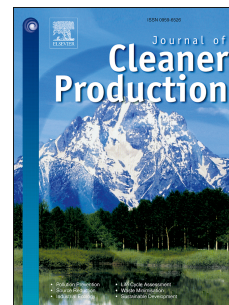# Journal Pre-proof

Thermodynamic evaluation and optimization of supercritical CO_2 Brayton cycle considering recuperator types and designs

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Thermodynamic evaluation and optimization of supercritical CO_2 Brayton cycle considering recuperator types and designs

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Abstract

The recompression supercritical carbon dioxide Brayton cycle (SCO₂-BC) has great potential for bottoming cycles among future energy conversion systems with comparatively high thermal efficiency. However, the recuperator type selections and optimal designs are a long-standing bottleneck of realizing maximum cycle efficiency and minimum cost. Reasonable collocation of recuperator types remains challenging. In this study, the thermodynamic–economic evaluation and optimization of recompression cycle were performed considering different recuperator types and designs. Printed circuit recuperator channel types include straight, zigzag, S-shaped, and airfoil fins. The design parameters include recuperator enthalpy efficiency and recompression fraction. The results indicate that low-temperature recuperator (LTR) with zigzag channel and high-temperature recuperator (HTR) with zigzag channel exhibit the best comprehensive cycle performance. The cycle thermal efficiency is more sensitive to the HTR enthalpy efficiency compared with the LTR. There is a local inflection point of the recompression fraction where the cycle efficiency is maximum and the exergy loss is minimum. The optimal solution of three-objective optimization considering efficiency, total cost, and exergy loss is more comprehensive than that of the two-objective optimization considering efficiency and cost. In the application of concentrated solar

power, the annual carbon dioxide emission can be directly reduced by 0.43 Gt through this optimization improvement. The results of the current study offer a promising route to high-efficiency and low-cost applications of SCO_2-BC by rational selection of recuperator types and designs.

Keywords: supercritical carbon dioxide recompression Brayton cycle; printed circuit recuperator; thermodynamic and economic evaluation; multi-objective optimization

1. Introduction

One of the most promising schemes would be the supercritical carbon dioxide Brayton cycle (SCO_2-BC) system in the field of clean energy power generation. Compared with the traditional steam-Ranking system, the SCO_2-BC possesses the characteristics of moderate working parameters, high thermal efficiency, simple and compact structure, flexible start–stop, wide temperature ranges, high safety, and cost saving (Guo et al., 2022). As a result of SCO_2's high density and less compressibility, compression work is significantly reduced. Consequently, compact turbomachines and heat exchangers can be used (Dostal et al., 2006). This cycle can help break the bottleneck of the development of existing power cycle technology. It has broad application prospects in thermal, nuclear, and solar power generation, energy storage system (Deng et al., 2020), coal-fired power plant (Liu et al., 2020), and others (Ahn et al., 2015). Nowadays, it has attracted much more attention as a research hotspot. Several research institutions, e.g., Sandia Nation Laboratory (Iverson et al., 2013), Tokyo Institute of Technology (Utamura et al., 2012), Korean Atomic Energy Research Institute (Al-Sulaiman and Atif, 2015), Xi'an Thermal Power Research Institute (Li et al., 2019), and Institute of Engineering Thermophysics of Chinese Academy of Sciences (Hu et al., 2020), successively performed the experimental loops of the SCO_2-BC and conducted its theoretical studies. It has become a potential way of energy saving and CO_2 resource utilization in the field of power generation, which also has great potential for process improvement and application in terms of cleaner and sustainable production.

To date, different SCO_2 cycle layouts have been proposed and developed, e.g., simple recuperation, recompression, intercooling, and multi-compression cycles. In recent time, the SCO_2 recompression cycle has been the most recognized cycle layout (Tong et al., 2022), which was

60　proposed to tackle the pinch-point conundrum of the recuperator. It had a very good performance in

61　the nuclear energy system (Dostal et al., 2006), part-load conditions (Yang et al., 2020), and solar

62　power system (Al-Sulaiman and Atif, 2015). Dyreby et al. (Dyreby et al., 2014; Dyreby, 2014)

63　developed a modeling framework to predict the design-point, off-design, and part-load

64　characteristics of different SCO_2-BCs. The results indicate that the recuperator used in the

65　recompression cycle is more balanced.

66　　　The aforementioned studies proved that the recompression cycle has a good application

67　prospect. Several studies have shown that recompression cycles are greatly influenced by the

68　performance of recuperators. The research showed that cycle thermal efficiency increased with an

69　increase in the total recuperator conductance (Dyreby, 2014; Neises and Turchi, 2014). In terms of

70　exergy losses, the recuperator effectiveness has a greater effect on the cycle exergy efficiency

71　(Padilla et al., 2015; Wang and He, 2017; Zhao et al., 2016). Besides, the cycle cost is closely related

72　to the recuperators, up to 90% of the cycle cost is in the heat exchangers (Fleming et al., 2013; Rao

73　et al., 2019).

74　　　The recuperator, one of the most crucial parts in the SCO_2 cycle, uses a printed circuit heat

75　exchanger (PCHE) on account of its high heat transmission efficiency, small volume, high

76　temperature, and pressure resistance endurance (Lian et al., 2021). To date, researchers have come

77　up with several types of channel patterns. They can be broadly divided into four types: straight,

78　zigzag, S-shaped, and airfoil-fin channels. Chen et al. (Chen et al., 2016) experimentally examined

79　the flow and heat transfer characteristics of a PCHE using a high-temperature helium test device. A

80　comparison was made between zigzag and straight channels in laminar and transition flow states.

81　The results indicated that a zigzag channel has a 2–3 times higher heat transfer coefficient than a

82　straight channel in laminar flow, and the multiple is 1.5–3 times in transition flow. However, the

83　zigzag channel inevitably results in high flow resistance. To improve this problem, researchers

84　developed the S-shaped fin and airfoil fin channels. Tsuzuki et al. (Tsuzuki et al., 2007) proposed

85　S-shaped fin and numerically evaluated its heat-transfer performance. The results indicated that the

86　pressure drop of the zigzag channel was 5 times that of the S-shaped fin with equal thermal–

87　hydraulic performance. Kim et al. (Kim, D.E. et al., 2008) proposed a NACA0020 airfoil fin and

88　found that the pressure drop was reduced to one-twelfth of that in a zigzag channel with the same

89　performance. Moreover, several studies have summarized the correlations of the Nusselt number

90　(*Nu*) and friction factor (*f*) for four different PCHE channel types, which are the basics for PCHE

91　performance analysis and design.

92　　　In the system analysis, the PCHE design is crucial for satisfying the requirements of the SCO_2

93　cycle. Currently, the widespread adoption of Pareto-based multi-objective optimization algorithms

94　promotes the development of cycle design. To optimize the economy of SCO_2 cycle, Zhu et al. (Zhu

95　et al., 2020) developed a new global optimization model using an improved genetic algorithm. In

96　comparison with the conventional unit, the optimized unit reduced its cost by 10.39%. In Zhao et

97　al.'s (Zhao et al., 2016) study, the exergy efficiency and cost per power were optimized based on

98　multi-objective optimization. While Rao et al. (Rao et al., 2019) conducted the optimization with

99　thermal efficiency and total cycle cost. The influences of recuperator's operating conditions are

100　discussed.

101　　　The heat source temperature, cycle pressure ratio, inlet temperature of turbine, compressor inlet

102　pressure, pinch-point of the heat exchangers, and recuperator conductance were the common

103　variables in the cycle analysis. However, few studies have been conducted on recuperator types and

104　designs, and multi-objective optimization with more than two objectives is rare. Besides,

105　comparisons between different types of PCHE for SCO_2-BC have not been carried out in the open

106　literature. The recuperator types and design parameters have a marked impact on the

107　thermodynamic–economic cycle analysis. For example, due to PCHE's high heat-transfer efficiency,

108　capital costs and recuperator volume can be reduced effectively. The low-resistance channels of the

109　PCHE could effectively reduce the pump work of the cycle.

110　　　Thus, the multi-objective optimization of the recompression cycle considering different

111　recuperator types and designs was performed in this study. A thermodynamic–economic model was

112　built and verified. The thermodynamic and economic performances of different printed circuit

113　recuperator types were compared, and the recuperator types of low-temperature recuperator (LTR)

114　and high-temperature recuperator (HTR) with the best comprehensive performance were

115　determined. In order to obtain the maximal thermal efficiency, minimum cost and exergy loss, the

116　recuperator enthalpy and recompression fraction were analyzed. Using the non-dominated sorting

117　genetic algorithm (NSGA-II), the two-objective optimization is carried out between thermal

118　efficiency and total cost; then, the three-objectives optimization is performed for efficiency, total

119　cost, and exergy loss. The technique for order preference by similarity to an ideal solution (TOPSIS)

120 method was used to get an optimal solution.

121 **2. Methods**

122   A representative recompression SCO_2-BC is illustrated in Fig. 1. The process is as follows:

123 First, SCO_2 flows through the primary heat exchanger (PHE) to reach a high temperature state

124 (5→6). Then, it passes into the turbine, which drives electric generators (6→7). The hot stream is

125 carried through pipes to the HTR (7→8) and LTR (8→9) and becomes a cold stream. At state 9, the

126 cold stream is divided into two streams. One enters the precooler (PC) (9→1) to cool further and is

127 compressed by the main compressor (1→2) to a high-pressure condition. Then, this stream is

128 preheated in the LTR (2→3). The other enters the recompressor (9→10), and the two streams merge

129 at state 4 (10+3→4). After preheating in the HTR (4→5), the SCO_2 stream loops into the PHE.
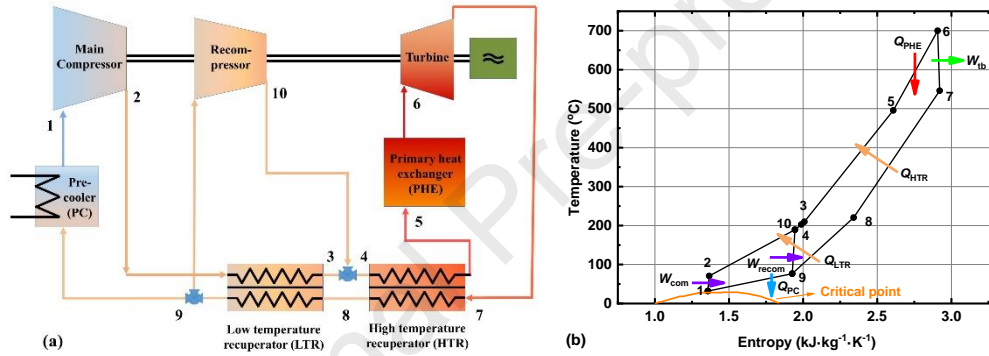
130

131 Fig. 1. Flow chart of SCO_2 recompression Brayton cycle.

132

133   A computational model of the recompression cycle was built using MATLAB codes. The SCO_2

134 thermophysical properties of the NIST REFPROP database (Lemmon, 2013) were applied to the

135 program. A thermodynamic model was created based on the one-dimension model and energy

136 balance. The following assumptions were illustrated:

137   (1)   The recompression cycle condition was the steady state.

