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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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shaft speed during the transient process have been presented in the paper where the mass flow rate is set to a constant value in this study, please see equation (6).

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 ω_{des} and Q_{des} are the angular velocity and the volume flow rate under the design condition, respectively."

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

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

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

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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
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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 m³ and 110 MW/26 h CAES in McIntosh, Alabama,
67 USA, with a cavern of approximately 500,000 m³ [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 CO₂ Rankine cycle, organic Rankine
82 cycles (ORCs), and subcritical steam Rankine cycle. Morandin et al. studied a TEES system
83 based on a transcritical CO₂ 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 CO₂ Rankine cycle wherein water was sprayed to cool/heat transcritical CO₂ 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 CO₂ heat pump cycle
93 during charging and the cascaded system of the transcritical CO₂ Rankine cycle and the subcritical
94 NH₃ 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].

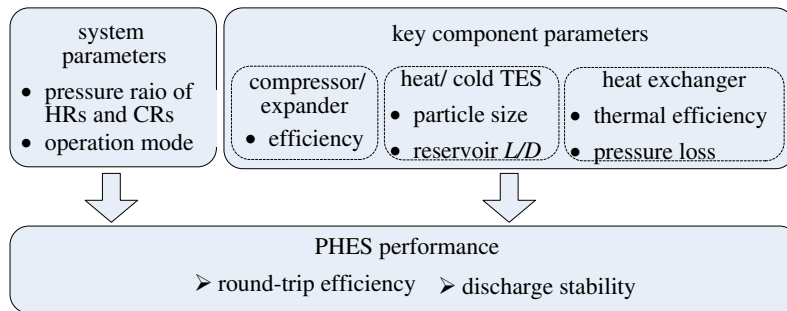
105 Using a single-phase gas as the working fluid, the Joule–Brayton-cycle based PHES
106 generally consists of cold (low-pressure) thermal energy storage (TES) reservoirs, hot
107 (high-pressure) TES reservoirs, and compressor–turbine-pairs, wherein the CRs and HRs are

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

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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

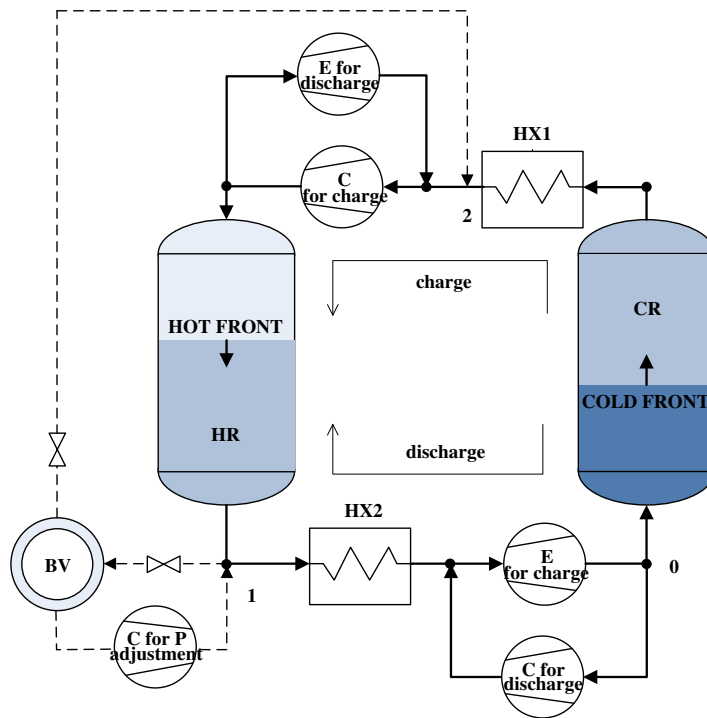
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charging/discharging to balance the gas mass changes in the two reservoirs. During the charging

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



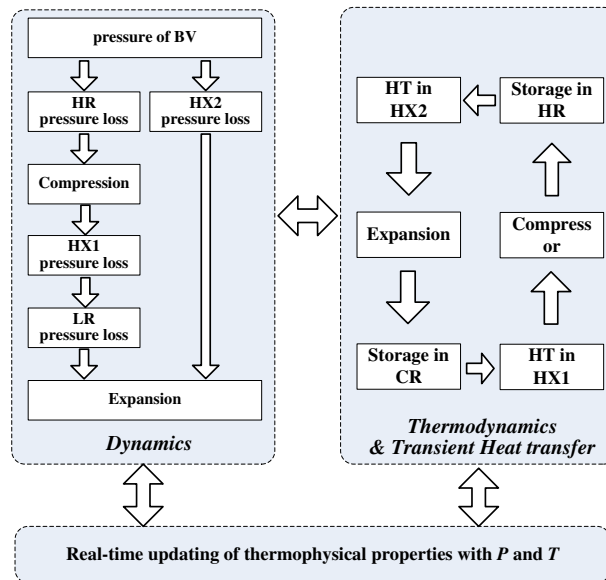
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183 Fig.2. Layout of the PHES system. BV = buffer vessel; C = compressor; E = expander; HX =
 184 heat exchanger; CR = cold reservoir; HR = hot reservoir.

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. Δp
 217 indicates the total pressure loss at each component, and β_c and β_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 η_c and η_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 κ is defined as $\kappa = (\gamma - 1) / \gamma$ and γ 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 ω_{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].

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

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

272

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

273

the i_{th} section, respectively, and ΔL ($\Delta L = L/n$) 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.

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

283 where \dot{m} and c_p are the mass flow rate and heat capacity ~~of the water and gas~~, and ε 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 kgm² 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~~ Under 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/s² and 0.0026 rad/s² 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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$$e_{\text{dis}}(t) = T'_{\text{e,in}}(t) \cdot \left(1 - r'_c(t)^{-k/\eta_c}\right) - T'_{\text{c,in}}(t) \cdot \left(r'_c(t)^{k/\eta_c} - 1\right) \quad (2046)$$

Where e_{chr} and e_{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).

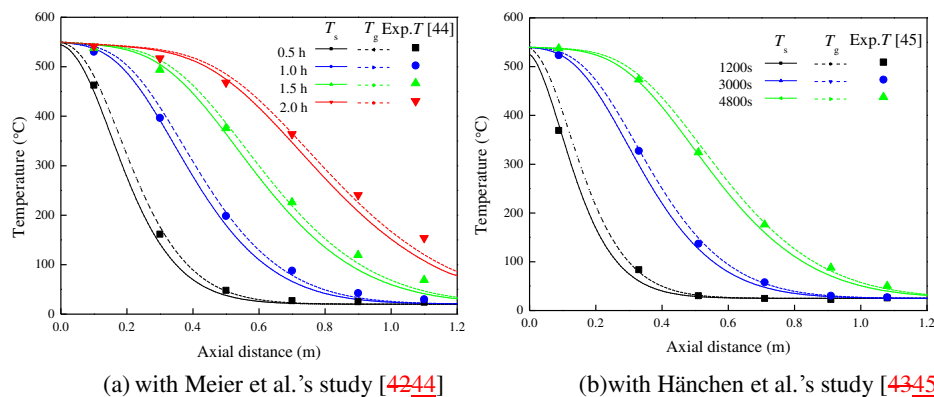
$$\theta = \frac{\text{Max}(e_{\text{dis}}(t)) - \text{Min}(e_{\text{dis}}(t))}{\text{Max}(e_{\text{dis}}(t))} \quad (48)$$

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

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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

Working gas	HP Pressure (MPa)	LP Pressure (MPa)	Average $c_{p,g}$ (J/kg/K)	Mass flow rate (kg/s)	Polytropic efficiency	ϵ of HXs	Δp of HP HXs (kPa)	Δp 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°C – 800°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

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

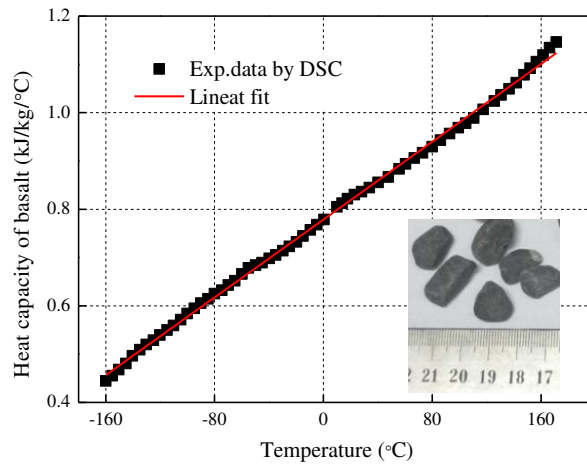


Fig.5. Dependence of heat capacity of basalt with temperature measured using DSC

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 ³)	Porosity	Average d_p (mm)	Total Volume (m ³)	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 ϵ - 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.

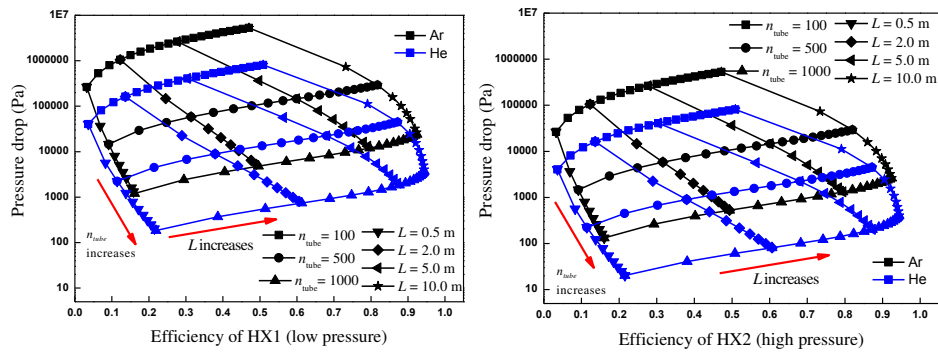


Fig.6. Efficiency versus pressure drop of the shell-and-tube heat exchangers

398

399

400

401 6.4 Result and Discussion

402 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

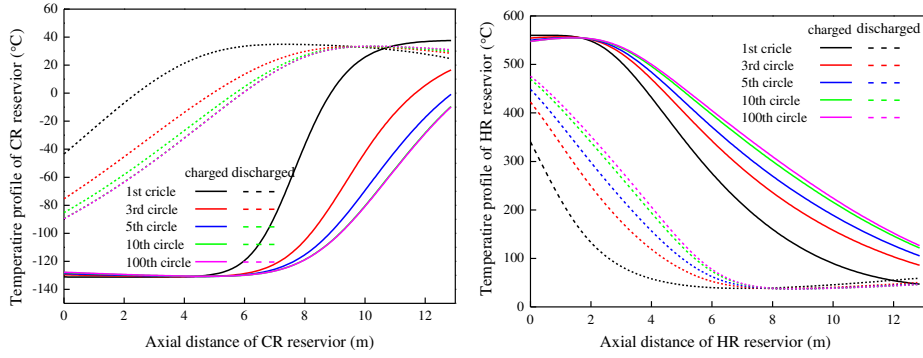
407 the 1st circle to the 100th circle are shown in figures 7(a) and 7(b), respectively. It can be observed

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 °C for all the gases and reservoirs which is deemed cyclically

420

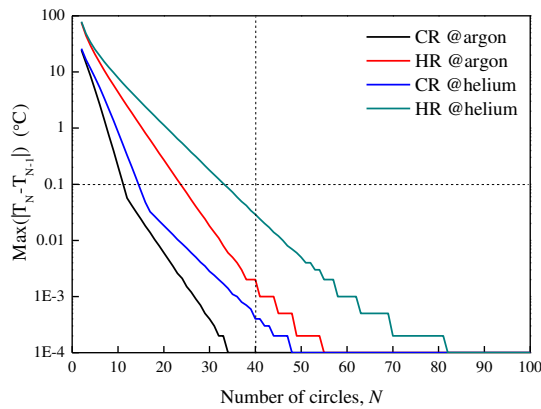
stable. According to this, the following analysis is based on the data of the 40th circles which have

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

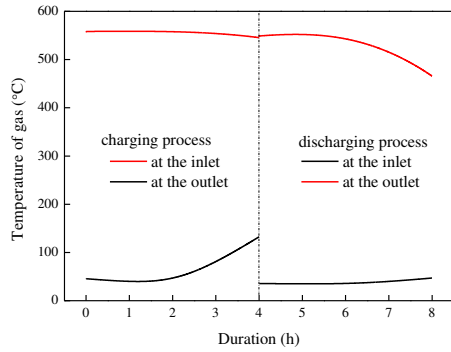
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Fig.8. Maximum temperature differences between circles versus the number of circles

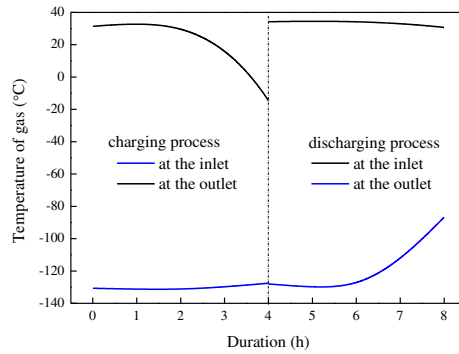
425 Under the cyclic stable state, figures 9(a) and 9(b) show the transient variation of the inflow
426 and outflow temperatures of the HR and CR during the charging and discharging, respectively,
427 when using argon as the working gas. This shows that the outflow temperature from the HR
428 increases 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 χ and the delivery working offset ratio θ using argon as the working gas is
446 39.3% and 71.0%, respectively, and the round-trip efficiency χ and delivery working offset ratio θ

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

448

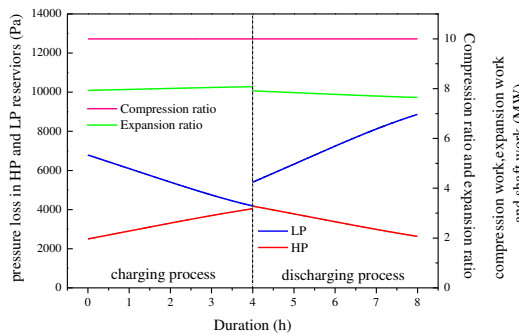


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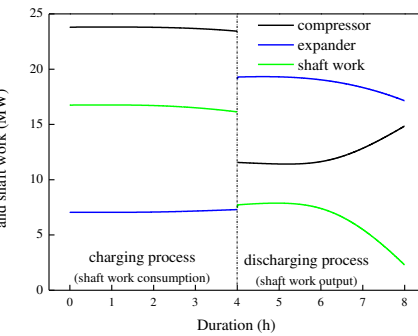
450 (a) inflow and outflow temperature of HP reservoir

(b) inflow and outflow temperature of LP reservoir



451

(c) pressure loss of the HP and LP reservoirs



452

(d) transient power variation of PHES

453

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

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

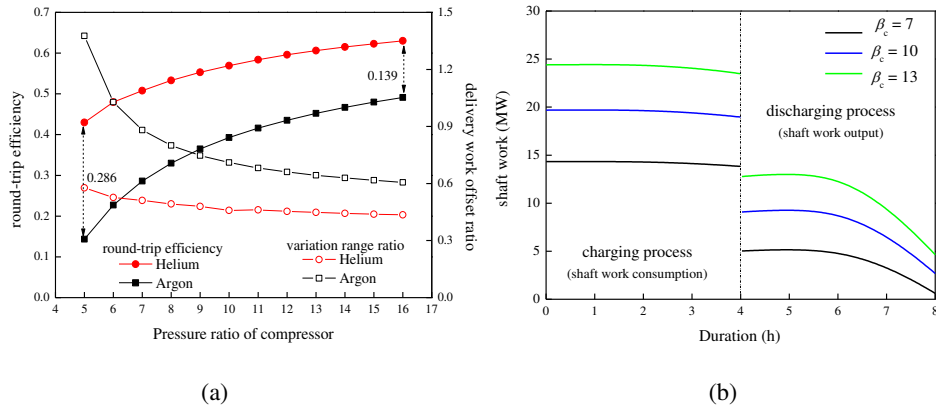
459 ~~6.4.2.1~~ Effect of compression ratio during charging and discharging

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 χ and the delivery working offset ratio θ 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 β_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 θ decreases with the compression ratio β_c , and the offset ratio θ 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 β_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 β_c of 16 for both argon and
 483 helium, which is within the permitted temperature range.



484
 485

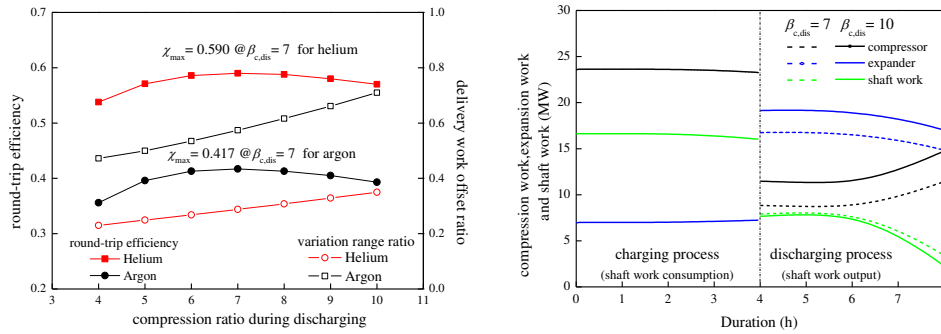
486 Fig.10. Impact of compression ratio during both charging and discharging

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 χ and the delivery
 495 working offset ratio θ . This result indicates that the round-trip efficiency χ increased
 496 first and then decreased with the discharging compress ratio and the maximum round-trip
 497 efficiency χ occurs at the discharging compress ratio of 7 for both argon and helium, the maximum
 498 round-trip efficiency χ 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 θ

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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.



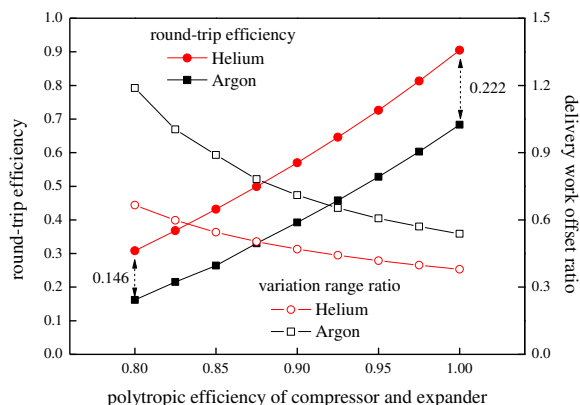
(a) (b)

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 χ 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 χ , 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 θ in figure 11 shows that the increase in the polytropic efficiency also
 519 improves the stability of the delivery power.



520

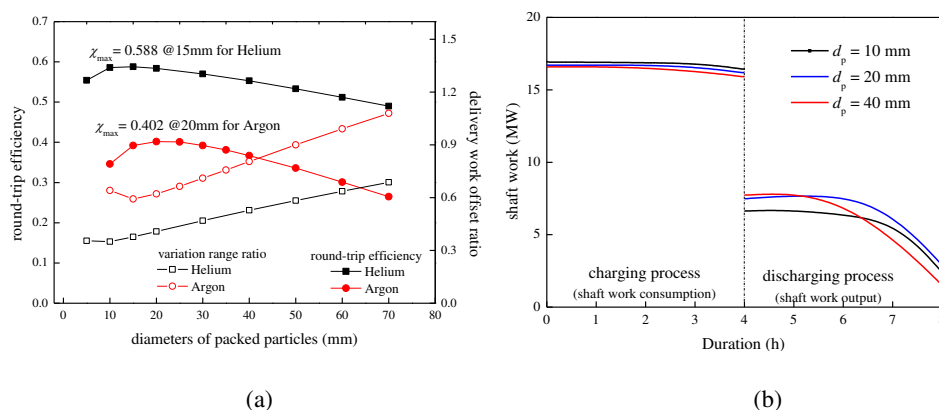
521 Fig.12. Impact of polytropic efficiency of compressor and expander

522 6.2.44.5 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 χ
 526 and the delivery working offset ratio θ . It can be observed that, the round-trip efficiency χ 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 θ . 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 20mm 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.



