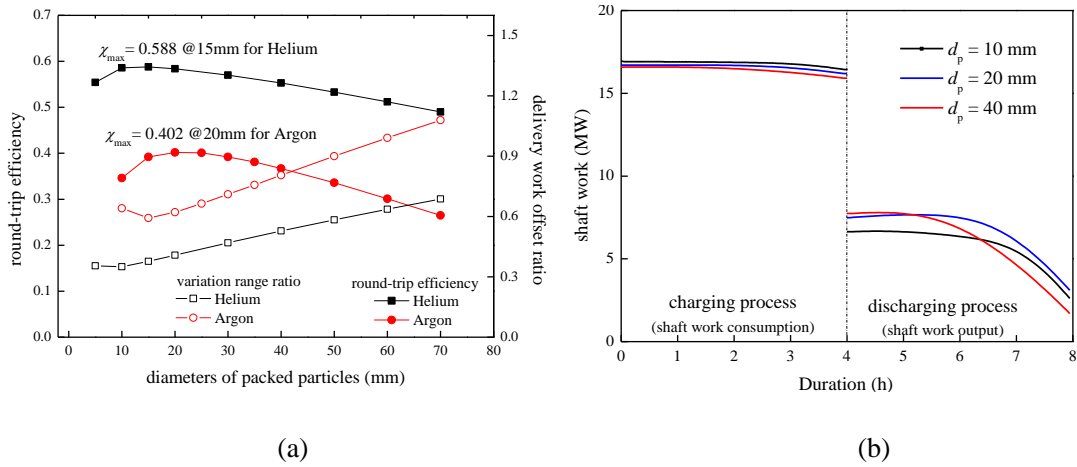
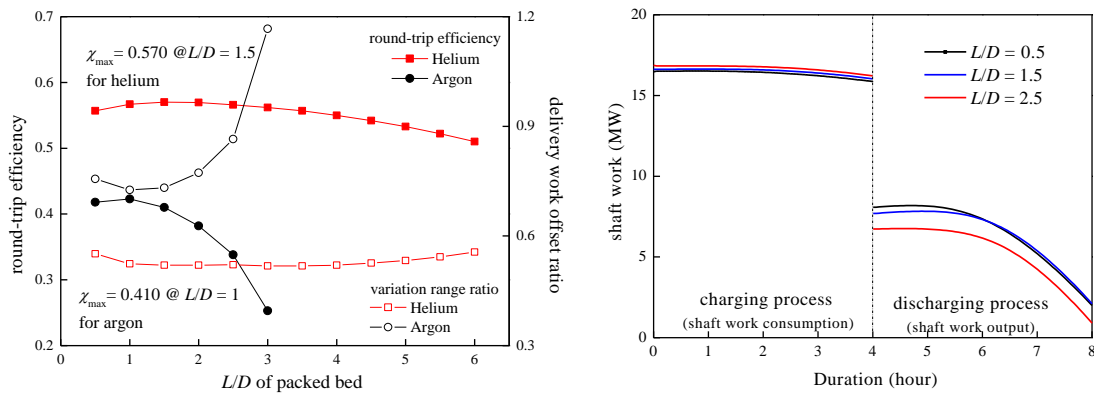


Fig.13. Impact of particle diameter of compressor and expander

#### 4.6 Effect of length-to-diameter ratio of reservoirs

533 As described in section 5, the volume of the designed HR and CR is  $460\text{ m}^3$  and  $740\text{ m}^3$ ,  
 534 respectively, for the 10 MW/4 h PHES system. For the cylindrical reservoirs with a fixed volume,  
 535 the length-to-diameter ratio  $L/D$  of the reservoirs is an important factor that influences the  
 536 pressure loss and heat transfer of the packed beds. Figure 14(a) shows the variation in the  
 537 round-trip efficiency  $\chi$  and the delivery working offset ratio  $\theta$  with the length-to-diameter ratio  
 538  $L/D$  of both the HR and CR, and the ranges of  $L/D$  are 0.5–3 for argon and 0.5–6 for helium. It can  
 539 be observed in figure 14(a) that the influence of  $L/D$  is rather gentle in the case of helium whereas

540 it is great in the case of argon. The round-trip efficiency  $\chi$  increases at the beginning and decreases  
 541 gradually with the increase in  $L/D$ , and a maximum round-trip efficiency of 41.0% and a  
 542 minimum discharging power offset ratio of 72.6% occurs at  $L/D = 1$  for argon; for helium the  
 543 maximum round-trip efficiency is 57.0% and the minimum discharging power offset ratio of 51.8%  
 544 occurs at  $L/D = 1.5$ . This is because a larger length-to-diameter ratio  $L/D$  would result in a larger  
 545 pressure loss and a relatively smaller proportion of the thermocline region in the packed beds  
 546 simultaneously, which is also a joint effect. Figure 14(b) shows the transient charging and  
 547 discharging power under the conditions of the length-to-diameter ratio  $L/D$  of 0.5, 1.5, and 2.5  
 548 using argon. During the charging process, the larger length-to-diameter ratio  $L/D$  results in  
 549 relatively higher charging power owing to the higher pressure loss; the discharging power is the  
 550 lowest at  $L/D = 2.5$  during the discharging process. However, the discharging power at  $L/D = 0.5$   
 551 is higher than that at  $L/D = 1.5$  during the discharging, and then declines fast and drops below that  
 552 at  $L/D = 1.5$ .



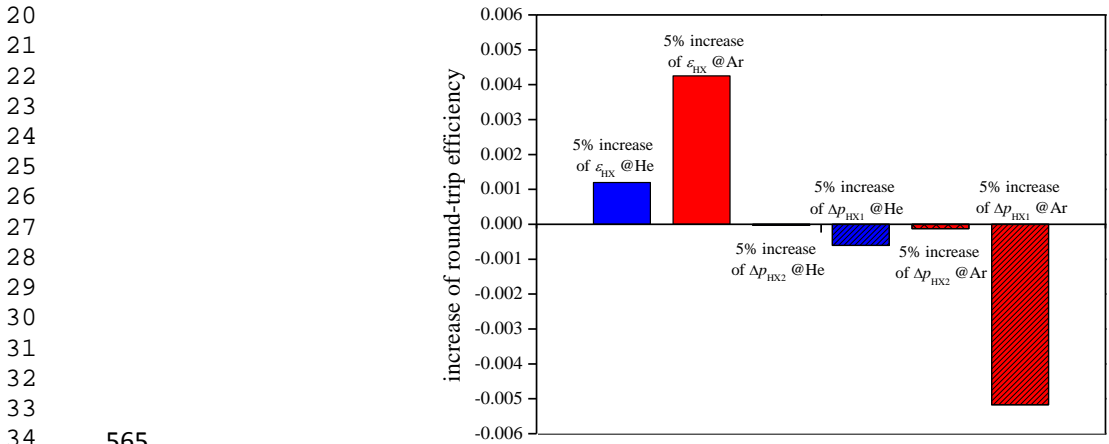
553 (a)  
 554 (b)  
 555 Fig.14. Impact of  $L/D$  of packed bed reservoirs

556 4.7 Effect of efficiency and pressure drop of heat exchangers

557 Figure 15 shows the round-trip efficiency variation of the PHES with a 5% increase in the



1 558 efficiency and pressure drop of the heat exchangers (including HX1 and HX2) based on the  
 2  
 3 559 parameters listed in tables 1 and 2. It can be observed that the increase in the heat transfer  
 4  
 5 560 efficiency of the heat exchangers improves the round-trip efficiency whereas the increase in the  
 6  
 7 561 pressure loss decreases the round-trip efficiency; the effect of the heat exchangers efficiency and  
 8  
 9 562 pressure drop on the PHES efficiency using argon is several times higher than that of helium; and  
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 11 563 the influence of the pressure loss of the low pressure heat exchanger (HX1) is more obvious than  
 12  
 13 564 that of the high pressure heat exchanger (HX2).



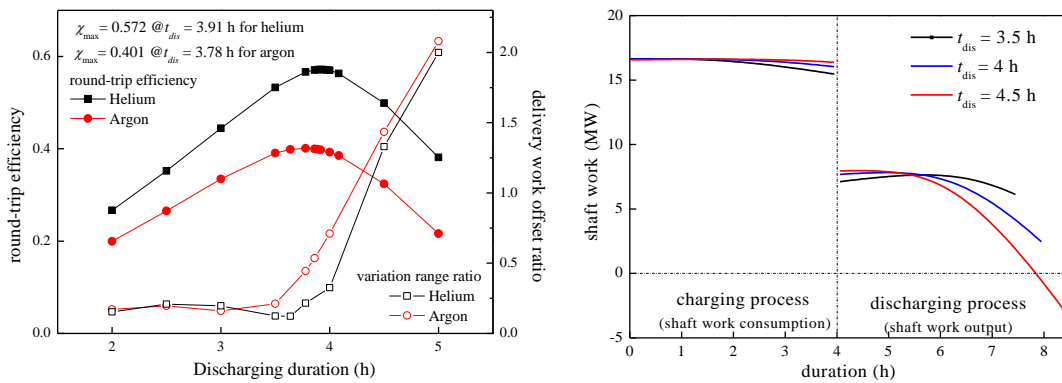
34 565  
 35  
 36 566 Fig.15. Impact of efficiency and pressure drop of heat exchangers

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 39 567 *4.8 Effect of discharging duration*

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 41  
 42 568 In the above analysis, each energy storage circle comprise a charging process of 4 h and a  
 43  
 44 569 discharging process of 4 h; however, an equal discharging and charging duration may not be  
 45  
 46 570 optimal for such a PHES system. Figure 16(a) shows the influence of the discharging time ranging  
 47  
 48 571 from 2 h to 5 h (one circle consists of a 4 h charging process and 2–5 h discharging process) on the  
 49  
 50 572 round-trip efficiency  $\chi$  and the delivery working offset ratio  $\theta$  using argon and helium, respectively.  
 51  
 52 573 From figure 15(a), it can be observed that the round-trip efficiency  $\chi$  increases at first and  
 53  
 54 574 then decreases with the discharging time. The best selection of the discharging duration is a few

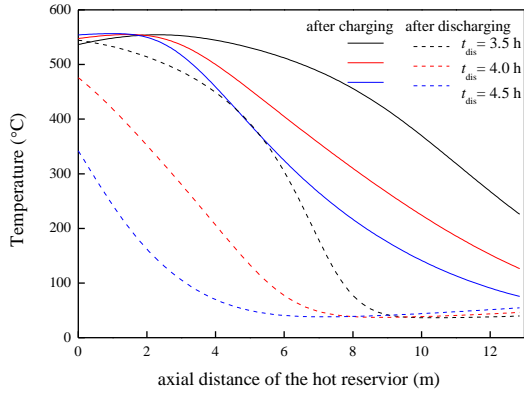
minutes shorter than the charging time such that the maximum round-trip efficiency of 40.1% occurs at the delivery duration of 3.78 h for argon, and the maximum round-trip efficiency is 57.2% at the delivery duration of 3.91 h for helium. The delivery working offset ratio  $\theta$  is relatively low (<20%) for a discharging duration less than approximately 3.5 h and then increases sharply.

Figure 16(b) shows the transient shaft power during the charging and discharging with the discharging duration of 3.5 h, 4 h and 4.5 h using argon. It can be observed that for the PHES system having a 3.5 h discharging duration has the most stable delivery power, and the obvious decline of the delivery power at the later stage of the discharging process can be observed with a longer discharging duration. Figure 16(c) shows the axial temperature profile of the hot TES reservoir at the end of the charging and discharging processes for the discharging durations of 3.5 h, 4 h and 4.5 h. It also shows that more exergy with a high temperature is stored in the hot TES reservoir in the PHES system in the case of the discharging duration of 3.5 h, and a relatively stable delivery thermal energy profile can be obtained during the discharging process, but it has the drawback of relatively unstable charging power, which can be reduced through the heat exchangers.



(a)

(b)



(c)

Fig.16. Impact of the discharging duration on the PHES behavior

### 5 Conclusions

In this paper, the use of the transient analysis method on the Joule–Brayton based PHES system is proposed for the coupling dynamics, thermodynamics and heat transfer process. The cyclic transient behavior of the 10 MW/4 h Joule–Brayton PHES system is studied using argon and helium as the working gases. Based on the round-trip efficiency and the variation range ratio of the delivery power, the mechanisms influencing PHES system and components parameters on the PHES system performance are further discussed. From the result of the analysis, the following conclusions can be obtained:

1. The delivery power clearly declines during the discharging process mainly owing to the thermal energy reduction from the packed bed TES reservoirs.

2. The gas resistance loss through the TES reservoirs and heat exchangers has a great influence on the system performance. In addition, helium, with small resistance losses, has an overwhelming advantage over argon for application in the PHES. The round-trip efficiency  $\chi$  of helium is 56.9%, which is much higher than 39.3%, which is obtained on using argon under the

1 610 design conditions. The PHES system using helium can also provide more stable electricity with  
2  
3 611 the delivery power offset ratio of 45.9% than that using argon with a delivery power offset ratio of  
4  
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6 612 71.0%.

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9 613 3. The increase in the pressure ratio and isentropic efficiencies would lead to an obviously  
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11 614 improvement in the round-trip efficiency and delivery stability. Furthermore, an appropriate  
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13 615 discharging compression ratio that is less than the charging compression ratio will aid in  
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17 616 improving the round-trip efficiency. For the 10 MW/4 h PHES system, the optimum round-trip  
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20 617 efficiency is obtained at the discharging compression ratio of 7 when the charging compression  
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23 618 ratio is 10.

24  
25 619 4. For the TES reservoirs, there exists optimal selections of particle sizes, ratios of length  
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28 620 –to–diameter, and discharging durations corresponding to the maximum round-trip efficiency and  
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31 621 preferable discharging power stability; this is mainly owing to the joint effects of the pressure loss,  
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34 622 heat transfer and thermodynamics.

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36 623 Further research is required for improving the improvement of the round-trip efficiency and  
37  
38  
39 624 discharging power stability and decreasing the costs, which will be the subject of the authors’  
40  
41  
42 625 future research.

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#### 46 627 **Conflict of Interest**

47  
48 628 The authors declare no conflict of interest.

