

Thermo-economic assessment and systematic comparison of combined supercritical CO₂ and organic Rankine cycle (SCO₂-ORC) systems for solar power tower plants

Gang Fan ^{a,b}, Jian Song ^{b,c}, Jiageng Zhang ^a, Zijun Fu ^a, Xiaoyu Gong ^a,

Yiping Dai ^{a,*}, Christos N. Markides ^{b,**}

^a Institute of Turbomachinery, Xi'an Jiaotong University, School of Energy and Power Engineering, Xi'an, 710048, China

^b Clean Energy Processes (CEP) Laboratory, Department of Chemical Engineering, Imperial College London, London SW7 2AZ, UK

^c Birmingham Centre for Energy Storage & School of Chemical Engineering, University of Birmingham, Birmingham B15 2TT, UK

* Corresponding authors: ypdai@mail.xjtu.edu.cn (Y. Dai),

** Corresponding authors: c.markides@imperial.ac.uk (C.N. Markides).

N.B.: This is the ACCEPTED version of this article. The final, published version of the article can be found at: <https://doi.org/10.1016/j.applthermaleng.2023.121715>

Abstract

Solar power towers (SPTs) integrated with thermal energy storage are promising solutions for solar energy utilisation. Supercritical CO₂ power cycles are acknowledged as an attractive option for dry-cooling SPT plants. However, abundant low-grade waste heat exists in the SCO₂ gas cooler, which is directly dissipated to the ambient. In order to further improve the performance of SCO₂-based SPT plants, organic Rankine cycle (ORC) systems can be introduced as a bottoming cycle subsystem. In this paper, an ORC subsystem is added to four different SCO₂ cycle layouts, i.e., recuperated (RE), recompression (RC), intercooling (IC), and partial cooling (PC) cycles, to form combined cycle systems for SPT applications. A parametric study reveals that the thermal efficiency of the power block (i.e., the whole power cycle system), and the salt temperature difference across the receiver are the determining factors of the thermo-economic performance of SPT-SCO₂-ORC plants. Design optimisation and annual performance evaluations are implemented using actual weather data in Delingha, China. Compared to the SPT plant with a standalone recuperated SCO₂ cycle system, the annual electricity generation is increased by 19% by integrating a bottoming ORC subsystem. The SPT-RC-ORC system produces the maximum electricity generation of 123 GW·h/year, while the SPT-RE-ORC system achieves the lowest levelised cost of electricity (LCOE) of 0.12 \$/kW·h and the minimum payback time of 8.8 years. A wide range of solar irradiance conditions is further considered to generalise the performance evaluation of such combined cycle systems, with results showing that with the highest solar irradiance investigated (3700 sunshine hours, 1000 W/m²·h), the LCOE of the SPT-RE-ORC plant can be as low as 0.07 \$/kW·h and the payback time is as short as 6 years. This study investigates optimal SCO₂-ORC configurations for SPT applications based on annual performance evaluations, and presents preliminary assessments of the thermo-economic potential of ushc SPT-SCO₂-ORC plants across a variety of solar resource conditions.

Keywords: organic Rankine cycle; solar power tower; supercritical CO₂ cycle; systematic comparison; thermo-economic assessment.

<i>A</i>	area, m ²
<i>B</i>	cost coefficient of heat exchangers
<i>C</i>	cost, \$
<i>D</i>	diameter, m
<i>d</i>	discount rate, %
<i>E</i>	electricity output, kW·h or MW·h
<i>F</i>	correction pressure factor
<i>h</i>	enthalpy, J/(kg·K); heat transfer coefficient, W/(m ² ·K)
<i>H</i>	height, m; hydraulic head, m
<i>i</i>	annual tax rate, %
<i>K</i>	cost coefficient of heat exchangers
<i>L</i>	length, m
<i>m</i>	mass flow rate, kg/s
<i>N</i>	number; rotation speed, rpm; plant life cycle, year
<i>Nu</i>	Nusselt number
<i>P</i>	working fluid pressure, Pa
<i>Pr</i>	Prandtl number
<i>Q</i>	heat flux, W
<i>R</i>	fouling resistance of heat exchanger, m ² ·K/W
<i>Re</i>	Reynolds number
<i>T</i>	temperature, K
<i>t</i>	operating hour, h
<i>U</i>	total heat transfer coefficient, W/(m ² ·K)
<i>v</i>	velocity, m/s
<i>W</i>	consumption power or output power, W
<i>x</i>	split mass ratio
<i>Y</i>	Stodola's constant, 1/(m ² ·s ² ·K)
ΔT	temperature difference, K
<i>Z</i>	component cost, \$

Greek symbols

ϕ	mass flow coefficient, m·s·K ^{1/2}
λ	thermal conductivity, W/(m·K)
ε	recuperator effectiveness; emissivity
σ	Stefan-Boltzmann constant
ρ	density, kg/m ³
τ	annual plant operation hours, h

Subscripts

a, b, 1, 2, ..., 13, 01, 02, ..., 06	point numbers in the schematic diagram
comp	compressor
cond	condenser
eva	evaporator
gen	generator
heli	heliostat field
in	inlet

lmt	log mean temperature difference
max	maximum value
out	outlet
pb	power block
pp	pinch point
rec	receiver
ref	reference year
s	isentropic
th	thermal
tur	turbine

Acronyms

<i>CEPCI</i>	chemical engineering plant cost index
CSP	concentrated solar power
<i>DNI</i>	direct normal irradiation
EPC	engineering and procurement cost
HTR	high-temperature recuperator
IC	intercooling SCO ₂ cycle
LTR	low temperature recuperator
MC	main SCO ₂ compressor
ORC	organic Rankine cycle
O&M	operating and maintenance cost
PC	partial-cooling cycle
PCHE	print circuit heat exchanger
<i>PEC</i>	purchase equipment cost
<i>PR</i>	pressure ratio
RC	recompression compressor and recompression SCO ₂ cycle
RE	recuperated SCO ₂ cycle
SCO ₂	supercritical carbon dioxide
SC	SCO ₂ compressor
<i>SIC</i>	specific investment cost
SPT	solar power tower
SRC	steam Rankine cycle
ST	SCO ₂ turbine
STHE	shell and tube heat exchanger
<i>TCC</i>	total capital cost
TES	thermal energy storage
TDC	total direct cost
<i>TIC</i>	total indirect cost
<i>LCOE</i>	levelised cost of electricity, \$/kW·h

1. Introduction

The world is planning to increase the use of renewables considering energy shortage and global warming issues, as well as the benefit from the continuous decrease in the costs of renewable energy technologies. According to the roadmaps to 2050 of the United States, the European Union, and China, the share of electricity generated from renewables will increase to 80% [1], 100% [2], and 80% [3], respectively. Solar energy can play a vital role in achieving these long-term strategic goals, of which photovoltaic (PV) and concentrated solar power (CSP) are two primary technologies. The former has the advantages of simplicity and relatively low capital costs [4], while CSP plants can provide dispatchable power via the integration of thermal energy storage (TES) systems [5]. Current CSP technologies are classified into four categories according to the optical types: parabolic trough, parabolic dish, linear Fresnel reflector, and solar power tower (SPT) [6], among which SPT technologies deliver the highest solar-to-electric efficiency and achieve the lowest nominal levelised cost of electricity (*LCOE*) [7]. Up to now, conventional steam Rankine cycle (SRC) systems have been adopted in SPT plants to generate electricity, which is a mature technology with a limited degree to further improve performance and reduce costs. Therefore, employing alternative power cycle systems is a key step to improving the market competitiveness for SPT plants.

Supercritical CO₂ (SCO₂) provides higher temperature operability than steam and allows compact component and system structure thanks to the high density, and hence is regarded as a promising working fluid for power cycle systems. In the 1960s, Feher EG [8] and Angelino [9] initially proposed power cycle systems using SCO₂. However, this technology attracted little attention over a long period due to the low technology readiness level. In 2004, Dostal proposed recompression SCO₂ Brayton cycle systems, which could avoid the pinch-point problem in recuperative heat exchangers, establishing the thermodynamic fundamentals of SCO₂ cycle systems [10], which put SCO₂ power cycle systems in the spotlight again. The advantages of SCO₂ cycle systems for SPT applications can be concluded as follows: (1) CO₂ shows “liquid-like” behaviour near the critical point (7.38 MPa, 31 °C), which reduces the compression work drastically and leads to higher cycle efficiency [11]; (2) the high density of SCO₂ working fluid offers advantages of much smaller equipment sizes than traditional steam Rankine cycle systems [12]; (3) the material costs of SCO₂ cycle systems can be reduced due to the lower corrosiveness [13]; (4) SCO₂ cycle systems can achieve high efficiency with dry-cooling and therefore can accommodate arid climate, which is vital for CSP systems located in droughty areas [14]. Consequently, SCO₂ cycle systems are identified as an attractive option in next-generation SPT plants, and further investigation is necessary to comprehensively understand this technology as well as the interaction with solar fields and thermal energy storage units.

Various layouts of SCO₂ Brayton cycle systems have been investigated in recent years, such as

76 pre-compression, partial-cooling, preheating, intercooling and split expansion SCO_2 cycle systems [15].
77 Crespi et al. [16] reviewed 42 standalone SCO_2 system layouts and 38 combined system configurations
78 for power generation, using thermo-economic performance as the indicator to select the most interesting
79 cycle layouts for SCO_2 power cycle systems coupled with TES in CSP applications [17]. Intercooling
80 and partial-cooling SCO_2 cycle systems have been proved to achieve cycle efficiency of over 50% for
81 dry-cooling CSP plants [18]. Compared with recompression SCO_2 cycle systems, partial-cooling SCO_2
82 cycle systems provided lower costs for recuperators and TES systems, which resulted in a lower
83 *LCOE* [19]. Several authors tried to add certain dopants to the raw carbon dioxide in the above CO_2
84 cycle systems, also known as blended transcritical CO_2 power cycle systems, to avoid efficiency
85 degradation at high ambient temperatures [20]. Amongst these, Hexafluorobenzene (C_6F_6) and
86 Titanium Tetrachloride (TiCl_4) were numerically confirmed as suitable additives in the SCARABEUS
87 project [21], which enabled thermal efficiency to exceed 50% at high environment temperatures.
88 Although these SCO_2 cycle systems demonstrate favourable performance in SPT applications,
89 significant heat loss still exists in the gas cooler [22].

90 Adding a bottoming power cycle subsystem to the SCO_2 cycle system to effectively utilise the
91 low-grade waste heat can further improve thermal efficiency. Organic Rankine cycle (ORC) is
92 considered a suitable choice due to its excellent performance for residual heat power generation [23].
93 Many researchers evaluated the thermo-economic performance of the combined SCO_2 -ORC power
94 systems [24]. Specifically, ORC systems were added as a bottoming cycle subsystem for recuperated
95 SCO_2 cycle systems, recompression SCO_2 cycle systems [22], partial-cooling SCO_2 cycle systems
96 [25], and intercooling SCO_2 cycle systems [26]. The results showed that the combined cycle system
97 provided a thermal efficiency increase of 2-13% compared to standalone SCO_2 cycle systems. Adding
98 ORC subsystems also enables flexible utilisations of multiple renewables. Song et al. performed
99 thermo-economic assessments of various configurations of combined SCO_2 cycle and ORC systems
100 for hybrid solar and geothermal power generation, and the annual performance evaluation showed
101 that the combined systems could achieve significant improvements, ranging from 22% to 45%, over
102 the total electricity generation of the standalone SCO_2 plant [27]. However, previous studies only
103 focus on the thermodynamic and economic performance of SPT- SCO_2 -ORC plants at the design stage.
104 This places a significant limitation on their results because such systems usually need to operate at
105 off-design conditions. Annual direct normal irradiation (*DNI*) differs radically in different regions,
106 and weather condition fluctuates during the daytime and determines the operating features of the
107 power block. The heat input to the power block, heat storage capacity and power output are
108 significantly influenced by *DNI* and air temperature variations.

109 Moreover, previous research has mainly focused on the power block in CSP plants, while the

interaction among the SPT, TES, and the power block is of interest and provides insights into such systems. It is noted that the plant efficiency is simultaneously affected by the optical field efficiency, receiver efficiency, and power block efficiency. In other words, each energy conversion process determines the overall thermodynamic performance of SPT systems. With respect to the economic performance, the capital cost of the heliostat field, which occupies 40-60% of the total investment cost of an SPT plant [28], is significantly affected by the thermal storage capacity and power block efficiency. Therefore, the object of this paper is to present an annual simulation for different SPT-SCO₂-ORC systems using thermal storage. The main novelties of this paper include: (1) an integrated model capturing the heliostat field, solar receiver, TES, and power block is established to evaluate the overall thermo-economic performance from a holistic perspective; (2) an ORC subsystem is added to four different SCO₂ cycle layouts, i.e., recuperated (RE), recompression (RC), intercooling (IC), and partial cooling (PC) configurations, to explore for the optimal combined cycle systems for SPT applications; and (3) real weather conditions and realistic off-design performance are considered to obtain accurate annual electricity generation.

The work is organised as follows: four combined SCO₂-ORC systems for SPT plants are proposed, along with comprehensive thermo-economic models, followed by a parametric study to investigate the interactive influence between the molten salt loop and power blocks. Afterwards, a case study, including design optimisation and annual performance evaluation, is performed based on the local weather data in Delingha, China. The performance of different SPT-SCO₂ plants with and without ORC systems are compared, and the optimal power system configuration is obtained. Furthermore, a wide range of solar irradiance conditions is considered to assess the potential of exploiting the proposed SPT-SCO₂-ORC plants in different global regions.

131

132 2. System configurations

The concentrated solar power plants considered in this study contain three major parts: the SPT, the TES, and the power block. The SPT subsystem consists of a heliostat field, a solar tower, and a molten salt receiver, as shown in Fig. 1. The TES subsystem is constituted of two molten salt thermal storage tanks (hot and cold) and ancillary facilities. The standalone SCO₂ cycle systems or combined SCO₂-ORC systems work as the power block.

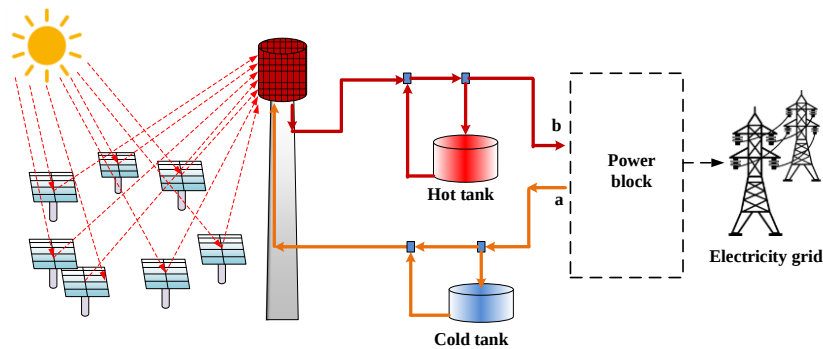
Four SCO₂ system layouts are investigated in this paper: the recuperated SCO₂ cycle system (RE), the recompression SCO₂ cycle system (RC), the intercooling SCO₂ cycle system (IC), and the partial cooling SCO₂ cycle system (PC). As shown in Fig. 2, the RE system configuration is derived from the simple Brayton cycle system by adding a recuperator to avoid discharging a large amount of energy to the ambient. However, a pinch-point problem caused by the significant difference in the heat capacity between the high-pressure and low-pressure SCO₂ streams exists in the recuperator [10]. The RC system

144 layout was proposed to mitigate this effect by dividing the recuperator into a high-temperature
 145 recuperator (HTR) and a low-temperature recuperator (LTR) [29]. Moreover, an additional compressor,
 146 i.e., the re-compressor, is included in the RC system, as shown in Fig. 3. Unlike the RE system
 147 configuration, a portion of the low-pressure working fluid CO₂ bypasses the SCO₂ cooler and is
 148 compressed by the re-compressor. This stream also bypasses the LTR after the compression process.
 149 This configuration provides a better match of heat capacity between the hot and cold streams of
 150 recuperators and effectively reduces the impacts of the pinch-point problem on cycle efficiency.