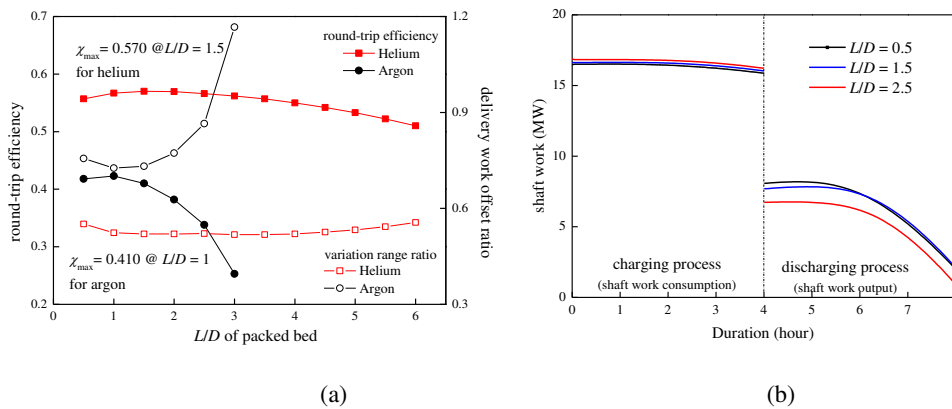
541
542 (a) (b)
543 Fig.13. Impact of particle diameter of compressor and expander

544 **6.2.5.4.6** Effect of length-to-diameter ratio of reservoirs

545 As described in section 5, the volume of the designed HR and CR is 460 m^3 and 740 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 χ and the delivery working offset ratio θ 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 χ 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 $L/D = 1.5$.



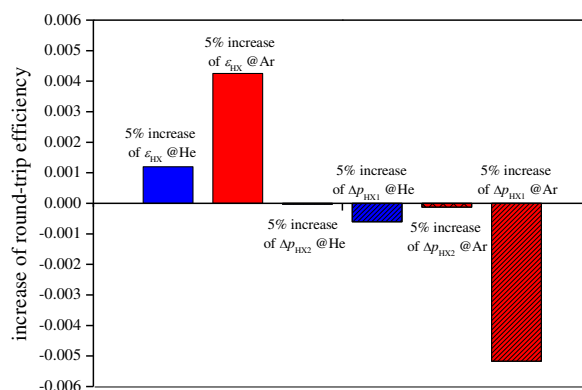
565 (a) 566 (b) 567 Fig.14. Impact of L/D of packed bed reservoirs

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**

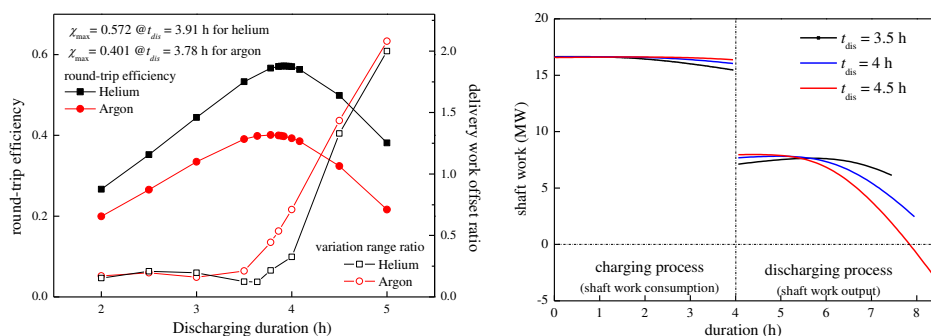
580 In the above analysis, each energy storage circle comprise a charging process of 4 h and a
 581 discharging process of 4 h; however, an equal discharging and charging duration may not be
 582 optimal for such a PHES system. Figure 16(a) shows the influence of the discharging time ranging
 583 from 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 χ and the delivery working offset ratio θ using argon and helium, respectively.
 585 From figure 15(a), it can be observed that the round-trip efficiency χ 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 θ 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.

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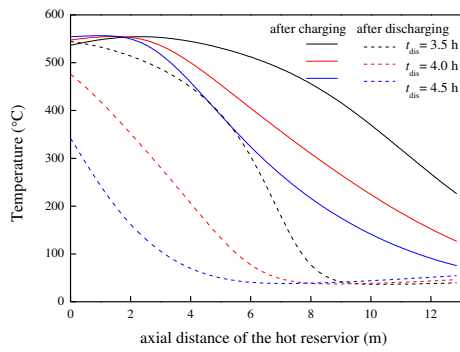


603

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 χ 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

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743
 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 ⁻¹ kg ⁻¹
<i>d</i>	Diameter of particles, m
<i>D</i>	Diameter of packed bed reservoir, m
<i>e</i>	Specific energy, J kg ⁻¹
<i>G</i>	Mass flow rate, kg s ⁻¹
<i>h</i>	Volumetric heat transfer coefficient, W m ⁻³ K ⁻¹

i	Number i	771
I	<u>Moment of inertia, kg m^2</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, $\text{m}^3 \text{s}^{-1}$</u>	
Re	Reynolds number	
t	Time, s	
T	Temperature, K	
β	Compression/expansion ratio of compressor/expander	
γ	Adiabatic exponent of gas	
ε	Efficiency of heat exchanger	
η	Polytropic efficiency of compressor/expander	
θ	Offset ratio of delivery power	
κ	Parameter, $(\gamma-1)/\gamma$	
μ	Dynamic viscosity, Pa s	
ρ	Density, kg m^{-3}	
Φ	Porosity of packed bed	
X	Round-trip efficiency	
ω	<u>Angular velocity, rad s^{-1}</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

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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53 62 The Bath County Pumped Storage Station in the USA is the largest PHS power station in the
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56 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 m³ and 110 MW/26 h CAES in McIntosh, Alabama,
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9 67 USA, with a cavern of approximately 500,000 m³ [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 CO₂ 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 CO₂ 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 CO₂ Rankine cycle wherein water was sprayed to cool/heat transcritical CO₂ 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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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].
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17 92 Wang and Zhang proposed and analyzed a PHES based on the transcritical CO₂ heat pump cycle
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20 93 during charging and the cascaded system of the transcritical CO₂ Rankine cycle and the subcritical
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23 94 NH₃ 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’ polytrophic 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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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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46 124 analyzed [30,31].

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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23 138 exchangers, and (5) argon or air is used as the working fluid.

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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
53 149 figure 1.

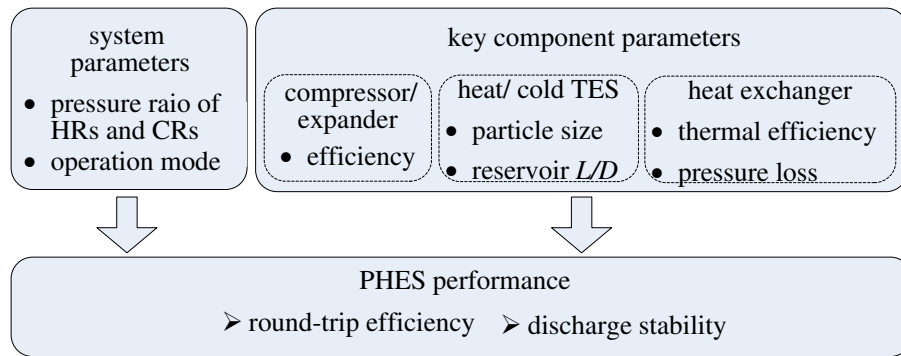


Fig.1. Parameters influencing on PHEs performance

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.

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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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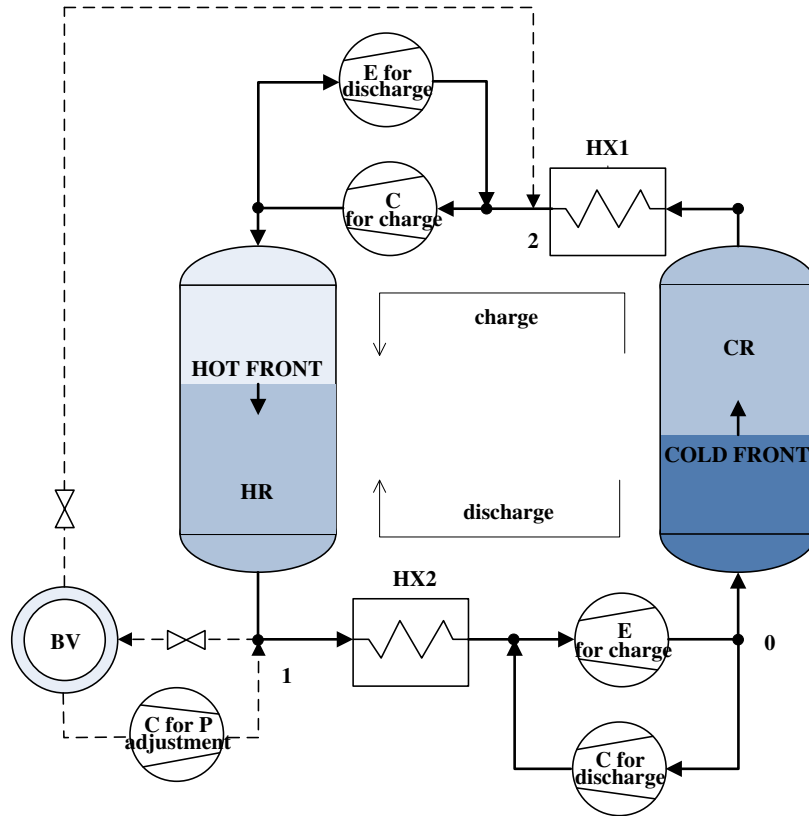


Fig.2. Layout of the PHES system. BV = buffer vessel; C = compressor; E = expander; HX = heat exchanger; CR = cold reservoir; HR = hot reservoir.

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.

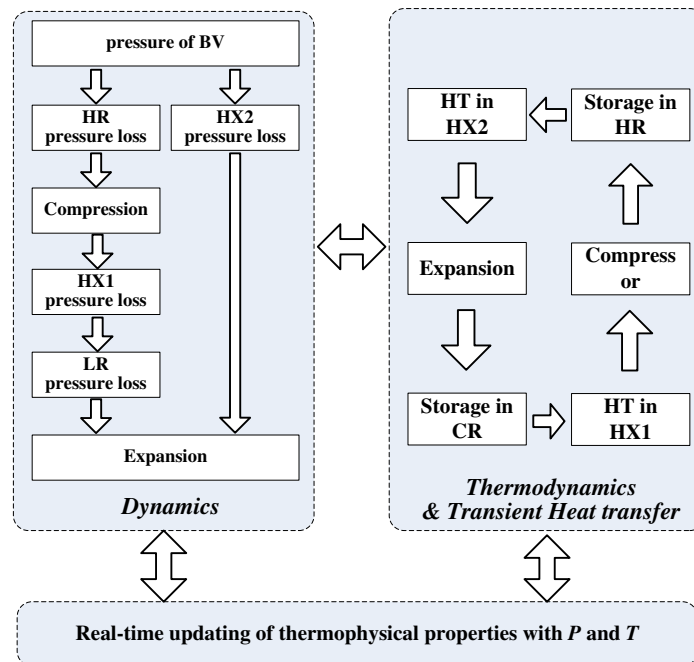


Fig.3. Coupling analysis of PHES during charging process

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. Δp
 indicates the total pressure loss at each component, and β_c and β_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 η_c and η_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 κ is defined as $\kappa = (\gamma - 1) / \gamma$ and γ 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

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

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

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

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

237 Where ω_{des} and Q_{des} are the angular velocity and the volume flow rate under the design condition,
 238 respectively.

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

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

246 For the fluid phase,

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

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$$248 \quad \frac{\partial T_g}{\partial t} + \frac{G}{\rho_g \varphi} \frac{\partial T_g}{\partial x} = \frac{h_v}{\rho_g c_{p,g} \varphi} (T_s - T_g) \quad (8)$$

249 For the solid phase,

$$250 \quad \frac{\partial T_s}{\partial t} = \frac{h_{v,\text{eff}}}{\rho_s c_s (1-\varphi)} (T_g - T_s) + \frac{k_{s,\text{eff}}}{\rho_s c_s (1-\varphi)} \frac{\partial^2 T}{\partial x^2} \quad (9)$$

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

$$256 \quad h_{v,\text{eff}} = \begin{cases} h_v & \text{for } Bi \leq 0.1 \\ \frac{1}{\frac{1}{h_v} + \frac{d_p^2}{60k_s(1-\varphi)}} & \text{for } 0.1 < Bi \leq 100 \end{cases} \quad (10)$$

$$257 \quad h_v = 1.45 \frac{Re^{0.7} k_g}{d^2} \quad (11)$$

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

$$259 \quad Bi = \frac{h_p d_p}{6k_s} \quad (12)$$

260 $k_{s,\text{eff}}$ is the effective thermal conductivity for the non-contiguous spherical particles in a
261 dispersion medium given by [38, 39]:

$$262 \quad \frac{k_s - k_{s,\text{eff}}}{k_s - k_g} \left(\frac{k_{s,\text{eff}}}{k_g} \right)^{\frac{1}{3}} = \varphi \quad (13)$$

263 which is solved by performing iteration.

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

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

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the i_{th} section, respectively, and ΔL ($\Delta L = L/n$) 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 ε 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 kgm² 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/s² and 0.0026 rad/s² 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_{chr} and e_{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)

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

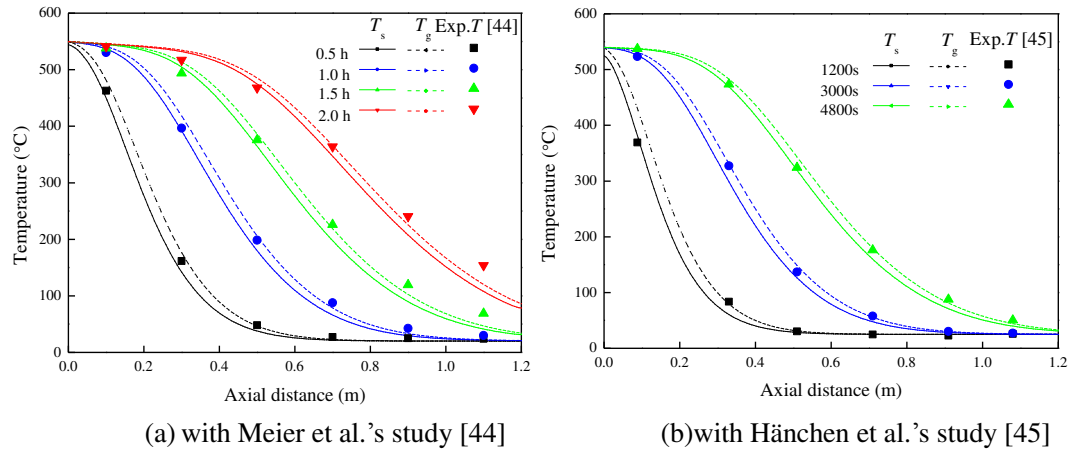
312 where \dot{m} is the mass flow rate through the compressors and expanders.

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.

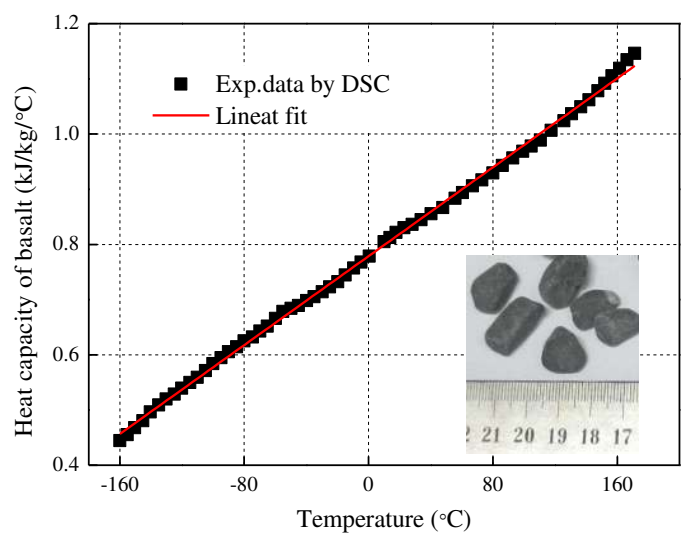
342 Table 1 Designed parameters of PHES system of 10 MW discharging power

Working gas	HP Pressure (MPa)	LP Pressure (MPa)	Average $c_{p,g}$ (J/kg/K)	Mass flow rate (kg/s)	Polytropic efficiency	ε of HXs	Δp of HP HXs (kPa)	Δp 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

1 345 basalt particles. The packed-bed TES is unstable and has a larger packed bed volume, which
 2
 3 346 results in a more stable output temperature but a higher cost and lower energy storage density. In
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 5
 6 347 consideration of the thermal front volume, the designed volumes of the HR and CR are selected to
 7
 8
 9 348 be twice the minimum design volume obtained using from the energy balance method
 10
 11 349 $V = 2Q / (\overline{\rho_s c_s} \Delta T)$. The detailed parameters of the HR and CR are shown in table 2. In this design,
 12
 13
 14 350 the basalt is chosen as the hot and cold TES material, as it has a good heat capacity and thermal
 15
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 17 351 stability within the temperature range of -196° C–800° C. Based on the TA Q2000 DSC, the heat
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 19
 20 352 capacity of basalt is found to be strongly dependent on the temperature as shown in figure 5, and
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 22
 23 353 the linear fit equation is given in equation (23).

24
 25 354
$$c_p(T) = 0.23 + 0.00201 \cdot T \tag{23}$$



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

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

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Reservoir	Pressure (MPa)	Density of solid material (kg/m ³)	Porosity	Average d_p (mm)	Total Volume (m ³)	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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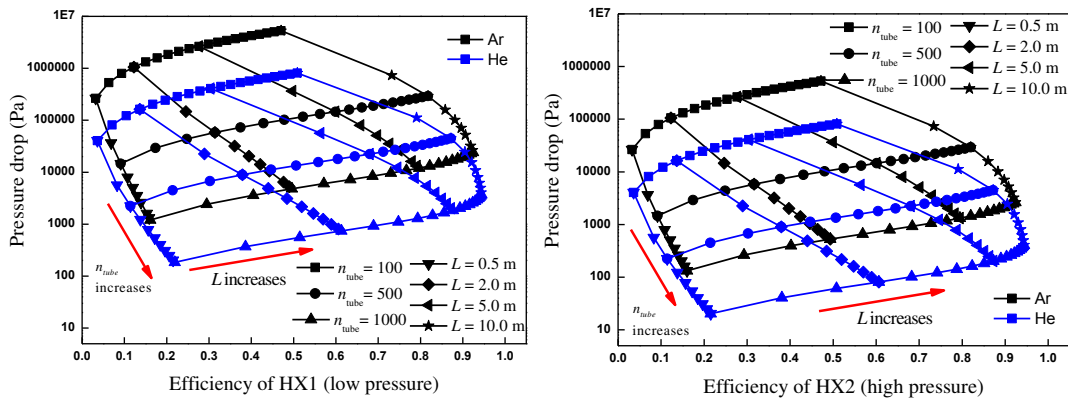
362 3.4.1 Heat exchangers design

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 ε - 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 1st circle to the 100th 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.