49 629

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730  
 731

732 **Nomenclature**

733 *Abbreviations*

BOT	Bottoming system	734
BV	Buffer vessel	735
CAES	Compressed air energy storage	736
CHEST	Compressed heat energy storage	737
CR	Cold Reservoir	738
DSC	differential scanning calorimetry	739
EES	Electrical energy storage	740
HP	High pressure	741
HR	Hot reservoir	742
HX	Heat exchanger	743
LNG	Liquefied natural gas	744
LP	Low pressure	745
NIST	National Institute of Standards and Technology	
ORC	Organic Rankine cycle	748
PHS	Pumped hydro storage	749
PHES	Pumped heat electricity storage	750
PTES	Pumped thermal electricity storage	751
TEES	Thermo-electrical energy storage	752
TEMA	Tubular Exchanger Manufacturers Association	753 754
TES	Thermal energy storage	755 756

757 *Symbols*

$Bi$	Biot number
$C$	Specific heat capacity, $J K^{-1} kg^{-1}$
$d$	Diameter of particles, m
$D$	Diameter of packed bed reservoir, m
$e$	Specific energy, $J kg^{-1}$
$G$	Mass flow rate, $kg s^{-1}$
$h$	Volumetric heat transfer coefficient, $W m^{-3} K^{-1}$
$i$	Number $i$

1	$I$	Moment of inertia, kg m <sup>2</sup>	758
2	$K$	Thermal conductivity, W m <sup>-1</sup> K <sup>-1</sup>	
3	$L$	Length scale of packed bed, m	
4	$m$	Mass of gas, kg	
5	$n$	Number	
6	$N$	Number of circles	
7	$P$	Power, W	
8	$Q$	Volume flow rate, m <sup>3</sup> s <sup>-1</sup>	
9	$Re$	Reynolds number	
10	$t$	Time, s	
11	$T$	Temperature, K	
12	$\beta$	Compression/expansion ratio of compressor/expander	
13	$\gamma$	Adiabatic exponent of gas	
14	$\varepsilon$	Efficiency of heat exchanger	
15	$\eta$	Polytropic efficiency of compressor/expander	
16	$\theta$	Offset ratio of delivery power	
17	$\kappa$	Parameter, $(\gamma-1)/\gamma$	
18	$\mu$	Dynamic viscosity, Pa s	
19	$\rho$	Density, kg m <sup>-3</sup>	
20	$\Phi$	Porosity of packed bed	
21	$X$	Round-trip efficiency	
22	$\omega$	Angular velocity, rad s <sup>-1</sup>	
23	<i>Subscript</i>		
24	0	Point 0	
25	1	Point 1	
26	c	Compressor	
27	chr	Charge	
28	des	Design	
29	dis	Discharge	
30	e	Expander	
31	eff	Effective	
32	g	Gas	
33	HP	High pressure	
34	HX1	Heat exchanger 1	
35	HX2	Heat exchanger 2	
36	i	Number i	
37	in	At the inlet	
38	LP	Low pressure	
39	p	Particle	
40	s	Solid	
41	w	Water	



Figure 1

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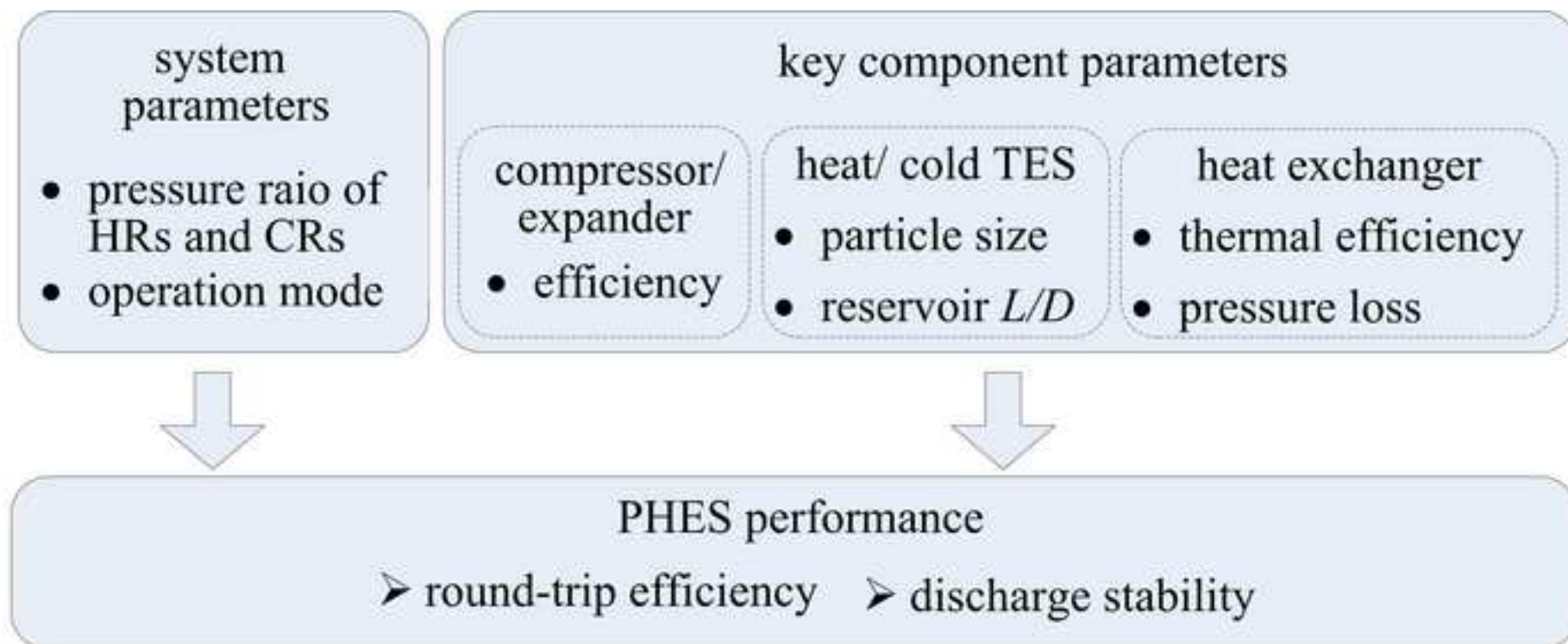


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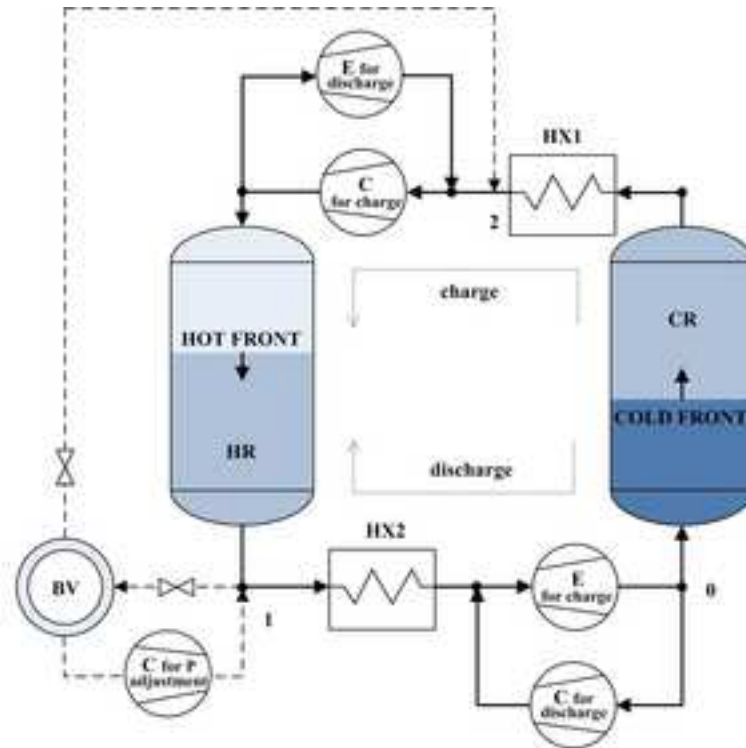


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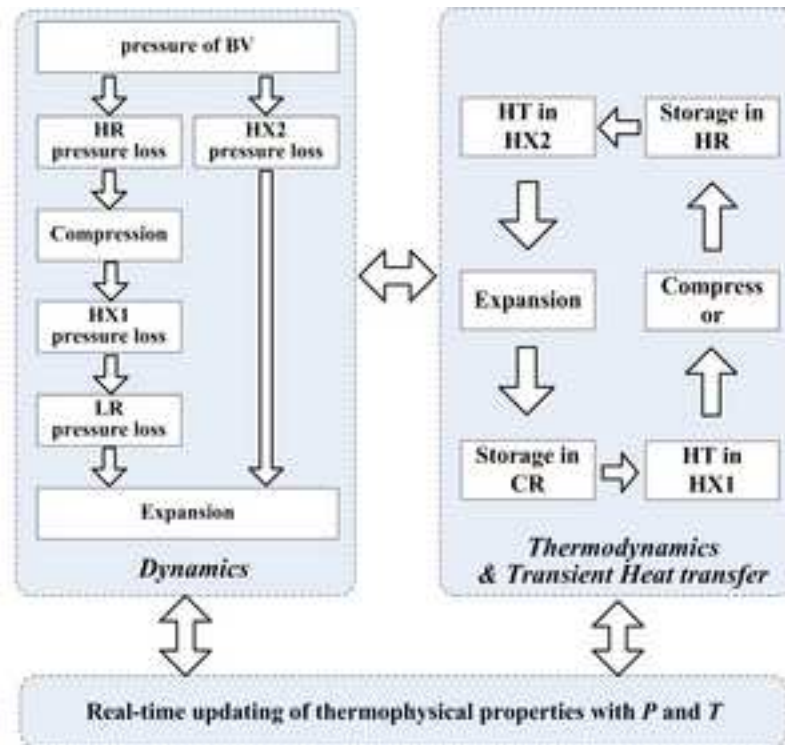


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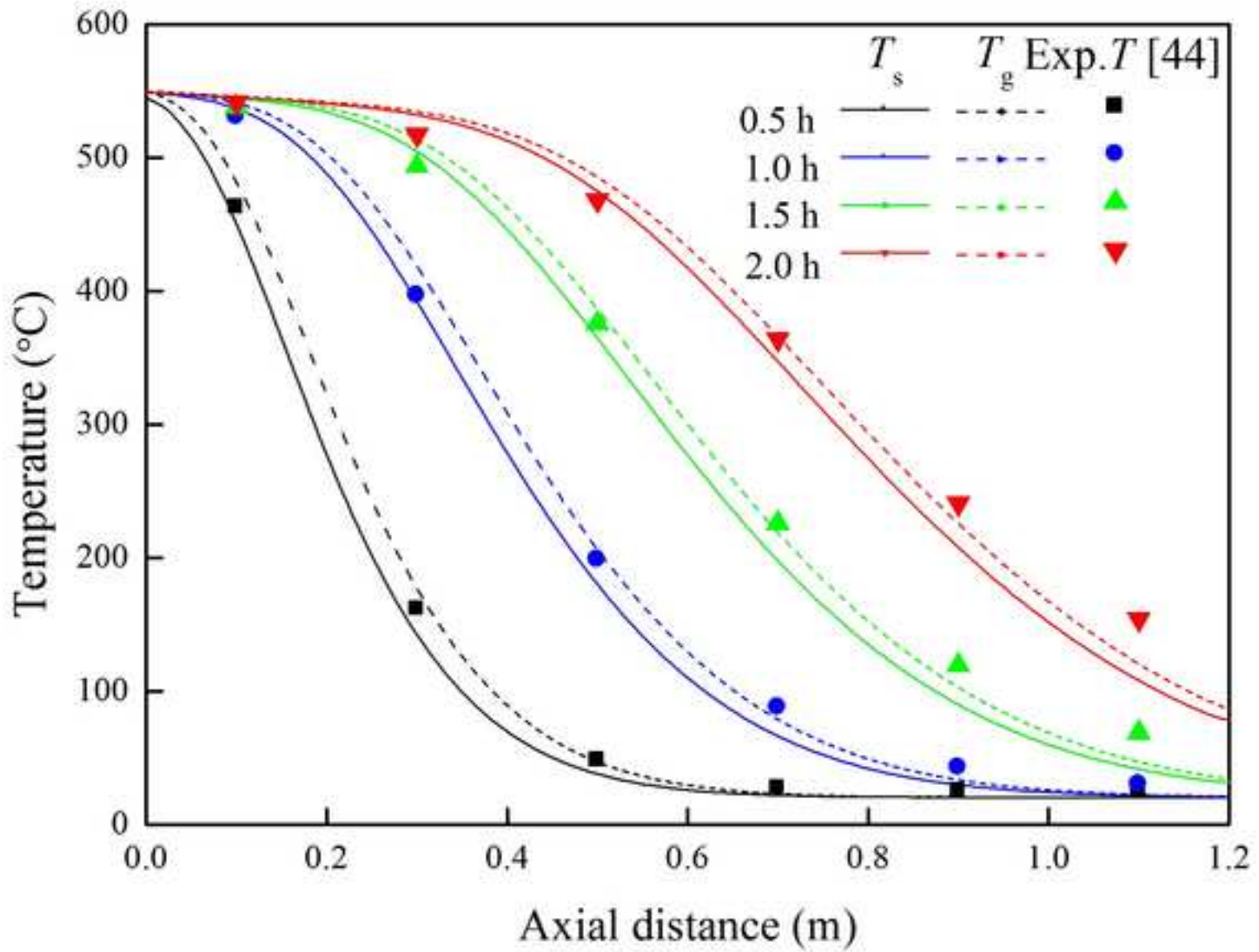