138   (2)   The compression and expansion processes were adiabatic. The efficiencies of the turbine,

139 main compressor, and recompressor were constant, i.e., isentropic efficiency.

140   (3)   Only the friction loss along the channel of the recuperator, which was the main pressure

141 loss of the recuperator, was considered.

142   (4)   The pressure loss and heat loss of the pipeline were ignored.

143   (5)   The pressure loss of the PHE and PC was constant under the same mass flow condition,

144 which was obtained from Dyreby's modeling framework (Dyreby, 2014). In this study, the mass

145 flow of SCO_2 was 74.968 kg·s^{-1}, and the pressure losses of the PHE and PC were 245 and 77 kPa,

146 respectively.

147 2.1 Thermodynamic model

148 2.1.1 Turbine model

149 Considering constant isentropic efficiency, the output power of a turbine model is defined as

150
$$w_{tb,isen} = h_{tb,in} - h_{tb,out,isen} ,$$
(1)

151
$$w_{tb} = w_{tb,isen} \cdot \eta_{tb} , \text{ and}$$
(2)

152
$$W_{tb} = m_{CO_2} \cdot w_{tb}$$
(3)

153 where $h_{tb,in}$ is the inlet specific enthalpy of the turbine; $h_{tb,out,isen}$ is the outlet specific enthalpy of the

154 turbine if the SCO_2 isentropic expands to the outlet pressure; $w_{tb,isen}$ is the isentropic specific work;

155 η_{tb} is the isentropic efficiency; w_{tb} is the specific work; m_{CO2} is the mass flow rate of SCO_2, which

156 is constant; W_{tb} is the total output work.

157 The outlet specific enthalpy ($h_{tb,out}$) is

158
$$h_{tb,out} = h_{tb,in} - w_{tb}$$
(4)

159 2.1.2 Compressor model

160 The compressors are also defined as constant isentropic efficiency, and the specific work (w_{comp})

161 is defined as

162
$$w_{comp,isen} = h_{comp,out,isen} - h_{comp,in} , \text{ and}$$
(5)

163
$$w_{comp} = w_{comp,isen} / \eta_{comp}$$
(6)

164 where $h_{comp,out,isen}$ is the outlet specific enthalpy if SCO_2 isentropic compresses to the outlet pressure,

165 $h_{comp,in}$ is the inlet specific enthalpy, $w_{comp,isen}$ is the isentropic specific work, and η_{comp} is the

166 isentropic efficiency.

167 The outlet specific enthalpy of compressor ($h_{comp,out}$) is

168
$$h_{comp,out} = h_{comp,in} + w_{comp}$$
(7)

169 The total compression power of the compressors is

170
$$W_{comp} = m_{CO_2} \cdot (1 - \phi_R) \cdot w_{mc} + m_{CO_2} \cdot \phi_R \cdot w_{rc}$$
(8)

171 where Φ_R is the recompression fraction, w_{mc} and w_{rc} are specific work of main compressor and

172   recompressor, respectively.

173   2.1.3 PHE and PC model

174   The heat transfer capacity of PHE (Q_{PHE}) is calculated using

175    $$Q_{PHE} = m_{CO_2} \cdot (h_{PHE,out} - h_{PHE,in})$$
(9)

176   where h_{PHX} is the specific enthalpy; subscripts of out and in indicate the outlet and inlet, respectively.

177   The heat transfer capacity of PC (Q_{PC}) is defined as

178    $$Q_{PC} = m_{CO_2} \cdot (1 - \phi_R) \cdot (h_{PC,in} - h_{PC,out})$$
(10)

179   where h_{PC} is the specific enthalpy.

180   The cycle thermal efficiency (η_{th}) is defined as

181    $$\eta_{th} = (Q_{PHE} - Q_{PC}) / Q_{PHE}$$
(11)

182   2.1.4 Recuperator model

183   In the present study, to accurately capture the influence of the SCO_2 variable property, the

184   recuperator was separated into several sub-heat exchangers connected in series along the channel

185   length (Fig. 2) (Guo, 2016). The heat balance calculation for each sub-heat exchanger depended on

186   the outlet and inlet states of the apiece sub-element. The PCHE periodic boundary condition makes

187   it possible to simplify the recuperator model to a single hot channel and a single cold channel. For

188   the recuperator model, initial input parameters were the inlet temperatures of the hot and cold

189   streams, inlet and outlet pressures of the hot and cold streams, enthalpy efficiency, mass flow-rate

190   of single channel, and geometric dimensioning. The number of sub-heat exchangers was set as 400
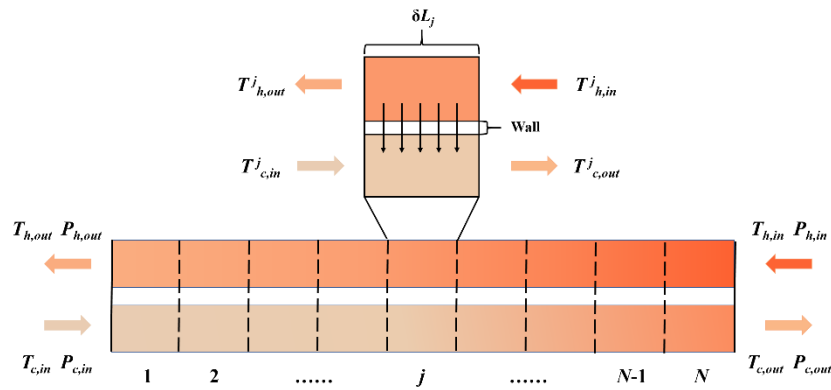
191   after the node independence was checked.

192

193   Fig. 2. Segmented counter-flow PCHE model.

194

195   The mass flow flux (G) is the mass flow rate per unit area, and the flow-rate of a single channel

196   (m_{ch}) is defined as

197    $$m_{ch} = G \cdot A_{cs}$$ (1

198   2)

199   where A_{cs} is the cross-sectional area of the single channel.

200   In this study, the heat transfer performance of the recuperator was evaluated by the maximum

201   enthalpy drop of SCO$_2$, which differed from the conditional calculation of efficiency based on

202   temperature. The effect of the pinch-point temperature was considered as well. The total heat

203   transfer capacity of model unit (Q_{ch}) is evaluated using the following equation:

204    $$Q_{ch} = \varepsilon \cdot \Delta h_{max}, \text{ and}$$ (13)

205    $$\Delta h_{max} = \begin{cases} \min\{m_{ch} \cdot (1-\phi_R) \cdot (h_{c,out}^* - h_{c,in}), m_{ch} \cdot (h_{h,in} - h_{h,out}^*)\} (\text{For LTR}) \\ \min\{m_{ch} \cdot (h_{c,out}^* - h_{c,in}), m_{ch} \cdot (h_{h,in} - h_{h,out}^*)\} (\text{For HTR}) \end{cases}$$ (14)

206   where ε is the enthalpy efficiency, $h_{c,out}^*$ is the cold outlet specific enthalpy when temperature is

207    $T_{h,in}$, $h_{c,in}$ is the cold inlet specific enthalpy, $h_{h,in}$ is the hot inlet specific enthalpy, and $h_{h,out}^*$ is the

208   hot outlet specific enthalpy when temperature is $T_{c,in}$.

209   The heat transfer capacity of sub-channel (δQ_{ch}) is equal for every sub-heat exchanger unit (N),

210   which can be calculated as

211    $$\delta Q_{ch} = Q_{ch} / N$$ (15)

212   According to the thermal equilibrium and heat transfer relationship of the sub-unit, the channel

213   length of the sub-unit (δL_j) is evaluated as

214    $$\delta L_j = \delta Q_{ch} / (k_j \cdot Pe \cdot \Delta T_j)$$ (16)

215   where Pe is the perimeter of the cross-section, ΔT_j is the node temperature difference between the

216   cold and hot streams, and k_j is the local comprehensive heat transfer coefficient obtained by:

217    $$k = (\frac{1}{h_h} + \frac{1}{h_c} + \frac{t}{\lambda_m})^{-1}$$ (16)

218   where t is the wall thickness, λ_m is the thermal conductivity of the metal material, which is 16.3

219   W·m^{-1}·K^{-1}, h is the convective heat transfer coefficient defined as

220
$$h = \lambda_{CO_2} \cdot Nu / D_{hy} \tag{18}$$

221 where λ_{CO2} is the thermal conductivity of fluid; Nu is the Nusselt number, which is determined using

222 Reynolds number (Re) and Prandtl number (Pr); and D_{hy} is the channel hydraulic diameter.

223 　　The temperature of the j+1 node is calculated as (Guo, 2016)

224
$$m_{ch}(h_{j+1} - h_j) = k \cdot (Pe/1000) \cdot \delta L_j \cdot (T_h^j - T_c^j) \text{ , and} \tag{19}$$

225
$$T^{j+1} = T^j + \frac{h_{j+1} - h_j}{c_p} \tag{20}$$

226 where c_p is the specific heat of SCO_2 in the j node.

227 　　After the element length of the sub-heat exchanger is got, the total pressure drop of the channel

228 (ΔP) is obtained by

229
$$\Delta P = \sum_{j=1}^{N} 2 \cdot \frac{f_j}{D_{hy}} \cdot \frac{m_{ch}^2}{\rho A_{cs}} \cdot \delta L_j \tag{21}$$

230 where f is the Fanning friction factor, and ρ is the density of SCO_2.