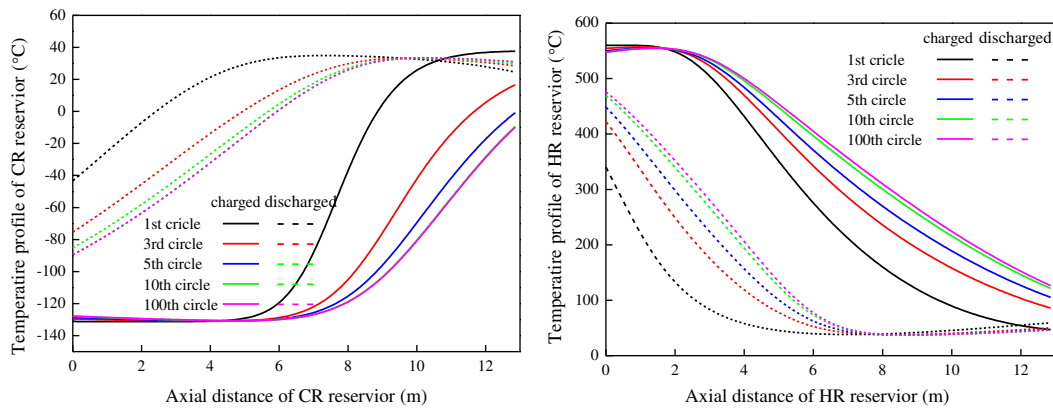
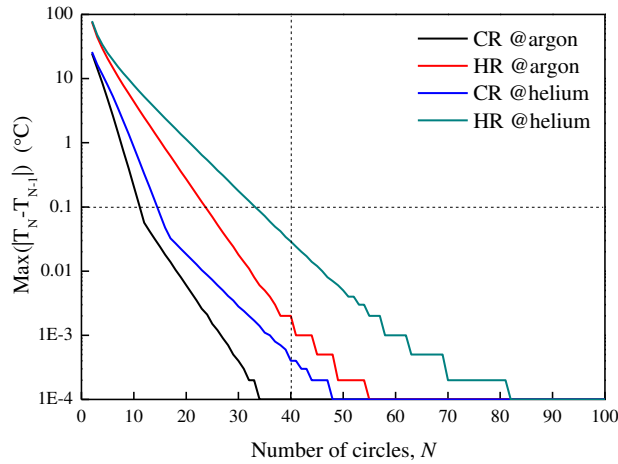


Fig.7. Cyclic behaviors of the HR and CR

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

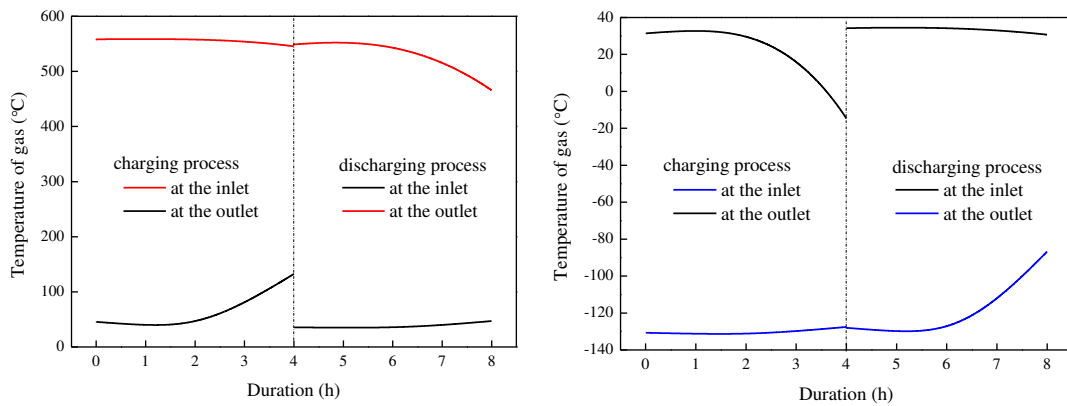
$$\Delta T_{\text{Max}}(N) = \text{Ma} \left(\frac{T_i}{\bar{T}} \right) \quad N=1, 2, 3, \dots \quad (24)$$



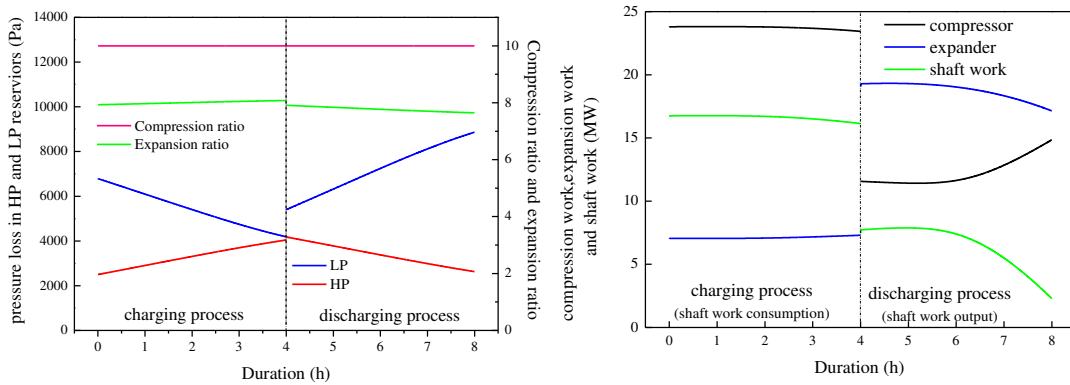
416
417 Fig.8. Maximum temperature differences between circles versus the number of circles

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 χ and the delivery working offset ratio θ using argon as the working gas is
 439 39.3% and 71.0%, respectively, and the round-trip efficiency χ and delivery working offset ratio θ
 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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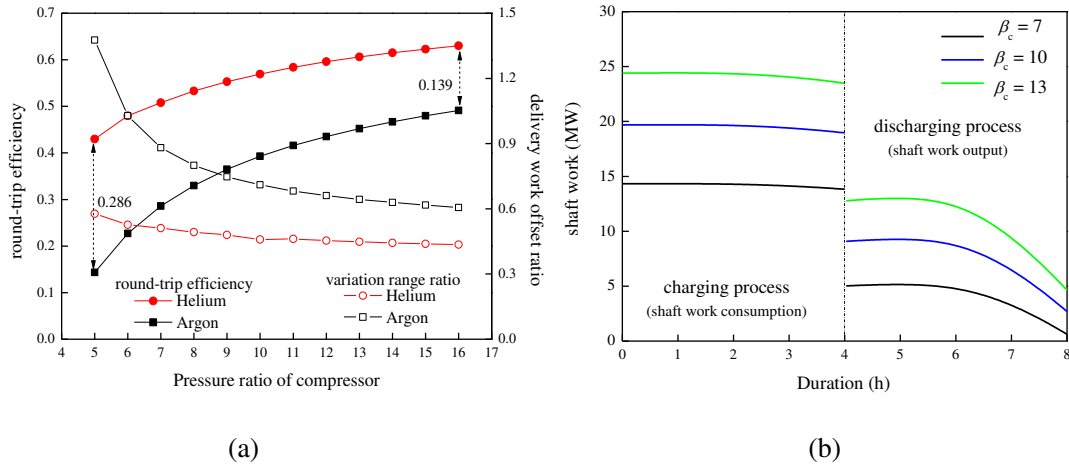
447 4.2 Effect of compression ratio during charging and discharging

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 χ and the
455 delivery working offset ratio θ 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 β_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 θ
462 decreases with the compression ratio β_c , and the offset ratio θ 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 β_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 β_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 χ and the delivery
 483 working offset ratio θ . This result indicates that the round-trip efficiency χ increased
 484 first and then decreased with the discharging compress ratio and the maximum round-trip
 485 efficiency χ occurs at the discharging compress ratio of 7 for both argon and helium, the maximum
 486 round-trip efficiency χ 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 θ
 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.

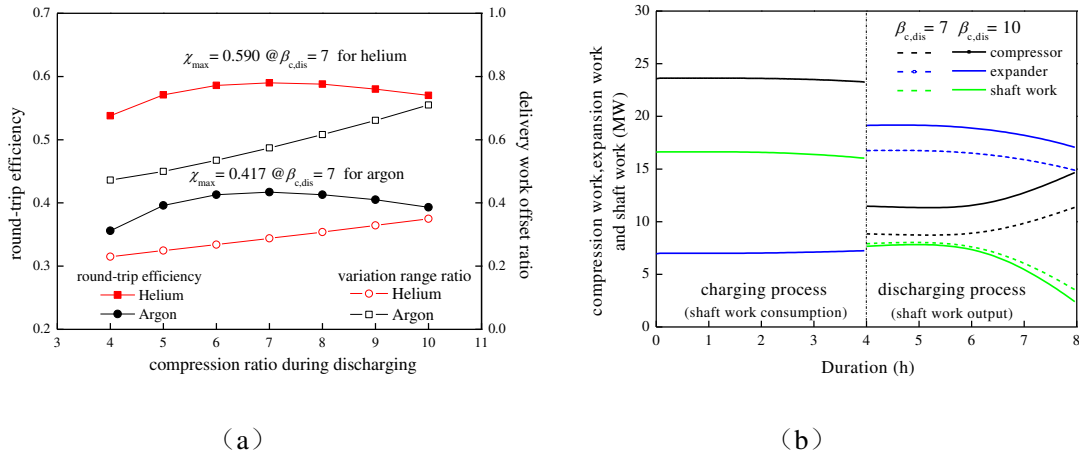


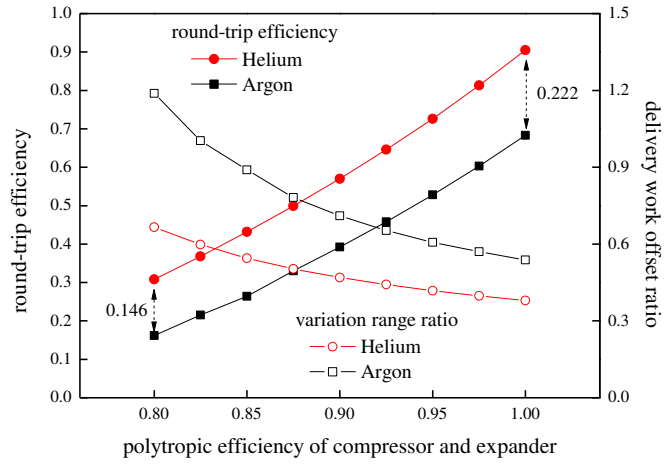
Fig.11. Impact of compression ratio during discharging (at $\beta_{c,chr} = 10$)

4.4 Effect of polytropic efficiency of both compressors and expanders

499 The plots of the round-trip efficiency χ 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 χ , 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 θ 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 χ
 514 and the delivery working offset ratio θ . It can be observed that, the round-trip efficiency χ 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 θ . 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 20mm 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.

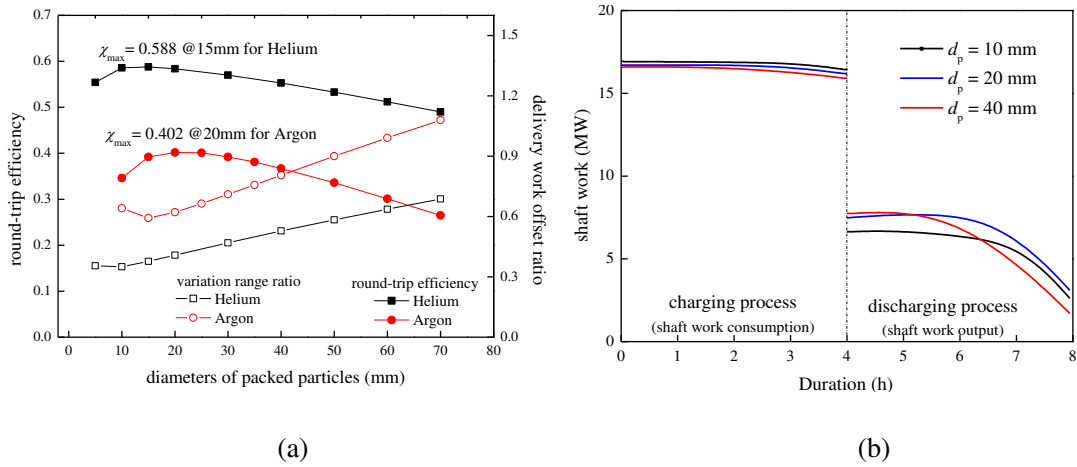
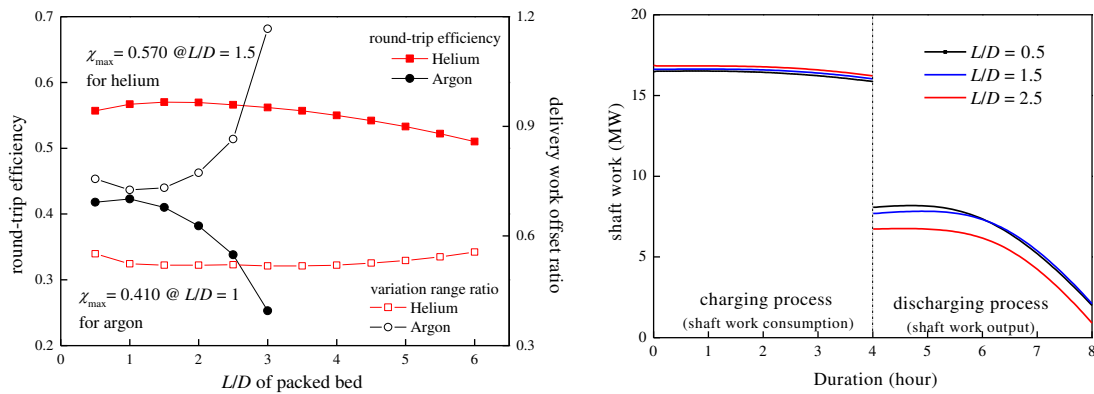


Fig.13. Impact of particle diameter of compressor and expander

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 m^3 and 740 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 χ and the delivery working offset ratio θ 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 χ 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

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

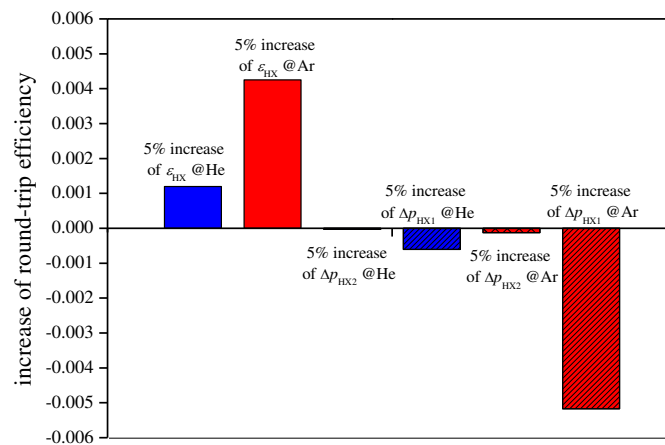


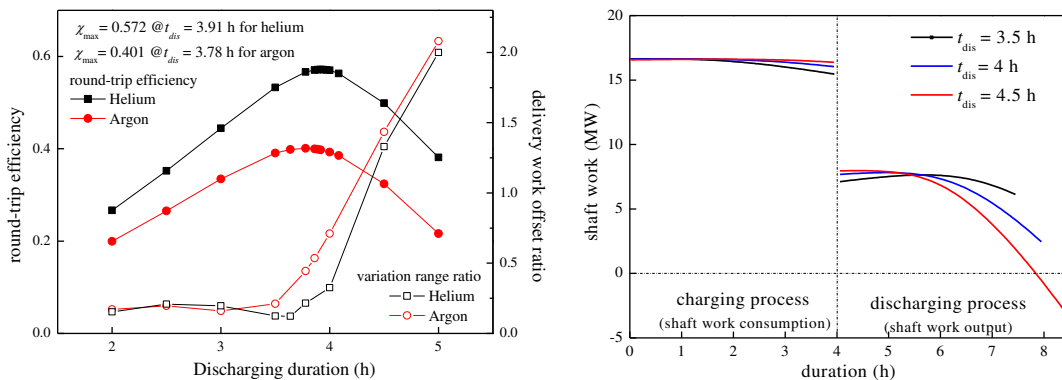
Fig.15. Impact of efficiency and pressure drop of heat exchangers

4.8 Effect of discharging duration

In the above analysis, each energy storage circle comprise a charging process of 4 h and a discharging process of 4 h; however, an equal discharging and charging duration may not be optimal for such a PHES system. Figure 16(a) shows the influence of the discharging time ranging from 2 h to 5 h (one circle consists of a 4 h charging process and 2–5 h discharging process) on the round-trip efficiency χ and the delivery working offset ratio θ using argon and helium, respectively. From figure 15(a), it can be observed that the round-trip efficiency χ increases at first and then decreases with the discharging time. The best selection of the discharging duration is a few

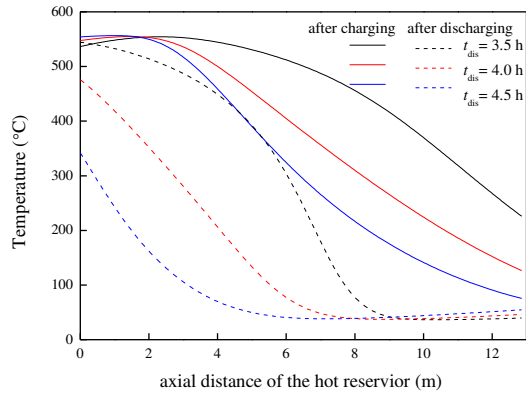
minutes shorter than the charging time such that the maximum round-trip efficiency of 40.1% occurs at the delivery duration of 3.78 h for argon, and the maximum round-trip efficiency is 57.2% at the delivery duration of 3.91 h for helium. The delivery working offset ratio θ is relatively low (<20%) for 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 χ 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
5
6 612 71.0%.

7
8
9 613 3. The increase in the pressure ratio and isentropic efficiencies would lead to an obviously
10
11 614 improvement in the round-trip efficiency and delivery stability. Furthermore, an appropriate
12
13 615 discharging compression ratio that is less than the charging compression ratio will aid in
14
15
16
17 616 improving the round-trip efficiency. For the 10 MW/4 h PHES system, the optimum round-trip
18
19
20 617 efficiency is obtained at the discharging compression ratio of 7 when the charging compression
21
22
23 618 ratio is 10.

24
25 619 4. For the TES reservoirs, there exists optimal selections of particle sizes, ratios of length
26
27
28 620 –to–diameter, and discharging durations corresponding to the maximum round-trip efficiency and
29
30
31 621 preferable discharging power stability; this is mainly owing to the joint effects of the pressure loss,
32
33
34 622 heat transfer and thermodynamics.

35
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.