151 Intercooling is a classical approach to decreasing the compression work of multi-stage
 152 compressors. It has been introduced into the recompression SCO₂ cycle system to further improve
 153 system efficiency. As illustrated in Figs. 4 and 5, there are two technical routes to integrate the
 154 intercooling process: recompression with main compressor intercooling [30] and recompression with
 155 pre-cooling [10]; these two cycle systems are also called “IC” and “PC” systems, respectively. The
 156 PC system involves a cooler and pre-compressor before the flow splits to the RC system, while the
 157 IC system includes an intercooler in the main compression line.

158 In order to achieve better performance for the power block, a bottoming ORC system is added to
 159 recover the heat rejected in the SCO₂ cooler. The SCO₂ and ORC subsystems are coupled via a shared
 160 heat exchanger. The heat recovery process occurs in the evaporator, and the condensed organic fluid
 161 turns into a vapour after absorbing the heat from the CO₂ stream. In this work, the molten salt SPT and
 162 TES system are integrated with the four specific combined SCO₂-ORC systems mentioned above, and
 163 the systems are named “STP-RE-ORC”, “STP-RC-ORC”, “STP-IC-ORC” and “STP-PC-ORC”, with
 164 schematic diagrams and corresponding *T-s* diagrams shown in Figs. 2, 3, 4, and 5, respectively.

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166

167

Fig. 1. Schematic diagram of an SPT plant.

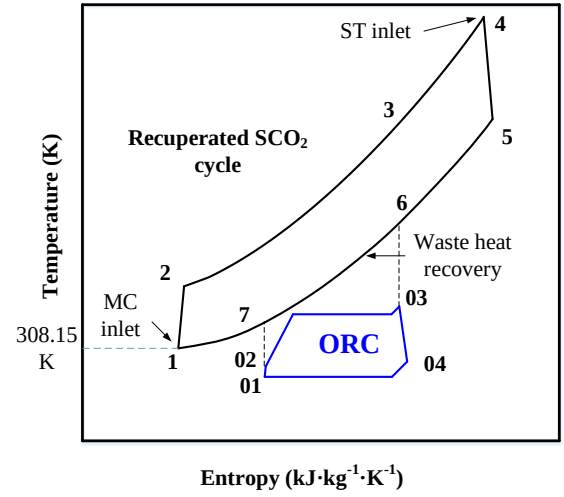
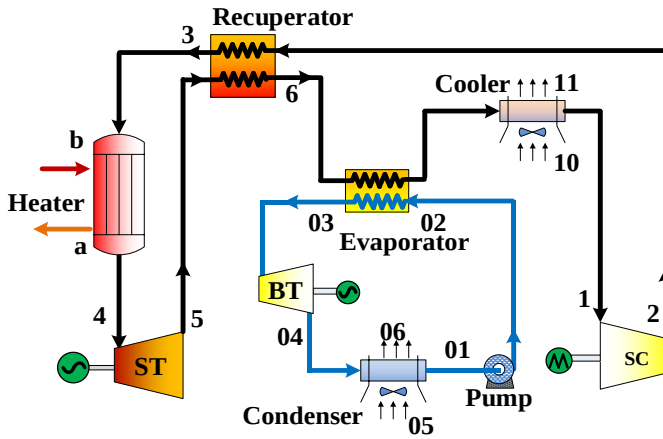


Fig. 2. (a) Schematic diagram of a RE-ORC power cycle system, and (b) corresponding T - s diagram.

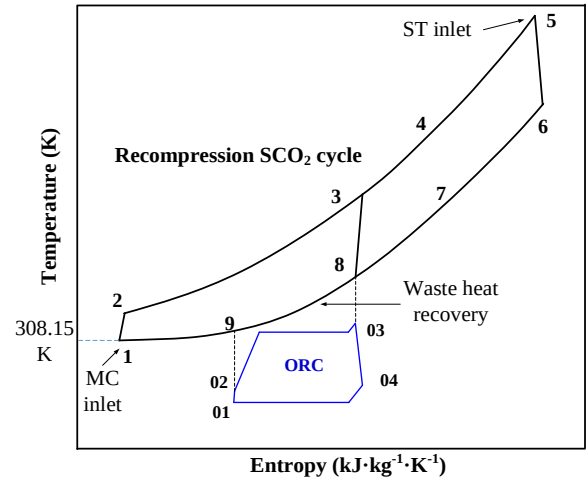
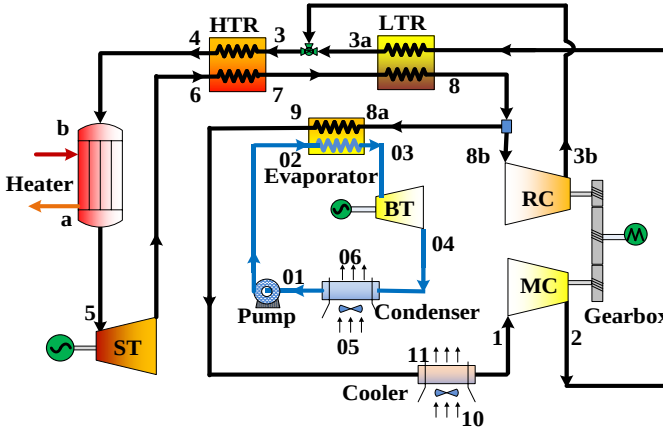


Fig. 3. (a) Schematic diagram of a RC-ORC power cycle system, and (b) corresponding T - s diagram.

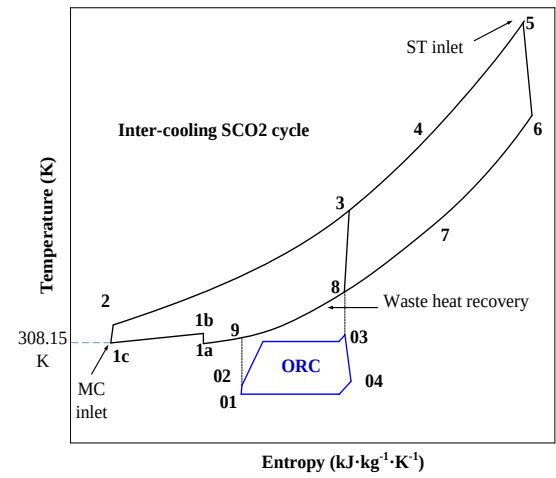
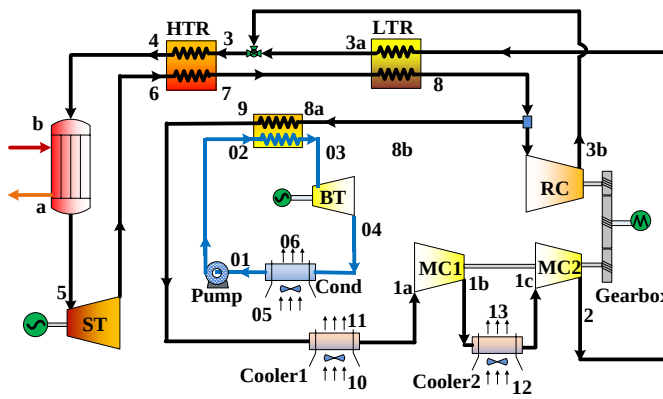


Fig. 4. (a) Schematic diagram of an IC-ORC power cycle system, and (b) corresponding T - s diagram.

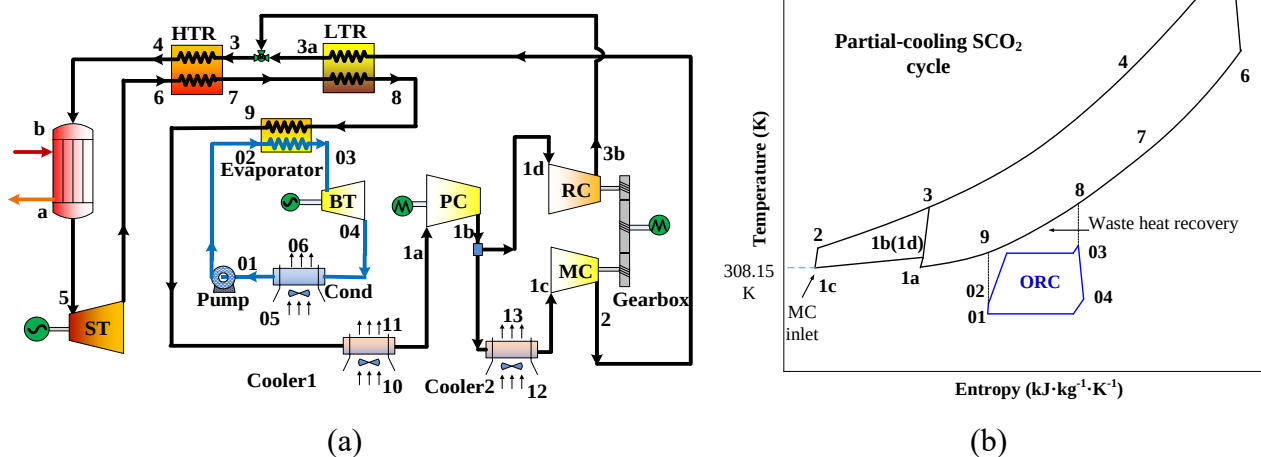


Fig. 5. (a) Schematic diagram of a PC-ORC power cycle system, and (b) corresponding T - s diagram.

3. Modelling methodology

Based on the principal assumption that the whole system always works at steady states, in-house MATLAB models have been developed to analyse systems' performance, with working fluid properties obtained from REFPROP 9.1.

3.1 Solar field and molten salt loop

The total solar energy reflected by the heliostat field is given by:

$$Q_{\text{heli.out}} = DNI \cdot A_{\text{heli}} \cdot N_{\text{heli}} \cdot \eta_{\text{heli}} \quad (1)$$

where $Q_{\text{heli,out}}$ is the output solar energy of the heliostat field, DNI is the direct normal irradiance, A_{heli} represents the area of each heliostat, N_{heli} is the total number of heliostats, and η_{heli} is the optical efficiency of the heliostat field.

The total energy collected by the receiver Q_{rec} equals $Q_{\text{heli,out}}$. The energy conservation equation of the solar receiver can be expressed as:

$$Q_{\text{rec.in}} = Q_{\text{rec.conv}} + Q_{\text{rec.rad}} + Q_{\text{rec.cond}} + Q_{\text{rec.ref}} + Q_{\text{rec.out}} \quad (2)$$

where $Q_{\text{rec,conv}}$, $Q_{\text{rec,rad}}$, $Q_{\text{rec,cond}}$ and $Q_{\text{rec,ref}}$ are the energy losses from convection, radiation, conduction, and reflection. The conductive loss $Q_{\text{rec,cond}}$ is neglected by assuming that the solar receiver has good thermal insulation. The solar receiver's output energy $Q_{\text{rec,out}}$ equals to the total heat transferred to the molten salt, the thermal energy transferred by convection in the receiver tubes, and the heat transferred by thermal conduction through the tube wall, respectively. It can be calculated from:

$$Q_{\text{rec,out}} = m_{\text{salt}} \cdot (h_{\text{salt,hot}} - h_{\text{salt,cold}}) \quad (3)$$

$$Q_{\text{rec,out}} = k_{\text{salt}} \cdot A_{\text{wall,in}} \cdot (T_{\text{wall,in}} - T_{\text{salt,ave}}) \quad (4)$$

$$Q_{\text{rec,out}} = \frac{2\pi \cdot \lambda_{\text{tube}} \cdot A_{\text{wall,in}} \cdot (T_{\text{wall,out}} - T_{\text{wall,in}})}{D_i \cdot \ln(D_o/D_i)} \quad (5)$$

where m_{salt} is the mass flow rate of molten salt in the receiver, $h_{\text{salt,hot}}$ and $h_{\text{salt,cold}}$ represent the enthalpies of the molten salt at the outlet state and inlet state, k_{salt} is the convective heat transfer coefficient of the molten salt, $A_{\text{wall,in}}$ is the sum of the internal surface area of the tube wall, $T_{\text{wall,in}}$ and $T_{\text{wall,out}}$ is the temperatures at the internal and external surface of tube walls, $T_{\text{salt,ave}}$ represents the average temperature of molten salt, λ_{tube} is the heat conductivity coefficient of the tube, and D_o and D_i are the external and internal diameters of the receiver tube.

The shape of the solar receiver is a cylinder. The overall convection loss $Q_{\text{rec,conv}}$ is expressed as:

$$Q_{\text{rec,conv}} = \pi \cdot k \cdot H_{\text{rec}} \cdot D_{\text{rec}} \cdot (T_{\text{wall,out}} - T_{\infty}) \quad (6)$$

where H_{rec} and D_{rec} are the height and diameter of the receiver, T_{∞} is the ambient temperature, and k is the convective heat transfer coefficient, which can be calculated from [31]:

$$k = [k_{\text{fc}}^{3.2} + k_{\text{nc}}^{3.2}]^{\frac{1}{3.2}} = [(\lambda \cdot Nu_D / D_{\text{rec}})^{3.2} + (\lambda \cdot Nu_H / H_{\text{rec}})^{3.2}]^{\frac{1}{3.2}} \quad (7)$$

where k_{fc} and k_{nc} are the forced and natural convective heat transfer coefficients. Nu_D and Nu_H are Nusselt numbers associated with different characteristic lengths, which can be calculated from [31]:

$$Nu_D = \begin{cases} 0.3 + 0.488 \cdot Re_D^{0.5} \cdot [1 + (Re_D/282000)^{0.625}]^{0.8} & \text{for } Re_D \leq 3.7 \times 10^5 \\ 0.53 \cdot (0.00257 \cdot Re_D^{0.98}) + 0.47 \cdot (0.0135 \cdot Re_D^{0.89}) & \text{for } 3.7 \times 10^5 < Re_D < 10^7 \\ 0.0455 \cdot Re_D^{0.81} & \text{for } Re_D \geq 10^7 \end{cases} \quad (8)$$

$$Nu_H = 0.049 \cdot \pi \cdot Gr^{0.33} \cdot \left(\frac{T_{\text{wall}}}{T_{\infty}}\right)^{-0.14} \quad (9)$$

The reflection loss of the solar receiver is calculated from [32]:

$$Q_{\text{rec,ref}} = (1 - \alpha_{\text{rec}}) \cdot Q_{\text{rec,in}} \quad (10)$$

where α_{rec} is the absorptivity of the receiver.

The radiation loss of the solar receiver is calculated from [32]:

$$Q_{\text{rec,rad}} = \varepsilon_{\text{rec}} \cdot \sigma \cdot A_{\text{rec}} \cdot (T_{\text{wall,out}}^4 - T_{\text{air}}^4) \quad (11)$$

where σ is the Stefan-Boltzmann constant, ε_{rec} is the receiver emissivity, and A_{rec} is the total surface

224 area of the solar receiver.

225 The receiver efficiency is expressed as:

226
$$\eta_{\text{rec}} = \frac{Q_{\text{rec,out}}}{Q_{\text{rec,in}}} \quad (12)$$

227 The hot molten salt splits into two streams after absorbing heat in the solar receiver: one part drives
228 the power system to generate electricity ($Q_{\text{ps,in}}$), and the redundant hot salt is stored in the tank (Q_{storage}).
229 Thermal storage efficiency, which is the ratio of the amount of energy sent to the thermal storage unit
230 and the amount of energy discharged to the steam generator for downstream use, is normally used to
231 evaluate the thermal insulation effects. In practical applications, thermal storage efficiency in
232 commercial plants can achieve 99.5% thanks to good thermal insulation [33], for example, the top and
233 side surfaces of tanks can be insulated with mineral wool blankets overlaid with fibreglass boards, and
234 foam glass insulation can be considered for the bottom. More technical details of insulated thermal
235 storage systems can be found in Ref. [33]. In the present work, good thermal insulation is considered
236 and the heat loss of the storage tanks is neglected, while the electricity generation can be easily re-
237 calculated if a certain thermal storage efficiency is considered, as the two values are linearly related.
238 Therefore, the energy balance equation of the molten salt thermal storage subsystem is expressed as:

239
$$Q_{\text{storage}} = Q_{\text{rec,out}} - Q_{\text{ps,in}} \quad (13)$$

240

241 3.2 Power block

242 The steady-state thermodynamic models for the main components of the power block are developed
243 based on the first law of thermodynamics, and the energy balance equations. Table 1 lists all
244 thermodynamic balance equations for the combined RE-ORC system, in which the recuperated SCO_2
245 cycle (RE) is selected as the topping system. Detailed thermodynamic models of other combined power
246 cycle systems are described in the Appendix.