231 　　In this study, the following PCHE structures of four different channel types were selected for

232 analysis: straight, zigzag, S-shaped fins, and airfoil fins. The models and size structures of the

233 channel unit are shown in Fig. A1, A3, A5, and A7, respectively. The computational formulas of the

234 hydraulic diameter, cross-sectional area, and perimeter for four different recuperator types are

235 shown in Table 1.

236 Table 1. Computational formula of the hydraulic diameter, cross-sectional area, and perimeter of

237 four different recuperator types.

	Hydraulic diameter (mm)	Cross-sectional area (mm²)	Perimeter (mm)
Straight	$D_{hy} = \frac{\pi}{2} \cdot D / (1 + \frac{\pi}{2})$	$A_{cs} = \pi \cdot (\frac{D}{2})^2$	$Pe = \pi \cdot (\frac{D}{2}) + D$
Zigzag	$D_{hy} = \frac{\pi}{2} \cdot D / (1 + \frac{\pi}{2})$	$A_{cs} = \pi \cdot (\frac{D}{2})^2$	$Pe = \pi \cdot (\frac{D}{2}) + D$
S-shape (Rao et al., 2019)	$D_{hy} = \frac{2 \cdot g_f \cdot H_{ch}}{g_f \cdot H_{ch}}$	$A_{cs} = g_f \cdot H_{ch}$	$Pe = 2.18 \cdot (g_f + H_{ch})$
Airfoil (Cui et al., 2018)	$D_{hy} = \frac{4 \cdot (L_b \cdot L_a - S_a) \cdot H_{ch}}{P_a \cdot H_{ch} + 2 \cdot (L_b \cdot L_a - S_a)}$	$A_{cs} = w_{ch} \cdot d_{ch}$	$Pe = \frac{P_a \cdot H_{ch} + 2 \cdot (L_b \cdot L_a - S_a)}{L_b}$

238

239   In addition, the correlations of Nu and f for four different channel types were investigated and

240 selected. The specific calculation correlations are expressed in Table 2. In this study, the mass flow-

241 rates of the LTR and HTR were fixed at 330 kg·m^{-2}·s^{-1} to ensure that Re is within the applicable

242 scope of Table 2. The Nu correlation of the airfoil fin was revised based on inlet Re.

243

244   Table 2. Correlations of Nu and f for four different recuperator types.

	Nu	f
Straight	$Nu = \dfrac{(f/2)\cdot(Re-1000)\cdot Pr}{1+12.7\cdot\sqrt{f/2}(Pr^{2/3}-1)}$ $2300\leq Re\leq10^6$, $0.6\leq Pr\leq10^5$	$f = \dfrac{(1.82lgRe-1.64)^{-2}}{4}$ $2300\leq Re\leq10^6$, $0.6\leq Pr\leq10^5$
Zigzag (Ngo et al., 2007)	$Nu = 0.184Re^{0.629}Pr^{0.317}$ $3500\leq Re\leq22000$, $0.75\leq Pr\leq2.2$	$f = 0.1924Re^{-0.091}$ $3500\leq Re\leq22000$, $0.75\leq Pr\leq2.2$
S-shape (Ngo et al., 2007)	$Nu = 0.174Re^{0.593}Pr^{0.43}$ $3500\leq Re\leq22000$, $0.75\leq Pr\leq2.2$	$f = 0.4545Re^{-0.34}$ $3500\leq Re\leq22000$, $0.75\leq Pr\leq2.2$
Airfoil	$Nu = 0.0318Re^{0.78}Pr^{0.4}$ $3500\leq Re\leq150000$, $0.6\leq Pr\leq0.8$ (Yoon et al., 2014)	$f = 0.08925(\dfrac{L_b}{L_f})^{-0.252}(\dfrac{W_f}{L_a})^{-0.255}Re^{-0.173(\frac{W_f}{L_a})^{-0.274}}$ $8000\leq Re\leq100000$ (Chu et al., 2017)

245   To examine the accurateness of the selected correlations, the theoretical calculation outcomes

246 of the correlations and literature results for each channel were compared. In the comparative analysis

247 of the straight channel (Meshram et al., 2016), the maximum temperature error and maximum

248 pressure drop error along the length were 1.9% and 3.8%, respectively (Fig. A2). As for the zigzag

249 channel (Meshram et al., 2016), the maximum temperature error and pressure drop error along the

250 length were 0.5% and 8.4%, respectively (Fig. A4). The maximum temperature and pressure

251 differences between the inlet and outlet errors of the S-shaped fin channel (Ngo et al., 2007) were

252 3.41% and 10%, respectively (Fig. A6). While in the comparative analysis of the airfoil fin channel

253 (Cui et al., 2018), the errors of Nu and pressure drop were 1.3% and 9.9%, respectively (Fig. A8).

254 The specific operating parameters of the aforementioned comparisons are shown in the

255 corresponding figures. These errors are permissible in engineering applications. According to the

256 aforementioned comparison results, *Nu* correlations and *f* correlations selected for the four channel

257 types are feasible.

258   The total channel length (*L*) is

259
$$L = \sum_{j=1}^{N} \delta L_j \qquad (22)$$

260 2.2 Economic model

261   In the economic analysis, the overall cycle cost includes the compressor, turbine, recuperator,

262 PHE, and PC components costs.

263   The compressor cost (C_{comp}) and turbine cost (C_{tb}) are functions of their output work based on

264 Carlson's study (Carlson et al., 2017):

265
$$C_{comp} = 6898 \cdot W_{comp}^{0.7865} \qquad (23)$$

266
$$C_{tb} = 7790 \cdot W_{tb}^{0.6842} \qquad (24)$$

267   The PHE cost (C_{PHE}) and PC cost (C_{PC}) are the functions of the average heat conductance based

268 on Carlson's study (Carlson et al., 2017):

269
$$C_{PHE} = 3500 \cdot KF_{PHE} \qquad (25)$$

270
$$KF_{PHE} = Q_{PHE} / (T_6 - T_5) \qquad (26)$$

271
$$C_{PC} = 2300 \cdot KF_{PC} \qquad (27)$$

272
$$KF_{PC} = Q_{PC} / (T_9 - T_1) \qquad (28)$$

273 where KF_{PHE} and KF_{PC} are the average heat conductance of the PHE and PC, respectively.

274   The recuperator cost is only a function of its volume, and the fabrication and interfacing costs

275 are not considered (Yoon et al., 2014). In this study, the recuperator metal density was 7940 kg·m⁻

276 ^{3}, and the cost of the metal material of the recuperator was 120 USD·kg⁻¹ (Kim, E.S. et al., 2008).

277 Thus, the cost of the recuperator (C_{recup}) can be calculated as

278
$$C_{recup} = 7940 \cdot 120 \cdot V \qquad (29)$$

279 where *V* is the recuperator volume. The recuperator volume calculation formulas for the four

280 channel types are shown in Fig. A1, A3, A5, and A7, respectively.

281   The total cycle cost (C_{cycle}) can be obtained as

282
$$C_{cycle} = C_{comp} + C_{tb} + C_{PHE} + C_{PC} + C_{recup} \tag{30}$$

283　2.3 Exergy loss model

284　In this study, the exergy efficiency (η_{ex}) was introduced to calculate the exergy loss of PHE and

285　PC. E_L is the exergy loss of different components. E is the exergy value of the different nodes, and

286　is calculated by:

287
$$E_i = (h_i - h_0) - T_0 \cdot (s_i - s_0) \tag{31}$$

288　where s denotes the specific entropy of the system node. The subscript 0 represents the

289　environmental status. i is the node in Fig. 1. The ambient temperature (T_0) was 20 °C, and the

290　ambient pressure (P_0) was 101.33 kPa.

291　The exergy efficiency (η_{ex}) of the heat exchanger can be written by:

292
$$\eta_{ex} = (E_{c,out} - E_{c,in}) / (E_{h,in} - E_{h,out}) \tag{32}$$

293　The exergy efficiency of PHE and PC was 80% (Cheng et al., 2021) and 50% (Zhang et al.,

294　2021), respectively. The exergy losses calculation formulas of every component are listed in Table

295　3.

296　Table 3. Exergy equation of each component.

Component	Equations
Compressor	$m_{CO_2}(1-\Phi_R)(w_{mc}+E_1-E_2)-E_{L,mc}=0$
Recompressor	$m_{CO_2}\Phi_R(w_{rc}+E_9-E_{10})-E_{L,rc}=0$
Turbine	$m_{CO_2}(E_6-E_7-w_{tb})-E_{L,tb}=0$
LTR	$m_{CO_2}(1-\Phi_R)(E_2-E_3)+m_{CO_2}(E_8-E_9)-E_{L,LTR}=0$
HTR	$m_{CO_2}(E_4+E_7-E_5-E_8)-E_{L,HTR}=0$
PHE	$m_{CO_2}(1-1/\eta_{ex,PHE})(E_5-E_6)-E_{L,PHE}=0$
PC	$m_{CO_2}(1-\eta_{ex,PC})(E_9-E_1)-E_{L,PC}=0$

297　2.4 Iteration process

298　The temperature and pressure states from nodes 1 to 10 were obtained by iterating the

299　aforementioned thermodynamic and economic models. The cycle efficiency, cost, and exergy loss

300　were evaluated. A detailed flowchart of computational course is illustrated in Fig. 3. The input

301　parameters included the performance parameters of the main components of the recompression

302　cycle and recuperator structural parameters. First, the important nodes of the initial temperature and

303 pressure were initialized, and then, according to the compressor model to obtain T_2. Next, T_8 was

304 initialized based on turbine model to obtain T_7. Based on the recuperator model, T_9, T_3, P_3, P_8, and

305 T_{10} were calculated based on the compressor model. T_4 was evaluated based on the principle of

306 energy conservation. Then, $T_8{}^*$, T_5, T_7, $P_5{}^*$, and $P_7{}^*$ were calculated according to the recuperator

307 model. We evaluated the values of T_8 and $T_8{}^*$ until their relative errors are lower than 0.001% and