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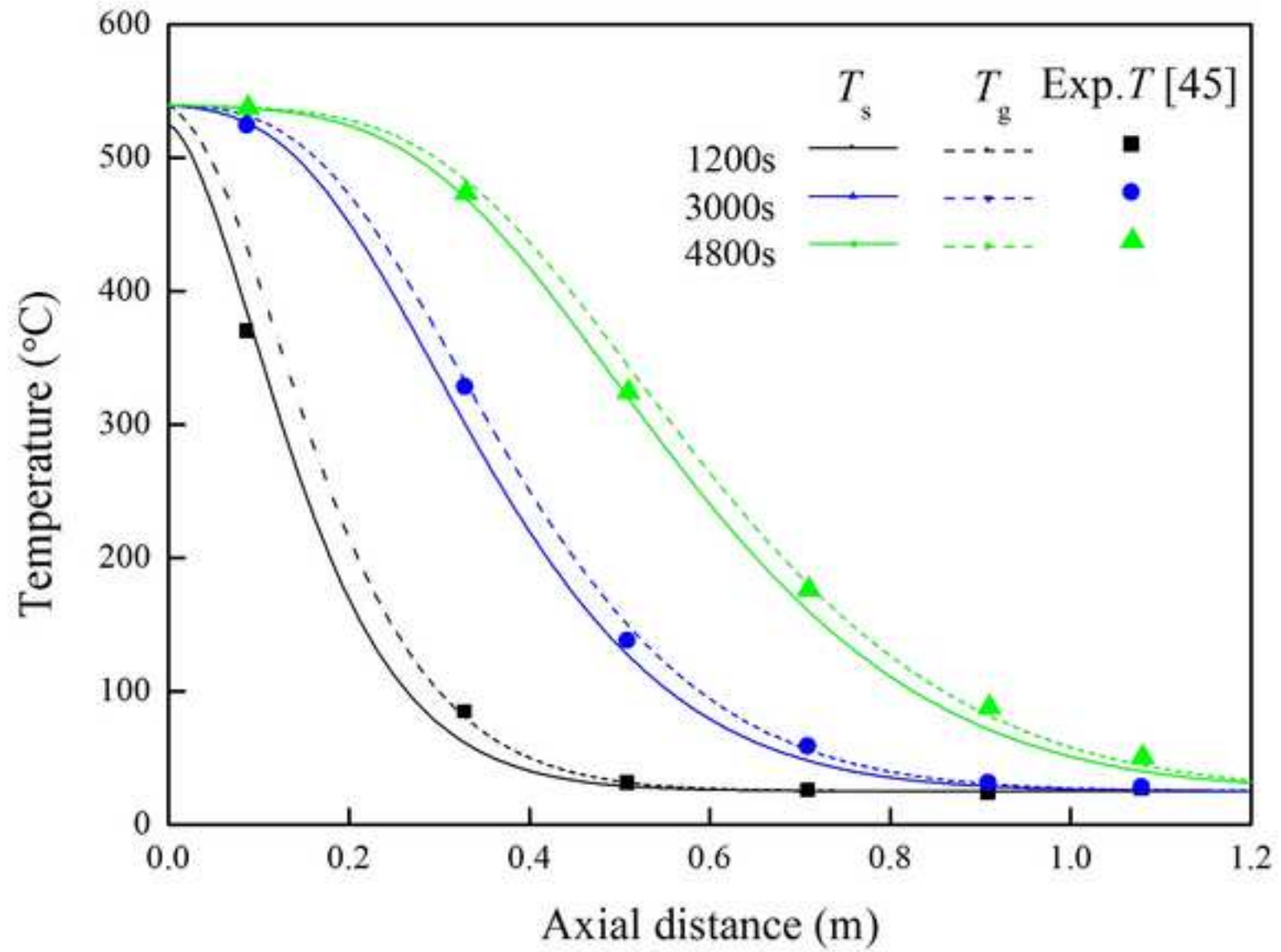


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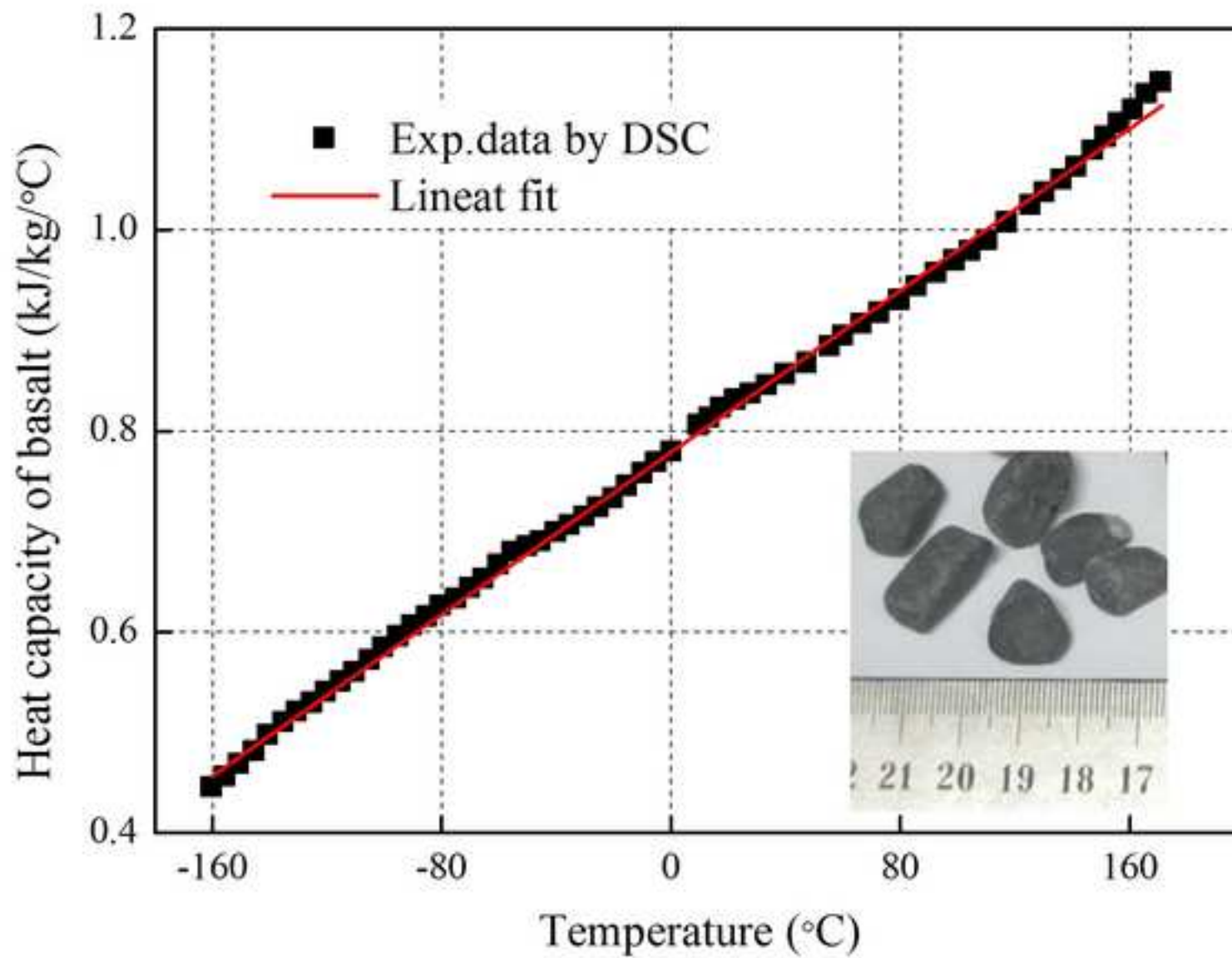


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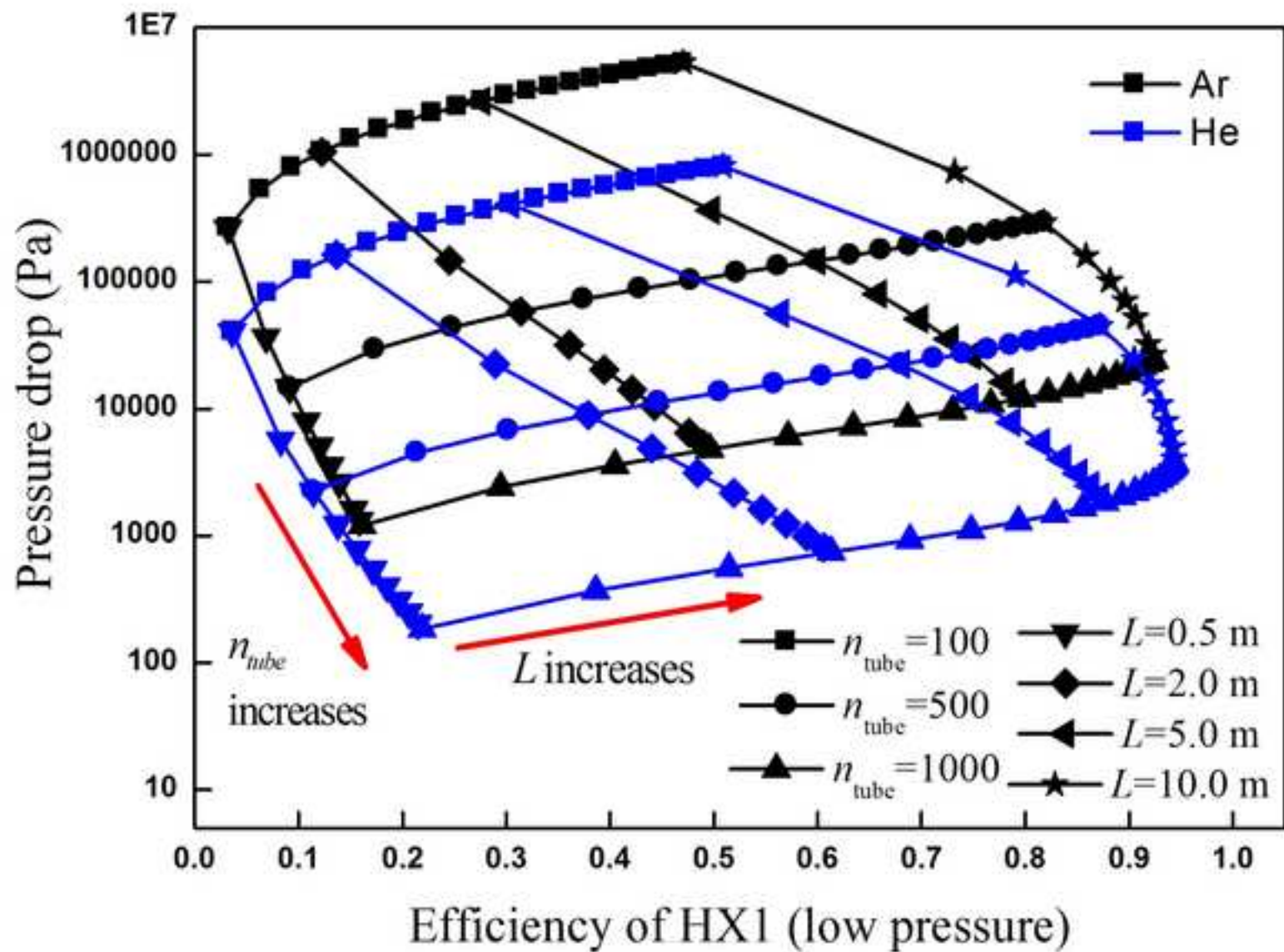


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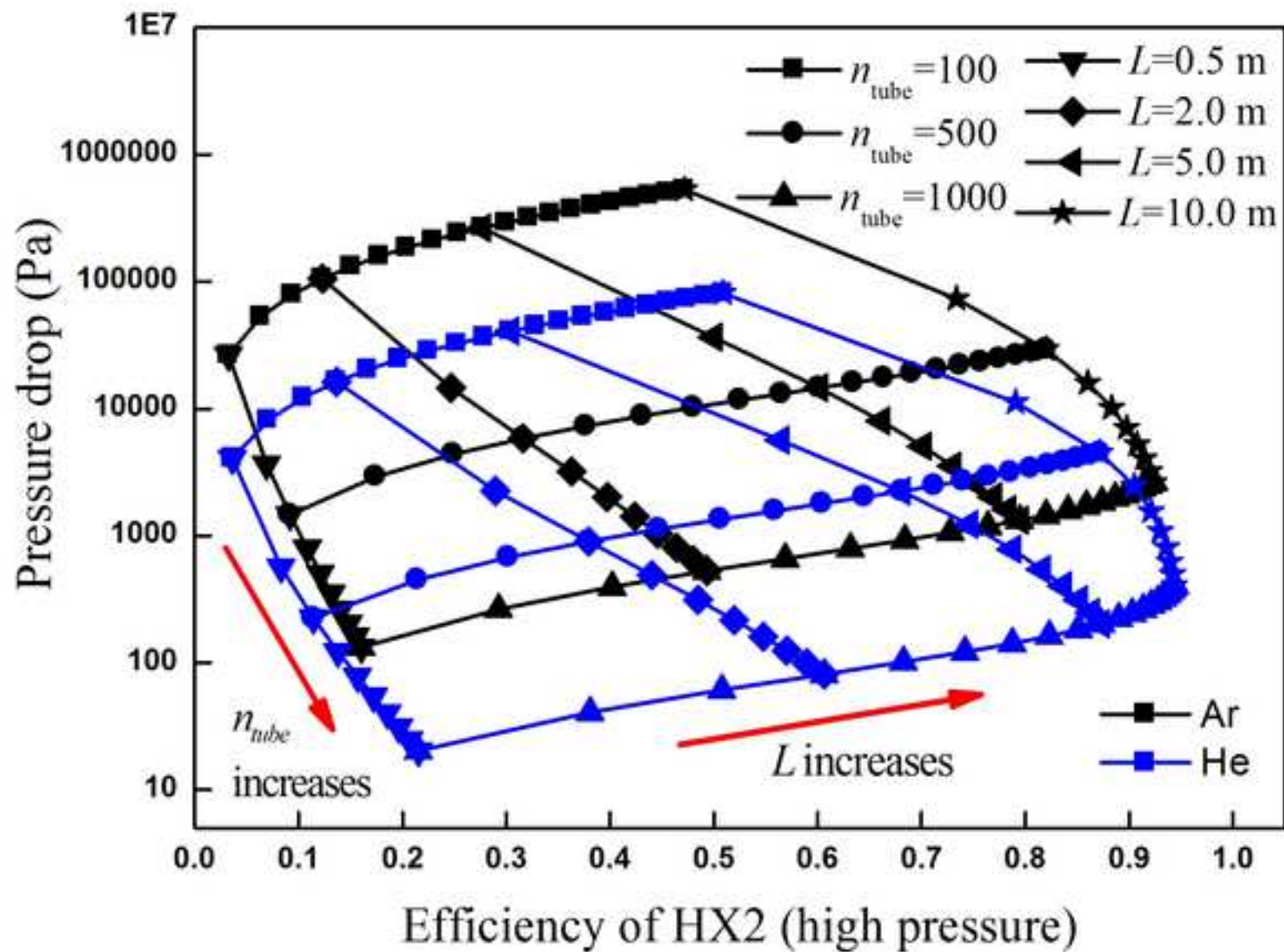