247 A simple ORC system works as the bottoming system in this work, with R245fa as the working
248 fluid, which has been considered in ORC systems in extensive studies thanks to the good thermal
249 performance [34], non-flammability and low leakage risks. The maximum operating pressure of R245fa
250 is limited to 3470 kPa and 160 °C to ensure operational safety [27].

251

Table 1. Thermodynamic models of the combined RE-ORC system.

Component	Thermodynamic balance equation
SCO ₂ turbine	$W_{\text{tur,SCO}_2} = m_4 \cdot (h_4 - h_5), \eta_{\text{tur,SCO}_2} = (h_4 - h_5) / (h_4 - h_{5s})$
ORC turbine	$W_{\text{tur,ORC}} = m_{04} \cdot (h_{03} - h_{04}), \eta_{\text{tur,ORC}} = (h_{03} - h_{04}) / (h_{03} - h_{04s})$
SCO ₂ compressor	$\eta_{\text{SC}} = (h_{2s} - h_1) / (h_2 - h_1), W_{\text{SC}} = m_1 \cdot (h_2 - h_1)$
ORC pump	$\eta_{\text{pump}} = (h_{02s} - h_{01}) / (h_{02} - h_{01}), W_{\text{pump}} = m_{01} \cdot (h_{02} - h_{01})$
Heater	$m_{\text{salt,b}} \cdot (T_{\text{salt,hot}} - T_{\text{salt,cold}}) = m_4 \cdot (h_4 - h_3)$
Recuperator	$h_5 - h_6 = h_3 - h_4$
Evaporator	$m_6 \cdot (h_6 - h_7) = m_{02} \cdot (h_{03} - h_{02})$
SCO ₂ -air cooler	$m_1 \cdot (h_7 - h_1) = m_{10} \cdot (h_{11} - h_{10})$ $W_{\text{fan,cooler}} = 0.5 \cdot m_{10} \cdot \Delta P_{\text{air}} \cdot (\rho_{10} - \rho_{11}) / \eta_{\text{fan}}$
Condenser	$m_{01} \cdot (h_{04} - h_{01}) = m_{05} \cdot (h_{06} - h_{05})$ $W_{\text{fan,cond}} = 0.5 \cdot m_{05} \cdot \Delta P_{\text{air}} \cdot (\rho_{05} - \rho_{06}) / \eta_{\text{fan}}$

254 3.3 Thermodynamic performance criteria

255 The thermal efficiency is used to evaluate the thermodynamic performance of the power block and is
256 defined as follows:

$$257 \quad \eta_{\text{th}} = \frac{W_{\text{net,total}}}{Q_{\text{heater}}} = \frac{W_{\text{net,total}}}{m_{\text{salt}} \cdot (T_{\text{salt,hot}} - T_{\text{salt,cold}})} \quad (14)$$

258 where Q_{heater} is the total energy input from the molten salt-SCO₂ heat exchanger, and $W_{\text{net,total}}$ is the
259 amount of electric power produced by the combined SCO₂-ORC system, which is calculated from:

$$260 \quad W_{\text{net,total}} = W_{\text{ST}} + W_{\text{BT}} - W_{\text{MC}} - W_{\text{RC}} - W_{\text{fan,cooler}} - W_{\text{fan,cond}} \quad (15)$$

261 The overall efficiency of the whole STP plant at the design condition is calculated from:

$$262 \quad \eta_{\text{plant,design}} = \eta_{\text{heli}} \cdot \eta_{\text{rec}} \cdot \eta_{\text{th}} = \frac{W_{\text{net,total}} + Q_{\text{storage}} \cdot \eta_{\text{th}}}{DNI \cdot A_{\text{heli}} \cdot N_{\text{heli}} \cdot \eta_{\text{heli}}} \quad (16)$$

263 where $\eta_{\text{heli}}, \eta_{\text{rec}}, \eta_{\text{th}}$ are the efficiencies of the heliostat field, solar receiver and the power block.

265 3.4 Economic performance criteria

266 The costs of the solar field, TES and main components of the power block can be calculated using
267 formulas listed in Table 2.

Table 2. Cost formulas used for the main components of the SPT-SCO₂-ORC plant.

Component	Cost formulas	CEPCI (Reference year) [30]
Heliostat field	$Z_{\text{heli}} = 156 \cdot A_{\text{heli}} \cdot N_{\text{heli}}$ [36]	776.9 (2021)
Power tower	$Z_{\text{tower}} = 3 \cdot 10^6 \cdot e^{0.0113 \cdot h_{\text{tower}}}$ [36] $h_{\text{tower}} = 72.31 \cdot e^{A_{\text{heli}} \cdot N_{\text{heli}} \cdot 8 \times 10^{-7}}$ [36]	776.9 (2021)
Solar receiver	$Z_{\text{receiver}} = 1.03 \cdot 10^8 \cdot (A_{\text{receiver}} / 1571)^{0.7}$ [36]	776.9 (2021)
TES	$Z_{\text{TES}} = 22 \cdot \left[284 / (T_{\text{salt,hot}} - T_{\text{salt,cold}}) \right] \cdot Q_{\text{storage}} \cdot t_{\text{storage}}$ [37]	776.9 (2021)
Compressor	$Z_{\text{comp}} = 71.1 \cdot m_{\text{comp}} / (0.93 - \eta_{\text{comp}}) \cdot PR_{\text{comp}} \cdot \ln(PR_{\text{comp}})$ [38]	381.7 (1996)
Turbine	$Z_{\text{tur}} = 479.3 \cdot m_{\text{tur}} / (0.93 - \eta_{\text{tur}}) \cdot \ln(PR_{\text{tur}}) \cdot (1 + e^{(0.036 \cdot T_{\text{tur,in}} - 54.4)})$ [38]	381.7 (1996)
Electrical generator	$Z_{\text{gen}} = 60 \cdot W_{\text{tur}}^{0.95}$ [39]	567.3 (2013)
Cooling fan	$Z_{\text{fan}} = 12400 \cdot (W_{\text{fan}} / 50)^{0.76}$ [40]	468.2 (2005)

270

271 The module costing technique is adopted to estimate the costs of other components in the power
 272 block. The equations for the purchase costs of these components are given by:

$$273 \quad Z = Z^0 \cdot F_{\text{bm}} \quad (17)$$

$$274 \quad \lg Z^0 = K_1 + K_2 \cdot \lg X + K_3 \cdot (\lg X)^2 \quad (18)$$

$$275 \quad \lg F_p = F_1 + F_2 \cdot \lg P + F_3 \cdot (\lg P)^2 \quad (19)$$

$$276 \quad F_{\text{bm}} = B_1 + B_2 \cdot F_m \cdot F_p \quad (20)$$

277 where Z^0 represents the cost at the base condition, F_{bm} stands for the correction factor, and F_m and F_p
 278 are material and operation pressure correction factors, respectively. The other coefficients K_1 - K_3 , F_1 -
 279 F_3 , B_1 , and B_2 are listed in Table 3, and the reference year is 2001.

280 The Chemical Engineering Plant Cost Index (CEPCI) updates the component cost. The
 281 relationship between the present capital cost and the reference cost is expressed as follows:

$$282 \quad Z_{\text{equip}} = Z_{\text{equip,ref}} \cdot \frac{CEPCI_{2021}}{CEPCI_{\text{ref}}} \quad (21)$$

283

Table 3. Coefficients for purchase cost estimation of heat exchangers and pumps [41].

Coefficient	Cooler and condenser	Heater	Evaporator	Pump
K_1	4.0336	4.6656	4.3247	3.3892
K_2	0.2341	-0.1557	-0.303	0.0536
K_3	0.0497	0.1547	0.1634	0.1538
F_1	-0.1250	-0.1250	-0.0016	-0.3935
F_2	0.1536	0.1536	-0.00063	0.3957
F_3	-0.02861	-0.02861	0.0123	-0.00226
B_1	0.96	0.96	1.63	1.89
B_2	1.21	1.21	1.66	1.35
F_m	1	3.5	1	1.6

The total capital cost or total installed cost (TCC) of the SPT plant can be calculated by functions listed in Table 4.

Table 4. Total capital cost (TCC) functions.

Cost	Subitem	Equations
Purchase equipment cost (PEC)	Solar field	$PEC_{\text{field}} = Z_{\text{heli}} + Z_{\text{tower}} + Z_{\text{receiver}}$
	TES	$PEC_{\text{TES}} = Z_{\text{TES}}$
	SCO ₂ system	$PEC_{\text{SCO}_2} = Z_{\text{heater}} + Z_{\text{ST}} + Z_{\text{SC}} + Z_{\text{HTR}} + Z_{\text{LTR}} + Z_{\text{cooler}}$
	ORC system	$PEC_{\text{ORC}} = Z_{\text{EVA}} + Z_{\text{BT}} + Z_{\text{pump}} + Z_{\text{condenser}}$
	Subtotal	$PEC_{\text{total}} = PEC_{\text{field}} + PEC_{\text{TES}} + PEC_{\text{SCO}_2} + PEC_{\text{ORC}}$
Contingency cost	/	$Cost_{\text{contin}} = 0.07 \cdot PEC_{\text{total}}$ [36]
Total direct cost (TDC)	/	$TDC = PEC_{\text{total}} + Cost_{\text{contin}}$
Total indirect costs (TIC)	Land cost	$Cost_{\text{land}} = 2.471 \cdot A_{\text{land}} = 2.471 \cdot A_{\text{heli}} \cdot N_{\text{heli}} / \varphi$ [36]
	EPC and owner cost	$Cost_{\text{E\&o}} = 0.13 \cdot TDC$ [36]
	Sales tax	$Cost_{\text{tax}} = 0.8 \cdot TDC \cdot r_{\text{tax}}$ [36]
	Subtotal	$TIC = Cost_{\text{land}} + Cost_{\text{E\&o}} + Cost_{\text{tax}}$
Total capital cost (TCC)	/	$TCC = TDC + TIC$

The total specific investment cost (SIC_{total}) of a SPT-SCO₂-ORC plant is calculated from:

$$SIC_{\text{total}} = \frac{PEC}{W_{\text{net,total}}} \quad (22)$$

The annual operating and maintenance (O&M) cost is given by:

$$Cost_{O\&M} = c_{O\&M} \cdot W_{net,total} \quad (23)$$

where $c_{O\&M}$ is the annual operating and maintenance cost per unit of power and $W_{net,total}$ is the total output power.

The levelised cost of electricity (*LCOE*) is chosen as the economic performance indicator to compare different configurations of the SPT-SCO₂-ORC plant from an economic point of view, and it is calculated from [8]:

$$LCOE = [TCC + \sum_{n=1}^N \frac{Cost_{O\&M} \cdot (1 + i_f)^{n-1}}{(1 + i_d)^n}] / \sum_{n=1}^N \frac{AEG}{(1 + i_d)^n} \quad (24)$$

$$AEG = 3600 \cdot W_{net,total} \cdot \tau \quad (25)$$

where *AEG* is the annual electricity generation, τ is the annual full-load operating hour, and the plant life cycle *N* is set to 25 years [42]; the inflation rate i_f is assumed to be 2.5% [43], and the discount rate i_d is taken as 5% [44].

The payback time (*PBT*) of the power plant is given by [44]:

$$PBT = \ln[\frac{TCC \cdot (i_f - i_d)}{C_{el}} + 1] / \ln[\frac{(1 + i_f)}{(1 + i_d)}] \quad (26)$$

where the electricity purchase price C_{el} is set to 0.19 \$/kW·h [45].

3.5 Model validation

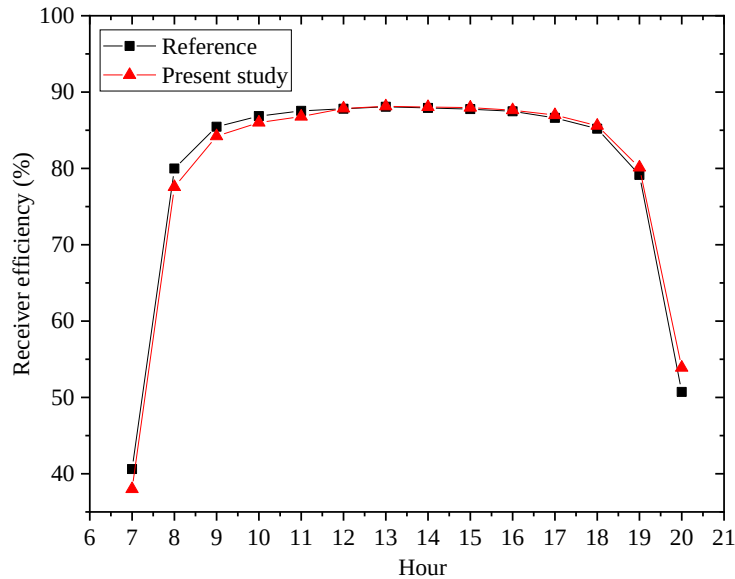
In order to validate the solar receiver model, simulation results of receiver efficiencies are compared to those from Ref. [32]. The input parameters are listed in Table 5, and the comparison results are shown in Fig. 7. It can be seen that the deviations between this work and the reference are less than 3% in the hourly simulation. The good agreements with the reference results indicate that the developed model is reliable in predicting the receiver performance. The SAM software [36], which specialises in SPT modelling, is used to obtain the heliostat field layout and efficiencies. Validation of the combined SCO₂-ORC model can be found in the authors' previous work [22]. A recompression SCO₂ cycle system with a simple ORC system, using five organic working fluids (Isobutane, R123, R114, R141b, RC318), is investigated under different pressures and evaporating temperatures. The power output and thermal efficiency calculated from our self-built models were compared with the performance indicators from Ref. [38]. The deviations were within 0.1% under the same operating parameters and organic working fluids, which indicates our model is accurate and sufficient for the simulation in this work.

322

Table 5. Input parameters for model validation of solar receiver [32].

Parameters	Values
Number of effective heliostats, N_{heli}	4946
Heliostat area, A_{heli}	144 m ²
Receiver diameter, D_{rec}	15 m
Receiver height, H_{rec}	19.5 m
Number of panels, N_{panel}	28
Number of tubes per panel, N_{tube}	40
Number of the flow path, N_{path}	2
Tube external diameter, D_o	42 mm
Tube internal diameter, D_i	40.4 mm
Receiver absorptivity, α_{rec}	0.94
Receiver emissivity, ε_{rec}	0.85

323



324

325 **Fig. 7.** Comparison of solar receiver efficiency predicted by the developed model with data reported
326 in Ref. [32].

327

328 **4. Parametric analysis**

329 Based on the above mathematic models, parameter analysis is performed for the solar field, TES and
330 power block. In this section, we investigate the effects of the key parameters, i.e., salt temperatures,
331 SCO_2 operating pressures and ORC evaporating pressure, on the systems' thermodynamic and
332 economic performance. The rated output power of the integrated SPT plants is 20 MWe, which
333 corresponds to a typical medium-scale solar power plant. A DNI of 850 W/m^2 and an environment
334 temperature of 25°C are considered design conditions. The solar multiple, which is defined as the

ratio of the maximal receiver output to the rated output power, is set to 2.8, so as to allow 24 hours of operation in summer. In the ORC subsystem, R245fa is selected as the working fluid. Tables 6 and 7 list the input parameters of the solar field, receiver and power block.

Table 6. Main input parameters of the solar field and receiver.

Parameters	Values
Annual heliostat field efficiency, η_{heli}	0.6 [46]
Receiver absorptivity, α_{rec}	0.94 [32]
Receiver emissivity, ε_{rec}	0.85 [32]
The ratio of receiver height to diameter, $H_{\text{rec}}/D_{\text{rec}}$	1.3 [47]
Number of panels, N_{panel}	28 [32]
Number of tubes per panel, N_{tube}	40 [32]
Number of the flow path, N_{path}	2 [32]

Table 7. Main input parameters of the power block.