308 the relative errors of P_5 and $P_5{}^*$, P_7 and $P_7{}^*$ until their relative errors are lower than 0.005%. It can

309 be implied that the thermodynamic iterative process converges. Next, according to the obtained state

310 parameter values of each node, the cost, thermal efficiency, and exergy loss were calculated using

311 the aforementioned formulas. Finally, the Pareto frontier of the optimization solution was obtained
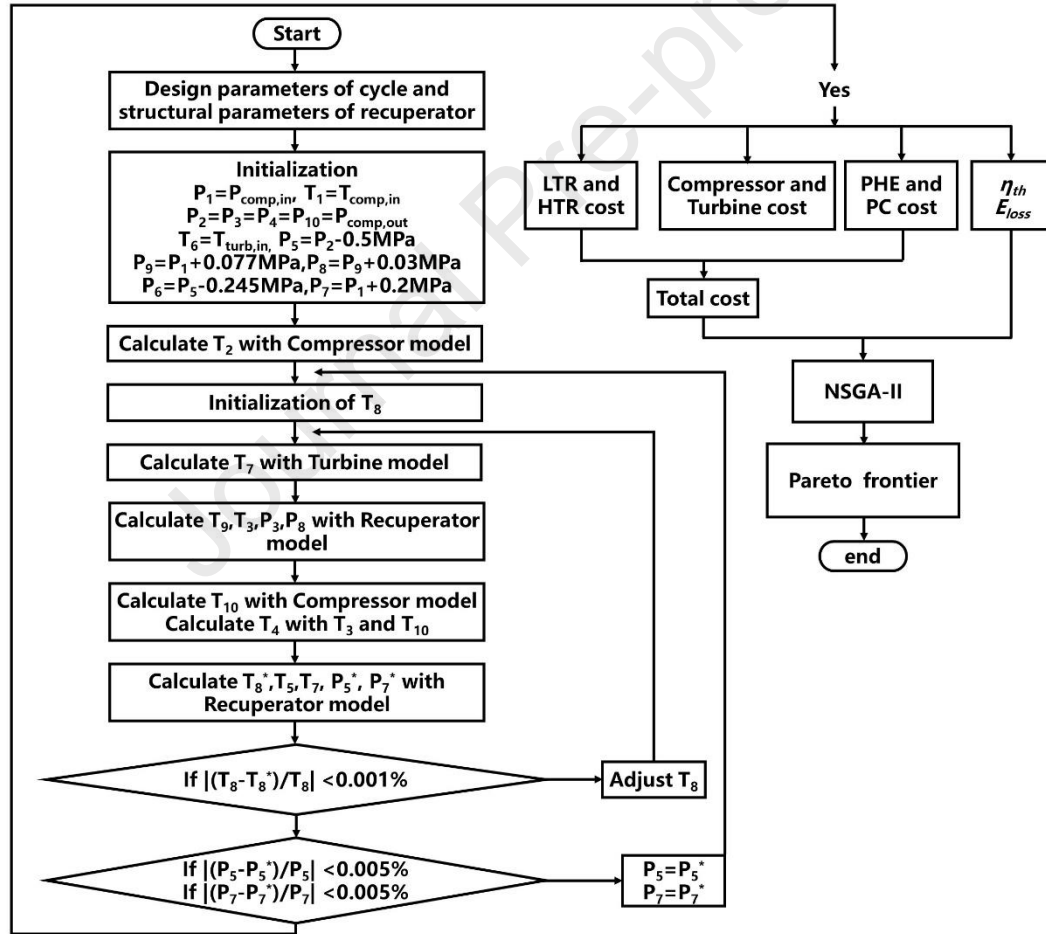
312 based on the NSGA-II.

313

314 Fig. 3. Flowchart of the iteration process for the thermodynamic–economic model.

315

316 2.5 Multi-objective optimization algorithm

317 In the present study, the most common NSGA-II algorithm was chosen to carry out multi-

318　objective optimization of cyclic performance. The optimization toolbox used herein was PlatEMO

319　3.5 (Tian et al., 2017), which can be downloaded from GitHub. It is an evolutionary multi-objective

320　optimization platform that has more than 150 open-source evolutionary algorithms. In PlatEMO,

321　the NSGA-II algorithm is developed to optimize two or three objective functions. Moreover, the

322　multi-objective optimization process is highly integrated. The population size and number of

323　evaluations are required to complete the optimization process on the application module GUI

324　interface. In the optimization process of this study, the population size was 70, and the number of

325　evaluations was 10000.

326　　　After obtaining the Pareto optimal frontier curve, it is necessary to analyze and

327　comprehensively evaluate the optimal frontier. The TOPSIS method is a resultful method commonly

328　used in multi-objective decision analysis. Its basic principle is to sort the gap between the evaluation

329　object and the optimal or worst solution. If the gap of optimal solution is minimum, and gap of worst

330　solution is maximum, it is the best solution. The algorithm content is detailed in Rao's (Rao et al.,

331　2019) and Jin's (Jin et al., 2022) studies. SPSSPRO (Suzhou Zhongyan Network Technology Co.),

332　a scientific platform of professional statistical services, was used to get the optimal solution of the

333　optimization results. SPSSPRO can directly perform online data processing, data analysis, and other

334　operations. It can also use various comprehensive evaluation algorithms with easy operation using

335　a GUI interface. In the TOPSIS analysis, the entropy weight method is adopted to obtain the optimal

336　solution. Entropy weights were automatically solved according to entropy information of the

337　algorithm. In the two-objective TOPSIS analysis, the weight of the cycle efficiency and cost were

338　42.67% and 57.33%, respectively. In the three-objective TOPSIS analysis, the weight of the cycle

339　efficiency, cost, and exergy loss was 34.6%, 30.6%, and 34.8%, respectively.

340　2.6 Model validation

341　　　To examine the accuracy of the recompression cycle thermodynamic model, the cycle

342　efficiency and PHE inlet temperature (T_5) at different turbine inlet temperatures were used to

343　compare the results calculated using Dyreby's modeling framework (Dyreby, 2014). The detailed

344　input parameters are listed in Table 4, where T is temperature, P is pressure, Φ_R is recompression

345　fraction. There is a one-to-one correspondence between the turbine inlet temperature, recompression

346　fraction, and mass flow-rate. The comparison results are displayed in Fig. 4. The maximum error is

347　2.5% for cycle efficiency and 0.5% for T_5. These errors may be caused by the deviation in the SCO_2

348 physical property parameter database and the construction of the algorithm. The comparative results

349 show that the model established in the present study can be performed for further thermodynamic

350 evaluation and multi-objective optimization.

351

352 Table 4. Input parameters for the model verification.

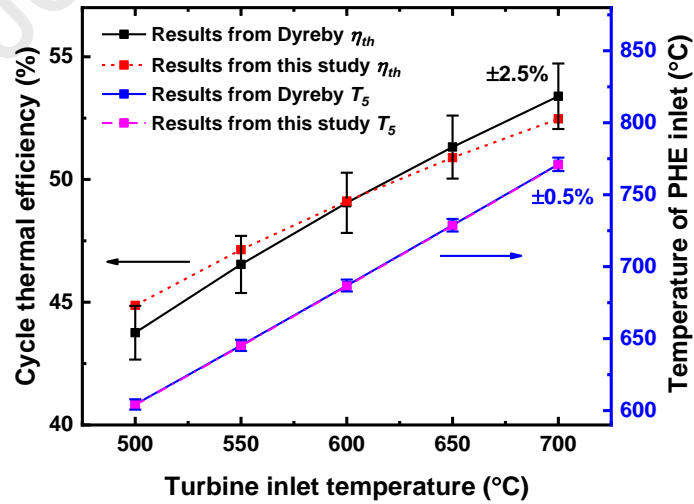| Input Parameter	Value	Unit
$T_{tb,in}$	500; 550; 600; 650; 700	°C
$T_{mc,in}$	32	°C
$P_{mc,in}$	7646; 7636; 7628; 7621; 7615	kPa
$P_{mc,out}$	25	MPa
η_{tb}	0.93	
η_{mc}	0.89	
η_{rc}	0.89	
ε_{LTR}	0.95	
ε_{HTR}	0.95	
Φ_R	0.3487; 0.355; 0.3604; 0.3646; 0.3682	
m_{CO_2}	106.94; 96.526; 88.028; 80.954; 74.968	kg/s

353

354

355 Fig. 4. Comparison of efficiency and T_5 with Dyreby's results (Dyreby, 2014).

356

357 **3. Thermodynamic and economic analysis**

358   In this section, the cycle performances of four types of channels were compared. Based on this,

359   a combination of HTR and LTR channels with optimal cycle performance was obtained. Then, the

360   influences of the recuperator enthalpy efficiency (ε) and recompression fraction (Φ_R) on cycle

361   efficiency, cost, and exergy loss were discussed.

362   3.1 Comparison of different channels and the optimal combination

363   The cycle performances of four different recuperator channels were contrasted under the same

364   conditions: $T_{tb,in}$=700 °C, $P_{mc,in}$=7615 kPa, Φ_R=0.3682, and m_{CO_2}=74.968 kg·s^{-1}. The hydraulic

365   diameter was 1.68 mm. The other parameters were consistent (Table 4). The size structures of the

366   four channels are presented in Appendix A. To further obtain the optimal combination of LTR and

367   HTR types, a cycle performance analysis of 16 combinations was performed. The comparison

368   results are listed in Appendix Table A1 and A2. In Table A1 and A2, L and H represent LTR and

369   HTR, respectively. The subscripts st, z, s and af represent the straight, zigzag, S-shaped fin and

370   airfoil fin, respectively. It can be observed that when the LTR and HTR are straight channels, the

371   thermal efficiency is the lowest, and the cost and exergy loss are the highest. This indicates that the

372   performance of the other three recuperator types is superior to that of the traditional straight channel.

373   The zigzag channels have maximal Nu and overall heat transfer coefficient, followed by airfoil fins,

374   S-shaped fins, and straight channels. This lead to the same variation trend of cycle efficiency, and

375   the contrary trend of cost and exergy loss. When the LTR is the type of zigzag, the thermal efficiency

376   is the highest, because the zigzag type possesses the maximal overall heat transfer coefficient. In

377   addition, when the LTR and HTR are zigzag channels, the lowest exergy loss is obtained. This is

378   because the pressure loss of the zigzag channel is the highest, which makes the pressure of P_7 enlarge,

379   resulting in a decrease in the exergy difference between the turbine inlet and outlet. Besides, the

380   total cycle cost is the lowest owing to the volume reduction with a more compact structure.