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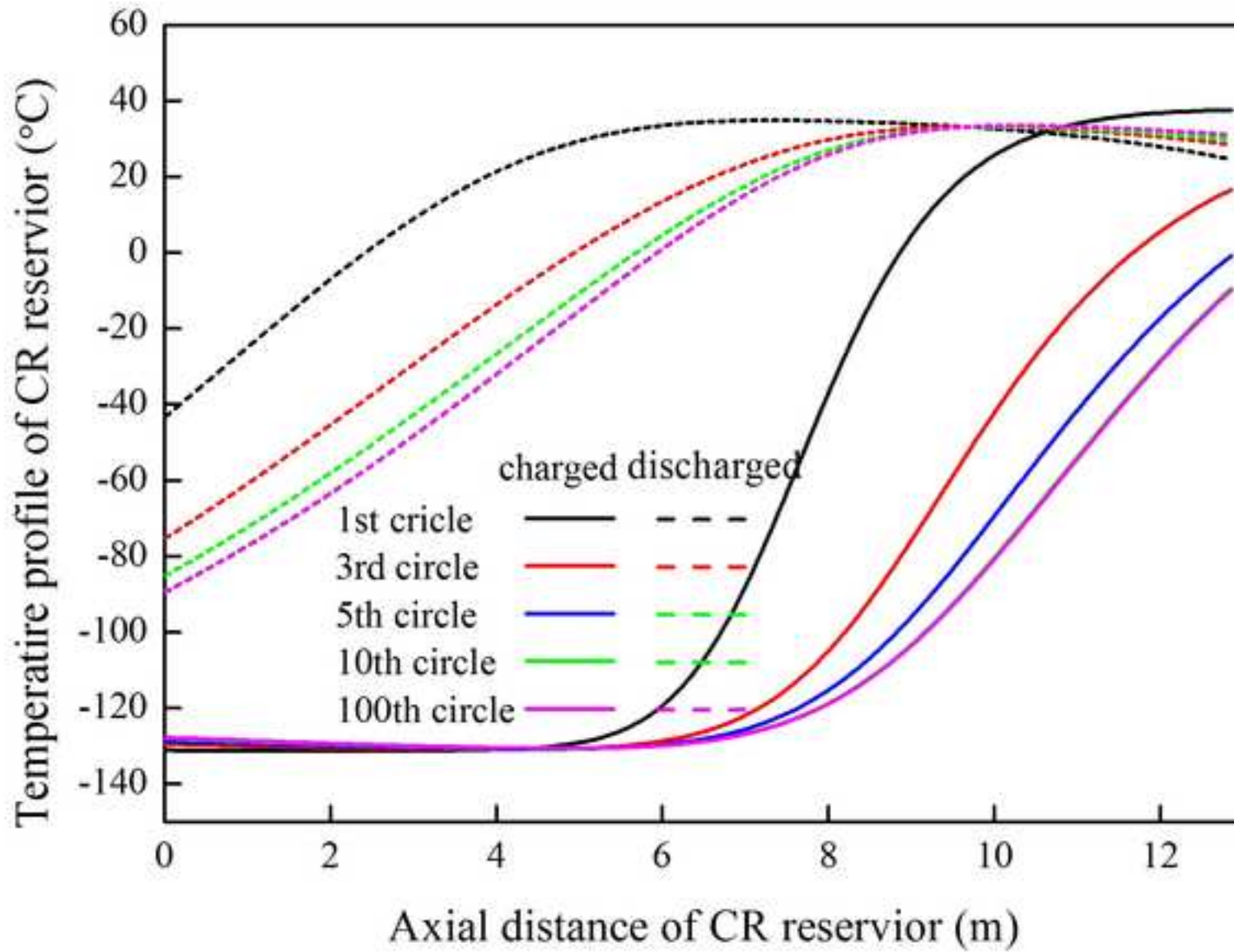


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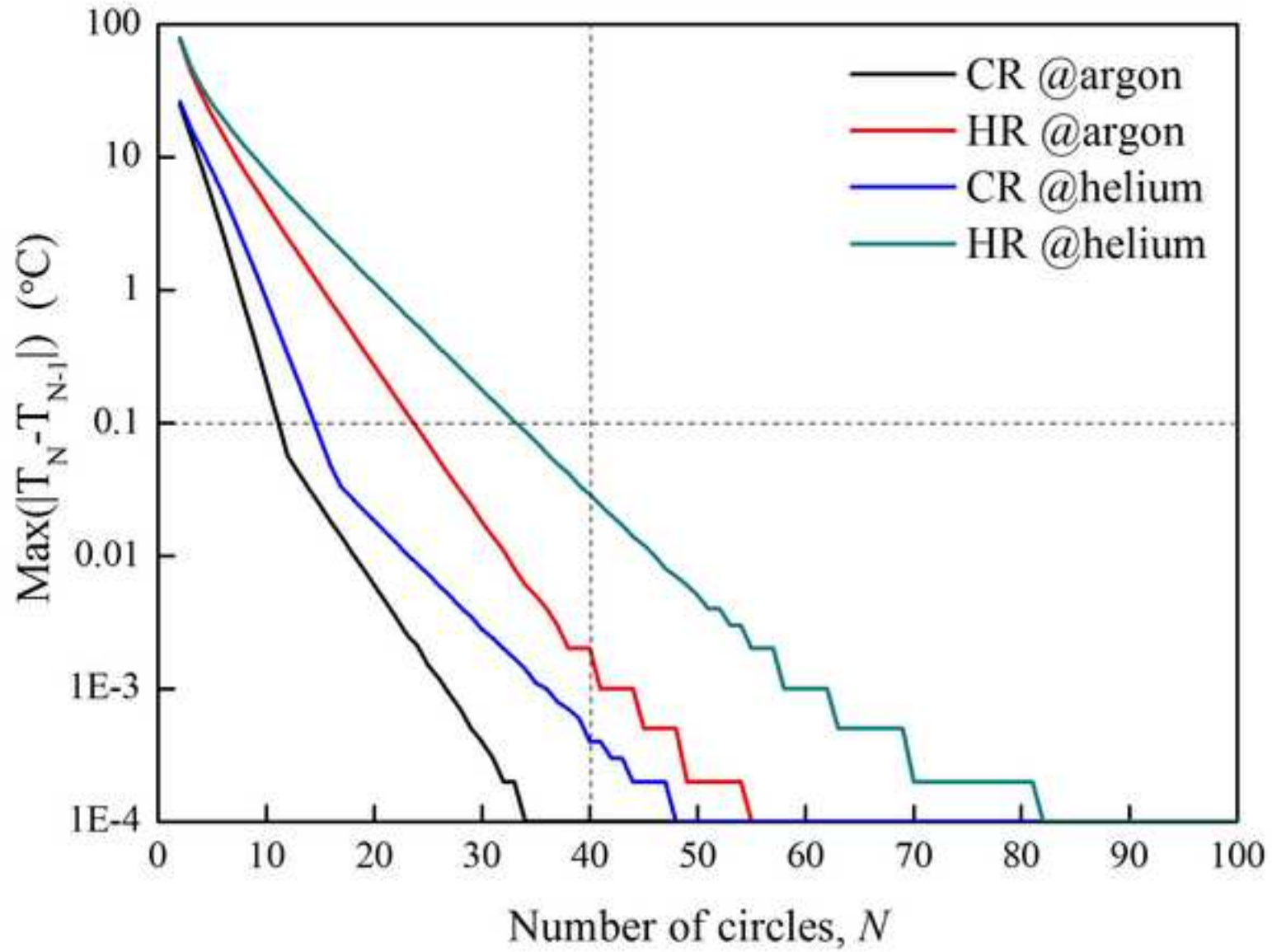


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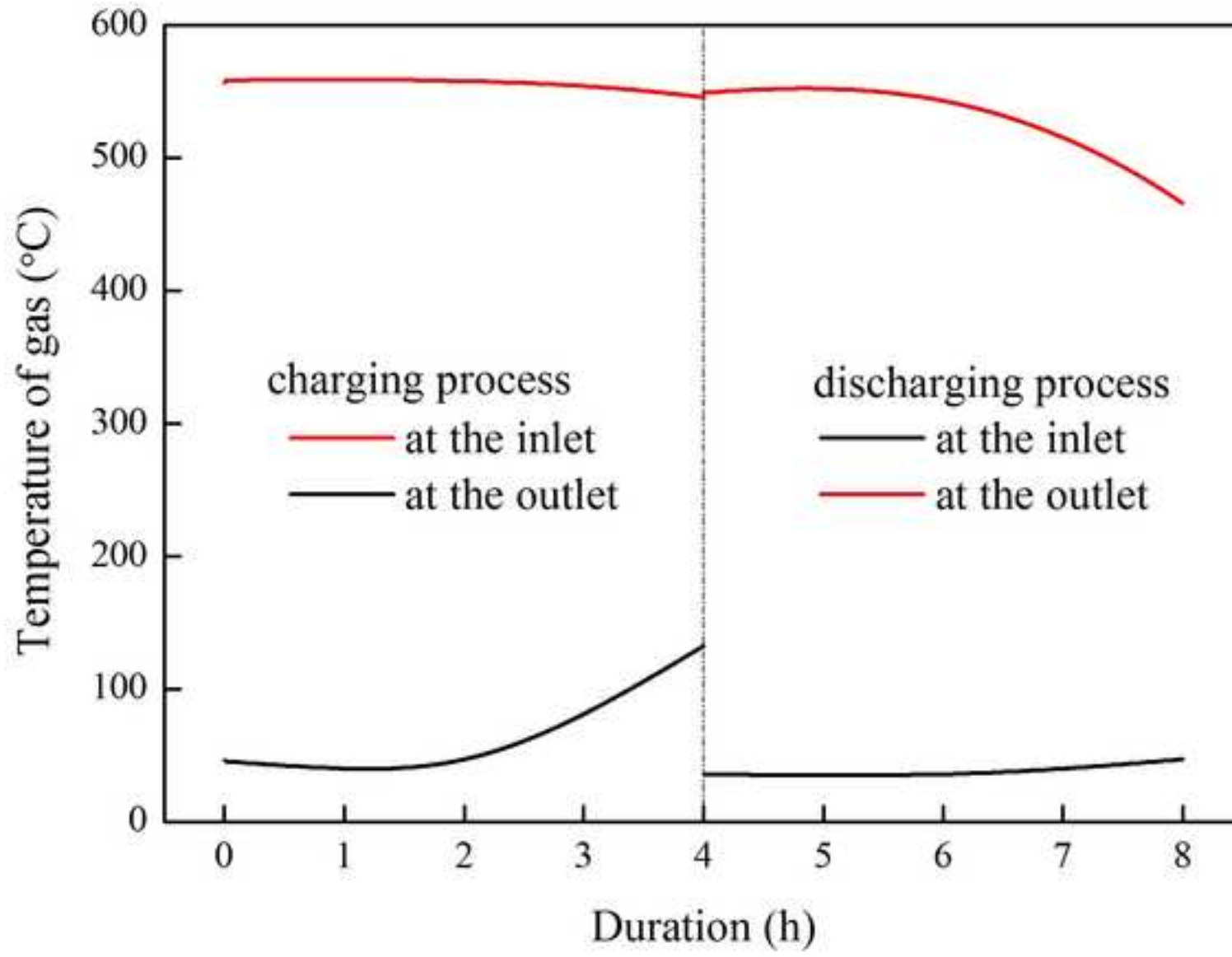


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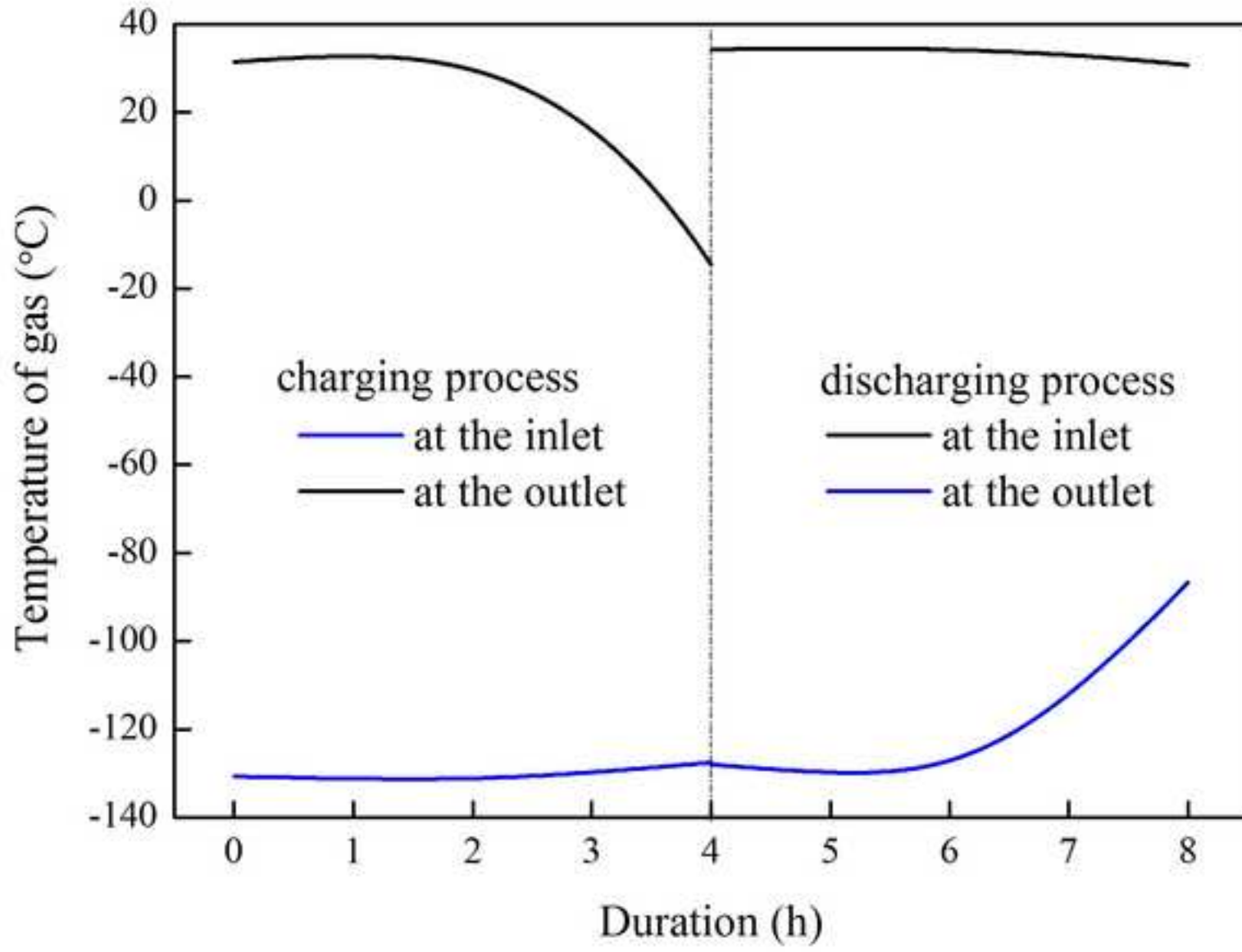