Parameters	Values
*Main compressor inlet temperature, T_1 (°C)	35 [48]
Compressor efficiency, η_{MC} , η_{RC} and η_{PC} (%)	85 [49]
SCO ₂ turbine efficiency, η_{ST} (%)	90 [49]
Bottoming cycle turbine efficiency, η_{BT} (%)	85 [50]
Generator efficiency, η_{gen} (%)	98 [51]
Pump efficiency, η_{pump} (%)	80 [48]
Cooling fan efficiency, η_{fan} (%)	60 [52]
Effectiveness of HTR, ε_{HTR}	0.9 [42]
Effectiveness of LTR, ε_{LTR}	0.9 [42]
PPTD of the air-cooled condenser, $\Delta T_{\text{pp,cond}}$ (°C)	10 [53]
PPTD of the evaporator, $\Delta T_{\text{pp,eva}}$ (°C)	5 [53]
Temperature difference at both heater ends, $\Delta T_{\text{pp,heater}}$ (°C)	15 [54]
Temperature difference at the cold end of SCO ₂ -air cooler, $\Delta T_{\text{pp,cooler}}$ (°C)	15 [55]
Overall heat transfer coefficient of the heater, U_{heater} (W/(m ² ·K))	330 [52]
Overall heat transfer coefficient of HTR, U_{HTR} (W/(m ² ·K))	3000 [50]
Overall heat transfer coefficient of LTR, U_{LTR} (W/(m ² ·K))	1600 [50]
Overall heat transfer coefficient of evaporator, U_{eva} (W/(m ² ·K))	1600 [42]
Overall heat transfer coefficient of air-cooled condenser, U_{cond} (W/(m ² ·K))	300 [52]
Overall heat transfer coefficient of sCO ₂ -air cooler, U_{cooler} (W/(m ² ·K))	77 [56]

* Note: For the intercooling cycle and the partial-cooling cycle, $T_{1a} = T_{1c} = 35$ °C.

344 A chloride ternary salt (mixture of 24.5 wt% NaCl, 20.5 wt% KCl and 55.0 wt% MgCl₂) is
 345 selected as the heat transfer fluid in the molten salt loop considering the wide operating temperature
 346 range, low cost, and high volumetric heat capacity [57]. The correlations of the thermal properties of
 347 the salt are summarised in Table 8.

348

349 **Table 8.** Thermal properties of the solar salt [52].

Property	Correlation
Density, ρ (kg/m ³)	$1899.3 - 0.43 \cdot (T - 273.15)$
Specific heat, c_p (J/(kg·K))	1180
Thermal conductivity, λ (W/(m·K))	$0.5423 - 0.0002 \cdot (T - 273.15)$
Absolute viscosity, μ (Pa·s)	$8.25 \cdot 10^{-6} \cdot e^{\frac{11870}{(1078 + T)}}$

350

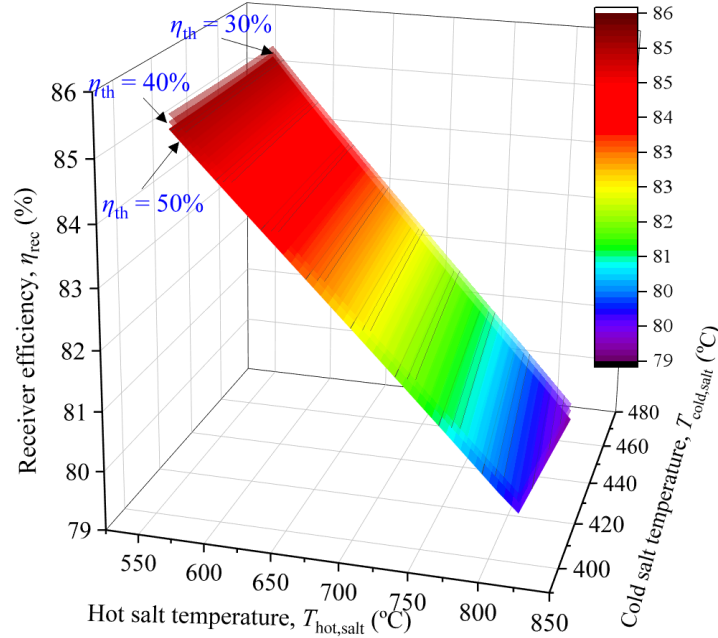
351 4.1 Solar field and TES system

352 The parametric analysis is performed in this section to investigate the effects of $T_{\text{hot,salt}}$, $T_{\text{cold,salt}}$, and
 353 η_{th} on the thermodynamic and economic performance of the solar field and TES subsystem. The
 354 power block thermal efficiency (η_{th}) is set as 30%, 40% and 50% to represent power cycle systems
 355 with different thermodynamic performance. The variation range of the $T_{\text{hot,salt}}$ and $T_{\text{cold,salt}}$ are 565-
 356 800 °C and 400-465 °C, respectively. The economic performance indicator considered here is the
 357 purchase equipment cost (PEC).

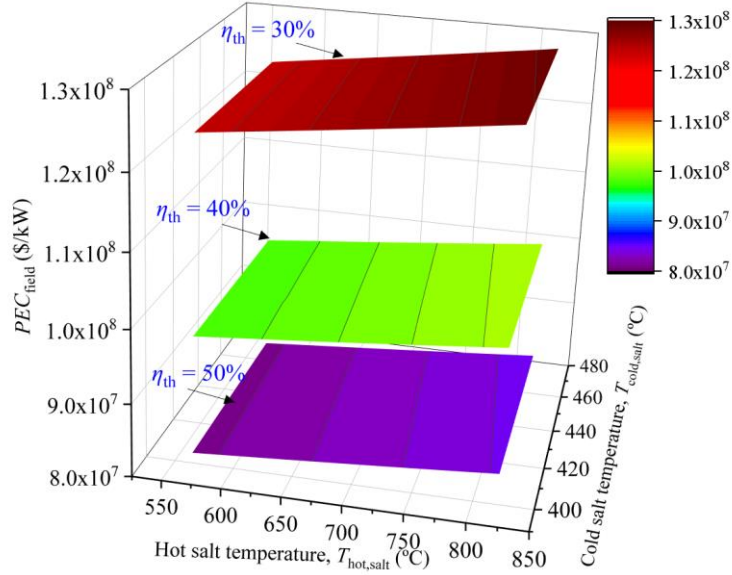
358 Figure 8 shows the variations of η_{rec} , PEC_{field} , and PEC_{TES} with the $T_{\text{hot,salt}}$, $T_{\text{cold,salt}}$ and η_{th} . With
 359 the decrement of $T_{\text{hot,salt}}$, the solar receiver can achieve better thermodynamic performance and the
 360 maximum receiver efficiency reaches 85%. It is also noted that the effects of $T_{\text{cold,salt}}$ and η_{th} on η_{rec}
 361 can be neglected. In addition, a larger solar field area and higher tower are required due to the drop
 362 in η_{rec} . Therefore, an increase in $T_{\text{hot,salt}}$ is associated with an increase in the solar field cost, as shown
 363 in Figure 8(b). It can be seen that increasing $T_{\text{hot,salt}}$ has no benefit on the solar field, but higher $T_{\text{hot,salt}}$
 364 can always result in better performance and lower cost of the power block. On the other hand, PEC_{field}
 365 decreases significantly as η_{th} increases. It shows that the solar field with η_{th} of 50% has the lowest
 366 PEC_{field} of \$ 8.1×10^7 , which is 34% lower than that of the case with η_{th} of 30%. Since the required
 367 heat input is inversely proportional to η_{th} for the given power output, the total heliostat area and the
 368 tower height increase drastically as η_{th} decreases, affecting the solar field cost. Therefore, improving
 369 the thermal efficiency of the power block not only boosts the solar-to-electricity efficiency of the
 370 whole plant but also dramatically reduces the total capital cost.

371 Figure 8(c) shows the variation of PEC_{TES} with $T_{\text{hot,salt}}$, $T_{\text{cold,salt}}$ and η_{th} , and it can be seen that

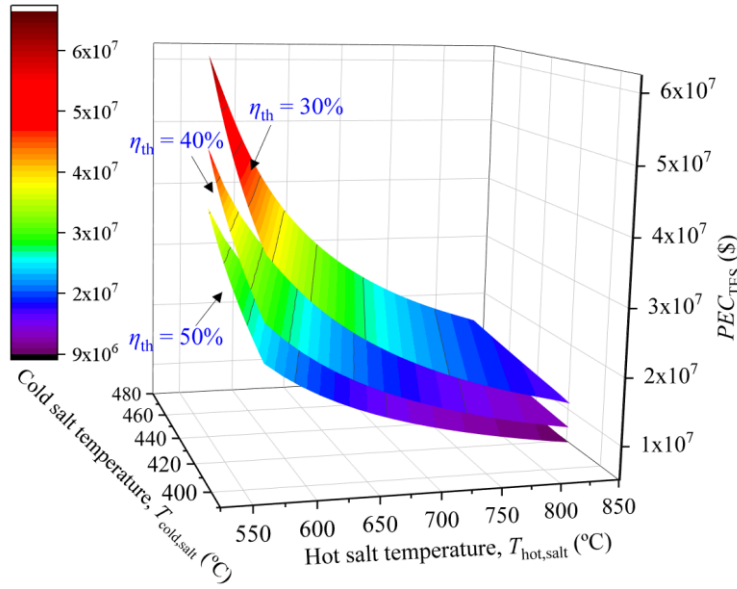
these three parameters can significantly affect the cost of the TES subsystem. As the power block thermal efficiency (η_{th}) increases from 30% to 40%, the purchase equipment cost of the TES system (PEC_{TES}) decreases by $\sim 25\%$. A 10% further increase in η_{th} is estimated to save 20% on the PEC_{TES} . This is because the required heat storage capacity decreases as the η_{th} increases within the same design power output, which also implies that TES cost is quite sensitive to the power block thermodynamic performance. Therefore, the cost of SPT plants can be reduced to some extent if the traditional steam Rankine power blocks are replaced by novel SCO_2 power blocks. On the other hand, the value of PEC_{TES} in this work is scaled from the commercial molten-salt power tower model using the widely used solar salt (mixture of 60 wt% $NaNO_3$ and 40 wt% KNO_3), which has a temperature difference between the hot salt and cold salt (ΔT_{salt}) of 284 °C, and a unit value of 22 $\$/kW \cdot h$ is used as suggested by the SAM software [36]. As seen in Fig. 8(c), PEC_{TES} shows a decreasing trend with an increase in ΔT_{salt} , which is the determining factor of the TES cost because the size of the thermal storage capacity is inversely proportional to ΔT_{salt} . The lowest cost of TES system can be achieved in the case of η_{th} , $T_{hot,salt}$, and $T_{cold,salt}$ equal to 50%, 800 °C and 400 °C, respectively. In summary, the values of $T_{hot,salt}$ and $T_{cold,salt}$ need to be carefully considered, and improving η_{th} is essential to achieve superior performance of the whole SPT plant.



(a)



(b)



(c)

Fig. 8. Variations of: (a) receiver efficiency η_{rec} , (b) purchase equipment cost of solar field PEC_{field} , and (c) purchase equipment cost of thermal energy storage system PEC_{TES} with different hot salt temperatures ($T_{hot,salt}$), cold salt temperatures ($T_{cold,salt}$), and thermal efficiencies of power block ($\eta_{th} = 30\%, 40\%, 50\%$). The rated output power of the SPT plant is 20 MWe, and the solar multiple is set as 2.8. The design DNI is 850 W/m^2 and the environment temperature is set as 25°C .

4.2 Combined SCO_2 -ORC systems

Effects of five key parameters, i.e., the hot salt temperature ($T_{salt,hot}$), the maximum pressure of the SCO_2 cycle (P_{max}), the minimum pressure of the SCO_2 cycle (P_{min}), the ORC evaporating pressure (P_{eva}), the

ratio of pressure ratio (RPR) on the thermal efficiency (η_{th}) and the purchase equipment cost (PEC_{pb}) of the power block are discussed in this section. Among these parameters, $T_{salt,hot}$ is the parameter of the molten salt loop, P_{max} , P_{min} , and RPR are parameters of the SCO_2 subsystem, and P_{eva} is the key parameter of the ORC subsystem. The ratio of pressure ratio, RPR , is only expressed for the cases with an intercooling SCO_2 cycle and the partial-cooling SCO_2 cycle, and it is defined as:

$$RPR = \frac{P_{max}/P_{inter}-1}{P_{max}/P_{min}-1} \quad (27)$$

where P_{inter} is the intermediate pressure in the SCO_2 compression process.

The following are set as baseline conditions for all the investigated layouts: $T_{salt,hot} = 565$ °C, $T_{salt,cold} = 445$ °C, $P_{max} = 25$ MPa, $P_{min} = 7.4$ MPa, $P_{eva} = 800$ kPa, and $RPR = 0.8$. When the effects of one parameter are studied, the other parameters are fixed at their basic values. Fig. 9 shows the variations of η_{th} and PEC_{pb} of the power block at various maximum pressures of the SCO_2 cycle subsystem (P_{max}). It can be seen that higher P_{max} enables better thermodynamic and economic performance of the combined power systems with the RE, IC, and PC SCO_2 layouts. In the case of RC-ORC layout, η_{th} increases first and then decreases as P_{max} increases, while the PEC_{pb} shows an opposite variation. The comparison results of the four proposed combined SCO_2 -ORC power systems indicate that the IC-ORC system yields the highest thermal efficiency when P_{max} is higher than 21 MPa, and the PC-ORC layout always provides the lowest PEC_{pb} .

Figure 10 shows the comparison of η_{th} and PEC_{pb} of different SCO_2 -ORC system configurations at various minimum pressures of the SCO_2 cycle subsystem (P_{min}). It shows that decreasing P_{min} benefits thermal efficiency and reduces unit cost for the RC-ORC layout and the PC-ORC layout. With respect to the RE-ORC layout, η_{th} decreases with the increase in P_{min} , while PEC_{pb} decreases with P_{min} in the low-pressure range and increases in the high-pressure range. For the RC-ORC layout, the highest η_{th} exists when P_{min} is 8 MPa, and decreasing P_{min} can result in a lower power block cost. Fig. 11 shows the comparison of η_{th} and PEC_{pb} with different SCO_2 -ORC system configurations at various hot salt temperatures ($T_{salt,hot}$). As expected, η_{th} increases with $T_{salt,hot}$ for all SCO_2 -ORC layouts. It also can be seen that the PEC_{pb} decreases with $T_{salt,hot}$ for all SCO_2 -ORC layouts, indicating that increasing $T_{salt,hot}$ is beneficial for both the thermodynamic and economic performance of the power block. More precisely, the RC-ORC system with $T_{salt,hot}$ of 800 °C achieves the highest η_{th} of 48% and the lowest PEC_{pb} of $\$1.7 \times 10^7$, which are 26% higher and 47% lower than that of the case with $T_{salt,hot}$ of 500 °C.

Figure 12 shows the effects of the ORC evaporating pressure (P_{eva}) on η_{th} and PEC_{pb} . It is noted that the operating ranges of P_{eva} for the RE-ORC and RC-ORC systems are larger than those for the IC-ORC and PC-ORC systems. This is because the P_{eva} of ORC working fluid is in proportion to the SCO_2 inlet temperature, which differs greatly in these combined SCO_2 -ORC systems. For the RE-ORC system,

the SCO_2 inlet temperature corresponding to the maximum P_{eva} is about 200 °C, while the maximum SCO_2 inlet temperature is only 80 °C for the PC-ORC system. It is also observed that η_{th} in all cases increases first to a maximum and then decreases. Moreover, higher P_{eva} results in higher PEC_{pb} , and PEC_{pb} of IC-ORC and PC-ORC layouts are sensitive to the change in P_{eva} . In particular, for the PC-ORC system, the PEC_{pb} increases by 10% when P_{eva} increases from 400 kPa to 1500 kPa. Fig. 13 demonstrates the effects of the ratio of pressure ratio (RPR) on η_{th} and PEC_{pb} . It is found that an optimum value of RPR exists for the IC-ORC and PC-ORC layouts when it ranges from 0.8 to 0.9. The PEC_{pb} of these two combined systems always increase with RPR , and the IC-ORC layout has a more significant increase.