43
44
45 626

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 ²	758
2	K	Thermal conductivity, W m ⁻¹ K ⁻¹	
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 ³ s ⁻¹	
9	Re	Reynolds number	
10	t	Time, s	
11	T	Temperature, K	
12	β	Compression/expansion ratio of compressor/expander	
13	γ	Adiabatic exponent of gas	
14	ε	Efficiency of heat exchanger	
15	η	Polytropic efficiency of compressor/expander	
16	θ	Offset ratio of delivery power	
17	κ	Parameter, $(\gamma-1)/\gamma$	
18	μ	Dynamic viscosity, Pa s	
19	ρ	Density, kg m ⁻³	
20	Φ	Porosity of packed bed	
21	X	Round-trip efficiency	
22	ω	Angular velocity, rad s ⁻¹	
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	

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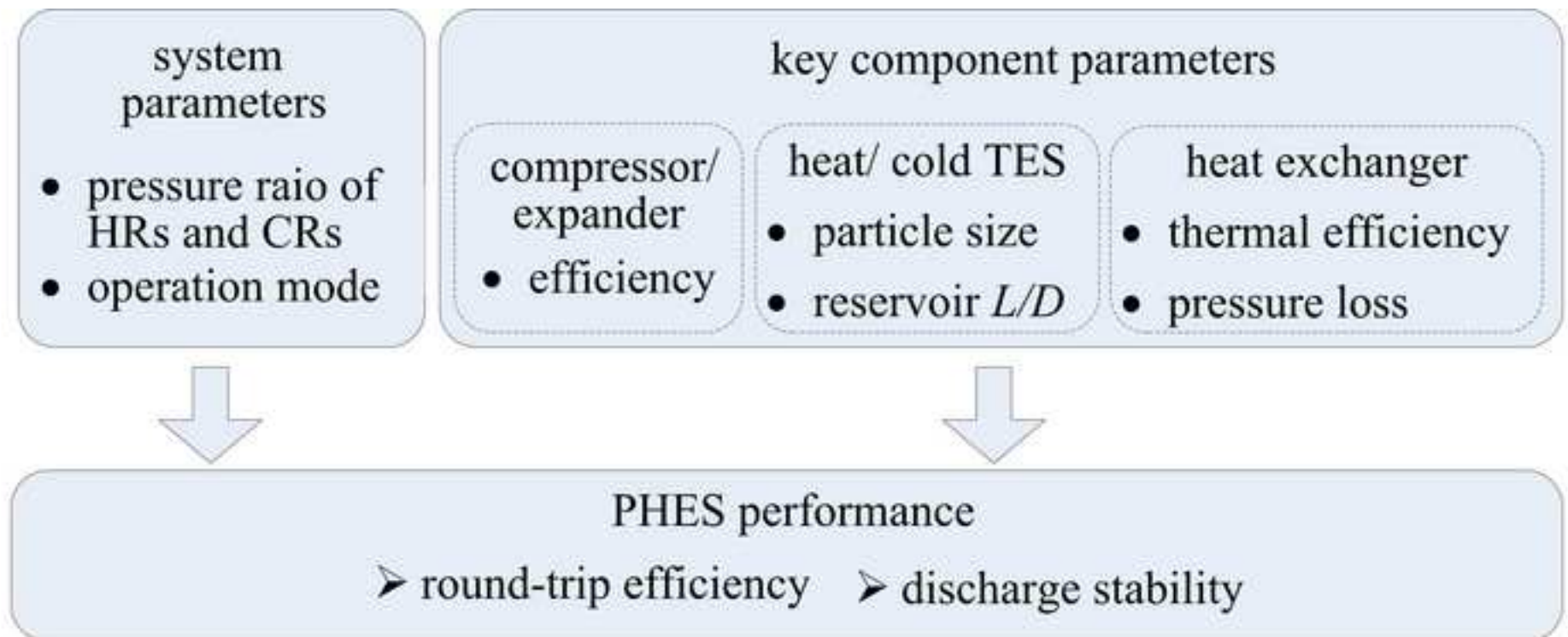


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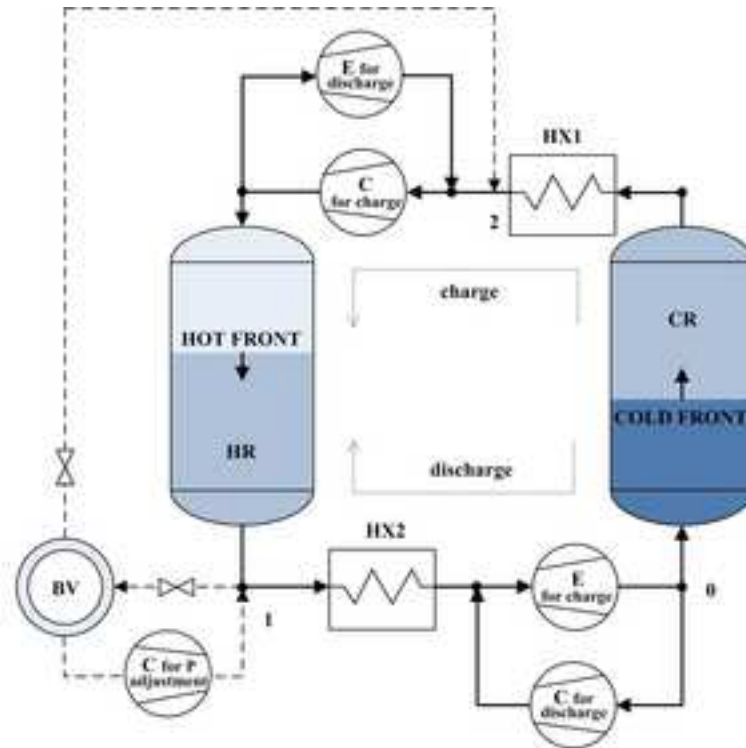


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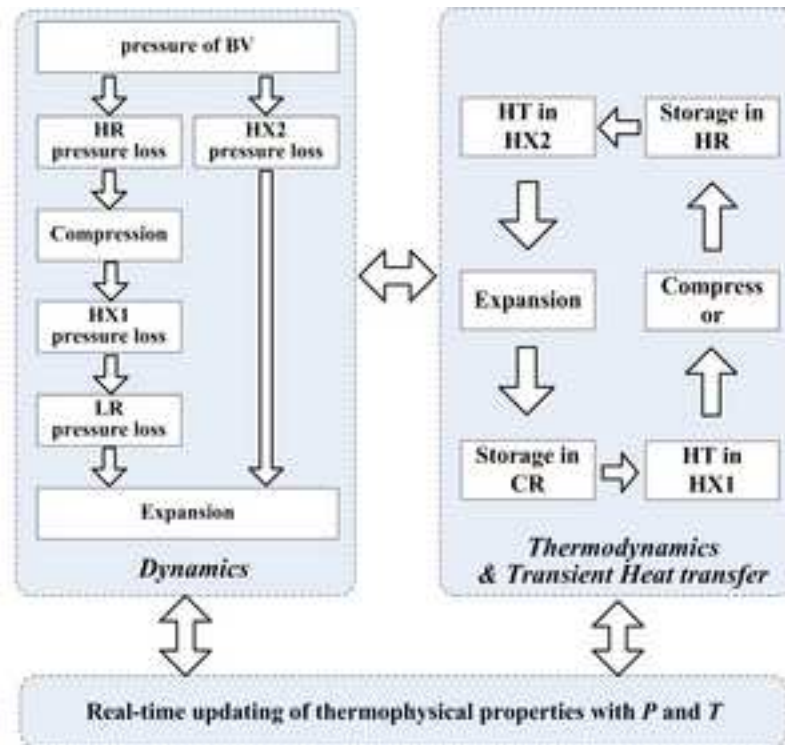


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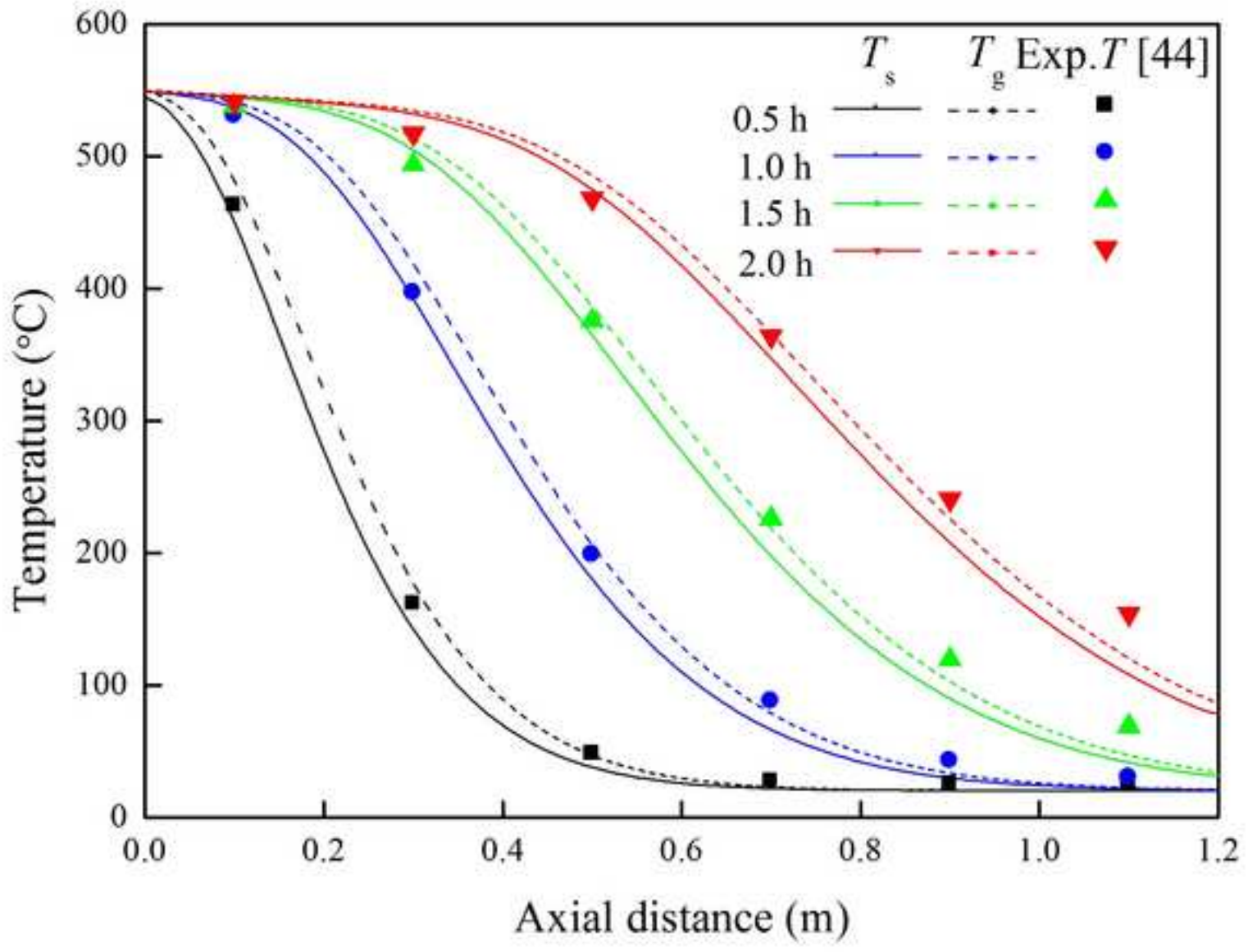


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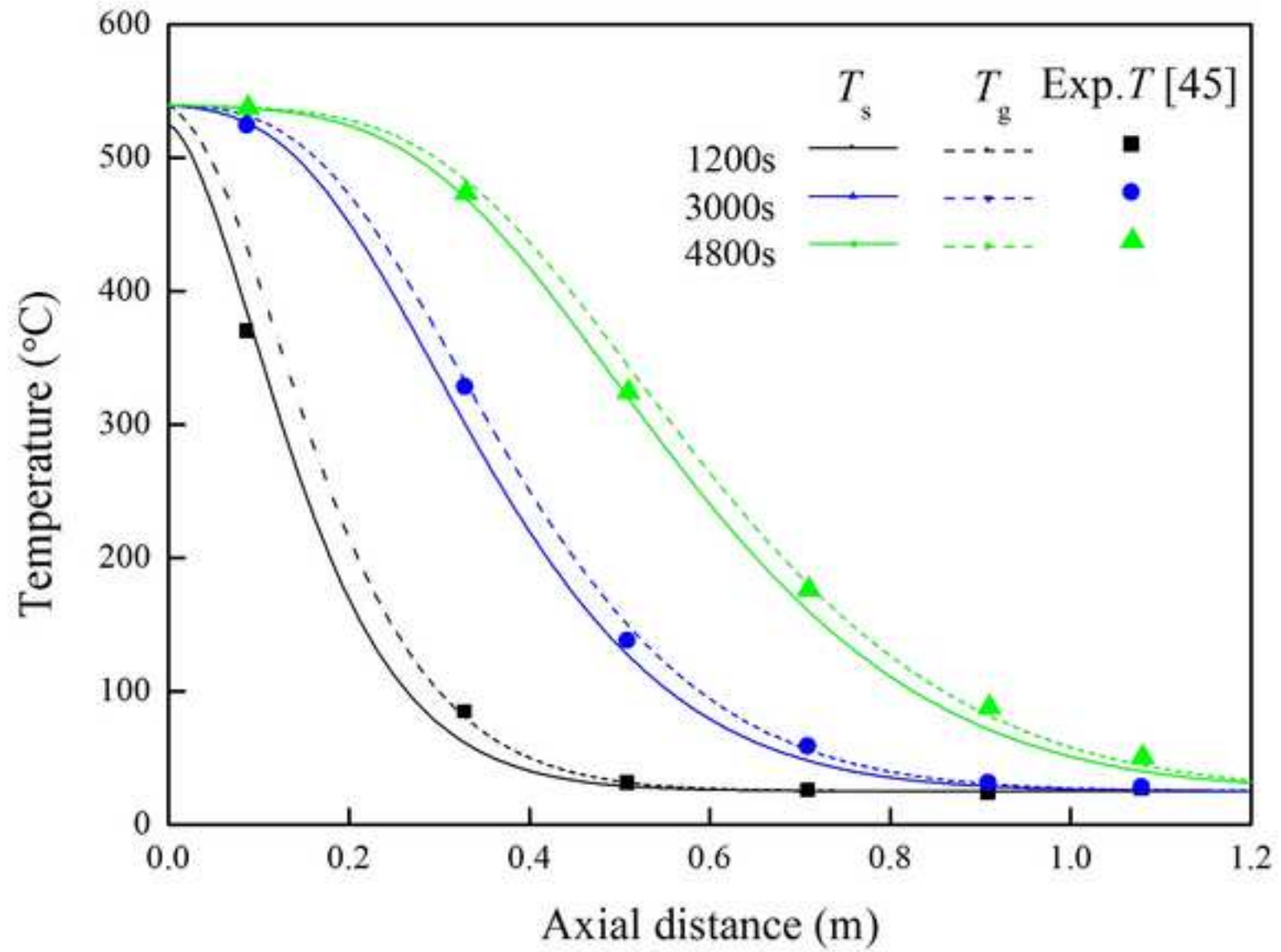


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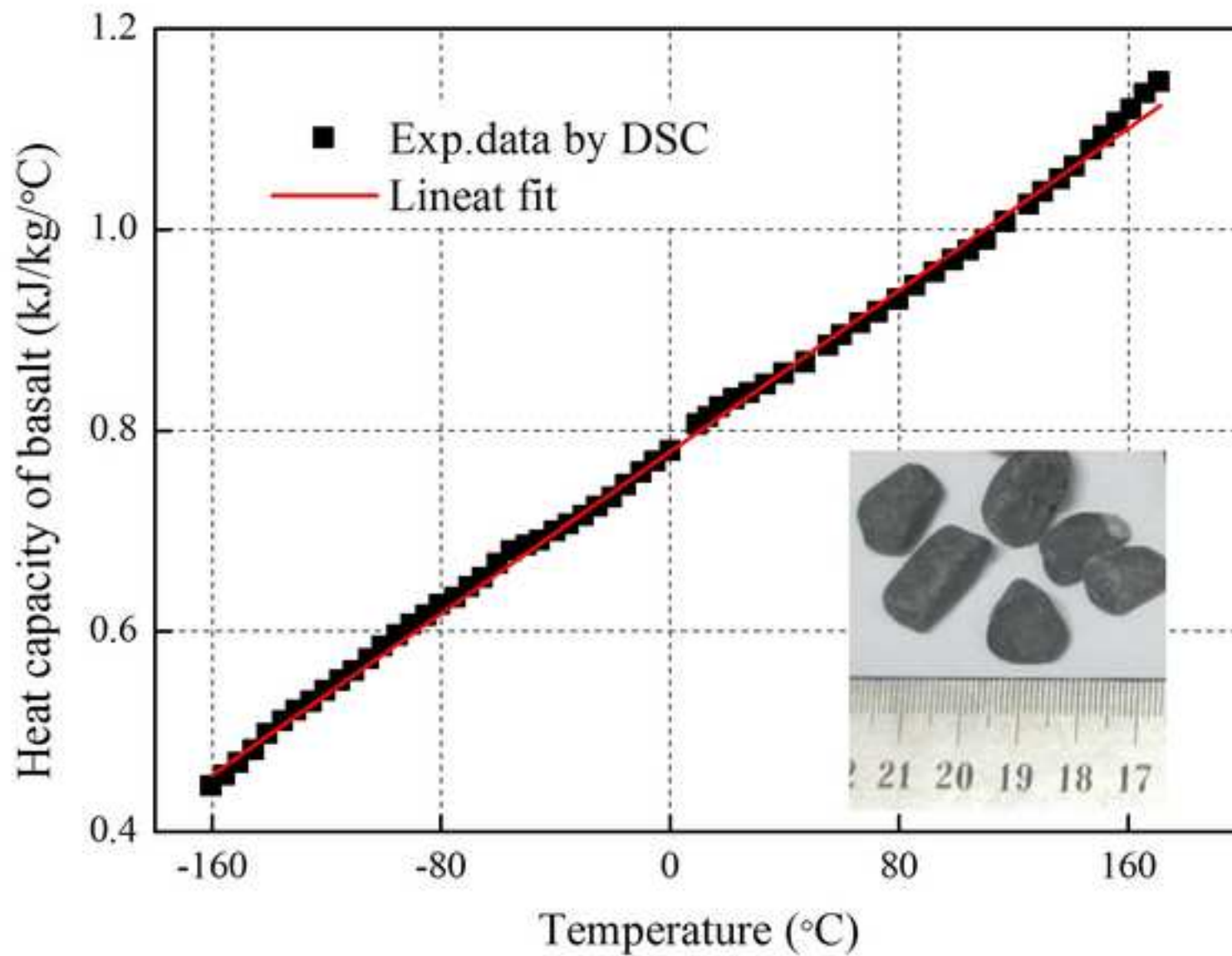


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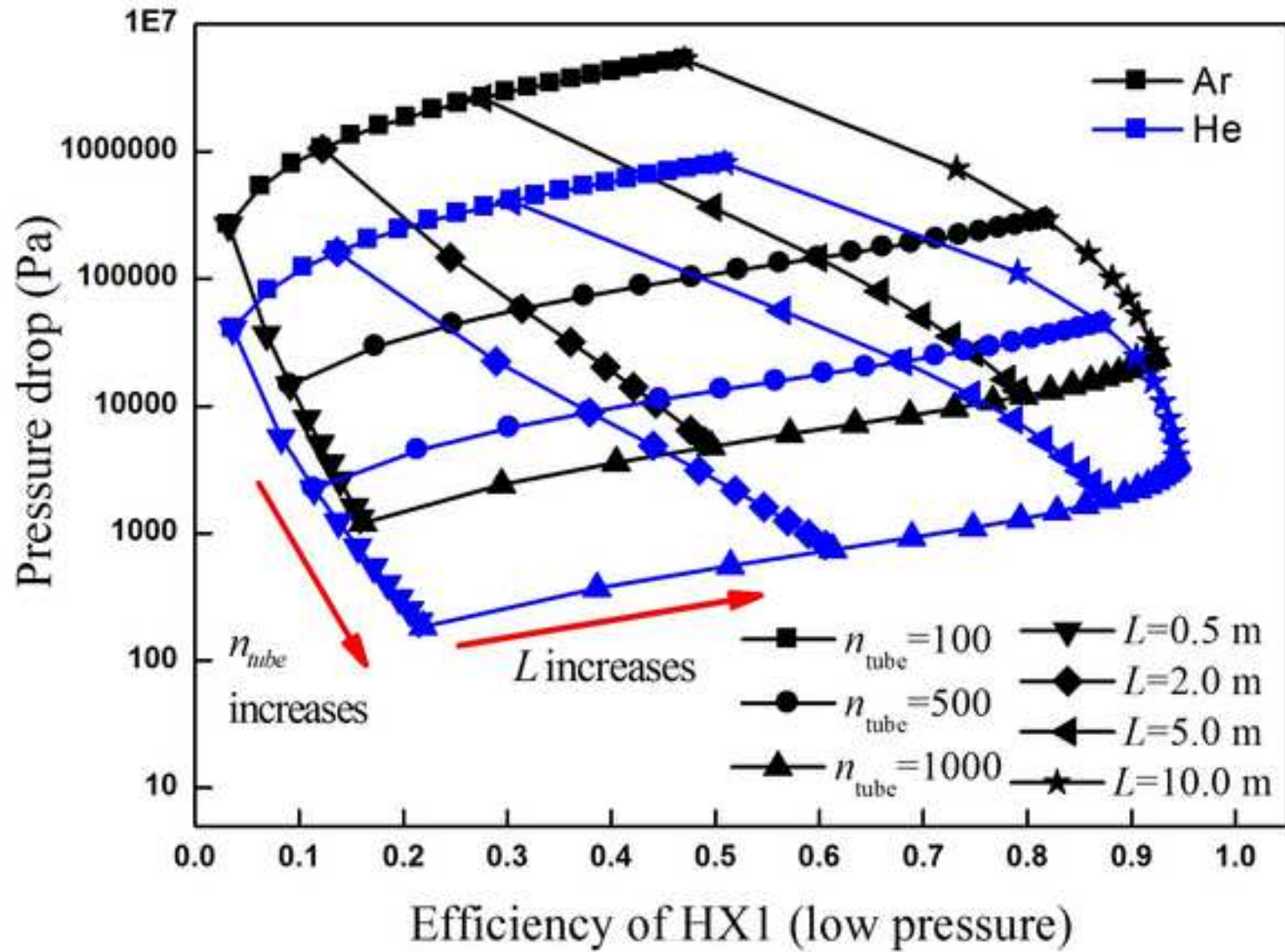