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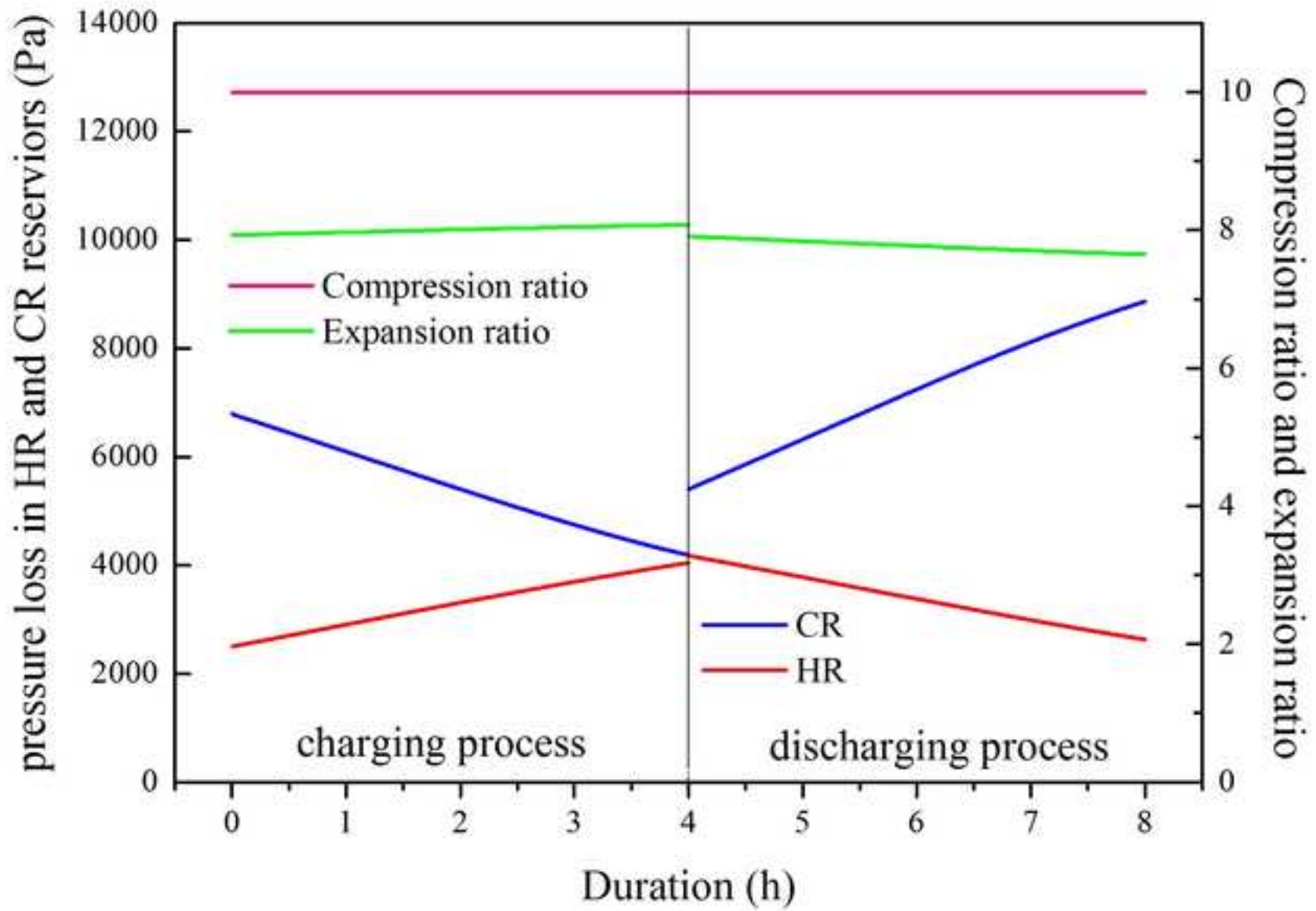


Figure 9d

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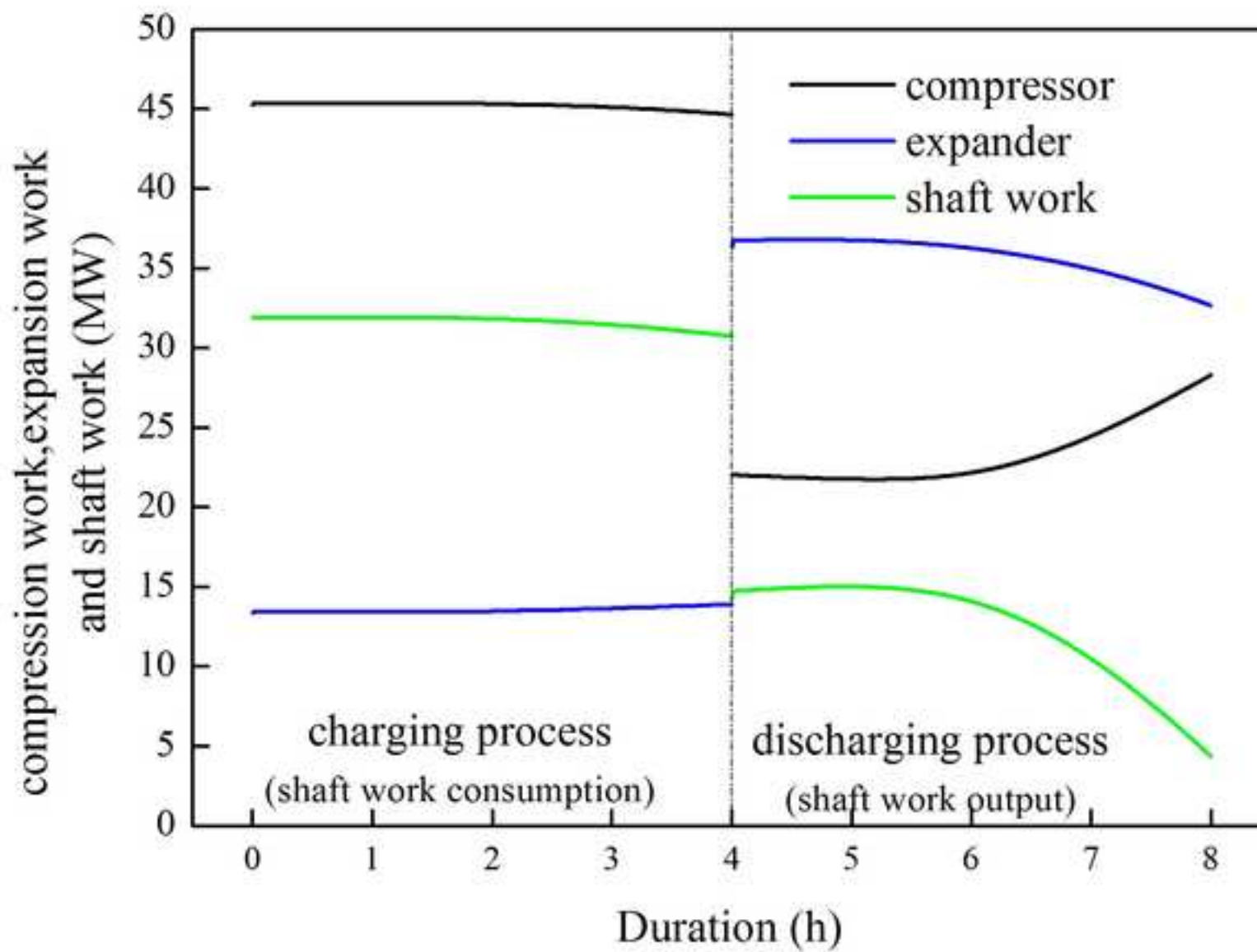


Figure 10a  
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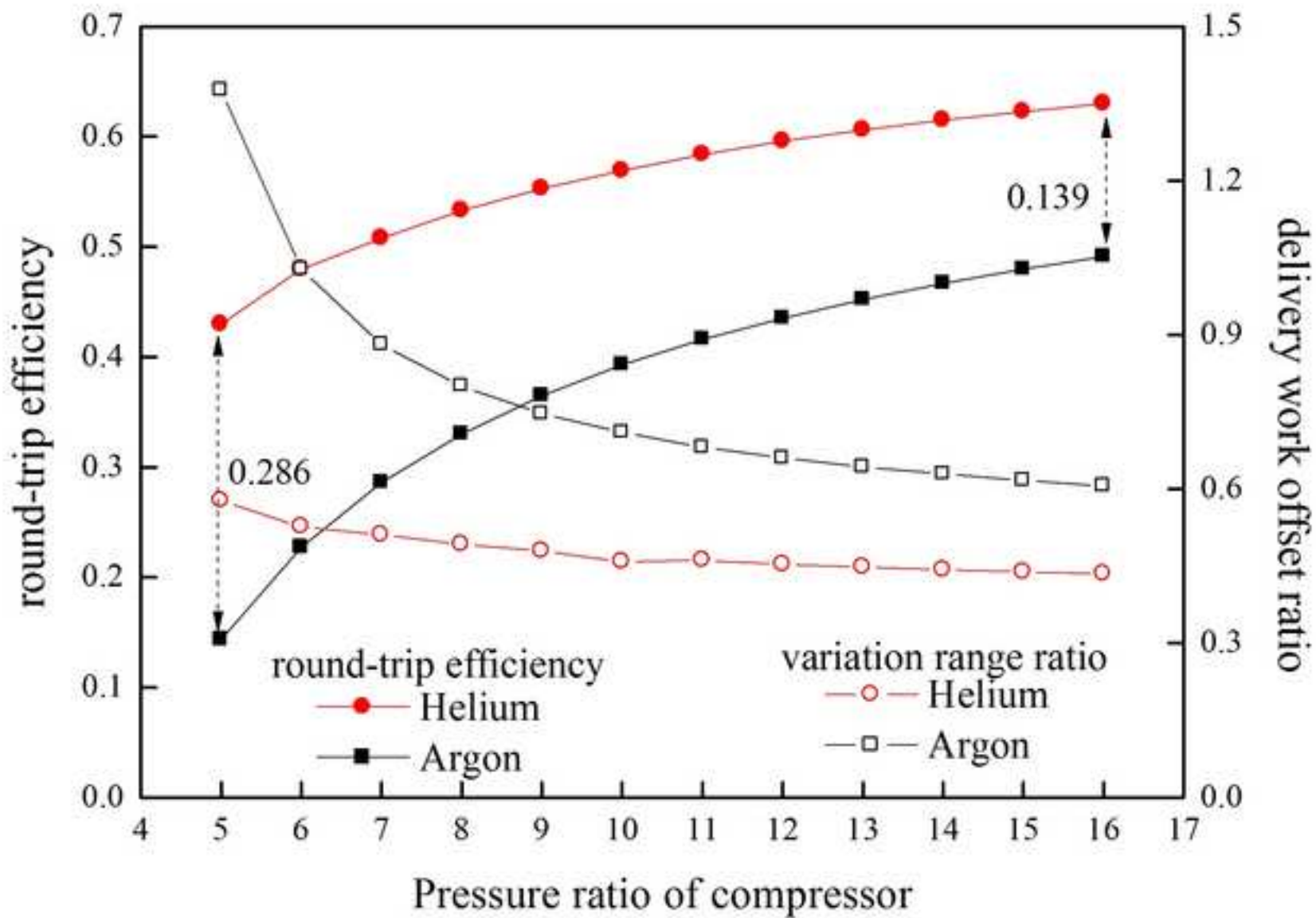




Figure 10b

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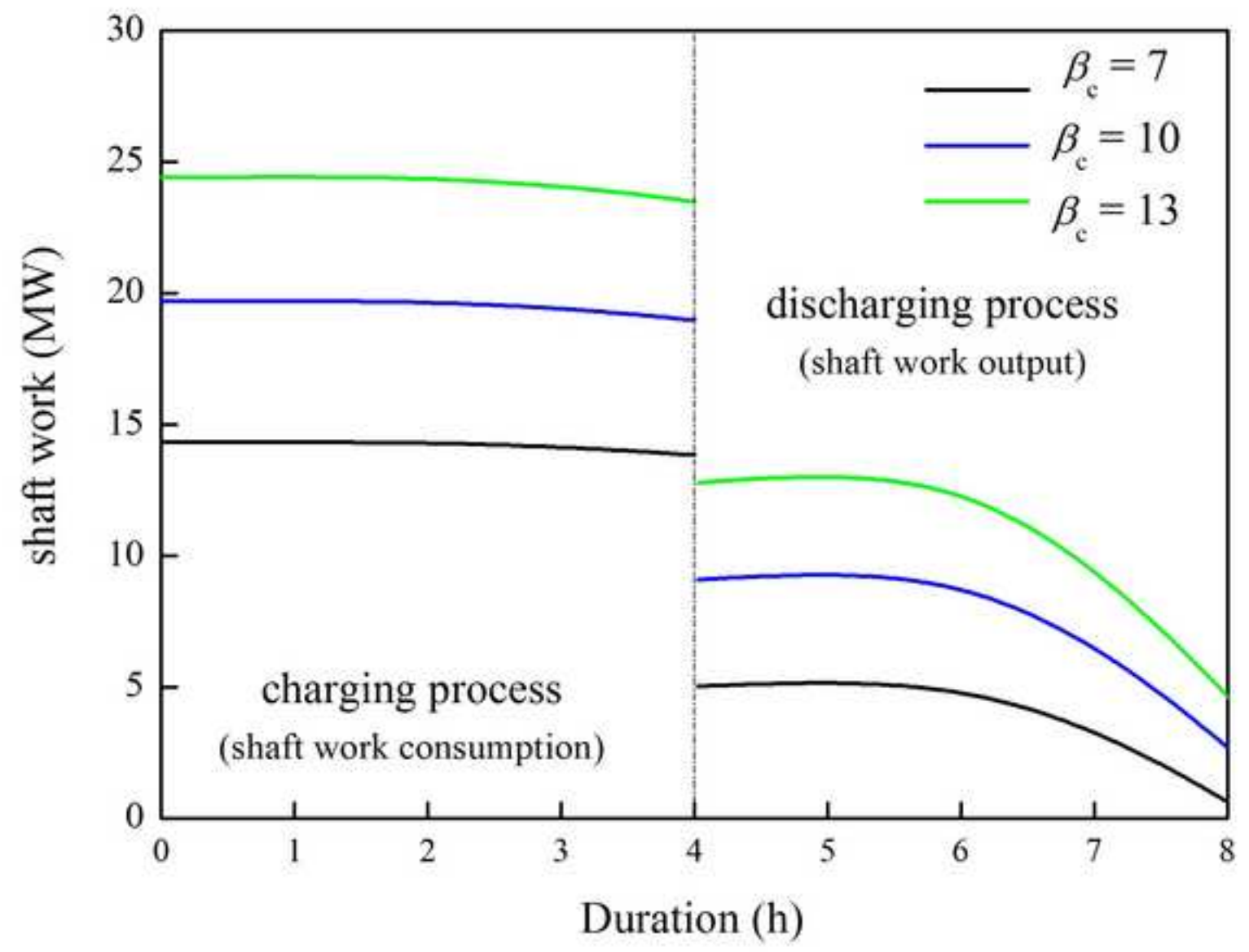