381   Therefore, the combination of the LTR with the zigzag channel and HTR with the zigzag channel is

382   considered for the recuperator analysis in the following section.

383   The statistics show that overall installed capacity of concentrated solar power worldwide

384   reached 6.3 GW in 2019 (Merchán et al., 2022). The efficiency of this paper is 4.6% higher than

385   that of steam Rankine cycle (47.94%) (Sharma et al., 2022). If the SCO$_2$-BC efficiency result of this

386   study replaces the traditional steam Rankine cycle, the annual CO$_2$ emission can be directly reduced

387   by 0.7 Gt.

388   3.2 Effect of recuperator enthalpy efficiency ε

389   Fig. 5(a) and 5(b) show the influences of different ε_{LTR} and ε_{HTR} on the efficiency, cost, and

390   exergy loss, respectively. The cycle efficiency and total cost increase with an increase in the ε.

391   However, the ε_{HTR} impacts the cycle efficiency more than ε_{LTR}. When the enthalpy efficiency

392   increases from 0.7 to 0.99, the cycle efficiency is influenced by HTR increases (from 44.2% to

393   54.3%), whereas that of LTR increases from 48.3% to 53.6%. It can be concluded that the high ε_{HTR}

394   is more important for cycle efficiency. Nevertheless, the ε_{LTR} is more sensitive to the cycle cost. The

395   main reason is that the LTR is a zigzag channel, and the increase in its efficiency results in an evident

396   increase of volume. Moreover, total cost sharply increases when the efficiency of the LTR and HTR

397   is greater than 0.95, which is the volume increase. The exergy loss of components decreases with

398   an increase in enthalpy efficiency because the greater the heat recovery with an increase in enthalpy

399   efficiency, the more useful energy can be recovered. Further analysis shows that the ε_{HTR}

400   significantly affects the exergy loss. This is partly because the LTR operates at a lower temperature

401   range than the HTR, and the change in enthalpy efficiency leads to a greater floating in heat recovery.

402   In addition, the aforementioned results imply that the optimized ε_{LTR} and ε_{HTR} are required to fulfil

403   higher thermal efficiency, lower exergy loss, and lower overall cost.
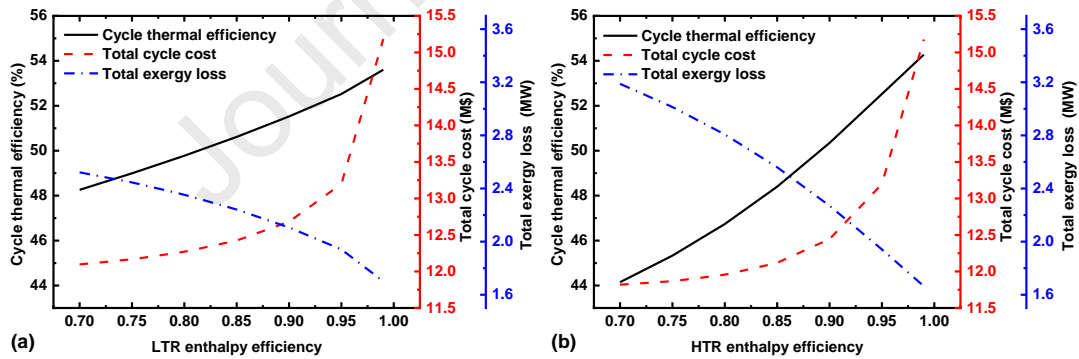

405   Fig. 5. Influences of ε on the cycle efficiency, cost, and exergy loss: (a) LTR; (b) HTR.

406

407   Fig. 6(a) and 6(b) show the impact of different LTR and HTR enthalpy efficiencies, respectively,

408   on the cost of different components. The recuperator cost is mainly affected by the change in

409   enthalpy efficiency and increases with an increase in efficiency. When the enthalpy efficiency is

410   greater than 0.95, the recuperator cost increases sharply because the enthalpy efficiency significantly

411   affects the recuperator volume. Meanwhile, the compressor cost gradually decreases with an

412   increase in enthalpy efficiency. The influence of the HTR enthalpy efficiency on the compressor

413   cost is evident. With the increase in the enthalpy efficiency, T_9 and T_{10} significantly reduce, which

414   results in a decrease in the compressor power consumption. In particular, the ε_{HTR} significantly

415   impacts the temperature change of T_9 and T_{10}. In summary, enthalpy efficiency negligibly affects
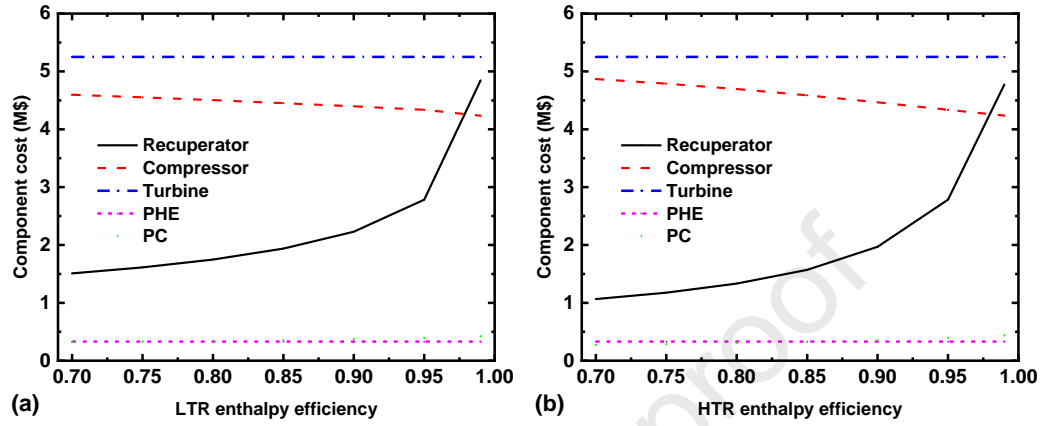
416   the costs of the turbine, PHE, and PC.

417

Fig. 6. Influences of ε on the component costs: (a) LTR; (b) HTR.

418

419

420   Fig. 7(a) and 7(b) show the influence of different LTR and HTR enthalpy efficiencies on the

421   exergy loss of different components. The LTR exergy loss is affected by the change in the HTR

422   enthalpy efficiency and gradually decreases with increasing the HTR enthalpy efficiency. The HTR

423   heat recovery increases with an increase in the HTR enthalpy efficiency, thereby reducing the LTR

424   heat load. The LTR enthalpy efficiency remains unchanged, and the LTR exergy loss changes are

425   consistent with those of the heat load. The HTR exergy loss decreases with an increase in the LTR

426   enthalpy efficiency because this increase reduces the heat load of the HTR. The HTR exergy loss

427   first increases and then decreases with increasing the HTR enthalpy efficiency. The increase is

428   related to the heat load, and the decrease is related to the heat exchange area, which sharply increases

429   with an increase in efficiency. The turbine exergy loss decreases with an increase in enthalpy

430   efficiency. When the enthalpy efficiency is greater than 0.95, the turbine exergy loss sharply

431   decreases The heat exchange area of the recuperator increases when the efficiency is greater than

432   0.95, and the pressure loss of the recuperator increases. This leads to a significant increase in P_7,

433   thus resulting in an increase in E_7, which can be determined by combining the formula in Table 3.

434   In summary, enthalpy efficiency negligibly affects the exergy loss of the compressor, PHE and PC.
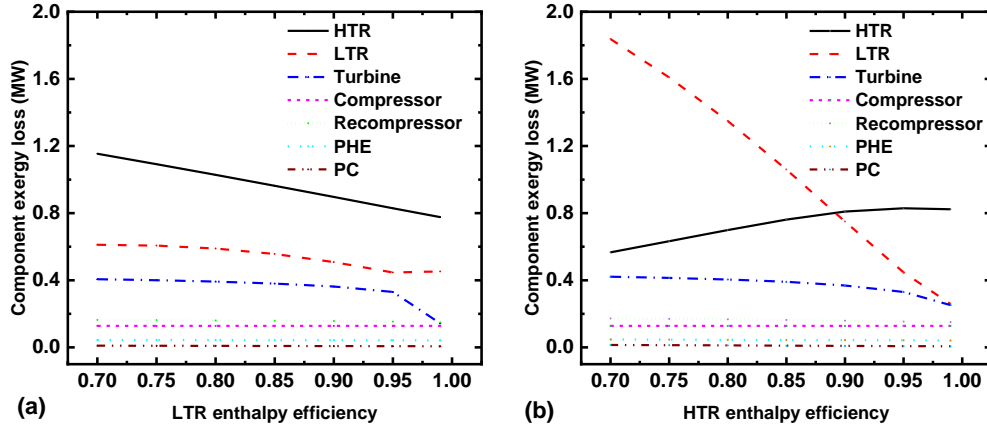
(a) LTR enthalpy efficiency **(b)** HTR enthalpy efficiency

Fig. 7. Effects of ε on the exergy loss of the components: (a) LTR; (b) HTR.