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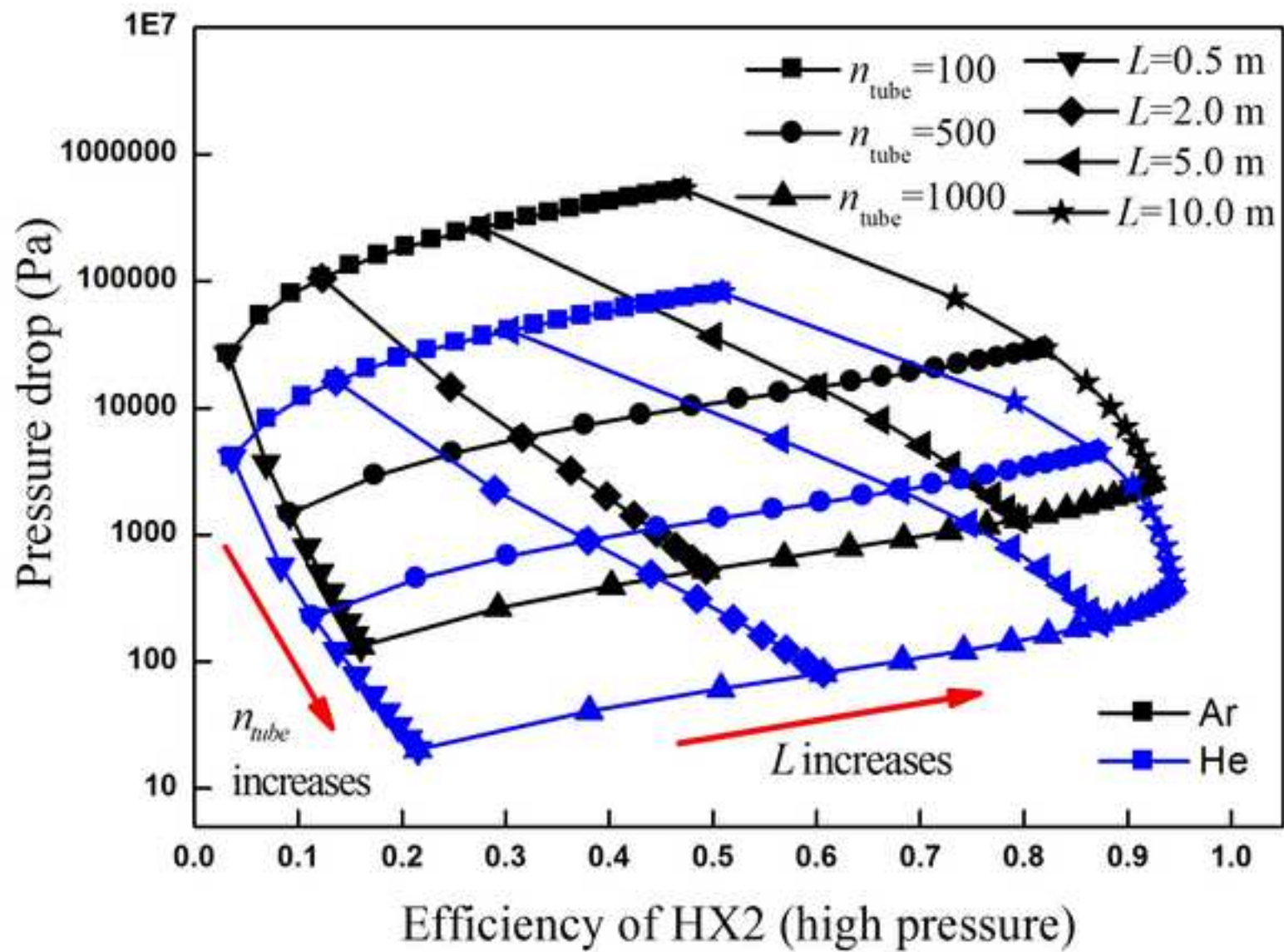


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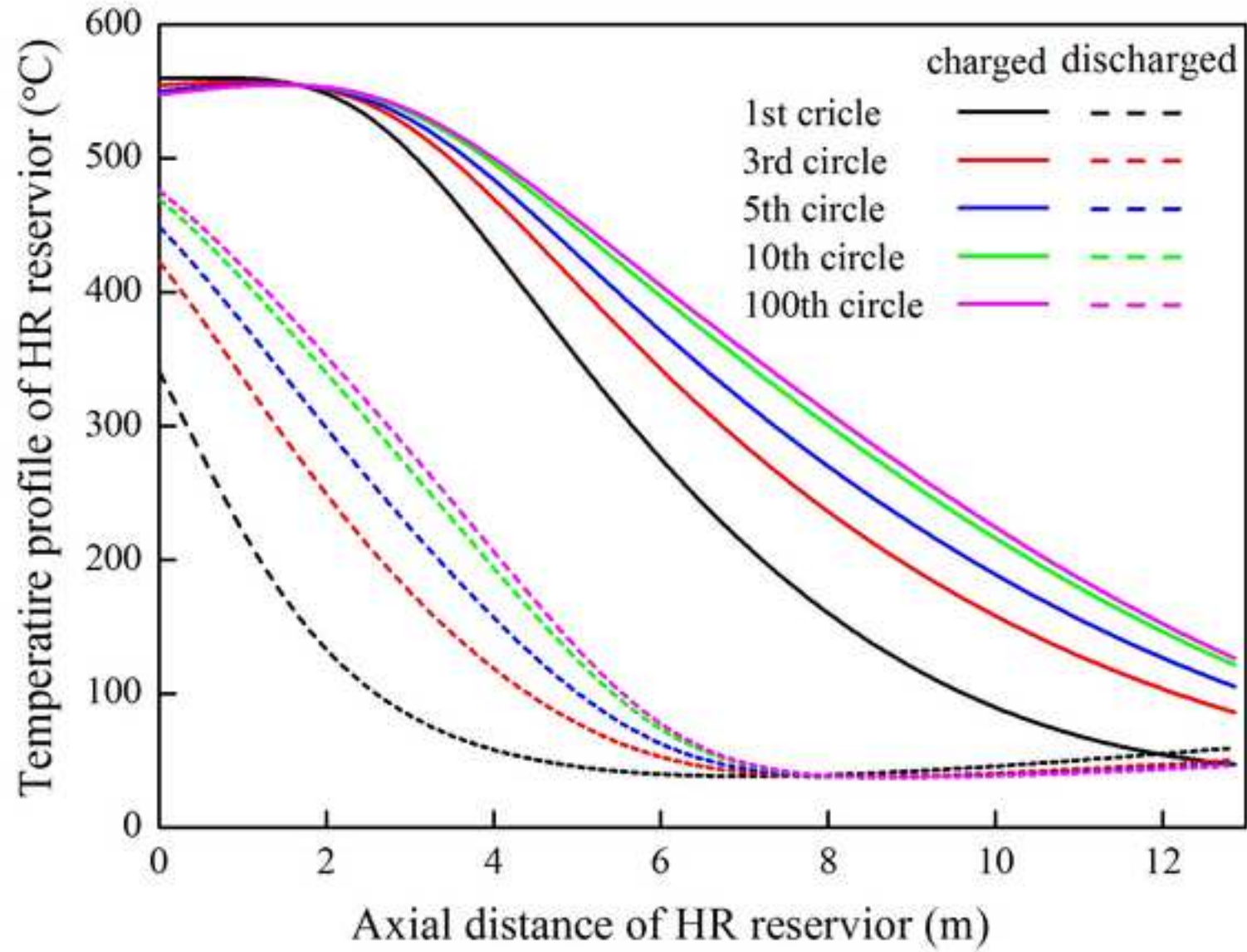


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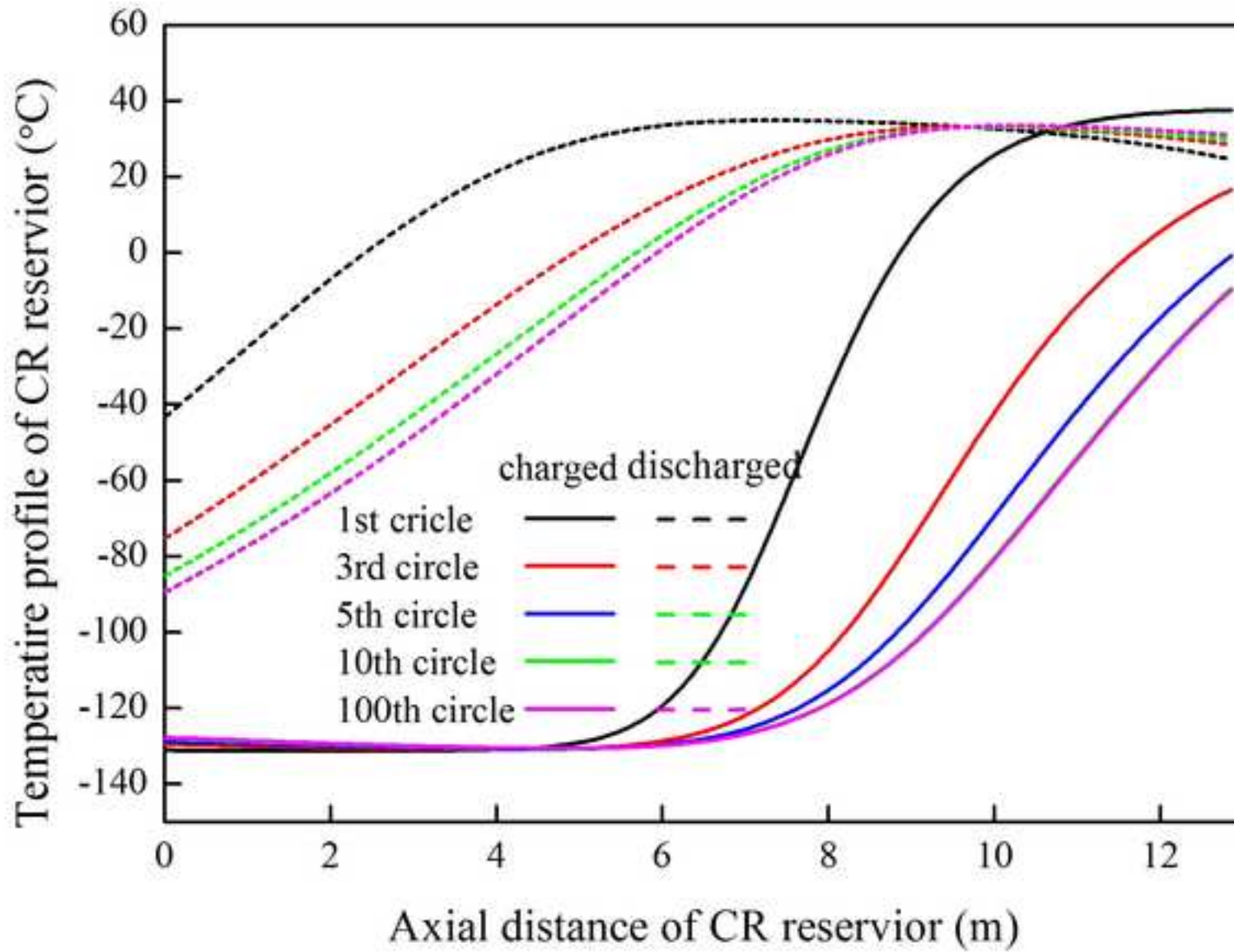


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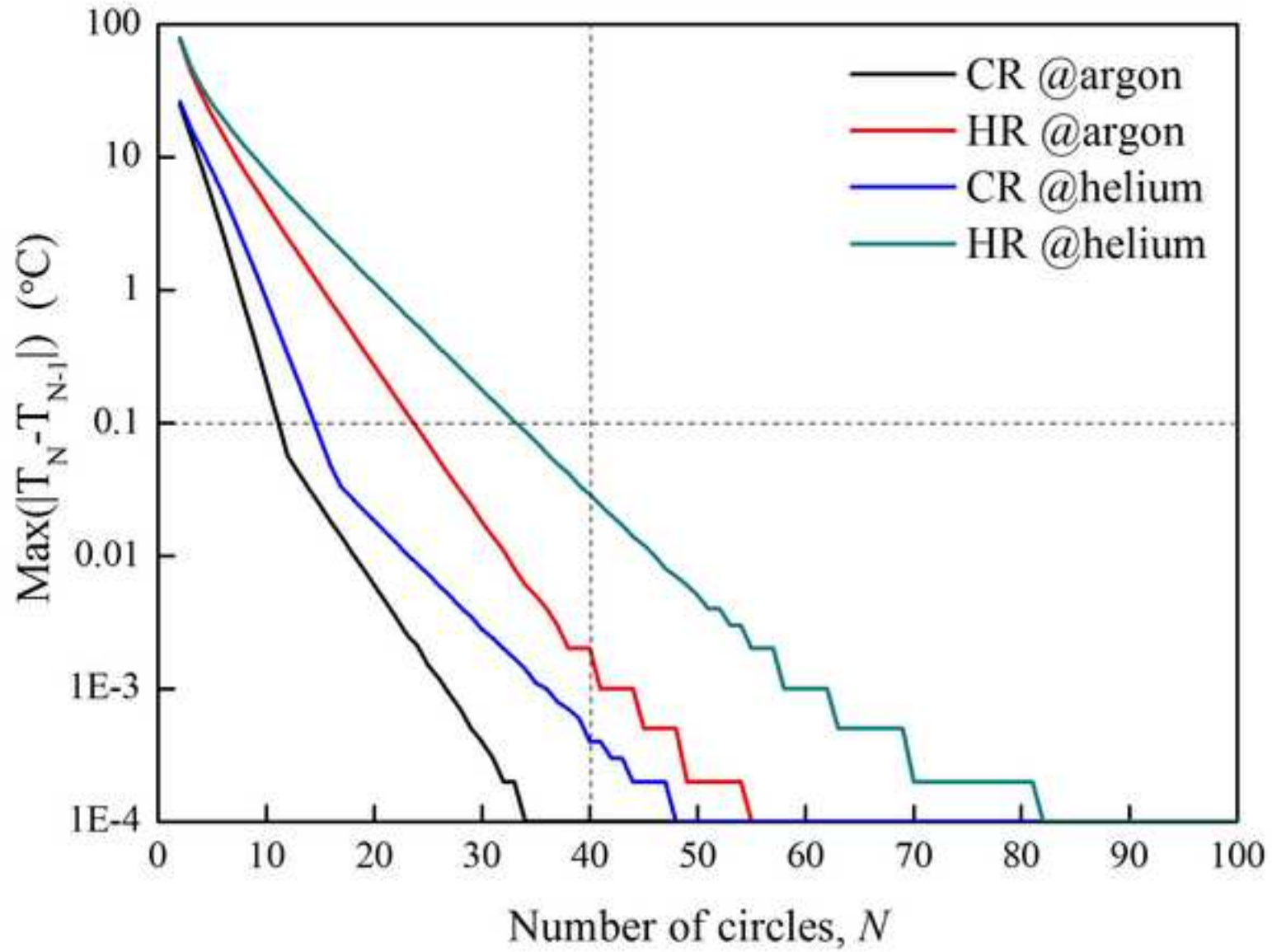


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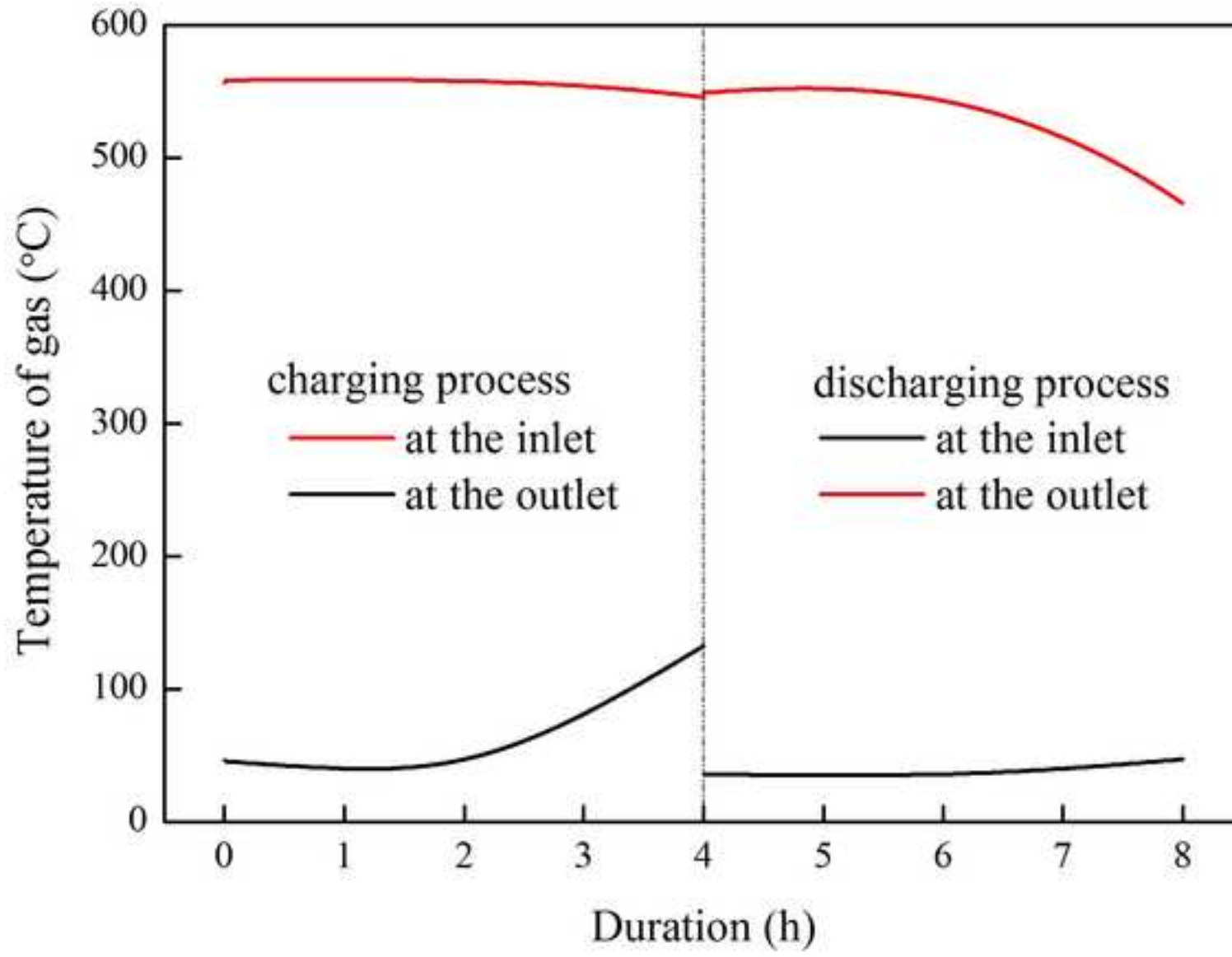