Figure 11a

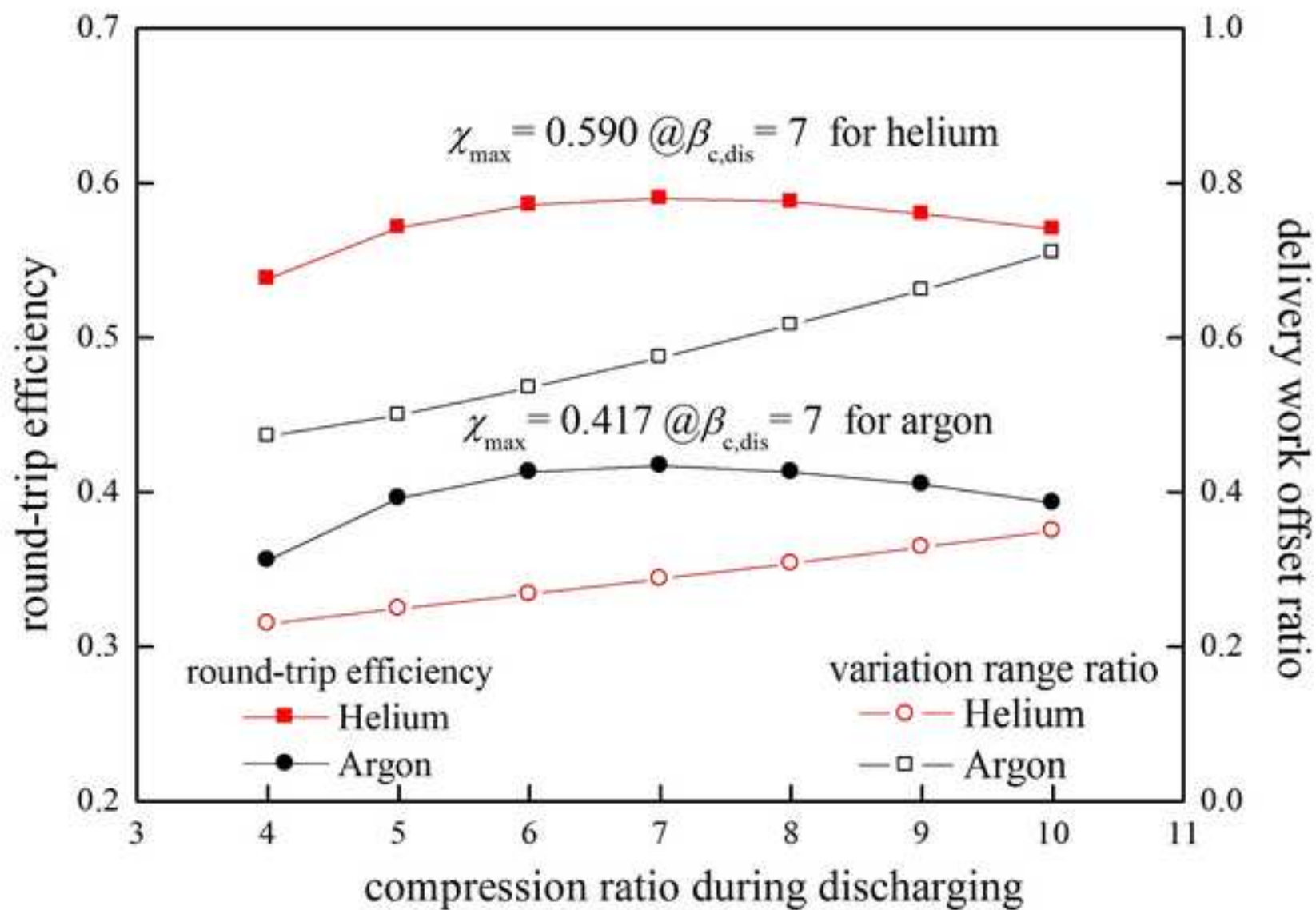
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Figure 11b  
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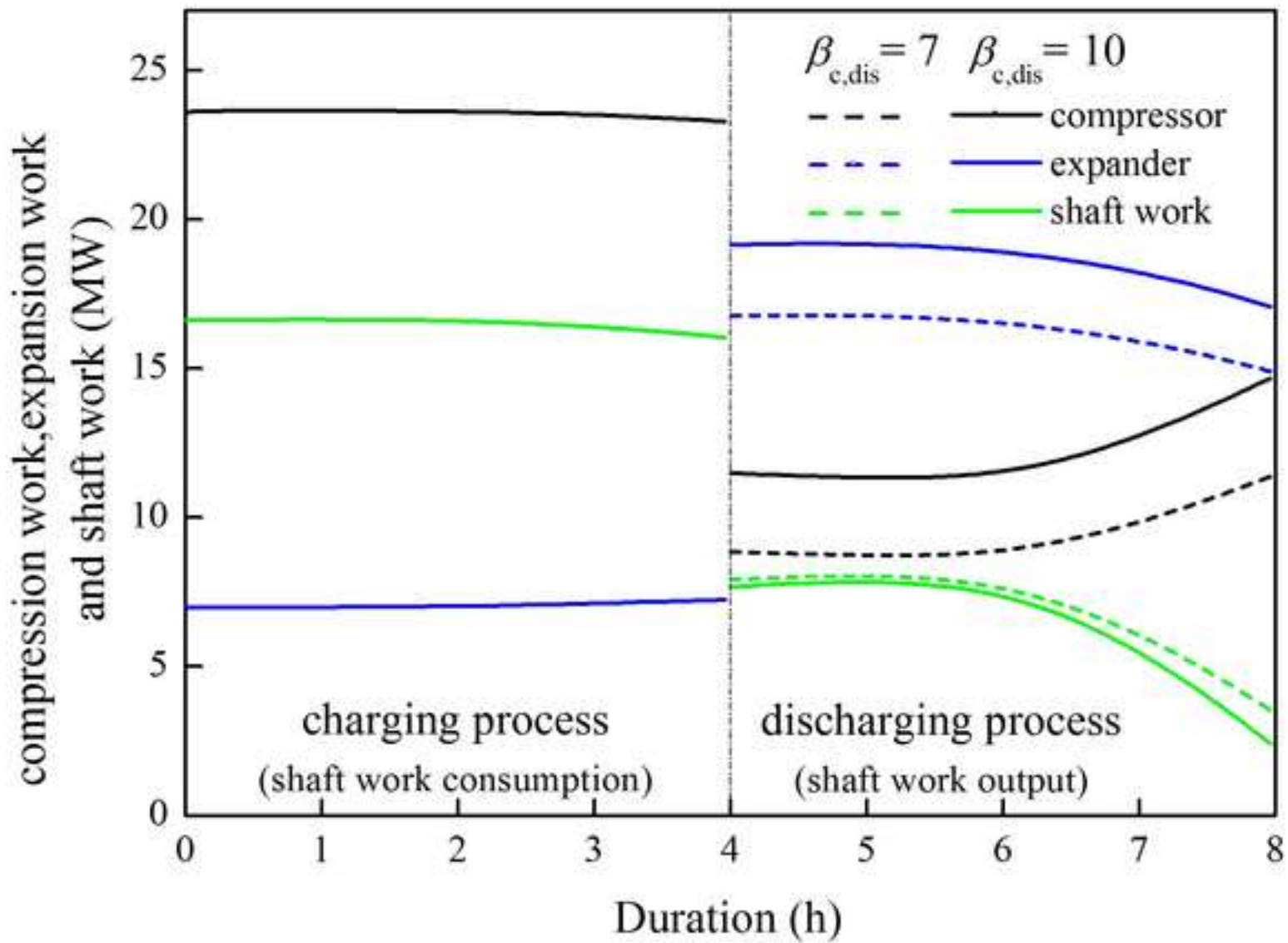


Figure 12  
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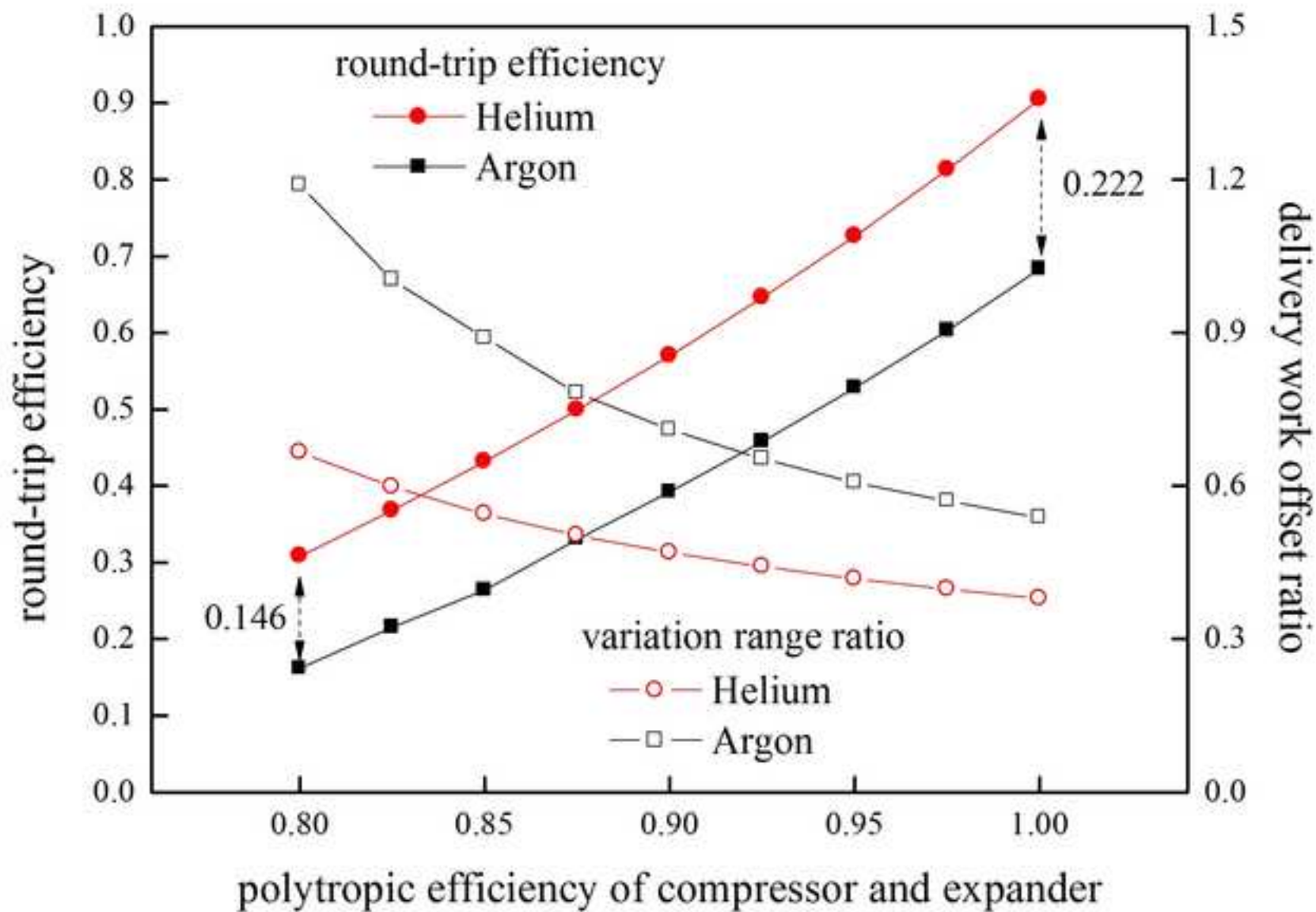


Figure 13a  
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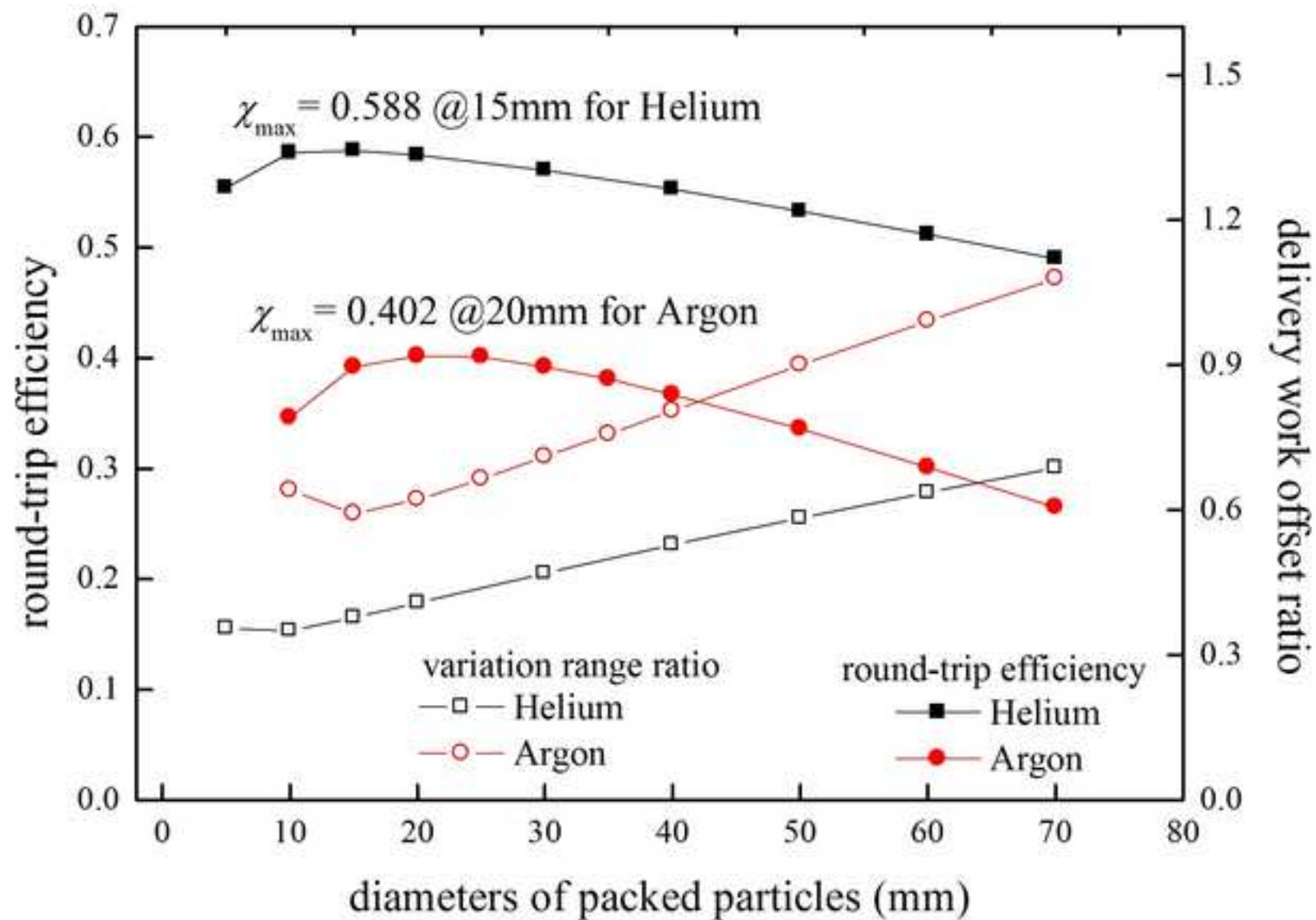