It can be concluded that the combined IC-ORC system yields the highest thermal efficiency, and the combined PC-ORC system provides the lowest specific investment cost under most conditions. Although the system configurations of IC-ORC and PC-ORC are complex due to the additional SCO_2 compressor and cooler, they have distinct advantages in terms of cost or thermal efficiency over the RE-ORC and RC-ORC layouts. In addition, the effects of P_{max} , P_{min} , P_{eva} , and RPR on the thermal efficiency and purchase equipment cost of the power block are not simply monotone. Moreover, the whole SPT plant performance also relies on the characteristics of the solar field and TES subsystems, and the incorporation abilities of different SCO_2 -ORC systems with the molten salt loop are of vital importance. Therefore, the performance coupling between the solar field, TES, and SCO_2 -ORC systems needs to be carefully considered and the key parameters need to be optimised at the design stage.

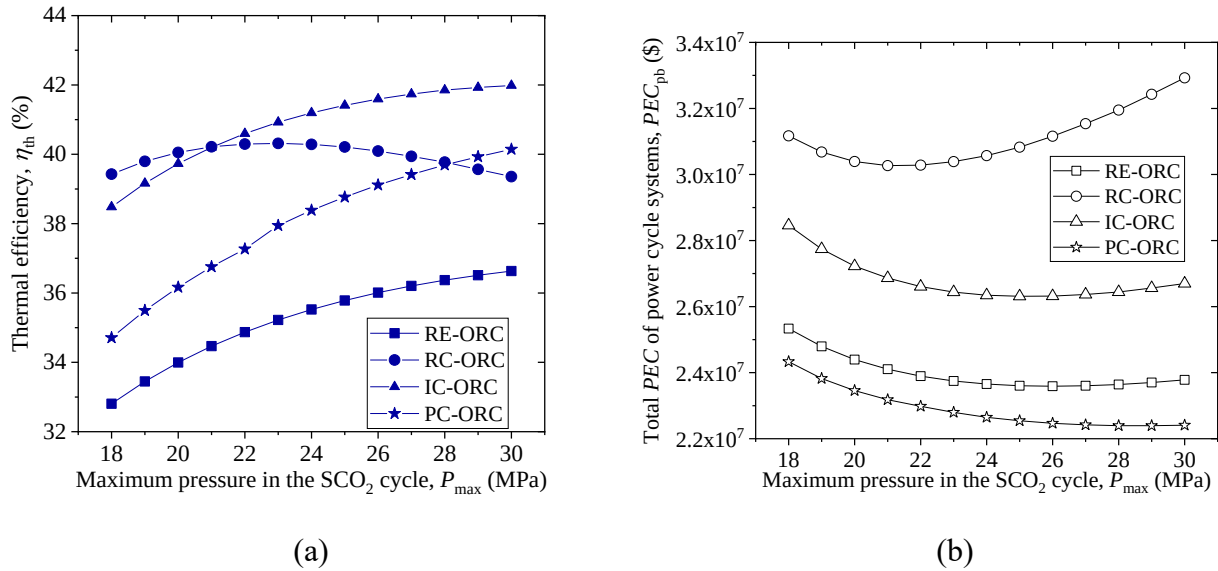
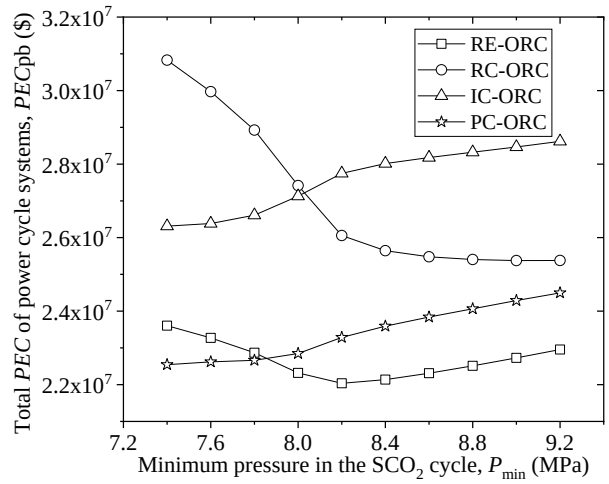
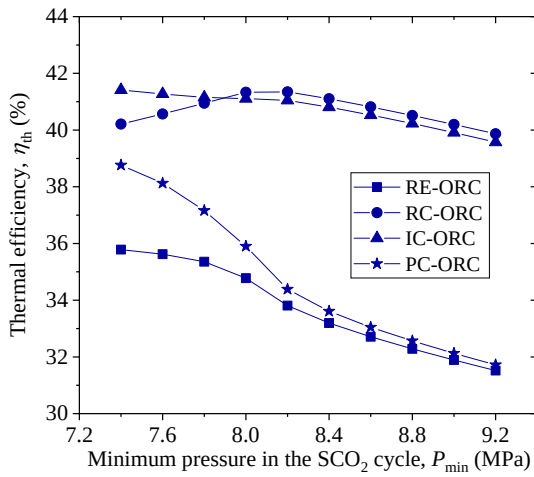


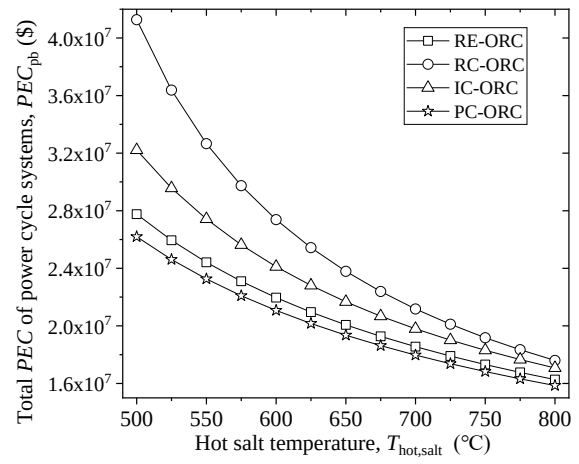
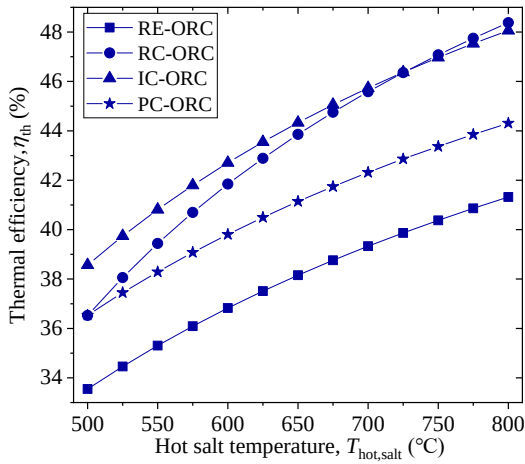
Fig. 9. Effects of the maximum pressure P_{max} on (a) thermal efficiency η_{th} , and (b) purchase equipment cost of power block PEC_{pb} , for different combined SCO_2 -ORC systems. Results correspond to design conditions that $T_{\text{salt,hot}} = 565$ °C, $T_{\text{salt,cold}} = 445$ °C, $P_{\text{min}} = 7.4$ MPa, $P_{\text{eva}} = 800$ kPa, and $RPR = 0.8$.



(a)

(b)

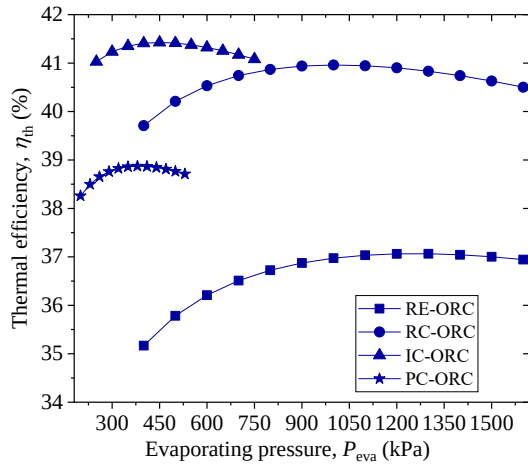
Fig. 10. Effects of the SCO_2 minimum pressure $P_{\min}$ on (a) thermal efficiency η_{th} , and (b) purchase equipment cost of power block PEC_{pb} , for different combined SCO_2 -ORC systems. Results correspond to design conditions that $T_{\text{salt,hot}} = 565^\circ\text{C}$, $T_{\text{salt,cold}} = 445^\circ\text{C}$, $P_{\text{max}} = 25\text{ MPa}$, $P_{\text{eva}} = 800\text{ kPa}$, and $RPR = 0.8$.



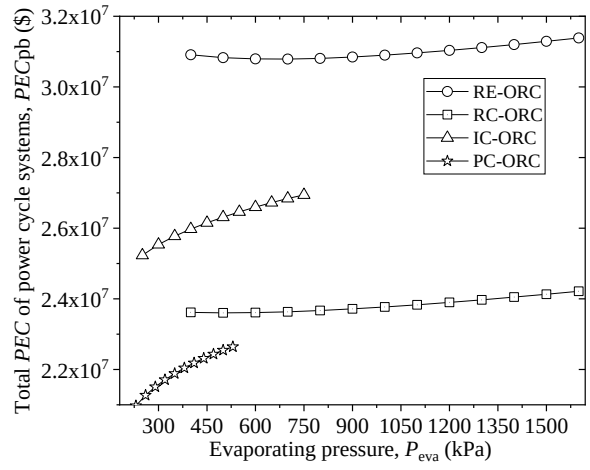
(a)

(b)

Fig. 11. Effects of the hot salt temperature $T_{\text{salt,hot}}$ on (a) thermal efficiency η_{th} , and (b) purchase equipment cost of power block PEC_{pb} for different combined SCO_2 -ORC systems. Results correspond to design conditions that $T_{\text{salt,cold}} = 445^\circ\text{C}$, $P_{\text{max}} = 25\text{ MPa}$, $P_{\min} = 7.4\text{ MPa}$, $P_{\text{eva}} = 800\text{ kPa}$, and $RPR = 0.8$.



(a)



(b)

Fig. 12. Effects of the ORC evaporating pressure P_{eva} on (a) thermal efficiency η_{th} , and (b) purchase equipment cost of power block PEC_{pb} for different combined SCO_2 -ORC systems. Results correspond to design conditions that $T_{salt,hot} = 565$ °C, $T_{salt,cold} = 445$ °C, $P_{max} = 25$ MPa, $P_{min} = 7.4$ MPa, and $RPR = 0.8$.

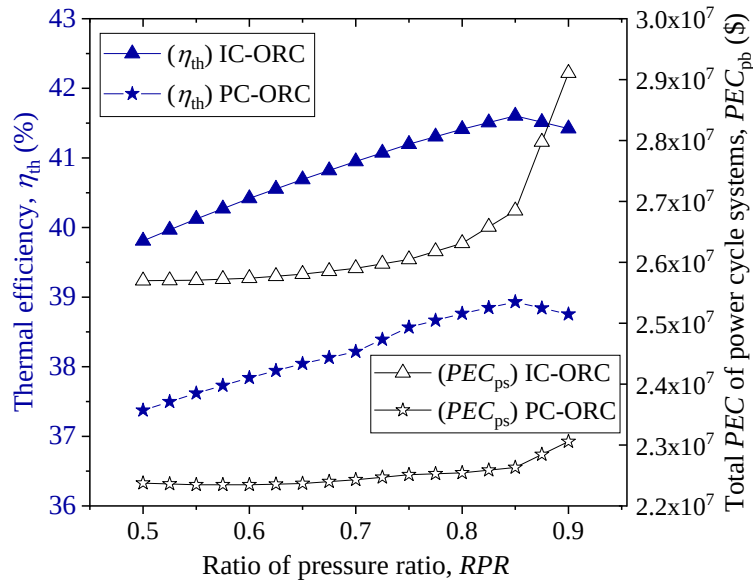


Fig. 13. Effects of the ratio of pressure ratio RPR to the thermal efficiency η_{th} , and purchase equipment cost of power block PEC_{pb} for different combined SCO_2 -ORC systems. Results correspond to design conditions that $T_{salt,hot} = 565$ °C, $T_{salt,cold} = 445$ °C, $P_{max} = 25$ MPa, $P_{min} = 7.4$ MPa, $P_{eva} = 800$ kPa.

5. Comparison of different STP- SCO_2 -ORC systems

A case study in Delingha (37.4°N, 97.4°E), China is considered for performance comparison of different STP- SCO_2 -ORC systems. The SPT plant design is performed at solar noon on the summer solstice [58], assuming a DNI of 850 W/m² and an environment temperature of 25 °C. To provide a fair performance comparison, all power blocks are integrated with the same solar field configuration

486 which derives from the GEMASOLAR plant located in Fuentes de Andalusia (Seville, Spain). The
 487 solar multiple is set to be 2.8. The main parameters of the solar field are listed in Table 9.

488

489 **Table 9.** GEMASOLAR plant main parameters [59, 60]

Parameter	Value
Number of heliostats	2650
Mirror aperture area	306660 m ²
Tower height	140 m
Receiver diameter	8.1 m
Receiver height	10.6 m
Storage size (extend power block operation without sun radiation)	15 hours

490

491 5.1 Design optimisation

492 It can be concluded from the parameter analysis study that $T_{\text{hot,salt}}$, $T_{\text{cold,salt}}$, P_{max} , P_{min} , P_{eva} , and RPR
 493 have a non-monotonic effect on the component efficiency and costs. Therefore, these parameters are
 494 selected as the decision variables for design optimisation, and for each integrated system, the vector
 495 of decision variables is presented as follows:

$$496 \quad \vec{X} = \begin{cases} (T_{\text{hot,salt}}, T_{\text{cold,salt}}, P_{\text{max}}, P_{\text{min}}, P_{\text{eva}}) & \text{for SPT-RE-ORC system} \\ (T_{\text{hot,salt}}, T_{\text{cold,salt}}, P_{\text{max}}, P_{\text{min}}, P_{\text{eva}}) & \text{for SPT-RC-ORC system} \\ (T_{\text{hot,salt}}, T_{\text{cold,salt}}, P_{\text{max}}, P_{\text{min}}, P_{\text{eva}}, RPR) & \text{for SPT-IC-ORC system} \\ (T_{\text{hot,salt}}, T_{\text{cold,salt}}, P_{\text{max}}, P_{\text{min}}, P_{\text{eva}}, RPR) & \text{for SPT-PC-ORC system} \end{cases} \quad (28)$$

497 The specific investment cost of the whole SPT plant (SIC_{total}), is selected as the optimisation objective:

$$498 \quad \min. SIC_{\text{total}}(\vec{X}) \quad (29)$$

499 which is subject to the following conditions:

$$500 \quad \begin{cases} 565 \leq T_{\text{hot,salt}}(^{\circ}\text{C}) \leq 800 \\ 290 \leq T_{\text{cold,salt}}(^{\circ}\text{C}) \leq 565 \\ P_{\text{min}} \leq P_{\text{max}}(\text{MPa}) \leq 30 \\ 7.38 \leq P_{\text{min}}(\text{MPa}) \leq P_{\text{max}} \\ 0.2 \leq P_{\text{eva}}(\text{MPa}) \leq 2 \\ 0 \leq RPR(-) \leq 1 \end{cases} \quad (30)$$

501 The thermo-economic optimisation for each system is conducted by using the genetic algorithms
 502 (GA) method, with detailed optimisation results listed in Table 10. The SPT systems with standalone
 503 SCO_2 systems are also included as a baseline case for comparison. The results indicate that all SPT- SCO_2 -
 504 ORC systems can achieve a decrease in SIC_{total} relative to the corresponding SPT plants with standalone

505 SCO₂ systems, and also, adding an ORC bottoming system is beneficial to the overall performance.

506 The SPT-IC-ORC system is found to have the lowest SIC_{total} of 7990 \$/kW, followed by the SPT-
 507 PC-ORC system of 8060 \$/kW, but their SIC_{total} reductions after adding bottoming ORC subsystems
 508 are quite small ($< 2\%$). Also, the SIC_{total} of the SPT-RC-ORC system just decreases by 3% relative to
 509 the SPT-RC system. The amount of performance improvement by adding bottoming ORC subsystems
 510 to the SPT-RC, SPT-IC and SPT-PC systems is not as expected. Fig. 14 shows a breakdown of power
 511 outputs by the topping SCO₂ subsystem and the bottoming ORC subsystem. It can be seen that the
 512 $W_{net,ORC}$ is relatively low at the optimal SIC_{total} condition for the cases of the SPT-IC-ORC and SPT-
 513 PC-ORC systems. By comparison, the ORC subsystem in the SPT-RE-ORC system yields a
 514 considerable amount of power output (2.8 MWe), which enables the SPT-RE-ORC system to achieve
 515 the largest SIC_{total} decrease (11%) by adding an extra ORC subsystem. This is because the ORC
 516 evaporation temperature of the SPT-RE-ORC plant is the highest among the four proposed integrated
 517 SPT systems, and the ORC subsystem can absorb the maximum amount of the waste heat from the
 518 SCO₂ working fluid, thereby improving the performance of the power block.