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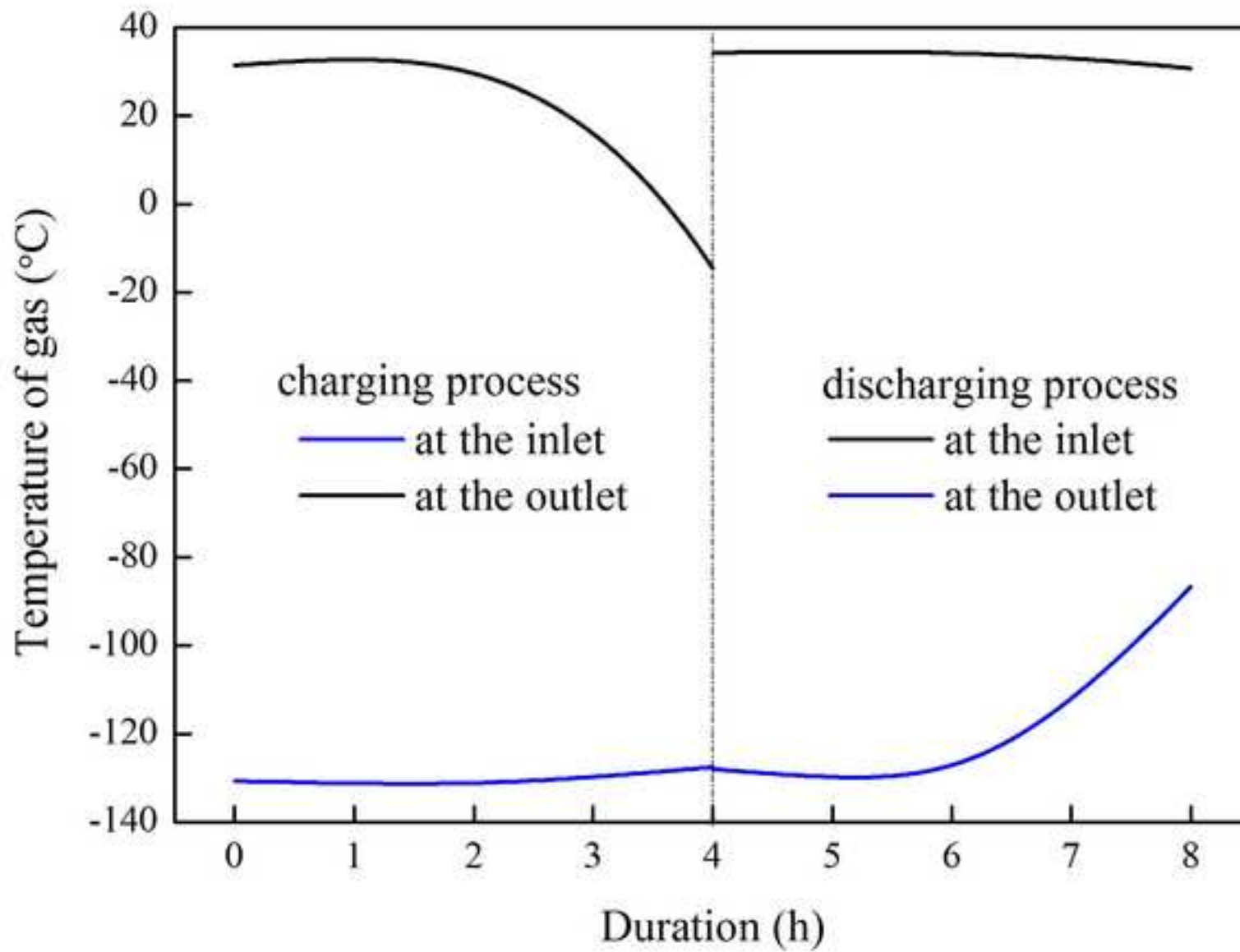


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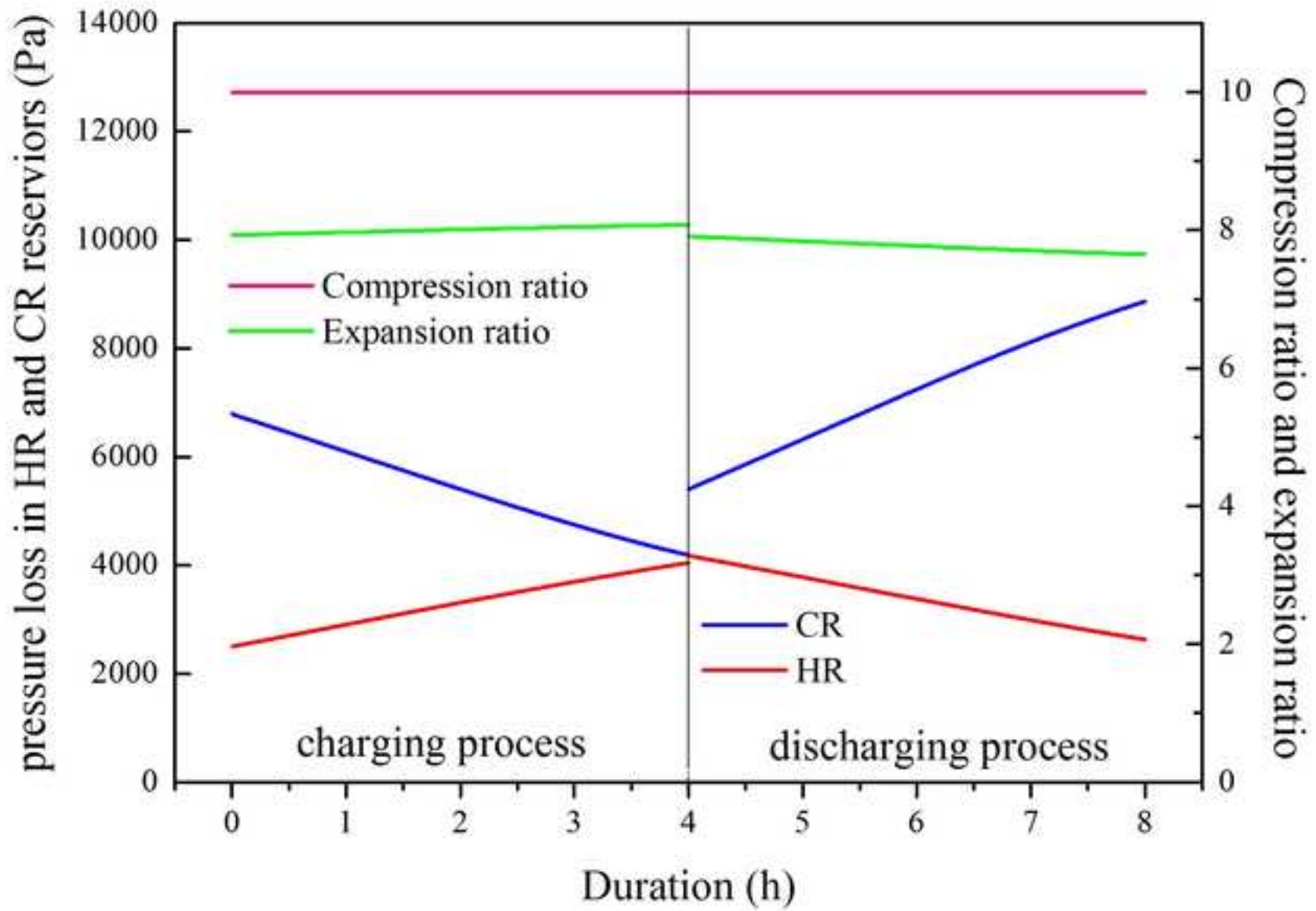


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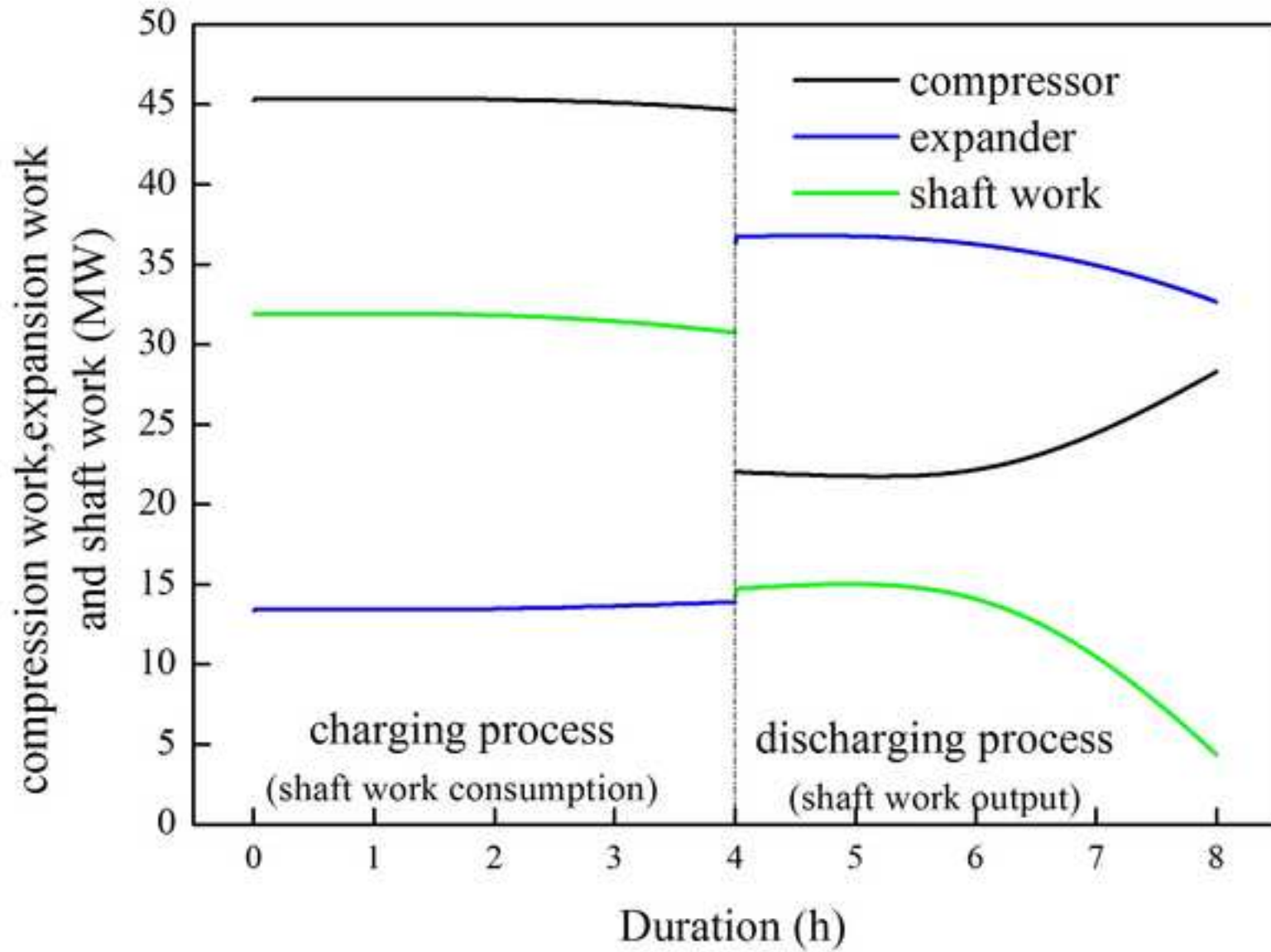


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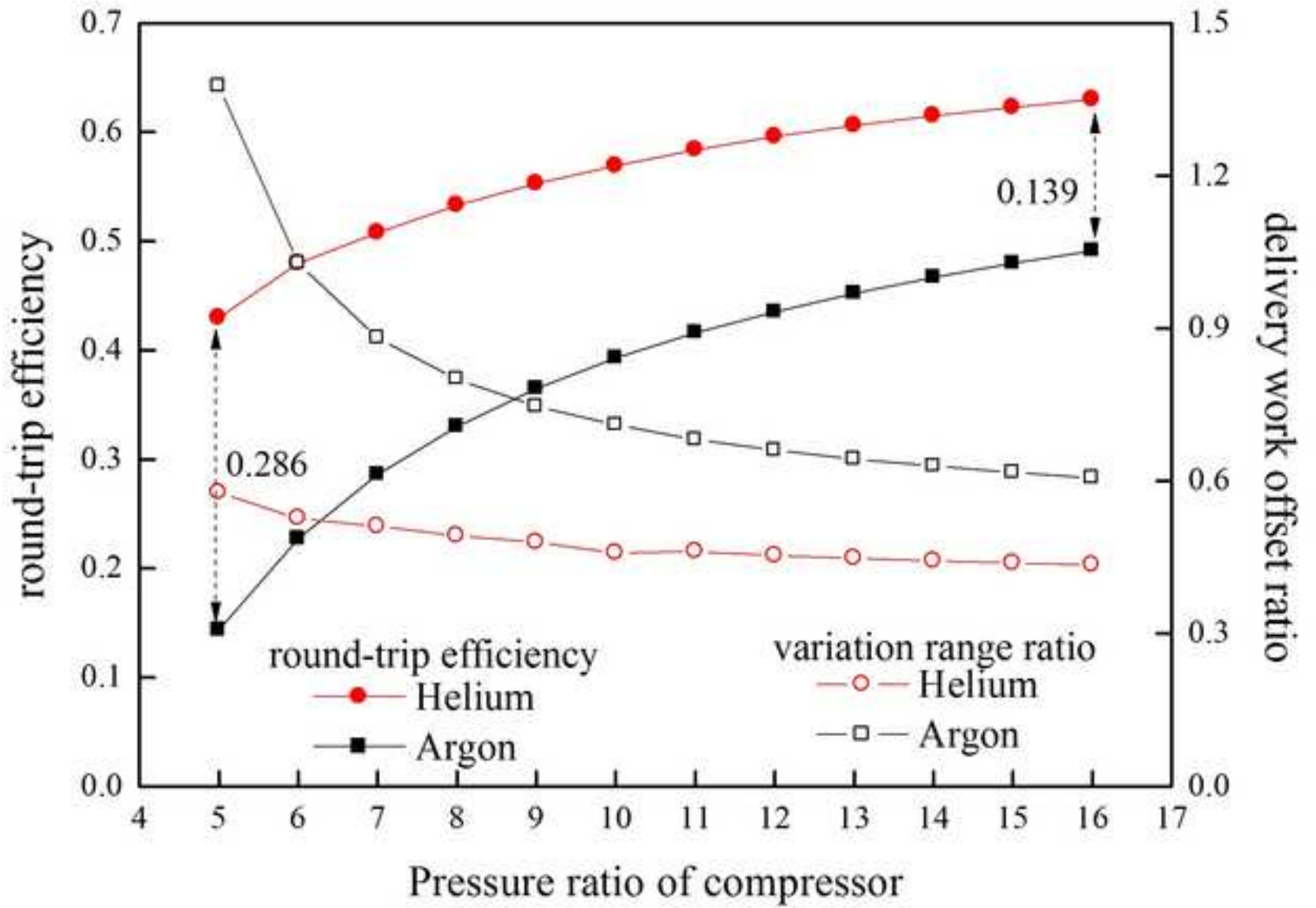


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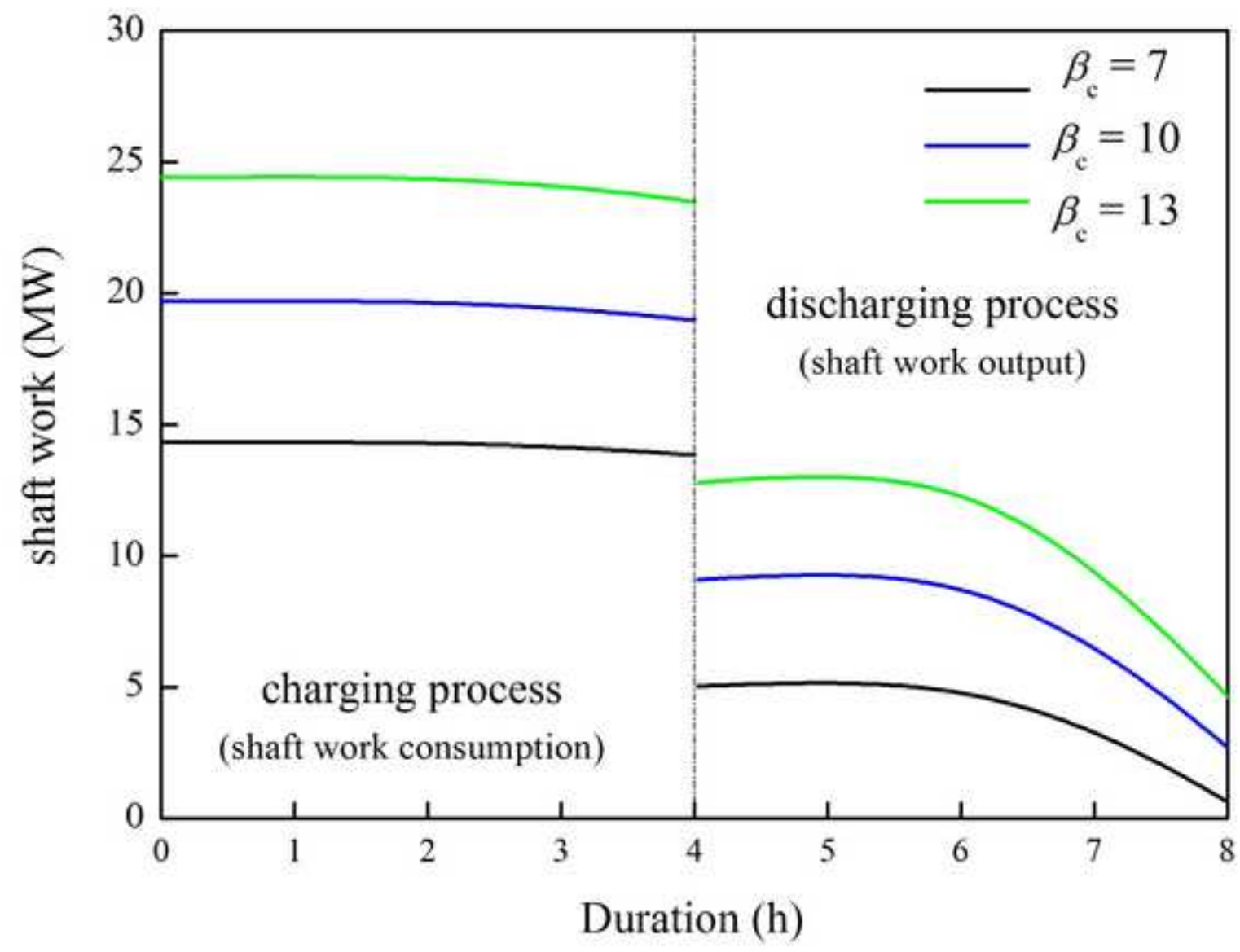