Figure 13b

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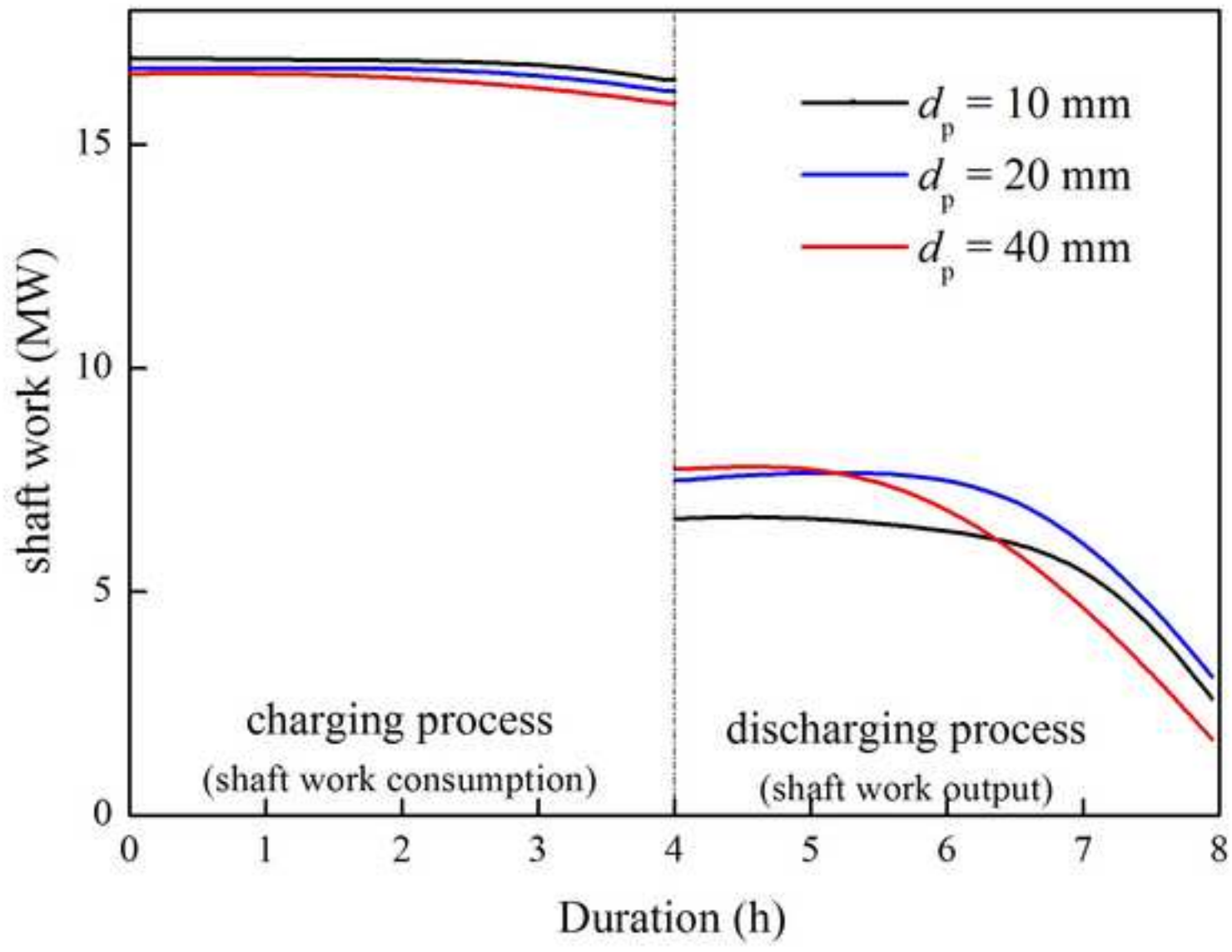


Figure 14a

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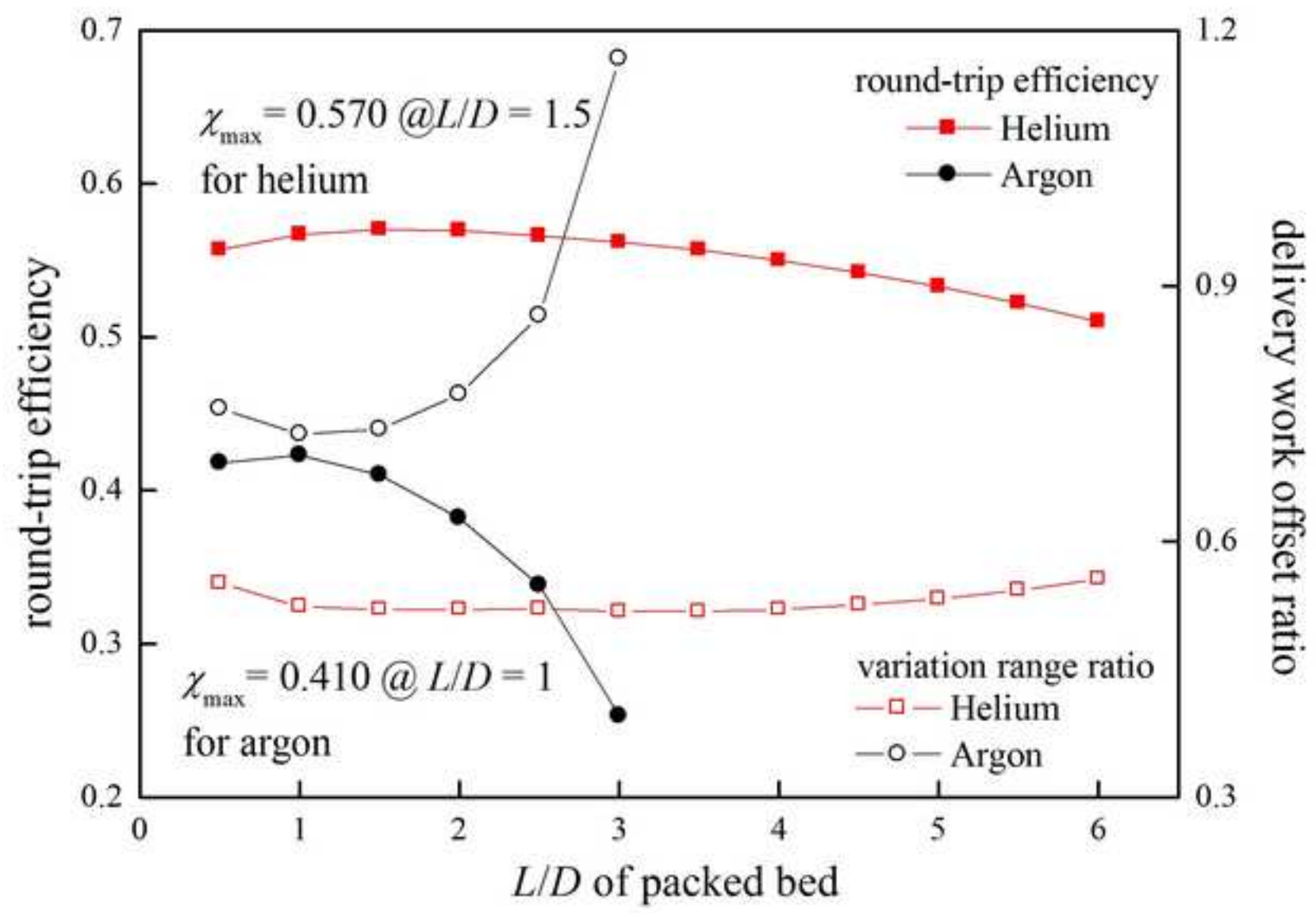


Figure 14b  
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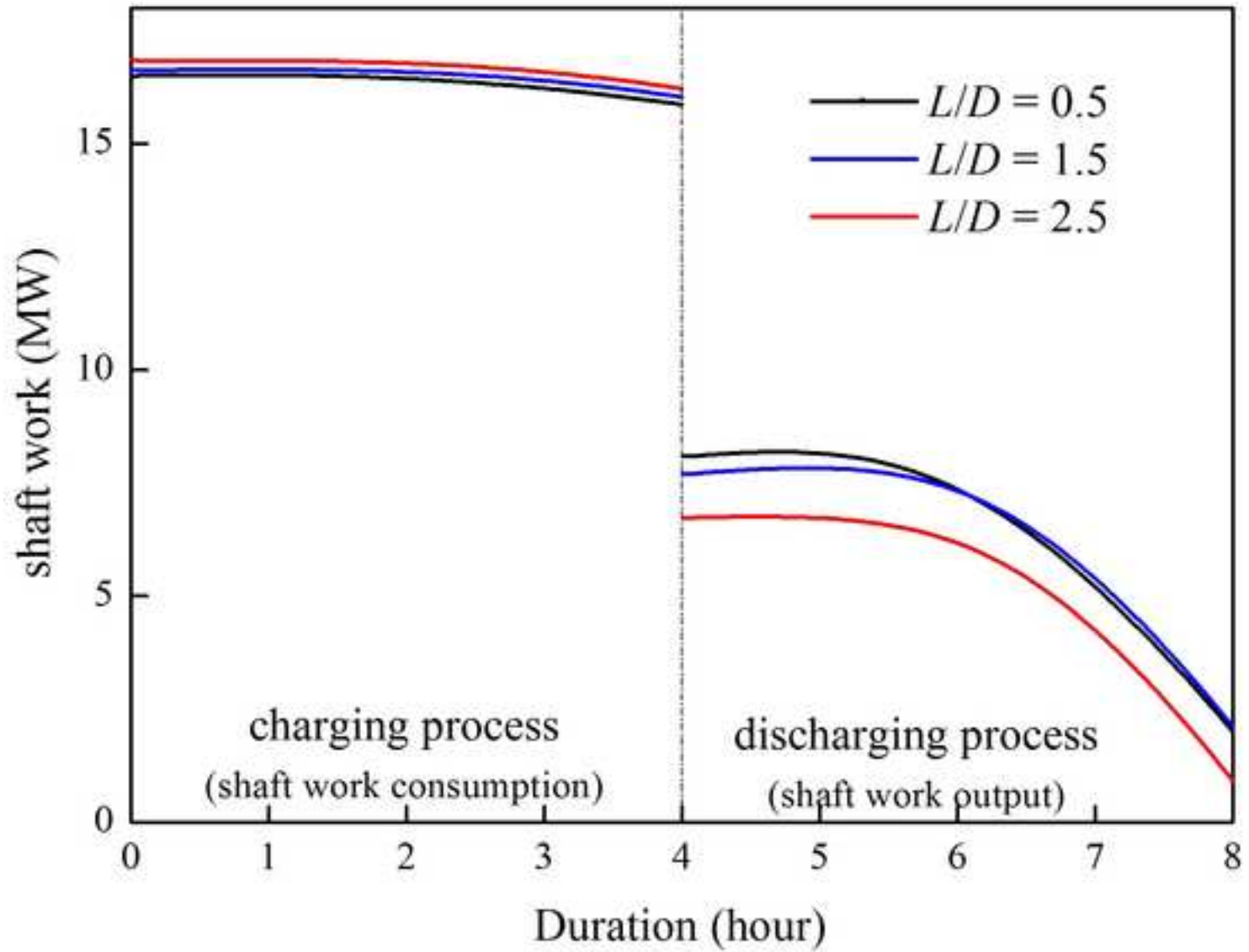




Figure 15

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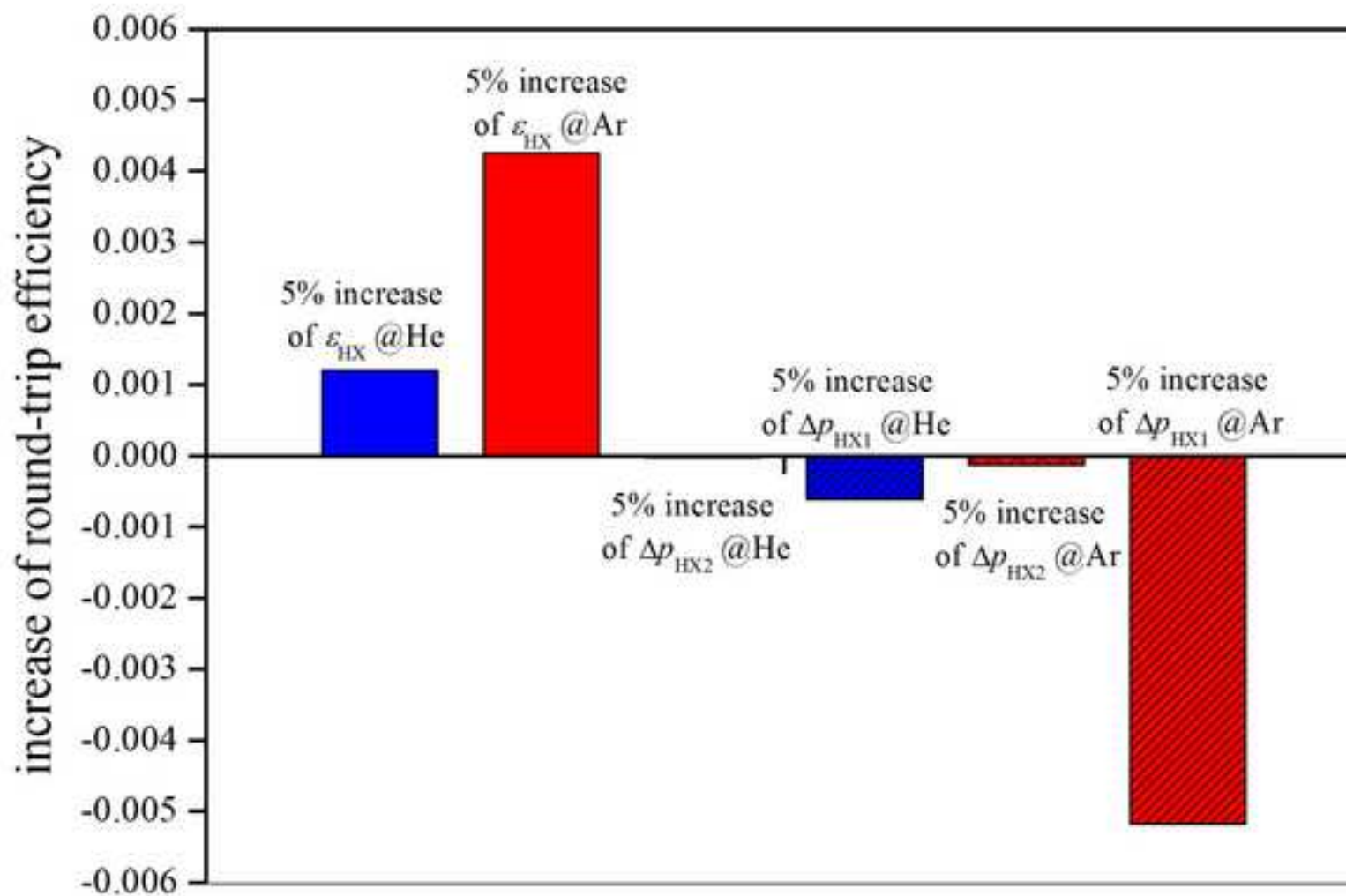