519

520 **Table 10.** Optimisation results of different SPT systems with environment temperature $T_{env} = 25$ °C.

		SPT- RE	SPT- RE- ORC	SPT- RC	SPT- RC- ORC	SPT-IC	SPT- IC- ORC	SPT- PC	SPT-PC- ORC
Optimal operating parameter	P_{max} (MPa)	30.0	30.0	29.9	29.8	29.1	29.7	29.5	29.6
	P_{min} (MPa)	8.2	7.4	9.5	8.7	7.5	7.6	7.7	7.5
	P_{eva} (MPa)	/	3.47	/	0.83	/	0.54	/	0.47
	$T_{hot,salt}$ (°C)	800	800	774	800	800	800	800	783
	$T_{cold,salt}$ (°C)	583	584	603	630	616	616	586	562
	RPR	/	/	/	/	0.65	0.71	0.71	0.73
Performance indicators	$\eta_{plant,design}$ (%)	20.1	23.0	23.5	24.2	23.9	24.3	22.7	23.2
	$W_{net,total}$ (MW)	18.7	21.4	21.9	22.5	22.2	22.6	21.2	21.6
	ITC (\$·10 ⁸)	1.72	1.74	1.84	1.83	1.81	1.80	1.73	1.74
	SIC_{total} (\$/kW)	9180	8150	8400	8160	8130	7990	8190	8060

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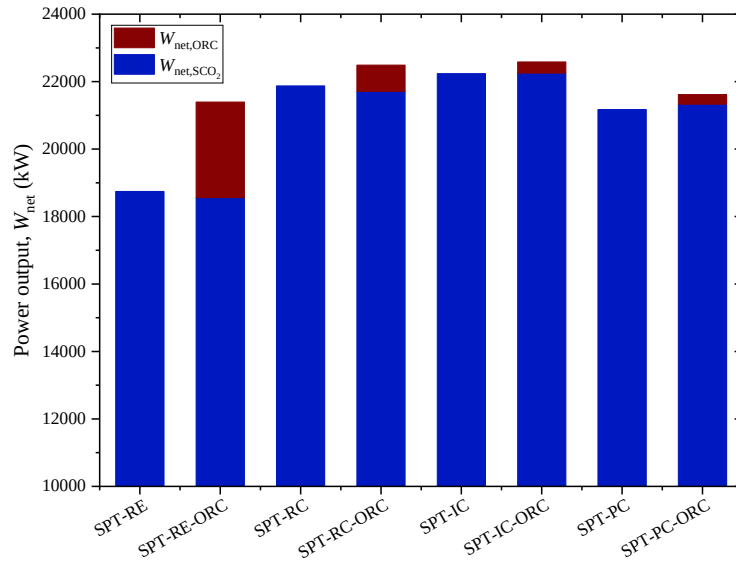
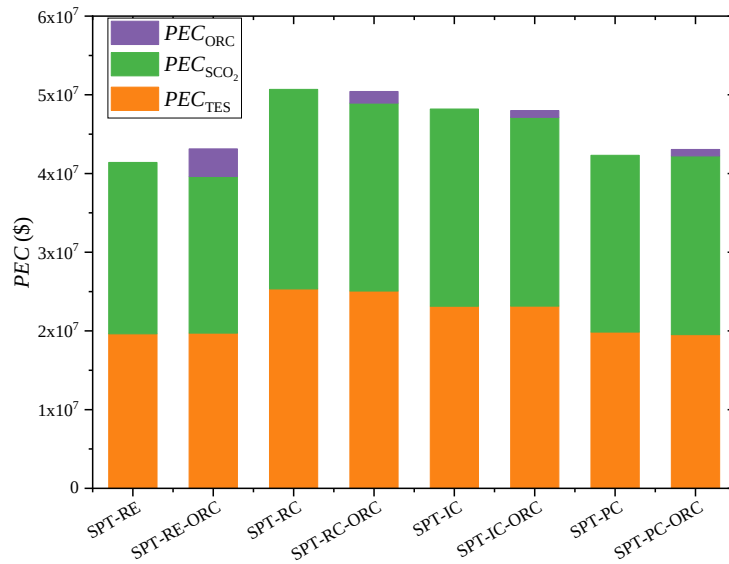


Fig. 14. Power output with breakdown contributed by the topping SCO_2 cycle subsystem and the bottoming ORC system of SPT plant integrated with different power cycle systems. The solar-array area and solar multiple are set to be 306660 m^2 and 2.8. The design DNI is 850 W/m^2 and the environment temperature is set as $25 \text{ }^\circ\text{C}$. Results correspond to the optimal conditions with the minimum specific investment cost of the whole plant shown in Table 10.

Furthermore, Fig. 15 shows the purchase equipment cost (PEC) breakdown of TES, SCO_2 and ORC subsystems, corresponding to the operating condition at which the SPT plants can achieve the lowest SIC_{total} . For SPT plants integrated with SCO_2 -ORC systems, the total PEC s do not show significant changes compared to those of the standalone SCO_2 systems. Using the SPT-RE-ORC system as a reference, the relative differences in PEC are 16.9%, 11.3% and -0.2% for the SPT-RC-ORC, SPT-IC-ORC and SPT-PC-ORC systems. As the salt temperature differences (ΔT_{salt}) in the SPT-RE-ORC and SPT-PC-ORC systems are lower, these two systems provide smaller salt storage tanks; thereby, their TES costs are relatively low.



538

539 **Fig. 15.** Purchase equipment cost (PEC) of subsystems for SPT plant integrated with different power
 540 cycle systems. The solar-array area and solar multiple are set to be 306,660 m² and 2.8. The design
 541 DNI is 850 W/m² and the environment temperature is set as 25 °C. Results correspond to the optimal
 542 conditions with the minimum specific investment cost of the whole plant shown in Table 10.

543

544 Fig. 16 shows the effect of the environment temperature on the optimal specific investment
 545 cost of the whole SPT plant. The environment temperature varies between 5 and 45 °C, and the DNI
 546 is fixed at 850 W/m². The results show that the specific investment costs of SPT plants will increase
 547 by more than 7.6% while the design environment temperature increases from 25 to 45 °C, which
 548 means that higher investment costs for SPT plants are needed in hot areas. The reasons can be
 549 summarised as follows: firstly, higher environment temperature corresponds to higher condensation
 550 pressures of the ORC system and results in a drop in the unit output work of the ORC turbine.
 551 Secondly, the SCO_2 main compressor deviates from its optimum operating condition if the
 552 environment temperature is higher than 25 °C. At relatively high environment temperatures ($T_{env} >$
 553 25 °C), the main SCO_2 compressor inlet deviates from its design operating condition and causes a
 554 drop in the SCO_2 system efficiency. Thirdly, environmental temperature is one of the major factors
 555 affecting the performance of air-cooled equipment. Any change in environment temperatures
 556 induces a change in the airflow rate in the cooler and condenser, which can lead to a change in the
 557 power consumption of cooling fans. Therefore, at high environmental temperatures, more power
 558 consumption is required, and less output power is produced.

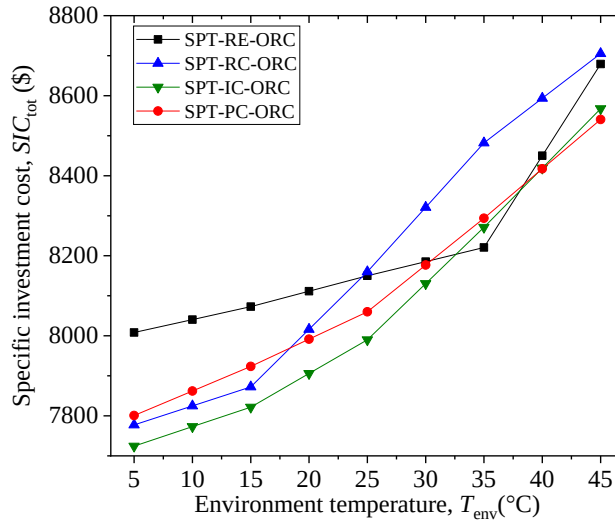
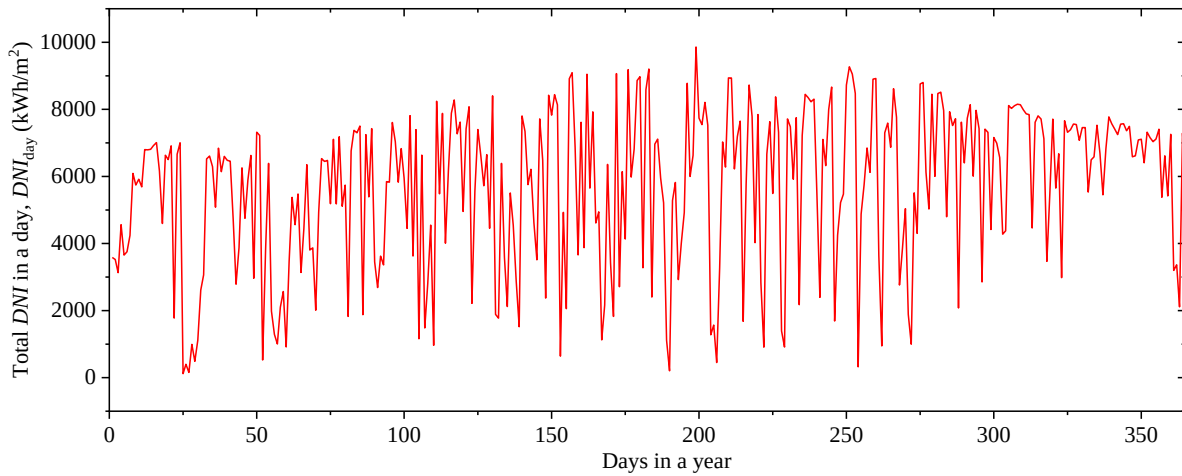


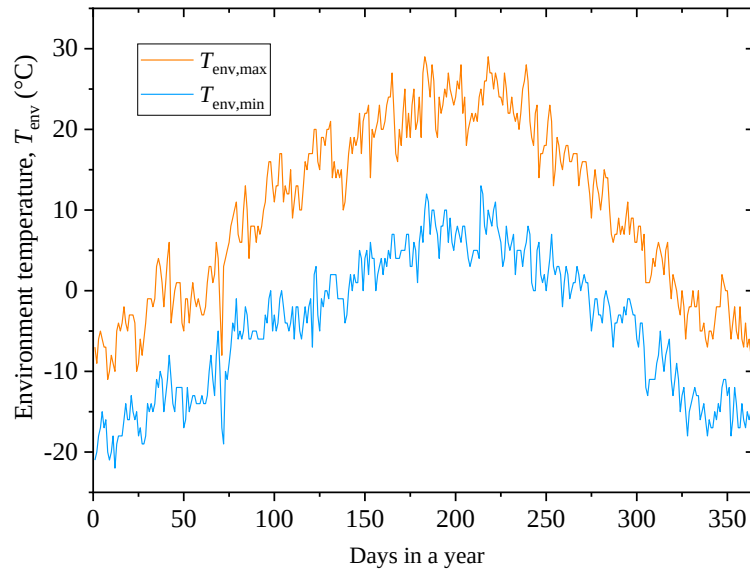
Fig. 16. Effects of environment temperature T_{env} to the optimal specific investment cost of the whole SPT plant SIC_{total} for different SPT systems.

5.2 Annual performance simulation

The annual performance of the proposed STP-SCO₂-ORC integrated systems is assessed based on real weather data in Delingha, China (37.4°N, 97.4°E), of which the environment temperature and DNI_{day} are shown in Fig. 17. System parameters correspond to the optimal design scheme with the lowest specific investment cost in Table 10. Off-design models are used to predict performance under different operating conditions. The annual electricity generation of the four proposed SPT-SCO₂-ORC systems is calculated based on real weather conditions.



(a)



(b)

Fig. 17. (a) Year-round direct normal insolation (DNI_{day}) fluctuation, and (b) maximum and minimum daily environment temperature in Delingha, China.

The total energy input to the solar field is directly proportional to the value of DNI . Therefore, the heat transferred from the solar receiver to the molten salt loop constantly changes with time. In addition, the SCO_2 compressor inlet temperature and the condensing temperature are influenced by the environment temperature. The output power of SPT plants changes frequently, and an off-design analysis should be implemented to obtain accurate electricity generation. Some assumptions are made to simplify the analysis: (1) weather conditions are constant for each hour and hourly-daily-monthly-yearly electricity generation is calculated in sequence; (2) the dynamic response of the systems are not taken into account and systems always runs in a steady state; (3) the SPT plants are designed to meet the base load and do not undertake the tasks of peaking and frequency modulation. The power block supplies the rated power when the heat input is sufficient, so as to address the maximum annual generating capacity of different SPT plants; and (4) the efficiencies of all SCO_2 compressors, turbines and pumps are kept constant at the design values, assuming that their deviations from the design condition will be small due to the adoption of variable-frequency speed control systems.

The geometric characteristics of the solar field and receiver, the design parameters of the power cycle system and the design conductance (UA) of heat exchangers are obtained in the design stage and considered known parameters. The heliostat field efficiency (η_{heli}) is obtained using the SAM software [36] for every hour and is determined by the heliostat field layout, geographical location and sun direction. The off-design performance of the solar receiver and heat exchangers is evaluated using the models in Section 3. For off-design analysis of heat exchangers, the correlation between the conductance (UA) and

the mass flow rates of hot fluid and cold fluid ($m_{\text{hot}}, m_{\text{cold}}$) can be expressed as [61]:

$$\frac{1}{U \cdot A} \propto \frac{1}{m_{\text{cold}}^{0.8}} + \frac{1}{m_{\text{hot}}^{0.8}} \quad (31)$$

Given the design and off-design mass flow rates of both the cold and hot fluids, the ratio of the off-design conductance UA_{off} and the design conductance UA_{design} can be calculated from [32]:

$$\frac{(U \cdot A)_{\text{off}}}{(U \cdot A)_{\text{design}}} = \left(\frac{m_{\text{cold,off}}^{0.8} \cdot m_{\text{hot,off}}^{0.8}}{m_{\text{cold,design}}^{0.8} \cdot m_{\text{hot,design}}^{0.8}} \right) \cdot \left(\frac{m_{\text{cold,design}}^{0.8} + m_{\text{hot,design}}^{0.8}}{m_{\text{cold,off}}^{0.8} + m_{\text{hot,off}}^{0.8}} \right) \quad (32)$$

Based on the above off-design analysis method for the SPT plant, the effect of environment temperature variation on the thermal efficiency of the power block can be assessed, as shown in Fig. 18. The ambient environment temperature varies from $-30\text{ }^{\circ}\text{C}$ to $30\text{ }^{\circ}\text{C}$, which correspond to the lower and upper-temperature limits in Delingha, China. It can be seen from the figure that the thermal efficiency (η_{th}) of the power block decreases as the environment temperature (T_{env}) increases. This is because an increase in T_{env} results in a decrease in ORC efficiency and an increase in the power consumption for SCO_2 compression.