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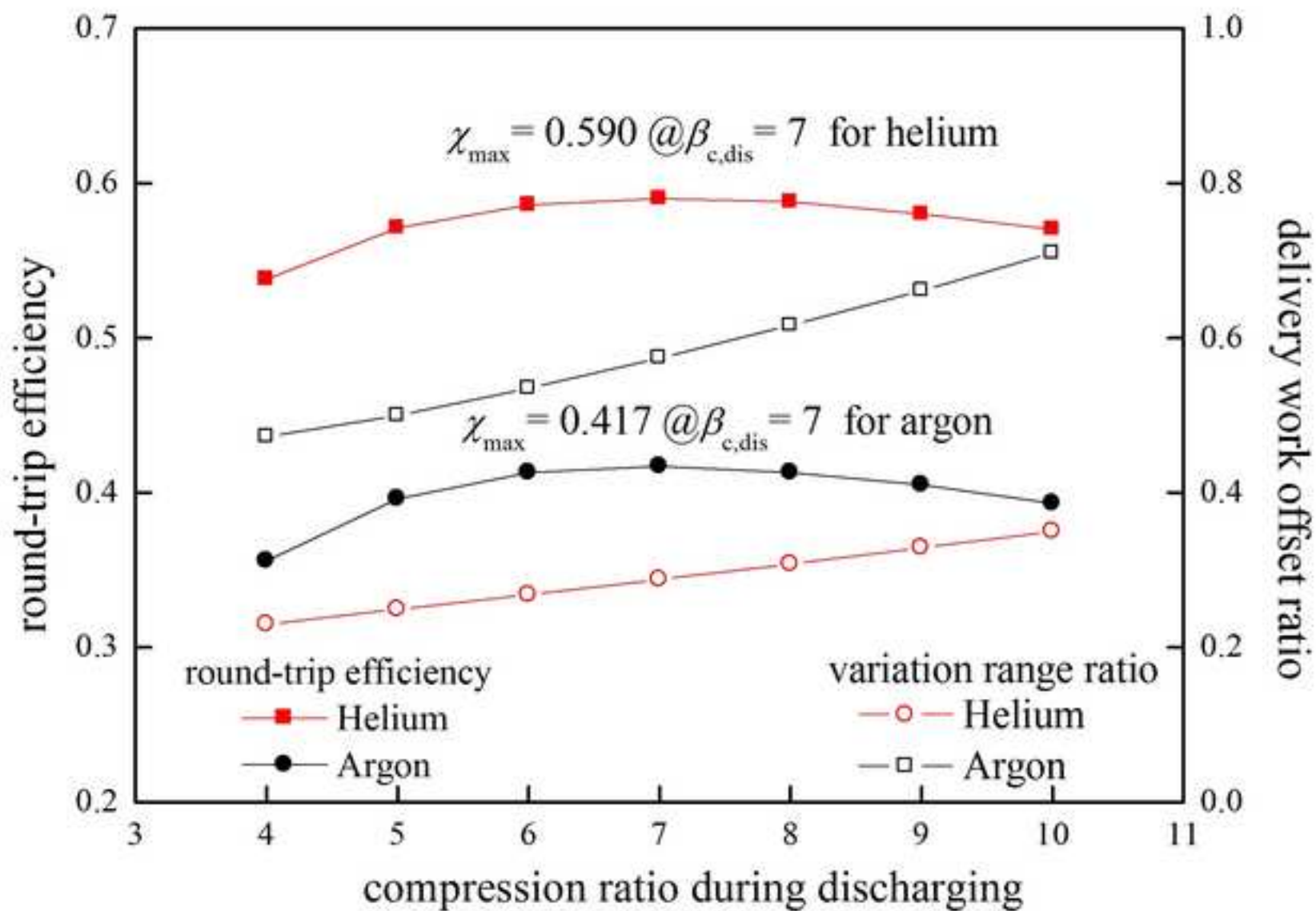


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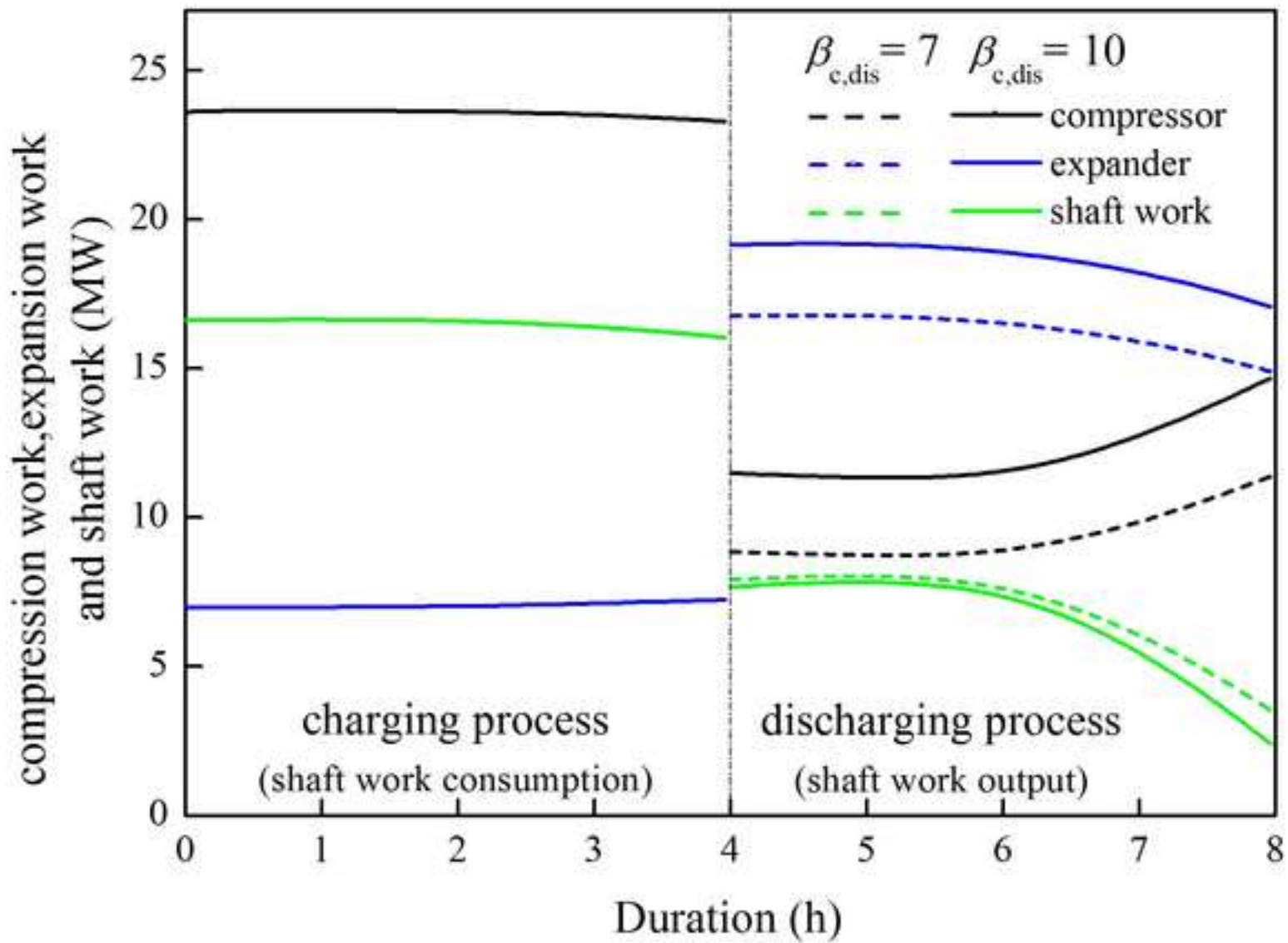


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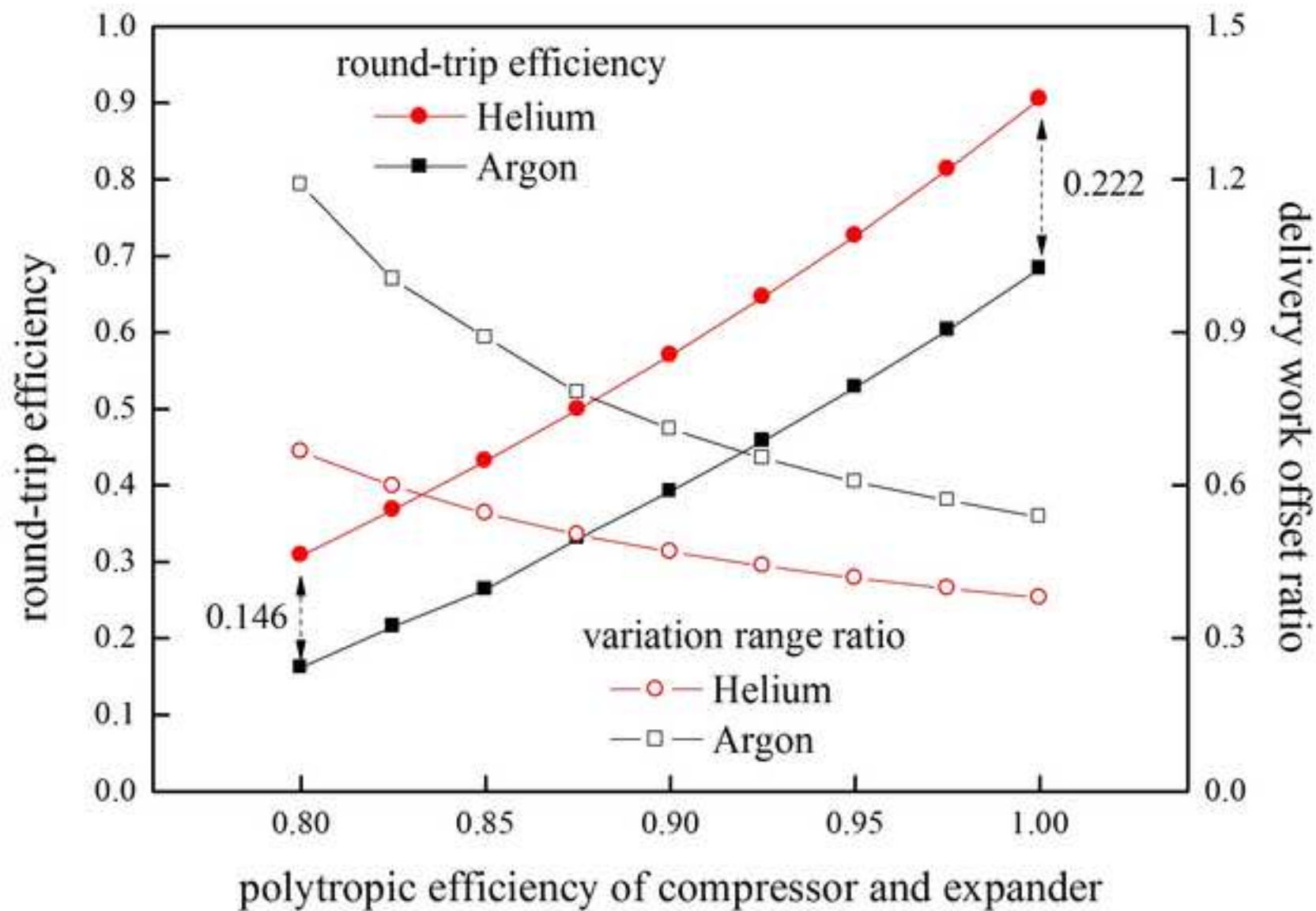


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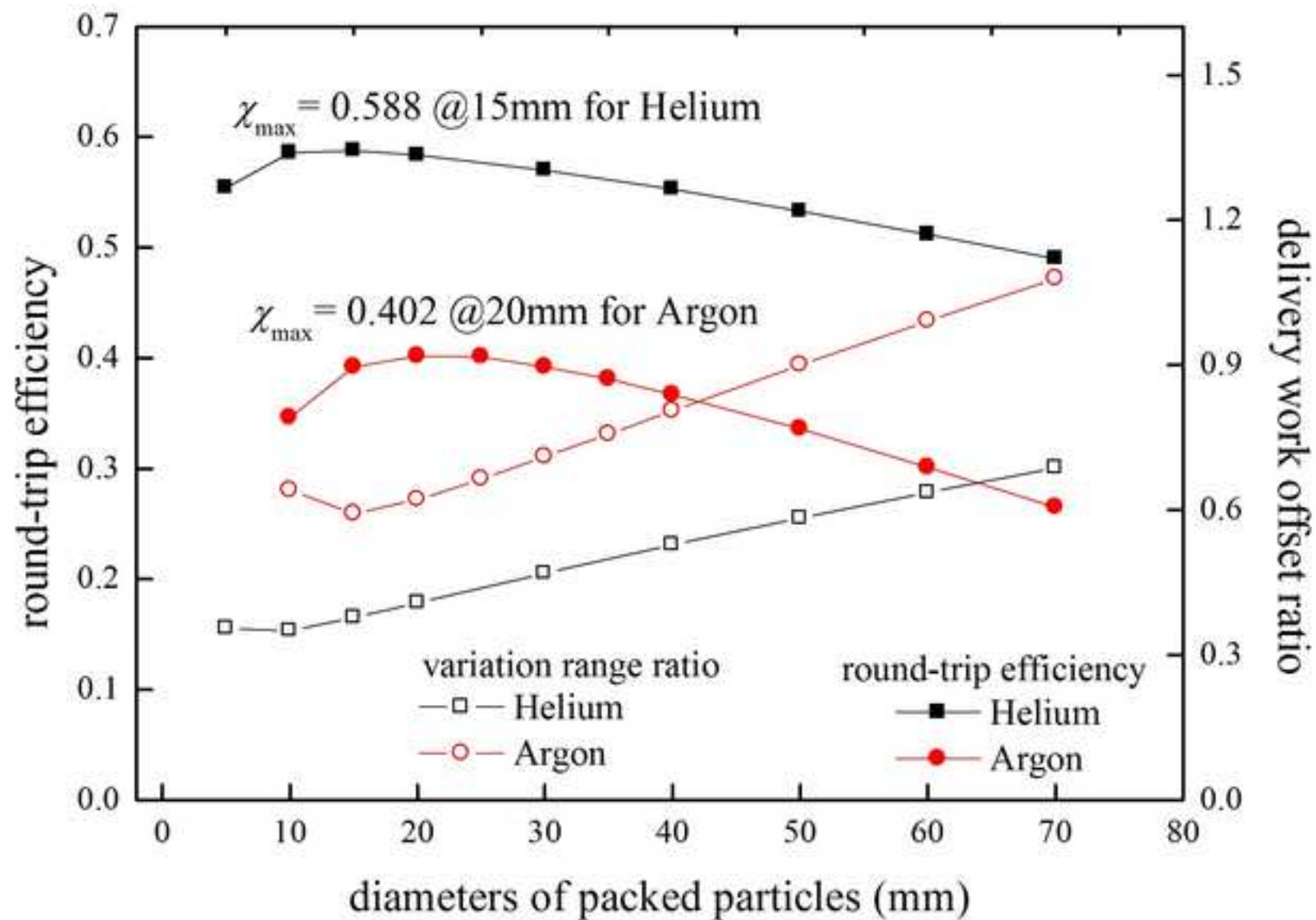


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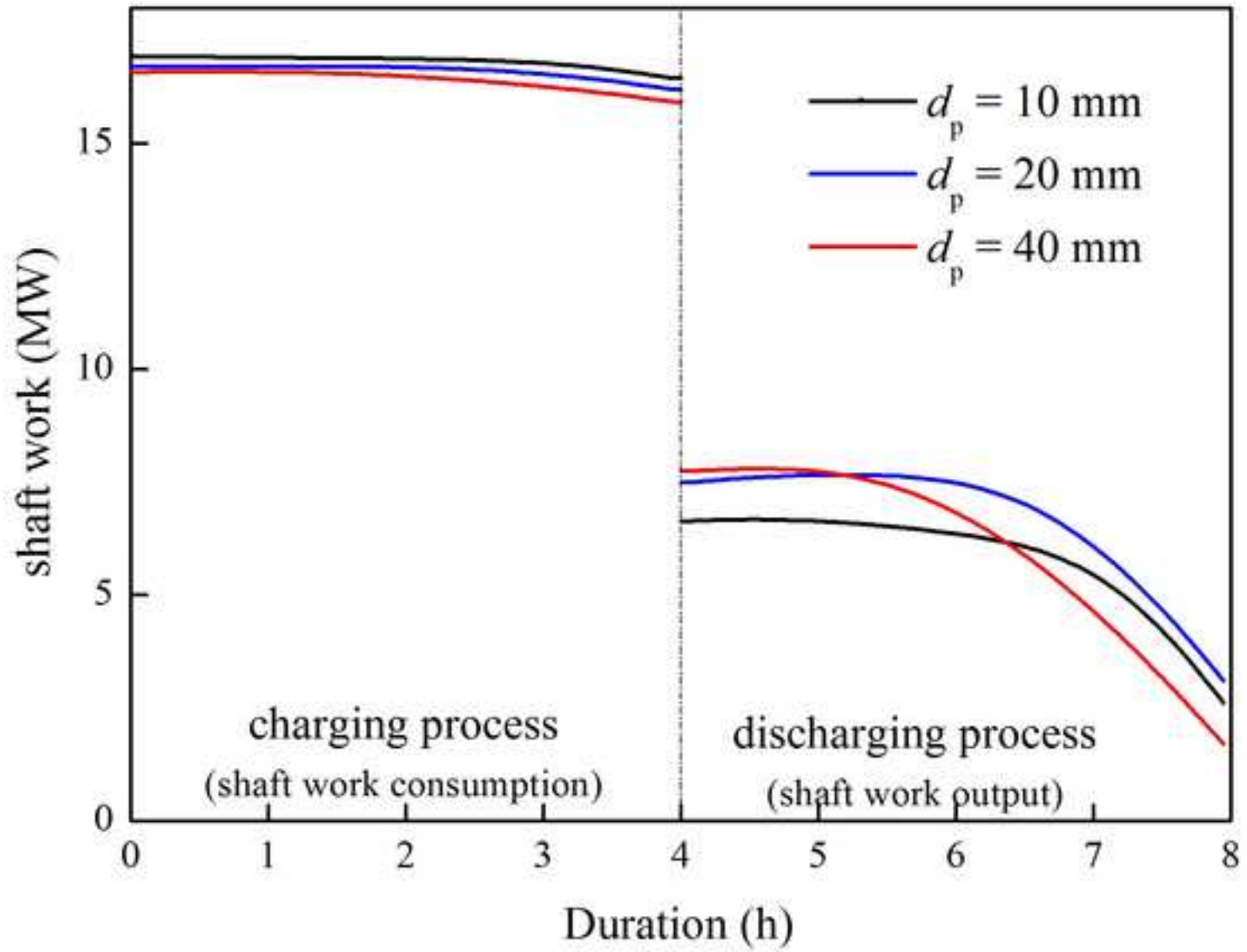


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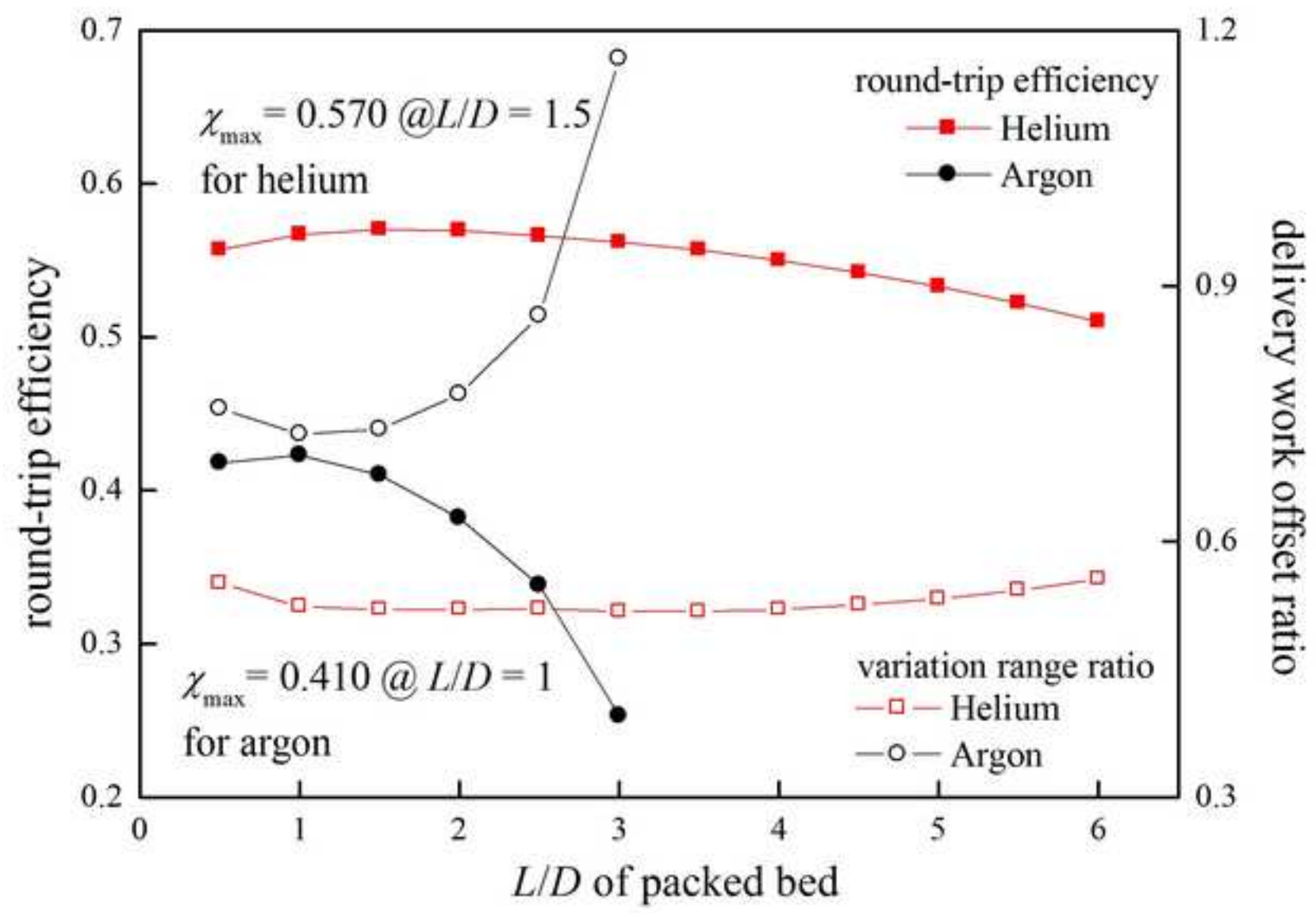