3.3 Effect of recompression fraction Φ_R

Fig. 8 shows the influences of different Φ_R on efficiency, cost, and exergy loss, respectively. When the recompression fraction is 0.33, the cycle efficiency is the highest, the exergy loss is the lowest, and the total cost locally peaks at this point. When the recompression fraction increases from 0.33 to 0.4, the total cycle cost decreases gradually. While the recompression fraction is greater than 0.42, the total cycle cost increases, and its maximum value is higher than the peak value when the recompression fraction is 0.33. This variation is dominated by the recompressor and precooler cost. As the Φ_R increases, recompressor cost increases and precooler cost reduces. The results list that when the recompression fraction is <0.4, high cycle efficiency and low component exergy loss incur a high cost. Therefore, an optimized recompression fraction is required to achieve high efficiency, low exergy loss, and low cost.
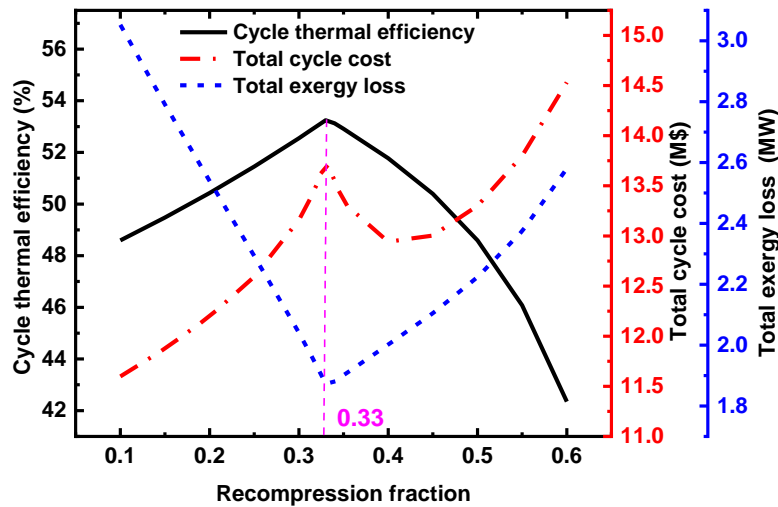

Fig. 8. Influences of Φ_R on the cycle efficiency, cost, and exergy loss.

451

Fig. 9(a) and 9(b) show the influence of different Φ_R on the cost and exergy loss of each

component, respectively. Recompression fraction significantly affects the recuperator cost. When

the recompression fraction is 0.33, the total recuperator cost has a maximum peak value. The

compressor cost gradually increases with an increase in the recompression fraction. The main reason

for this is that much more SCO_2 flowed through the recompressor with increasing recompression

fraction, which increases its compression work. The PC cost gradually decreases with an increase

in the recompression fraction. The heat transfer quantity of PC decreases with an increase in the

recompression fraction owing to the mass that flows through the PC reduces. The cost of PC falls

off due to the decrease of the heat transfer area. In Fig. 9(b), as to the exergy losses of the HTR,

LTR, and turbine, an inflection point exists when the recompression fraction is 0.33. The LTR exergy

loss gradually increases with an increase in recompression fraction, whereas the HTR exergy loss

exhibits the opposite trend. This is due to the fact that with increasing recompression fraction, the

heat recovery quantity by LTR increases, whereas the heat recovery through HTR decreases. Under

the condition of invariable efficiency, the exergy loss changes with the variation in the heat recovery

quantity. Moreover, when the recompression fraction is 0.33, the turbine exergy loss is the lowest,

which implies that it can do the most useful work. With an increase in the recompression fraction,

the mass of SCO_2 flowing into the recompressor increases, whereas the mass in the main compressor

decreases. This leads to a decrease in the exergy loss of the main compressor with an increase in the

recompression fraction, whereas the exergy loss of the recompressor is inverse. This conclusion can

be deduced by combining the relevant calculation formulae presented in Table 3.
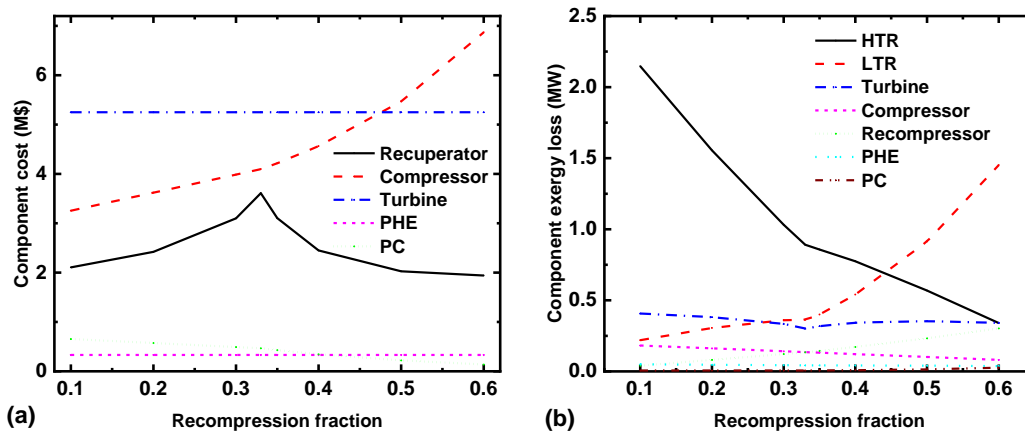

472

Fig. 9. Influences of Φ_R on the (a) component cost and (b) component exergy loss.

474

4. Multi-objective optimization

476   The aforementioned consequences state that the cycle efficiency, cycle cost, and exergy loss

477 are significantly affected by the recuperator enthalpy efficiency and recompression fraction.

478 However, they are in a mutually restrictive relationship. Thus, multi-objective optimization was

479 executed to balance the relationship between them, and to obtain the optimal cycle performance. In

480 the optimization of this section, the recompression fraction varied from 0.2 to 0.5, and the enthalpy

481 efficiency of the LTR and HTR varied from 0.8 to 0.96. The other parameters were the same as

482 previous ones.

483 4.1 Two-objective optimization

484   Fig. 10 shows the Pareto frontier curve of the two-objective optimization between cycle

485 efficiency and cost. Five typical points were labeled based on different decisions. The target values

486 corresponding to these five points and corresponding parameter values are listed in Table 5. As

487 shown in Fig. 10, within the selection range of optimization parameters, the value of cycle efficiency

488 ranges from 44.50% to 53.77%, and the total cost ranges from M\$10.71 to M\$14.50. The maximum

489 cycle efficiency is at point E, while the cost is the highest. The lowest cycle efficiency is at point A,

490 where the cost is the lowest. The TOPSIS optimal solution is point C, where the cycle efficiency is

491 49.37% and the cycle cost is M\$11.58. As shown in Table 5, when the three optimization parameters

492 (ε_{LTR}, ε_{LTR} and Φ_R) exhibit the lowest values, the corresponding case refers to point A. When the

493 enthalpy efficiency is the highest and the recompression fraction is close to 0.33, the corresponding

494 case is point E. Point C corresponds to the TOPSIS optimal solution, indicating that the HTR should

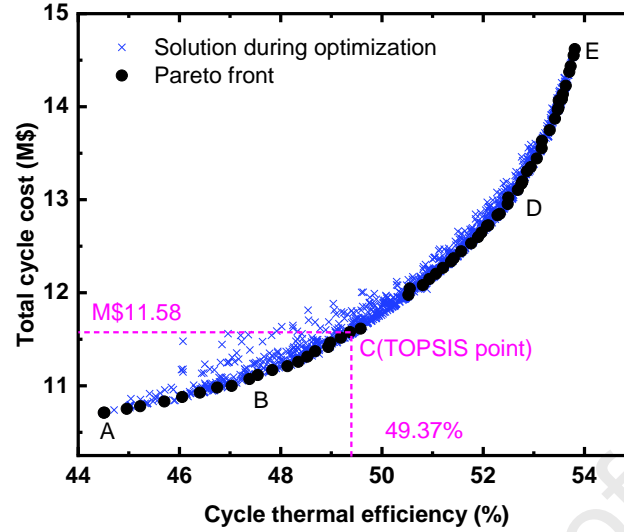495 maintain high enthalpy efficiency.

496

497   Fig. 10. Pareto frontier curve of the two-objective optimization between cycle efficiency and cost.

498

499   Table 5. Five typical points and corresponding parameter values with regard to the Pareto frontier

500   of two objectives.

Point	Decision variables			Two objectives	
	Φ_R	ε_{LTR}	ε_{HTR}	η_{th} (%)	C_{cycle} (M\$)
A	0.2	0.8	0.8	44.50	10.71
B	0.22	0.88	0.90	48.53	11.31
C	0.22	0.87	0.95	49.37	11.58
D	0.31	0.93	0.96	52.50	13.02
E	0.34	0.96	0.96	53.77	14.50

501

502   4.2 Three-objective optimization

503   To further consider the influence of exergy loss of the main components on the cycle

504   performance analysis, Fig. 11 shows the Pareto frontier curve of three-objective optimization among

505   cycle efficiency, exergy loss of the components, and cost. As shown in Fig. 11, four typical points

506   were labeled based on different decisions. The target values corresponding to these four points and

507   corresponding parameter values are illustrated in Table 6. Within the selected range of the

508   optimization parameters, the variation range of cycle efficiency and cost is consistent with that of

509   the two-objective optimization, and the exergy loss of the components is between 1.77 and 3.7 MW.

The maximum cycle efficiency is at point IV, where the overall cost is the highest and the exergy loss is minimal. The minimum cycle efficiency is at point I, where the cost is the lowest and exergy loss is the maximum. The TOPSIS solution is point III, where the cycle efficiency is 52.15 %, the cycle cost is M$12.76, and the exergy loss is 2.05 MW. As listed in Table 6, when the optimization parameters are the lowest, the corresponding case relates to point I. When the enthalpy efficiency is the highest and the recompression fraction is close to 0.33, the corresponding case is point IV. The TOPSIS optimal solution indicates that the HTR must maintain high enthalpy efficiency and the recompression fraction is closer to the local peak point (0.33). Meanwhile, Φ_R is the main parameter to compare the results of three-objective optimization with two-objective optimization. The TOPSIS optimal Φ_R of three-objective optimization is closer to the inflection point (0.33) than that of two-objective optimization, and the TOPSIS optimal η_{th} of three-objective optimization is larger than that of two-objective optimization. The results show that the three-objective optimization is more comprehensive due to introducing the second law of thermodynamics, and its TOPSIS optimal solution is more reasonable.