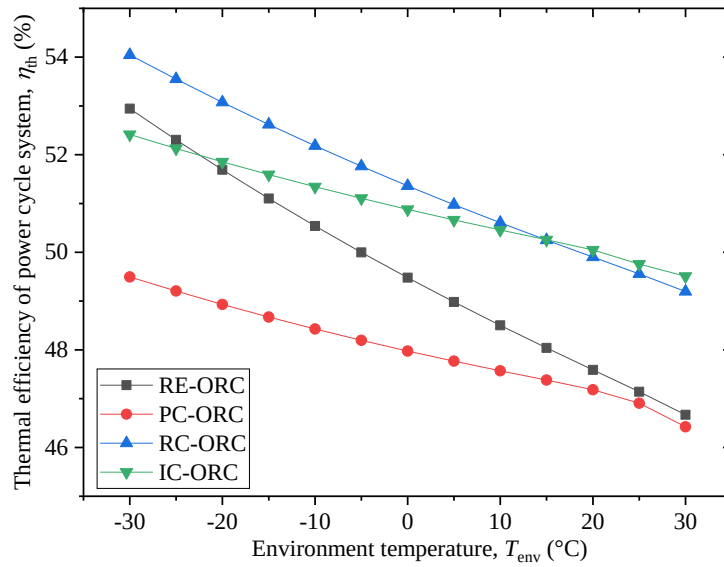
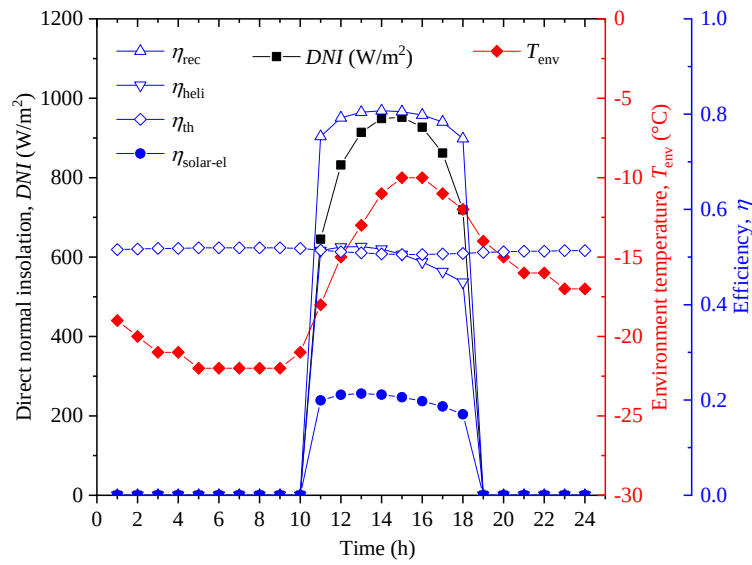


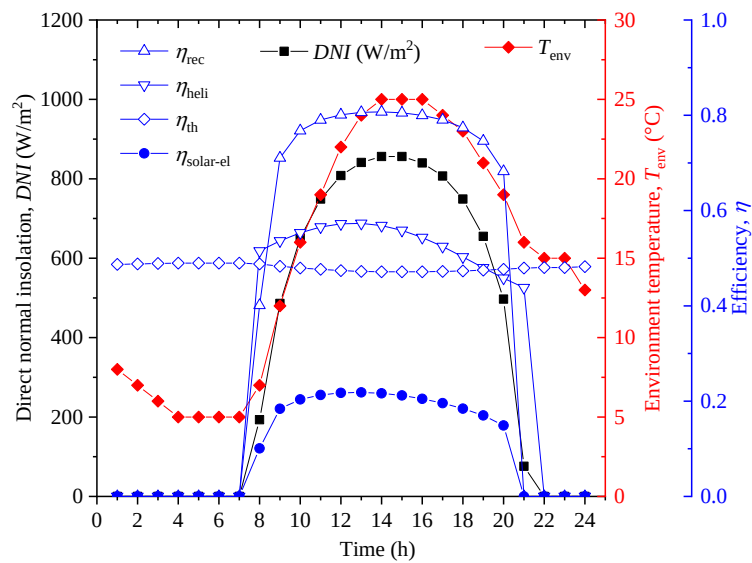
Fig. 18. Effect of environment temperature on the thermal efficiency of different combined SCO_2 -ORC power systems. System design parameters correspond to the optimal conditions with the minimum specific investment cost of the whole plant shown in Table 10.

In this section, we take the SPT-RE-ORC system as an example to show the annual electricity generation calculation process. Parameters such as T_{env} , DNI , the power block thermal efficiency (η_{th}), receiver efficiency (η_{rec}), heliostat field efficiency (η_{heli}), and plant design efficiency ($\eta_{\text{plant,design}}$) over

618 a typical winter and summer day are shown in Fig. 19. Lower T_{env} in winter results in higher η_{th} due
 619 to the decrease in power consumption of cooling fans and the increase in power output of the ORC
 620 expander. Therefore, η_{th} of the SPT-RE-ORC can be as high as 51% on a typical winter day (12th
 621 January). On the other hand, the sunlight lasts 9 hours on the selected winter day and 14 hours on the
 622 summer day. During the summer, the heliostat field receives more solar energy; thus, the SPT plant
 623 has a longer operating time. In this case, η_{hel} and η_{rec} are determined by DNI over a day, and the real-
 624 time solar-to-electricity efficiency ($\eta_{\text{solar-el}}$) is positively associated with the DNI . The hourly-daily
 625 electricity generation can be obtained based on the above data and the solar field parameters.



(a)



(b)

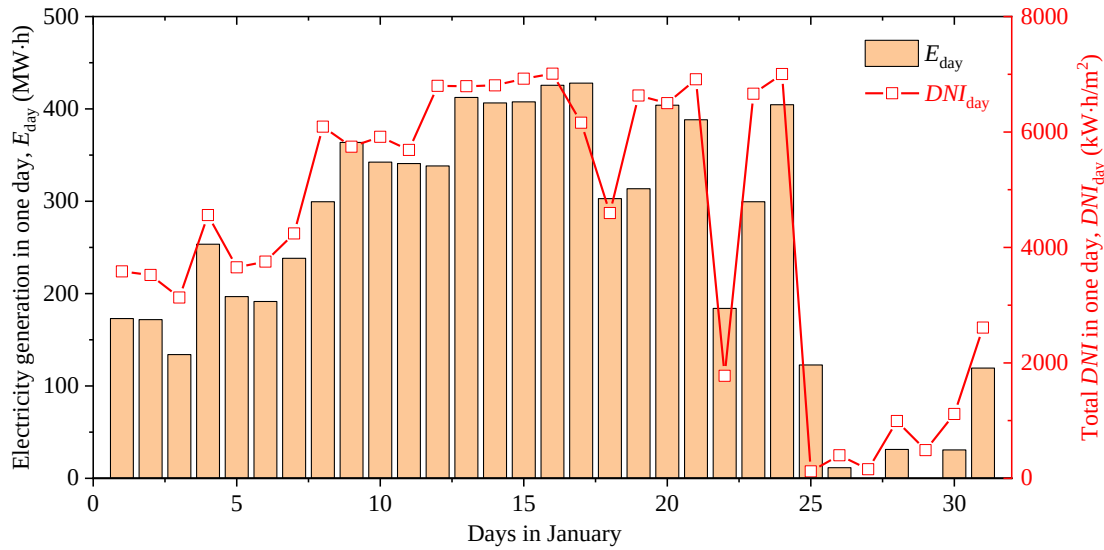
629 **Fig. 19.** Variation of ambient conditions and system efficiency on: (a) a typical winter day (12th
 630
 631

January), and (b) a typical summer day (21st June) of an SPT-RE-ORC system in Delingha, China.

633

Figures 20 and 21 show the daily direct normal insolation (DNI_{day}) and electricity generation (E_{day}) in January and June, respectively. The E_{day} always follow the trend of the change in DNI_{day} , which shows dramatic fluctuations due to the unstable weather conditions. The maximum DNI_{day} in January and June are $7003 \text{ kW}\cdot\text{h}/\text{m}^2$ and $9185 \text{ kW}\cdot\text{h}/\text{m}^2$, respectively. The DNI difference between the summer and winter can be explained by the different sun positions and amount of solar hours. Therefore, on a clear day, the peak value E_{day} in June is higher than that in January. Figure 22 shows the monthly electricity generation and operation hours of the SPT-RE-ORC system. It can be seen that the maximum monthly generating capacity (E_{month}) appears in November, while the SPT plant has the lowest E_{month} in February, which is linked to the Delingha climate. The monthly total DNI value (DNI_{month}) reaches the maximum value of $209 \text{ MW}\cdot\text{h}/\text{m}^2$ in November while the DNI_{month} in February is $128 \text{ MW}\cdot\text{h}/\text{m}^2$. The full-load operation hour is calculated by dividing the rated power output into the monthly electricity generation. It can be seen that the electricity generation of the SPT-RE-ORC plant exceeds $11000 \text{ MW}\cdot\text{h}$ in October, November and December. The total electrical energy over a year reaches $119.8 \text{ GW}\cdot\text{h}$, and the full-load operation time is 5600 h .

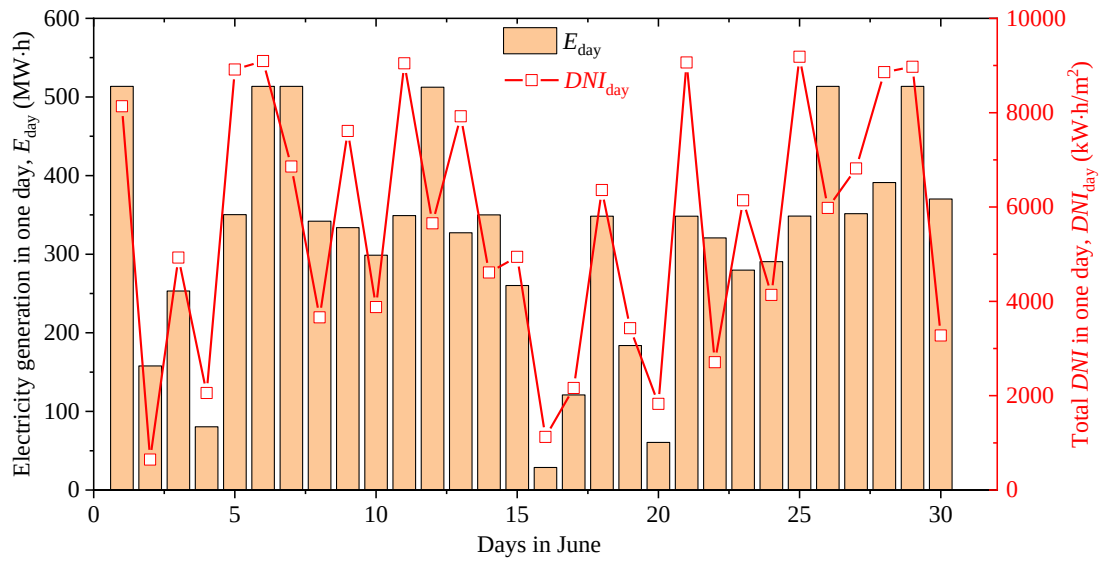
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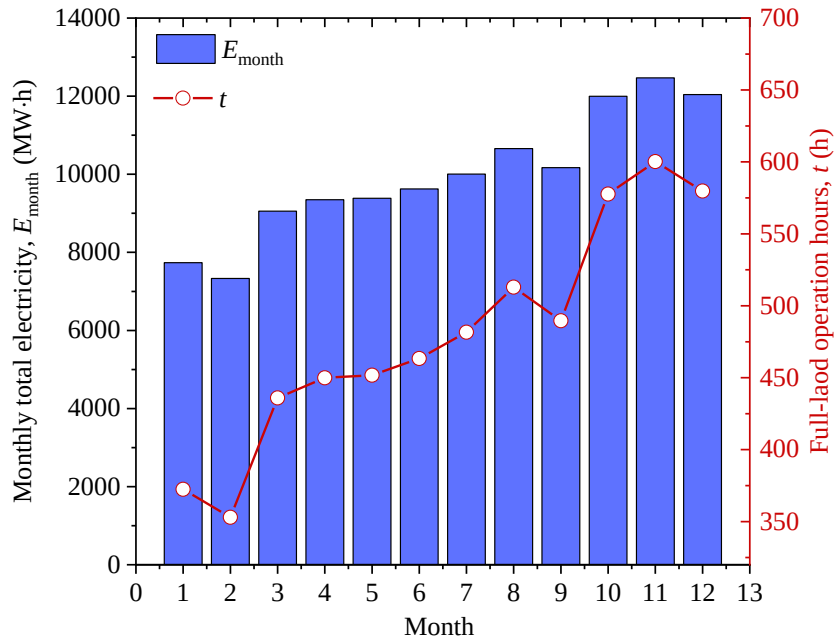
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Fig. 20. Daily direct normal insolation and electricity generation of an SPT-RE-ORC system in Delingha, China, in January.

651



652
 653 **Fig. 21.** Daily direct normal insolation and electricity generation of an SPT-RE-ORC system in
 654 Delingha, China, in June.

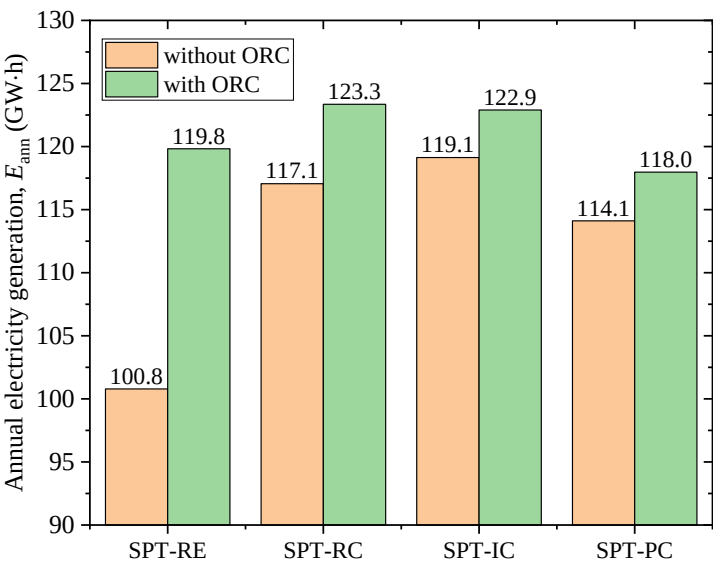


655
 656 **Fig. 22.** Monthly electricity generation and operation hours of an SPT-RE-ORC system in Delingha, China.
 657

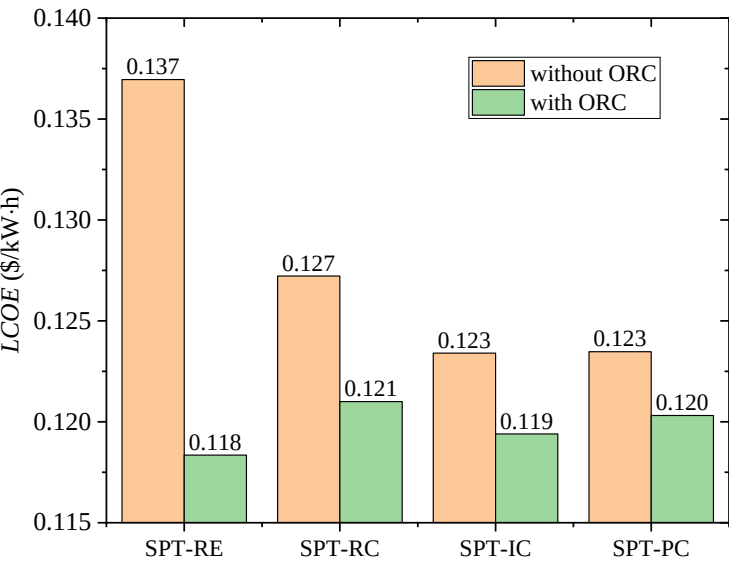
658 The annual electricity generation of four proposed SPT-SCO₂-ORC systems, along with comparisons
 659 with different SPT-SCO₂ systems, are shown in Fig. 23(a), which indicates that an improvement (from 3%
 660 to 19%) in the electricity generation can be achieved by adding a bottoming ORC subsystem to the power
 661 block. The levelised cost of electricity (*LCOE*) and payback time of these SPT-integrated systems are
 662 calculated using the actual annual electricity generation data. As shown in Fig. 23(b), it is not surprising
 663 that the SPT systems integrated with combined SCO₂-ORC systems have lower costs, which benefits from

the improvement in energy conversion efficiency. As shown in Fig. 23 (c), the capital cost of the SPT-SCO₂ plant integrated with ORC systems can be paid back in a shorter period, which further proves that the combined SCO₂-ORC systems are promising technologies for SPT plants.