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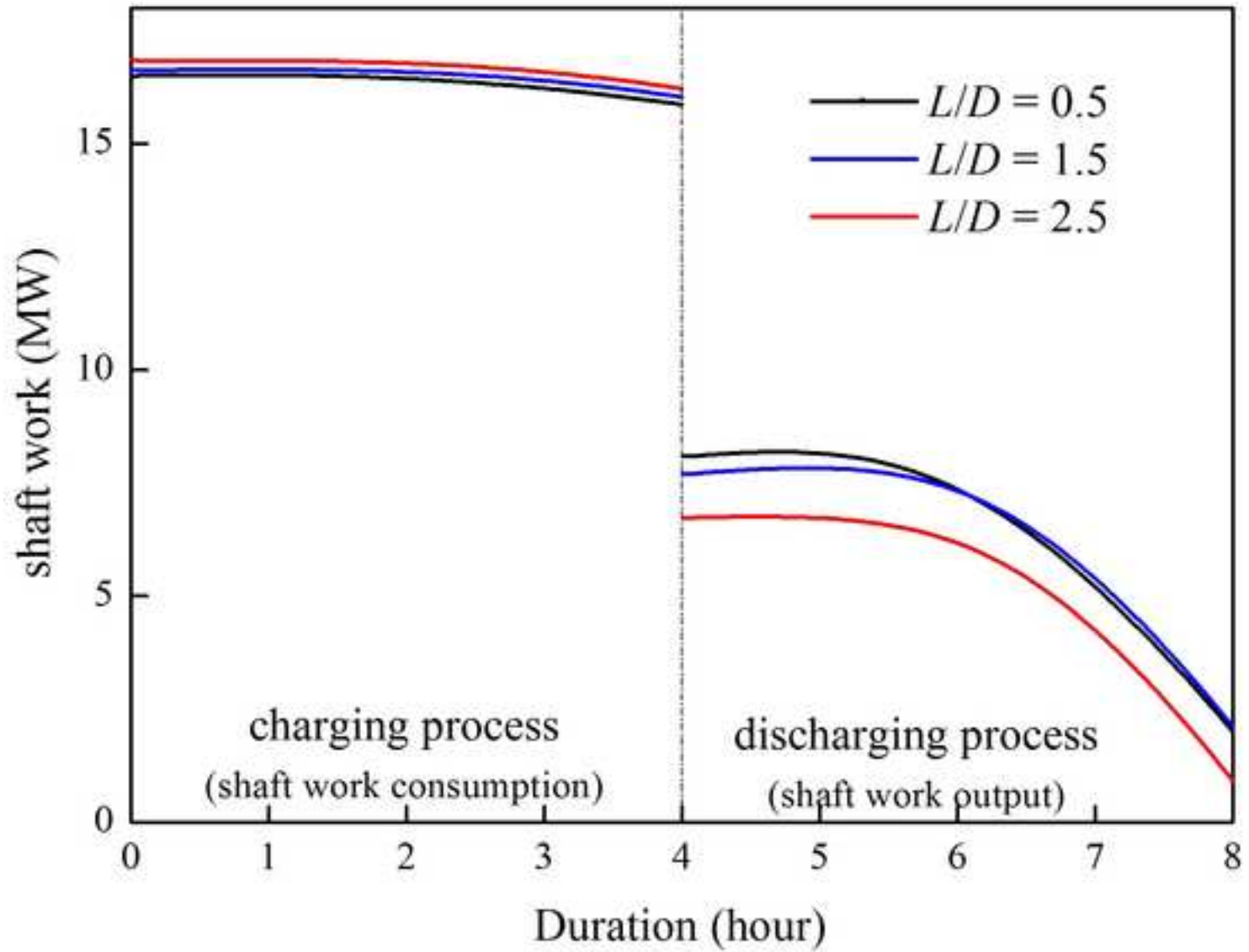


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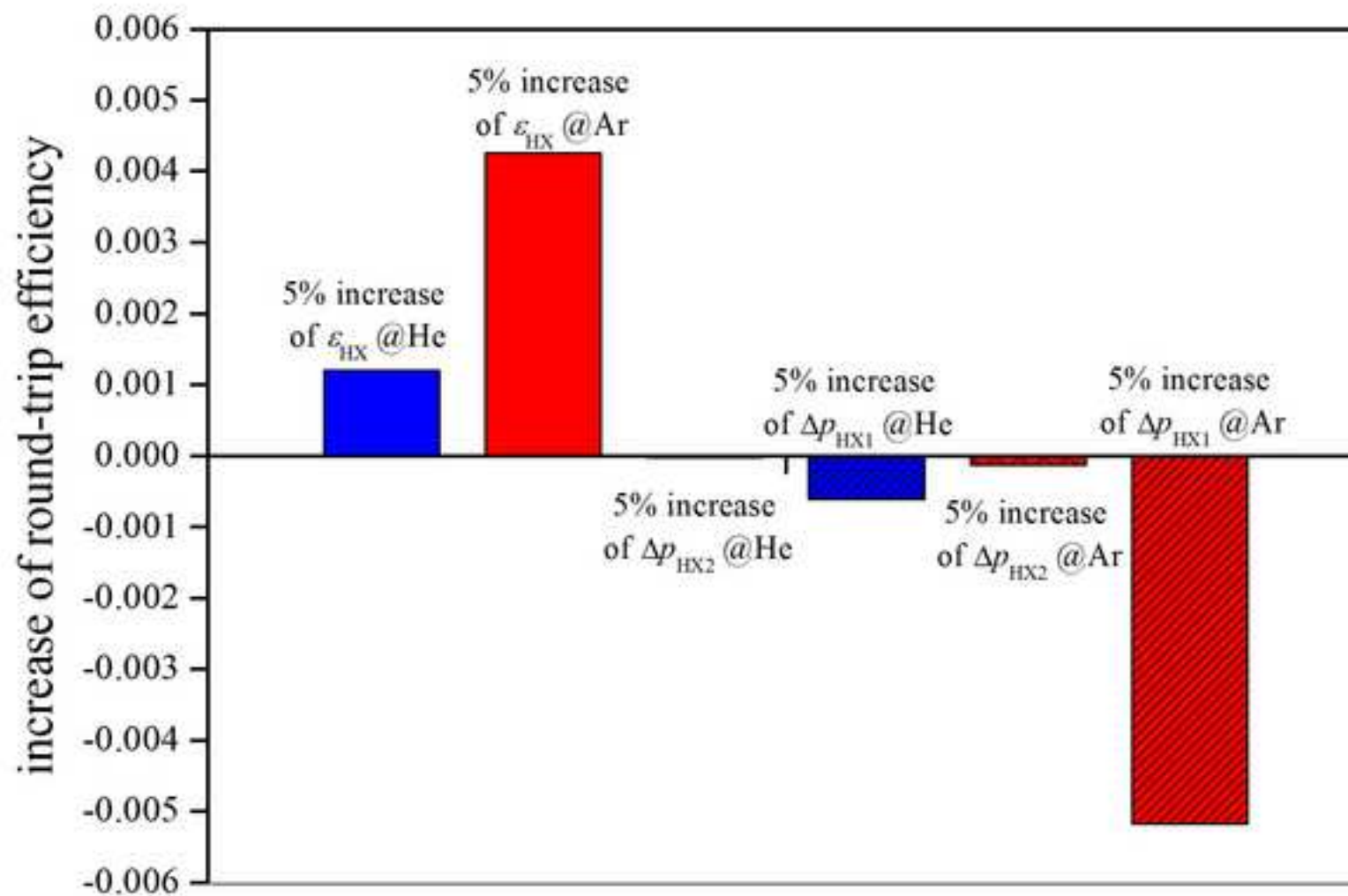


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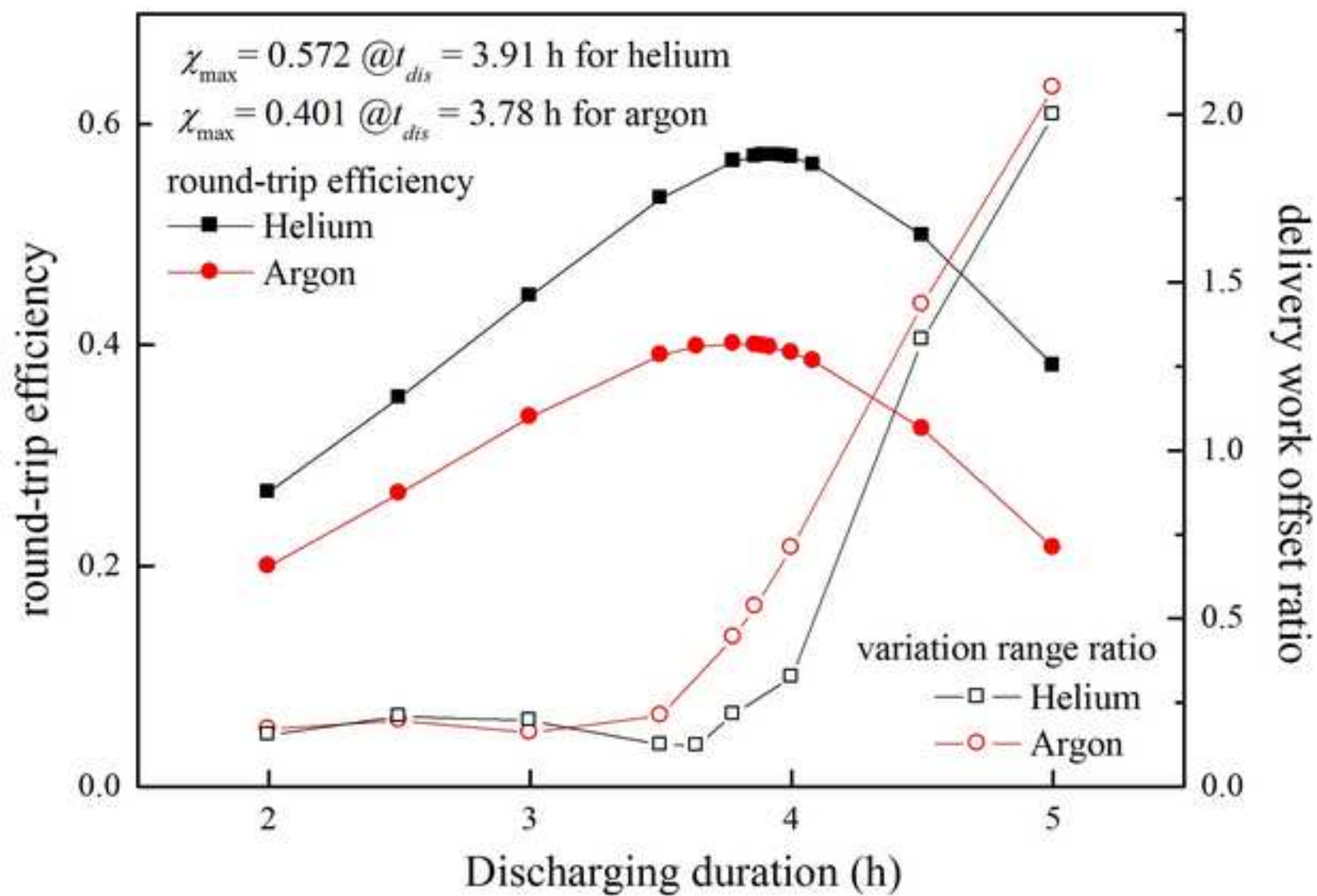


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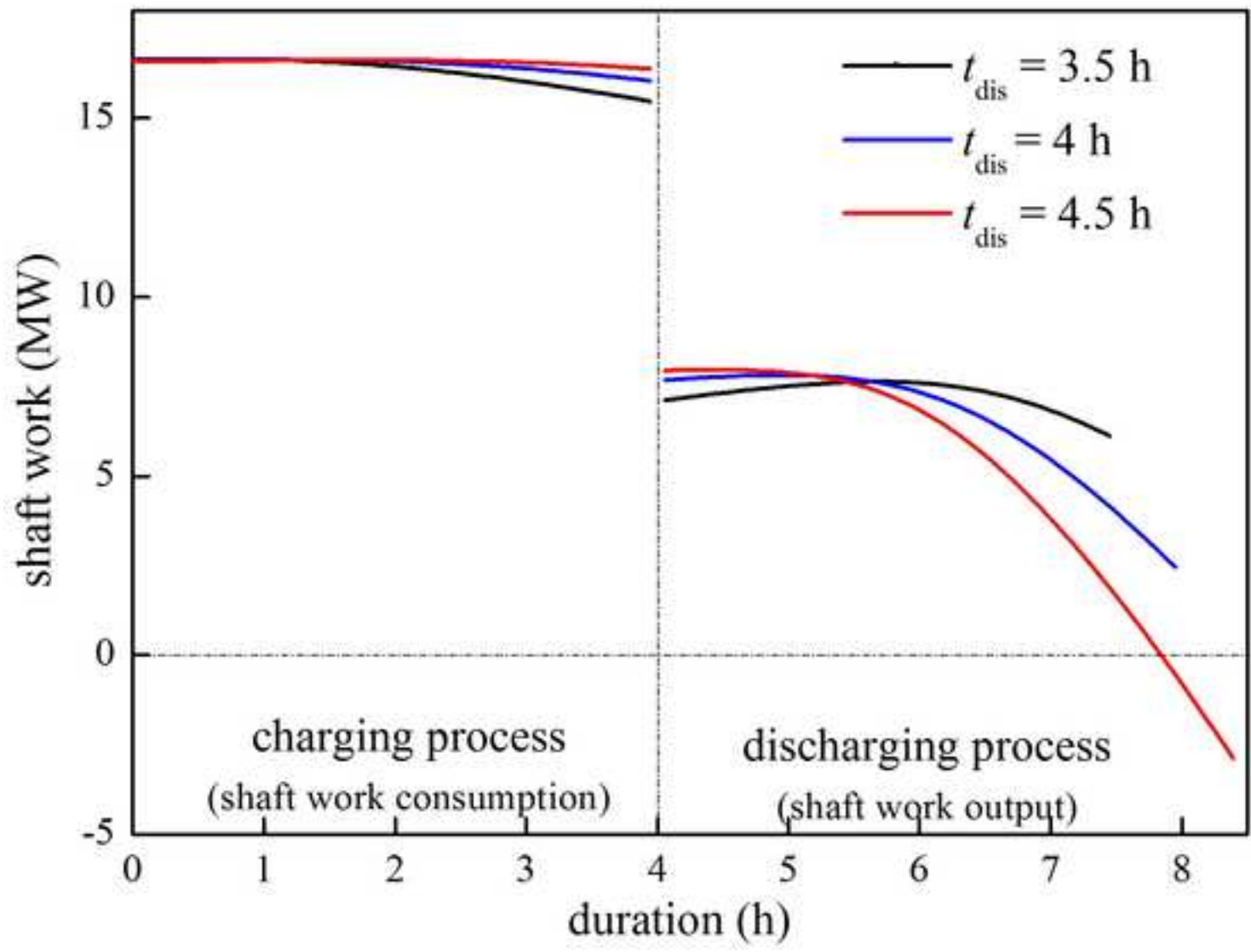


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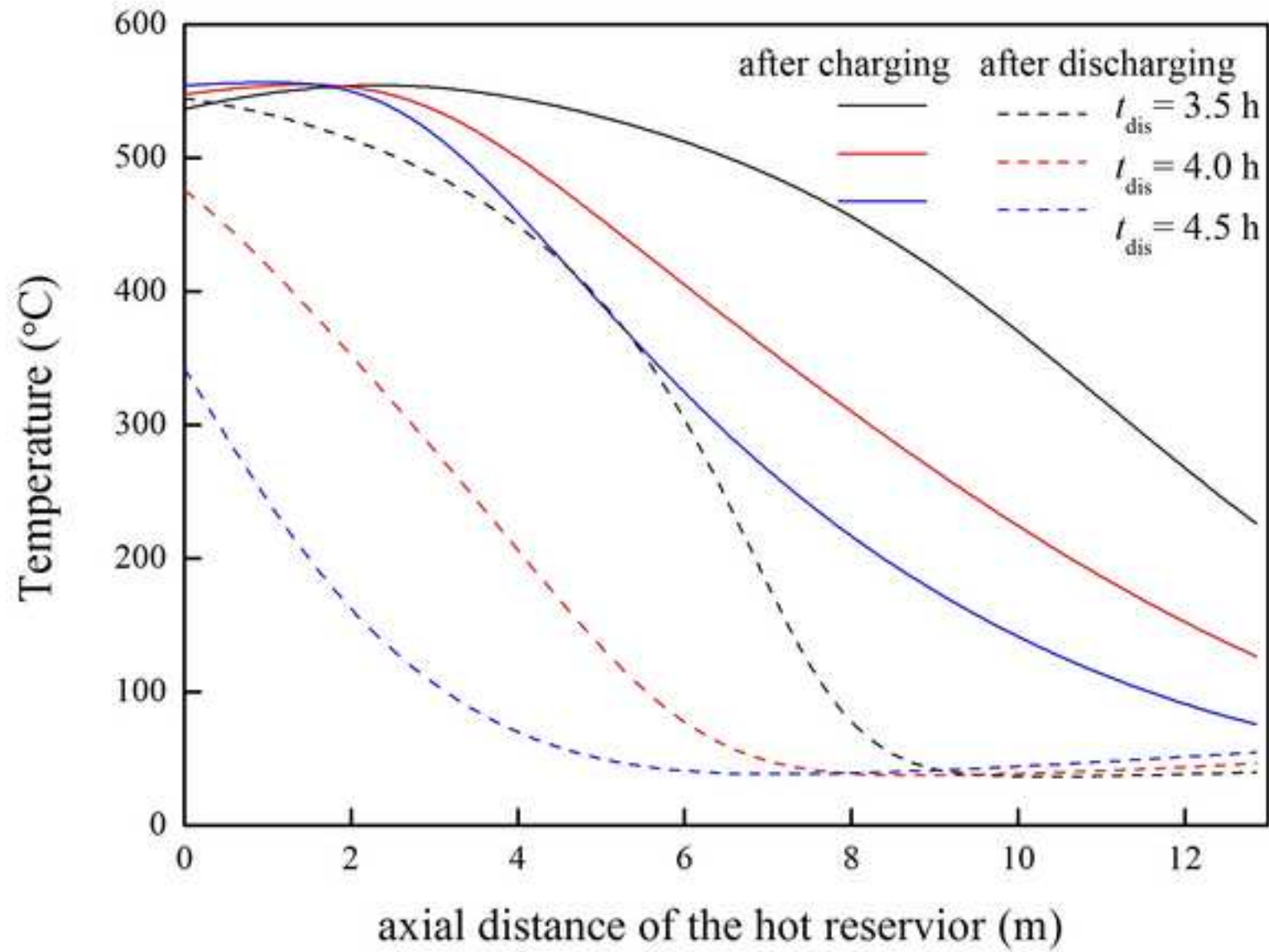


Table 1 Designed parameters of PHES system of 10 MW discharging power

Working gas	HP Pressure (MPa)	LP Pressure (MPa)	Average $c_{p,g}$ (J/kg/K)	Mass flow rate (kg/s)	Polytropic efficiency	ε of HXs	Δp of HP HXs (kPa)	Δp 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 ³)	Porosity	Average d_p (mm)	Total Volume (m ³)	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.