Moreover, the TOPSIS cycle efficiency solution of three-objective optimization exceeds that of two-objective optimization by 2.78%. Based on the analysis in Section 3.1, in the application of concentrated solar power, the annual CO_2 emission can be directly reduced by 0.43 Gt through this optimization improvement. The results of current study are of great significance to solve the problems of environment and energy.
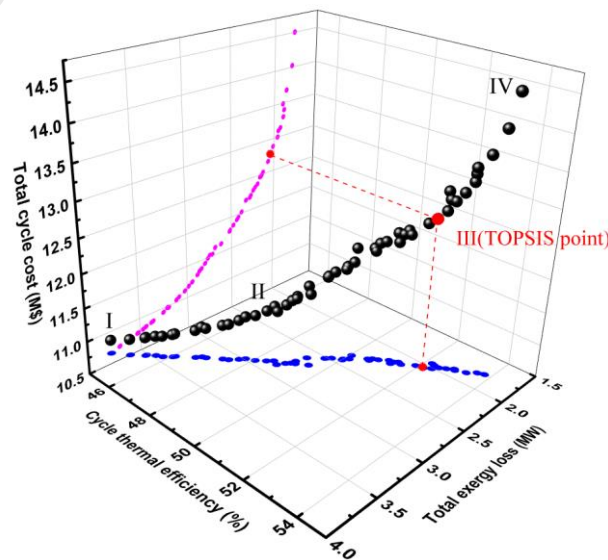

Fig. 11. Pareto frontier curve of the three-objective optimization among the cycle efficiency, total

531   exergy loss, and total cycle cost.

532

533   Table 6. Four typical points and corresponding parameter values with regard to the Pareto frontier

534   of three objectives.

Point	Decision variables			Three objectives		
	Φ_R	ε_{LTR}	ε_{HTR}	η_{th} (%)	C_{cycle} (M\$)	E_{loss} (MW)
I	0.2	0.8	0.8	44.50	10.71	3.7
II	0.2	0.81	0.94	48.06	11.22	2.94
III	0.34	0.89	0.96	52.15	12.76	2.05
IV	0.34	0.96	0.96	53.77	14.50	1.77

5. Conclusions

536   Based on the typical SCO$_2$ Brayton recompression cycle process, a thermodynamic and

537 economic mathematical model coupling the recuperator performance was developed and verified.

538 Thermodynamic–economic analysis and multi-objective optimization were performed considering

539 different types and designs of printed circuit recuperator. Recuperator channel types included

540 straight, zigzag, S-shaped fins and airfoil fins. Design parameters included the ε_{LTR}, ε_{HTR}, and Φ_R.

541 Their influences on the thermodynamic and economic performance were analyzed. Multi-objective

542 optimization computation was carried out by adopting NSGA-II algorithm and TOPSIS method. We

543 conclude on the following points:

544   (1) The correlations of Nusselt number (Nu) and Fanning friction factor (f) for the four different

545 channel types were selected and verified with corresponding literature results, the errors were within

546 10%. The cycle performances of the four different recuperator channel types were contrasted with

547 same design parameters. The combination of the LTR with a zigzag channel and HTR with a zigzag

548 channel exhibits the best overall cycle performance when the channel hydraulic diameter is the same.

549   (2) The enthalpy efficiency of the recuperator and recompression fraction significantly affects

550 cycle efficiency, cost, and exergy loss. The thermal efficiency is more sensitive to the HTR enthalpy

551 efficiency than that of the LTR. The exergy losses of the main components are reversed. The LTR

552 enthalpy efficiency is more sensitive to the cycle cost. There is a local inflection point of the

553 recompression fraction (Φ_R=0.33) where the cycle efficiency is maximum and exergy loss is

554 minimum. The total cost is the local peak value at the inflection point.

555 (3) Two-objective optimization considering cycle efficiency and cost was performed. The

556 TOPSIS optimal solution was η_{th}=49.37%, and C_{cycle} = 11.58 M\$. Then, the three-objective

557 optimization considering cycle efficiency, cost, and exergy loss was conducted. The TOPSIS

558 optimal solution is η_{th}=52.15%, C_{cycle} =12.76 M\$, and E_{loss} = 2.05 MW. The optimal solution of

559 three-objective considering the efficiency, total cost, and exergy loss is more comprehensive than

560 that of two-objective optimization. In the application of concentrated solar power, the annual carbon

561 dioxide emission can be directly reduced by 0.43 Gt through optimization efficiency improvement.

562 The results of the current study offer a promising route to high-efficiency and low-cost

563 applications of SCO_2-BC by rational selection of recuperator types and designs. Moreover, the

564 influence of recuperator types and designs on the off-design, part-load, and dynamic characteristics

565 of SCO_2-BC will be further discussed in our future studies.

566

572

573 **Data availability**

574 The processing algorithm used to run simulations (as well as details on its' usage) can be

575 found in the following Github repository: https://github.com/linjin2016/SCO2BCthermodynamic.

576

577 **Appendix. Nomenclature**

578 A_{cs} cross-sectional area of the channel, mm^2

579 C cost, \$

580 c_p specific heat capacity, kJ·kg^{-1}·K^{-1}

581 D_{hy} hydraulic diameter of channel, mm

582 E exergy value, kW

583 E_L exergy loss of different components, kW

584 f Fanning friction factor

585   G mass flow-flux, $kg \cdot m^{-2} \cdot s^{-1}$

586   h specific enthalpy, kJ/kg, or convective heat transfer coefficient, $kW \cdot m^{-2} \cdot K^{-1}$

587   k overall heat transfer coefficient, $kW \cdot m^{-2} \cdot K^{-1}$

588   KF average heat conductance, $kW \cdot K^{-1}$

589   L total channel length, m

590   m mass flow-rate, $kg \cdot s^{-1}$

591   N sub-heat exchanger element

592   Nu Nusselt number

593   Pe perimeter of a cross section, mm

594   Pr Prandtl number

595   Q heat transfer capacity, kW

596   s specific entropy, $kJ \cdot kg^{-1} \cdot K^{-1}$

597   t wall thickness, mm

598   T temperature, °C

599   V volume of recuperator, m^3

600   w specific work, $kJ \cdot kg^{-1}$

601   W total work, kJ

602

603   *Greek symbols*

604   δL_j channel length of sub-element, m

605   ΔP total pressure drop in the channel, kPa

606   δQ_{ch} total heat transfer rate of the sub-channel unit, kW

607   ΔT_j node temperature difference between the cold and hot streams, K

608   ε enthalpy efficiency

609   η_c isentropic efficiency of the compressor

610   η_{tb} isentropic efficiency of the turbine

611   η_{th} cycle thermal efficiency

612   λ thermal conductivity, $W \cdot m^{-1} \cdot K^{-1}$

613   ρ density, $kg \cdot m^{-3}$

614   Φ_R recompression fraction

615

616   *Subscripts*

617   *af*   airfoil fin

618   *c*   cold fluid

619   *ch*   channel

620   CO_2 carbon dioxide

621   *comp*   compressor

622   *h*   hot fluid

623 *in*   inlet

624 *isen*   isentropy

625 *m*   metal material

626 *out*   outlet

627 *PC*   precooler

628 *PHE*   primary heat exchanger

629 *r*   recompressor

630 *RC*   recompression

631 *recup*   recuperator

632 *s*   S-shaped fin

633 *st*   straight channel

634 *tb*   turbine

635 *z*   zigzag channel

636 *Abbreviations*

637 SCO$_2$ supercritical carbon dioxide

638 PCHE   printed circuit heat exchanger

639

640 **Appendix A**

641

Parameters	Dimension (mm)
height *H*	4
width *W*	3.3
diameter *D*	2.75

Volume:

$$V = \frac{H \cdot W}{1000000} \cdot L \cdot \frac{m_{CO_2}}{m_{ch}}$$

642
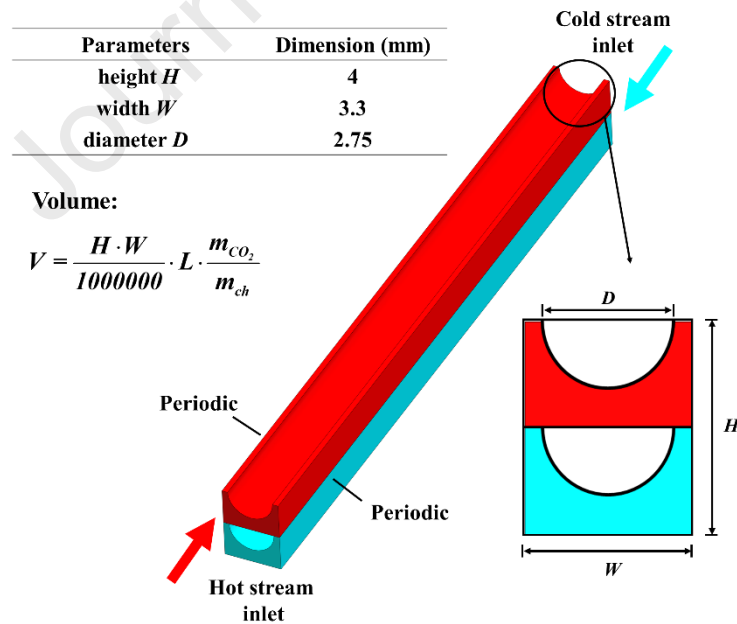
643 Fig. A1. Schematic diagram of a single straight PCHE unit.
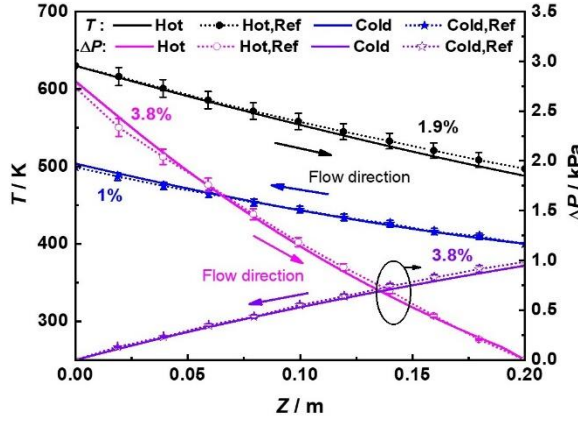
644

645

646   Fig. A2. Verification of the correlations between the theoretical results and simulation results for a

647   straight PCHE.