Figure 16a

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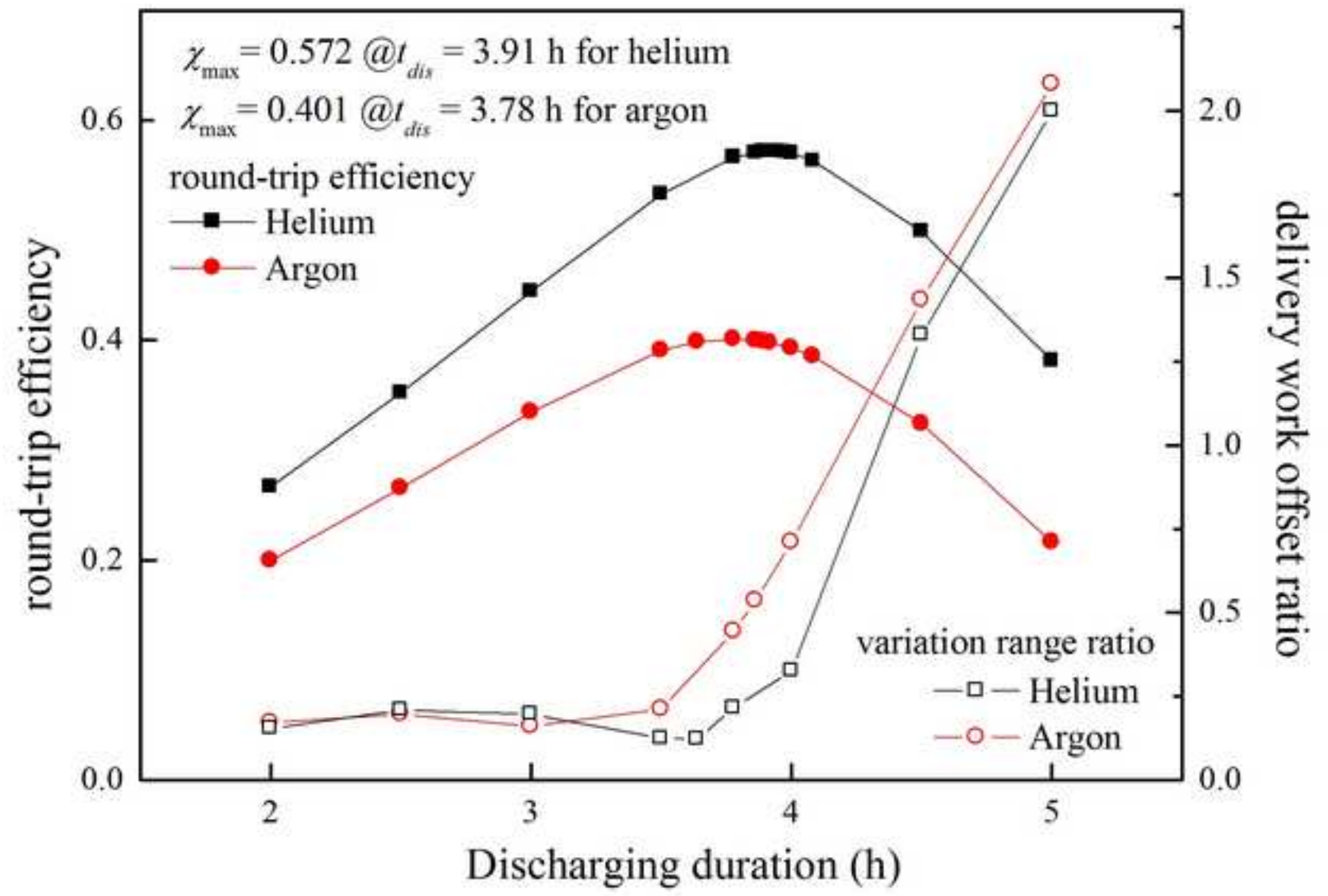


Figure 16b

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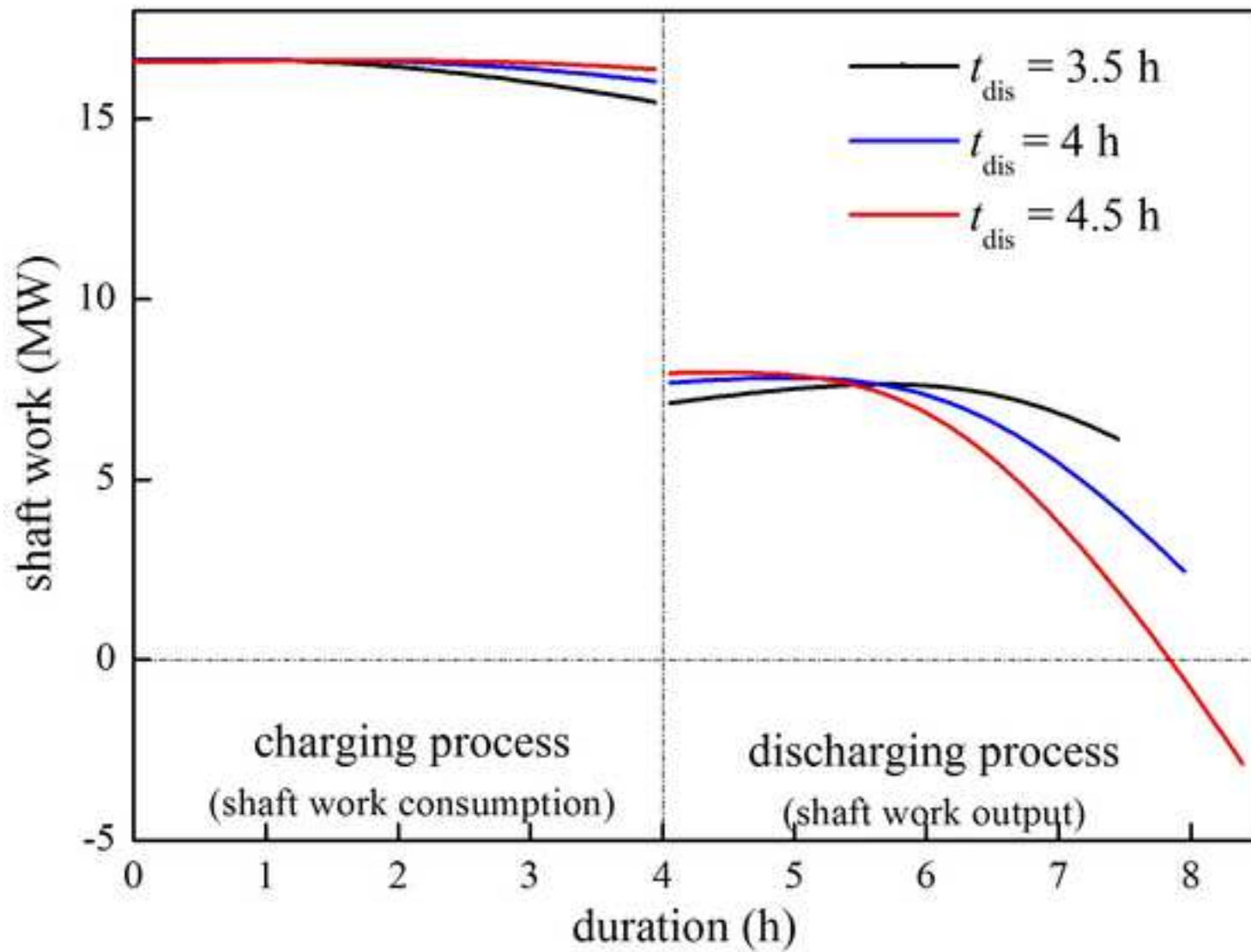


Figure 16c  
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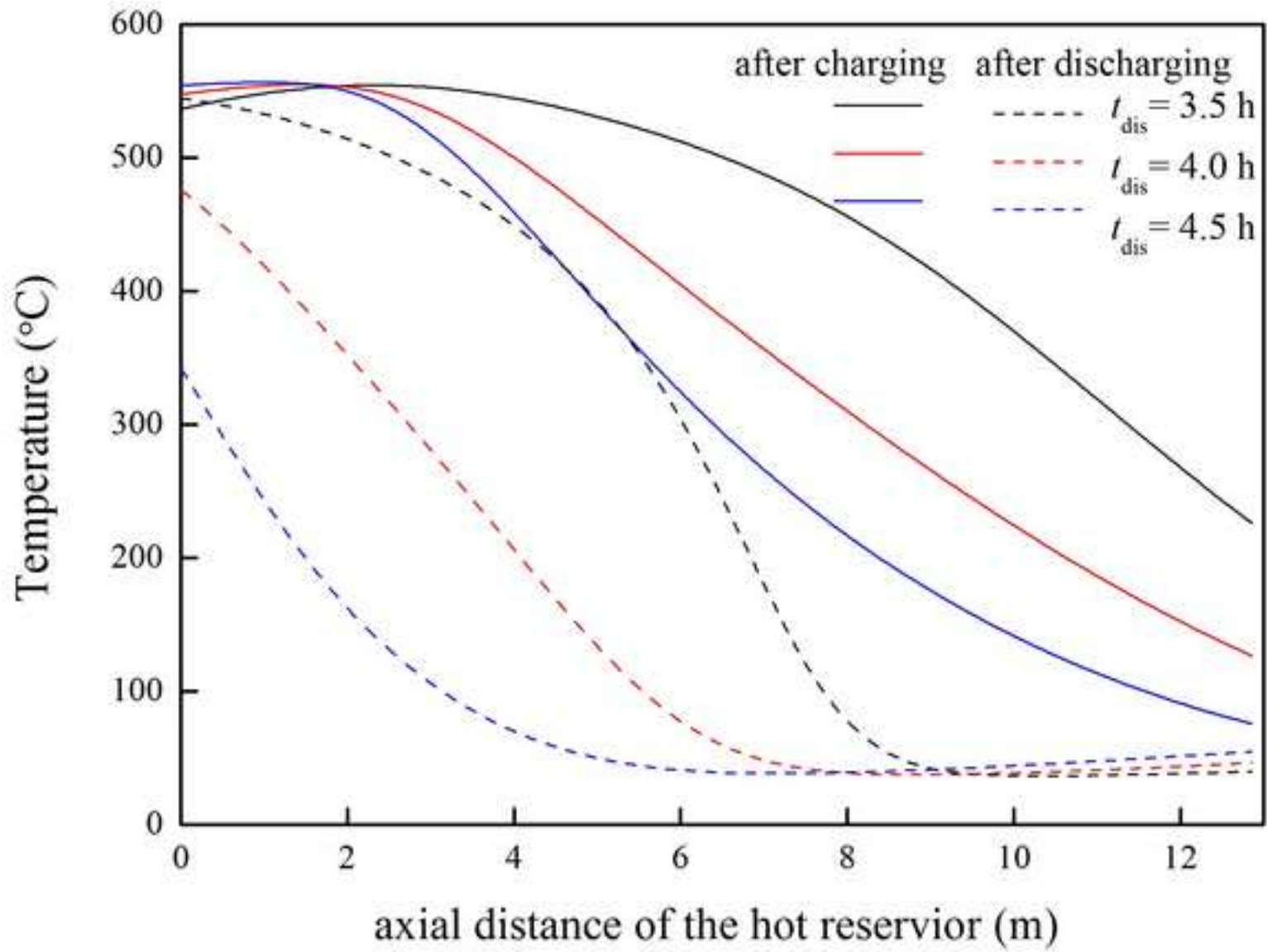


Table 1 Designed parameters of PHES system of 10 MW discharging power

Working gas	HP Pressure (MPa)	LP Pressure (MPa)	Average $c_{p,g}$ (J/kg/K)	Mass flow rate (kg/s)	Polytropic efficiency	$\varepsilon$ of HXs	$\Delta p$ of HP HXs (kPa)	$\Delta p$ of LP HXs (kPa)	Cooling water temperature (K)
Argon	1.05	0.105	525	85.1	0.9	0.9	3	20	300
Helium	1.05	0.105	5193	8.6	0.9	0.9	0.3	2	300

Table 2 Hot and cold reservoir details for 10 MW/4 h PHES system  
(the total volume is twice the minimum design volume)

Reservoir	Pressure (MPa)	Density of solid material (kg/m <sup>3</sup> )	Porosity	Average $d_p$ (mm)	Total Volume (m <sup>3</sup> )	$L$ (m)	$D$ (m)
Heat	1.05	5175	0.35	30	460	10.96	7.31
Cold	0.105	5175	0.35	30	740	12.86	8.56

## Highlights

- The transient analysis method for PTES system is proposed.
- The cyclic transient of 10MW/4h Joule-Brayton PTES is studied.
- Both the round-trip efficiency and delivery stability of the PTES are discussed.
- Helium has the overwhelming advantage above argon as the working gas.
- Impact of particle sizes and length to diameter ratio of packed bed was analyzed.