648

649   Fig. A3. Schematic diagram of a single zig PCHE unit.
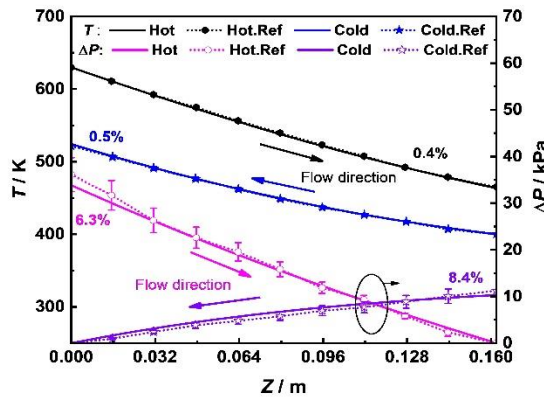
650

651

652   Fig. A4. Verification of the correlations between the theoretical results and simulation results for a

653   zigzag PCHE.

654

Parameters	Dimension (mm)
fin width W_f	0.8
fin pitch in x direction P_x	7.565
fin pitch in y direction P_y	3.426
fin gap g_f	2.4
channel depth H_{ch}	1.3
channel thickness t_p	1.8

Volume:

$$V = \frac{(g_f + w_f) \cdot t_p}{1000000} \cdot L \cdot \frac{2 \cdot m_{CO_2}}{m_{ch}}$$

655

656 Fig. A5. Schematic diagram of an S-shaped fin PCHE unit.

657

Experimental conditions: length of channel is 650 mm

Case	Hot side			Cold side		
	T_{in} (K)	P_{out} (kPa)	m (kg/s)	T_{in} (K)	P_{out} (kPa)	m (kg/s)
1	393.57	6004.55	0.0421	321.42	10017.96	0.0246
2	393.53	5995.41	0.01435	308.13	7742.94	0.00961

Correlations used in this study

$$Nu = 0.174 Re^{0.593} Pr^{0.43}$$

$$f_F = 0.4545 Re^{-0.34}$$

$(3500 \leq Re \leq 23000;$
$0.75 \leq Pr \leq 2.2)$

Comparison result

Case 1	Experimental results	Theoretical calculation	Error
ΔT_h	69.32 K	66.96 K	3.41%
ΔP_h	35.45 kPa	34.90 kPa	1.56%
ΔT_c	63.32 K	61.20 K	3.35%
ΔP_c	22.04 kPa	23.49 kPa	6.58%
Case 2	Experimental results	Theoretical calculation	Error
ΔT_h	82.5 K	80.51 K	2.41%
ΔP_h	2.59 kPa	2.85 kPa	10.0%
ΔT_c	83.06 K	81.38 K	2.0%
ΔP_c	7.06 kPa	7.19 kPa	1.84%

658

659 Fig. A6. Verification of the correlations between the theoretical results and experimental results for
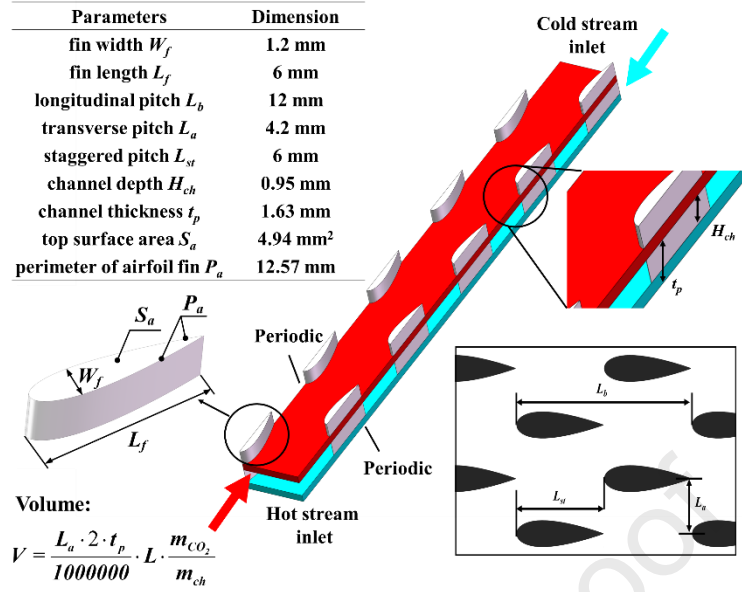
660 S-shaped fin.

661

Parameters	Dimension
fin width W_f	1.2 mm
fin length L_f	6 mm
longitudinal pitch L_b	12 mm
transverse pitch L_a	4.2 mm
staggered pitch L_{st}	6 mm
channel depth H_{ch}	0.95 mm
channel thickness t_p	1.63 mm
top surface area S_a	4.94 mm²
perimeter of airfoil fin P_a	12.57 mm

Volume:

$$V = \frac{L_a \cdot 2 \cdot t_p}{1000000} \cdot L \cdot \frac{m_{CO_2}}{m_{ch}}$$

Fig. A7. Schematic diagram of an airfoil fin unit.

Simulation conditions: length of channel is 122 mm

Case	Hot side			Cold side		
	T_{in} (K)	P_{out} (MPa)	m (g/s)	T_{in} (K)	P_{out} (MPa)	m (g/s)
1	553.05	2.52	0.4335	381.05	8.28	1.8912
2	553.05	2.52	0.4335	381.05	8.28	1.4184

Comparison result

Case 1	Simulation results	Theoretical calculation	Error
Nu_c	65.89	66.78	1.3%
ΔP_c	674.15 Pa	641 Pa	4.9%
Case 2	Simulation results	Theoretical calculation	Error
Nu_c	53.39	53.13	0.5%
ΔP_c	429.6 Pa	387 Pa	9.9%

Correlations used in this study

$$Nu = 0.0318 Re^{0.78} Pr^{0.4} \quad (3000 \le Re \le 150000; \ 0.6 \le Pr \le 0.8)$$

$$f_D = 0.357 (\frac{L_b}{L_f})^{-0.252} (\frac{W_f}{L_a})^{-0.255} Re^{-0.173(\frac{W_f}{L_a})^{0.274}} \quad (8000 \le Re \le 100000)$$

Fig. A8. Verification of the correlations between the theoretical results and simulation results for an airfoil fin PCHE.

Table A1. Cycle performance with different recuperator types (L_{st} and L_z).

Performance	$L_{st}H_{st}$	$L_{st}H_z$	$L_{st}H_s$	$L_{st}H_{af}$	L_zH_{st}	L_zH_z	L_zH_s	L_zH_{af}
LTR pressure drop of cold side (kPa)	5.96	5.96	5.96	5.96	44.35	44.35	44.35	44.35
LTR pressure drop of hot side (kPa)	21.38	21.38	21.38	21.38	177.38	177.38	177.38	177.38
HTR pressure drop of cold side(kPa)	9.19	65.99	19.40	16.99	9.26	66.54	19.56	17.14
HTR pressure drop of hot side (kPa)	32.92	247.4	70.32	61.79	32.40	243.62	69.22	60.82
LTR overall heat transfer coefficient (W·m^{-2}·K^{-1})	648.1	648.1	648.1	648.1	1005.8	1005.8	1005.8	1005.8
HTR overall heat transfer coefficient (W·m^{-2}·K^{-1})	755.9	1194.0	778.6	954.6	756.11	1194.3	778.75	954.77

LTR *Nu* of cold side	38.35	38.35	38.35	38.35	63.75	63.75	63.75	63.75
HTR *Nu* of cold side	51.30	81.68	53.16	64.68	51.32	81.70	53.18	64.70
LTR pinch point (K)	10.36	10.36	10.36	10.36	9.56	9.56	9.56	9.56
HTR pinch point (K)	17.14	17.14	17.14	17.14	17.15	17.15	17.15	17.15
LTR volume (m³)	2.26	2.26	2.26	2.26	1.51	1.51	1.51	1.51
HTR volume (m³)	1.98	1.27	1.48	1.35	1.99	1.27	1.49	1.35
Cycle thermal efficiency (%)	52.44	52.43	52.44	52.44	52.55	52.54	52.55	52.55
Total cost (M$)	14.36	13.68	13.88	13.75	13.65	12.97	13.18	13.04
Main component exergy loss (MW)	1.96	1.96	1.96	1.96	1.94	1.93	1.94	1.94

Table A2. Cycle performance with different recuperator types (L_s and L_{af}).

Performance	L_sH_{st}	L_sH_z	L_sH_s	L_sH_{af}	$L_{af}H_{st}$	$L_{af}H_z$	$L_{af}H_s$	$L_{af}H_{af}$
LTR pressure drop of cold side (kPa)	13.12	13.12	13.12	13.12	11.02	11.02	11.02	11.02
LTR pressure drop of hot side (kPa)	43.97	43.97	43.97	43.97	41.51	41.51	41.51	41.51
HTR pressure drop of cold side(kPa)	9.20	66.08	19.43	17.02	9.20	66.06	19.42	17.01
HTR pressure drop of hot side (kPa)	32.84	246.8	70.16	61.65	32.85	246.90	70.18	61.66
LTR overall heat transfer coefficient (W·m⁻²·K⁻¹)	678.8	678.8	678.8	678.8	805.58	805.58	805.58	805.58
HTR overall heat transfer coefficient (W·m⁻²·K⁻¹)	755.9	1194.1	778.6	954.6	755.91	1194.1	778.58	954.58
LTR *Nu* of cold side	44.70	44.70	44.70	44.70	47.73	47.73	47.73	47.73
HTR *Nu* of cold side	51.30	81.68	53.17	64.68	51.30	81.68	53.17	64.68
LTR pinch point (K)	10.24	10.24	10.24	10.24	10.26	10.26	10.26	10.26
HTR pinch point (K)	17.14	17.14	17.14	17.14	17.14	17.14	17.14	17.14
LTR volume (m³)	1.66	1.66	1.66	1.66	1.56	1.56	1.56	1.56
HTR volume (m³)	1.98	1.27	1.48	1.35	1.98	1.27	1.48	1.35
Cycle thermal efficiency (%)	52.46	52.44	52.45	52.45	52.45	52.44	52.45	52.45
Total cost (M$)	13.79	13.11	13.31	13.18	13.70	13.01	13.22	13.09
Main component exergy loss (MW)	1.96	1.95	1.95	1.95	1.96	1.95	1.96	1.96

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796

1. Printed circuit recuperator types and designs are included in recompression cycle model.

2. Thermo–economic performances are assessed with multi-objective optimization method.

3. Effects of recuperator effectiveness and recompression fraction are significant.

4. Optimal recuperator design parameters depend on the preferred objective.

5. The solution of three-objective optimization is more comprehensive than that of two-objective.