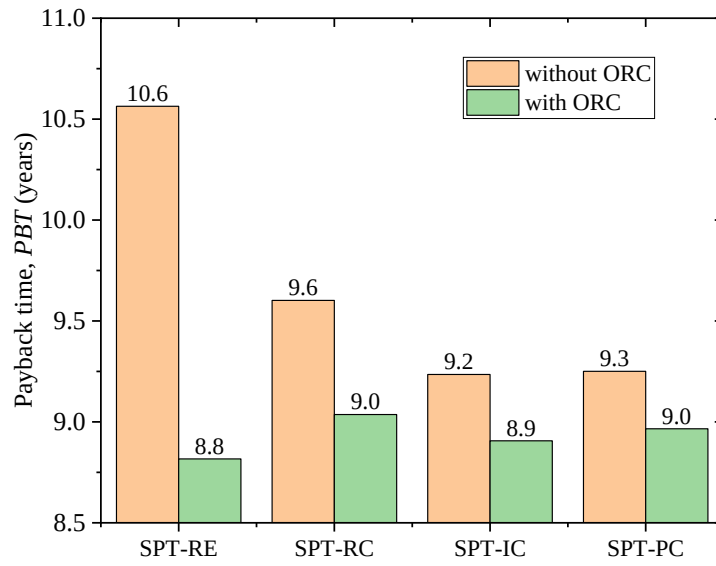
Among the eight cases, the STP-RE-ORC layout has the lowest *LCOE* of 0.12 \$/kW·h, followed by the STP-IC-ORC layout and the STP-RC-ORC layout. In addition, the SPT-RE-ORC system has the simplest structure, and the operational controls and procedures are also much simpler than other systems. Therefore, the combined RE-ORC system is preferred and can be considered in practical applications.



(a)



(b)



(c)

Fig. 23. Comparison of (a) annual electricity generation, (b) $LCOE$, and (c) payback time for different SPT- SCO_2 systems with and without ORC subsystem, using the weather data in Delingha, China.

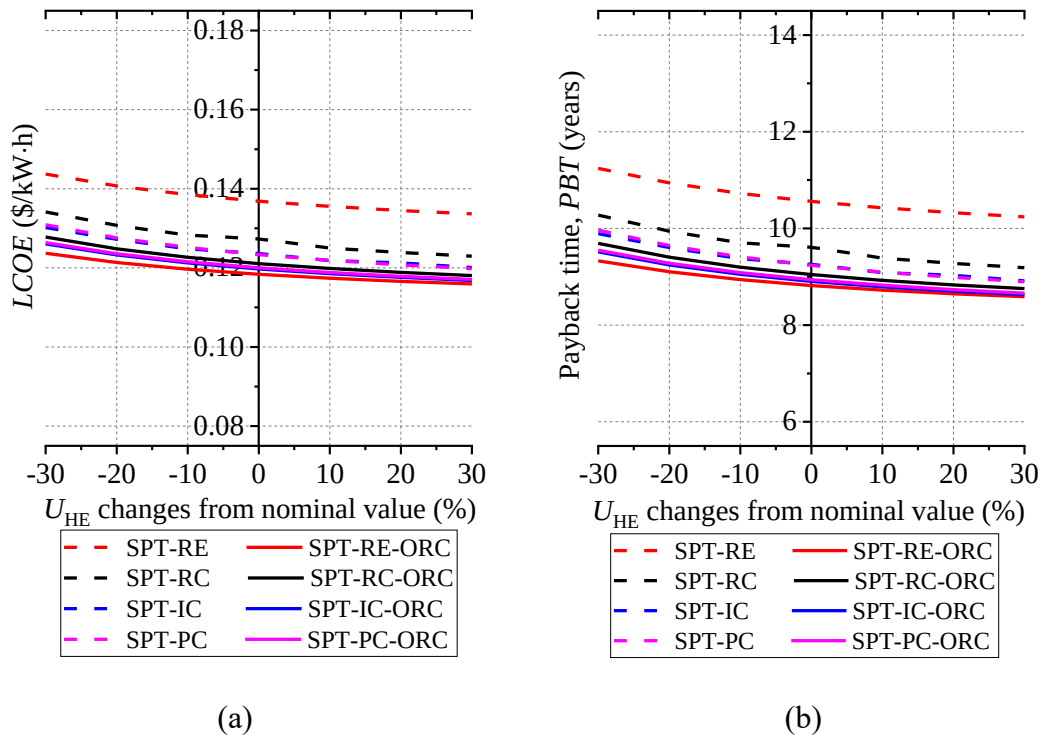
5.3 Sensitivity analysis

As the SCO_2 power blocks are emerging technology and therefore not commercially mature, the investment cost data is not sufficiently accurate. Therefore, the sensitivity analysis of economic performance is conducted to address the uncertainty in the modelling variables, including the total purchase equipment cost (PEC), the discount rate (i_d), the inflation rate (i_f) and the overall heat transfer coefficient of heat exchangers (U_{HE}). At the design stage, the overall heat transfer coefficient of heat exchangers (U_{HE}) is the input parameter, which is a reference value. The design values of U_{HE} are related to the detailed structural parameters and vary in a given range, which induces the uncertainty of heat exchanger cost. In addition, most cost correlations of the SCO_2 equipment are derived from the process industries and the accuracy is not enough, leading to the calculation error of the total purchase equipment cost (PEC). The discount rate (i_d) and inflation rate (i_f) usually have different values in different engineering projects. It is estimated that the uncertainty of these variables ranges from -30 to +30% of the nominal values [62].

The uncertainty analysis results that the effects of U_{HE} , PEC , i_d , and i_f on levelised cost of electricity ($LCOE$) and payback time (PBT) are presented in Figs. 24, 25, 26 and 27, respectively. The economic performance is most sensitive to PEC and for every 10% by which the PEC changes from the nominal value, the $LCOE$ and PBT change by about 8.7% and 11.0%, respectively. This is because the SPT plant construction is a capital-intensive project. The next most influential factor

699 in the SPT economic performance is i_d , which leads to a change in $LCOE$ of 4.3% and PBT of 2.9%
 700 for a 10% change in its value. For every 10% change in U_{HE} , the $LCOE$ and PBT change by about
 701 1.5%. Finally, i_f has little impact on the economic performance of SPT plants. The sensitivity
 702 analysis results indicate that the accuracy of equipment cost correlations should be improved when
 703 the central tower solar thermal technology and SCO_2 power system become mature in the future.
 704 Also, stakeholders should give careful consideration to the interest charged to the loan when
 705 evaluating the economic performance of SPT plants.

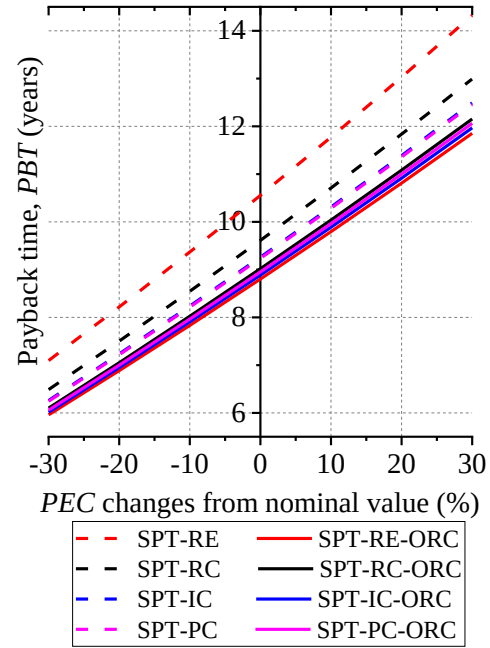
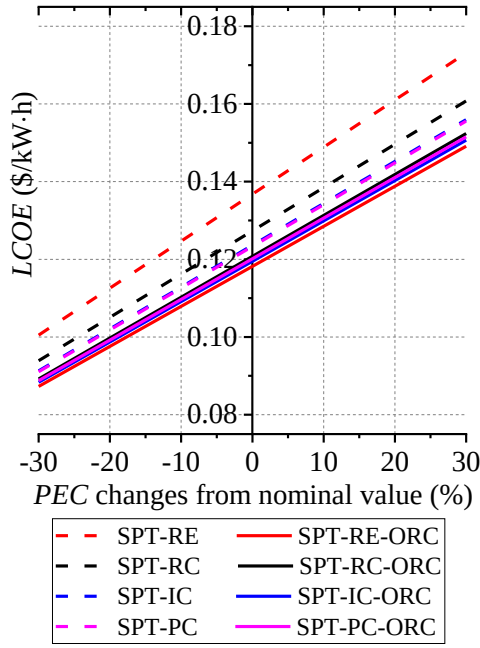
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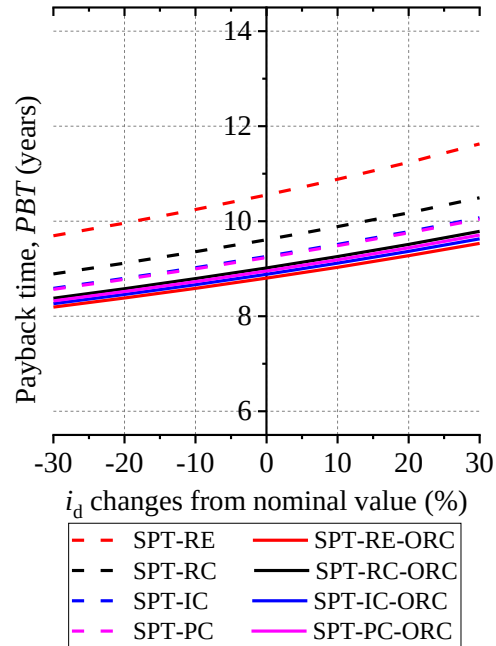
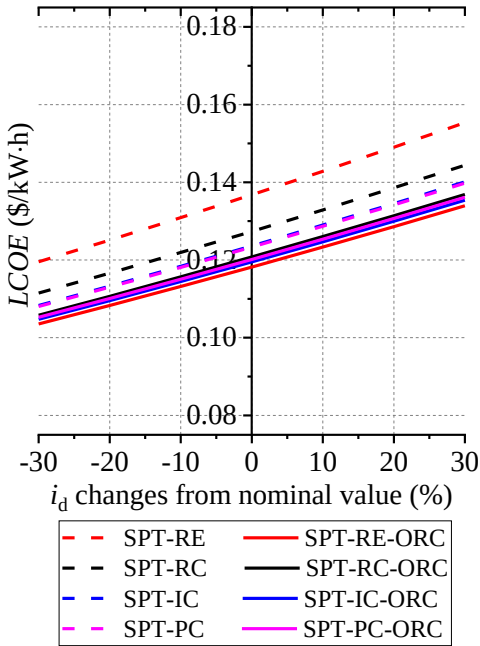
709 **Fig. 24.** Sensitivity analysis with the overall heat transfer coefficient of heat exchangers (U_{HE}).
 710 Analysis results of (a) levelised cost of electricity, $LCOE$, and (b) payback time, PBT for different
 711 SPT- SCO_2 systems with and without ORC subsystem, using the weather data in Delingha, China.



(a)

(b)

Fig. 25. Sensitivity analysis with the total purchase equipment cost (PEC). Analysis results of (a) levelised cost of electricity, $LCOE$, and (b) payback time, PBT for different SPT- SCO_2 systems with and without ORC subsystem, using the weather data in Delingha, China.



(a)

(b)

Fig. 26. Sensitivity analysis with the discount rate (i_d). Analysis results of (a) levelised cost of electricity, $LCOE$, and (b) payback time, PBT for different SPT- SCO_2 systems with and without ORC subsystem, using the weather data in Delingha, China.

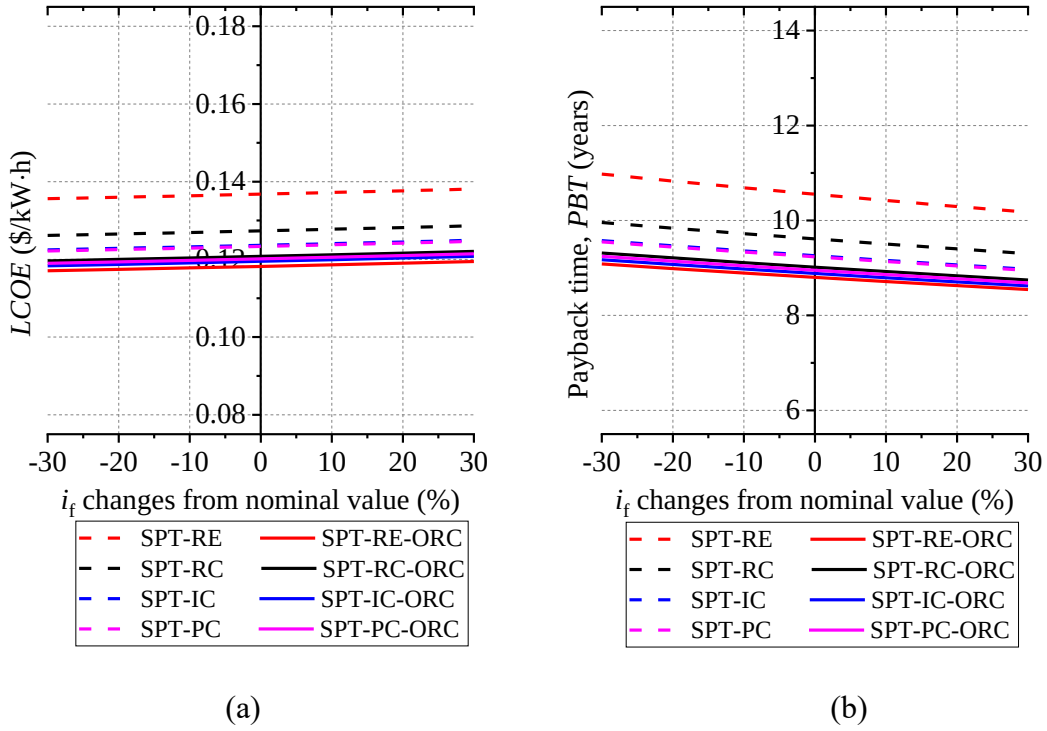


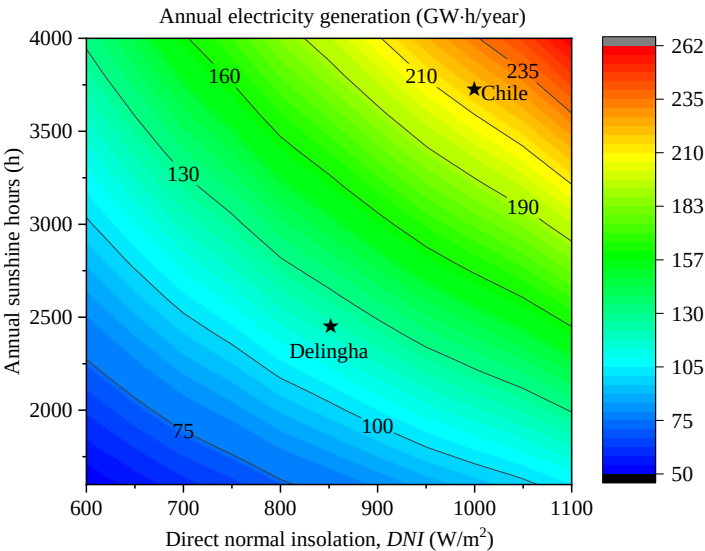
Fig. 27. Sensitivity analysis with the inflation rate (i_f). Analysis results of (a) levelised cost of electricity, $LCOE$, and (b) payback time, PBT for different SPT-SCO₂ systems with and without ORC subsystem, using the weather data in Delingha, China.

6. Generalised multi-climate analysis

The annual direct normal solar irradiance in Delingha as a case study above is around 2,080 kW·h/m²/year (~2450 sunshine hours, 850 W/m²/h), while it can reach 2900 kW/m²/year in Australia and even >3700 kW/m²/year in Chile [62]. In this section, the performance of SPT plants in multi-climate regions is focused, with DNI and annual sunshine hours taken as input parameters [64].

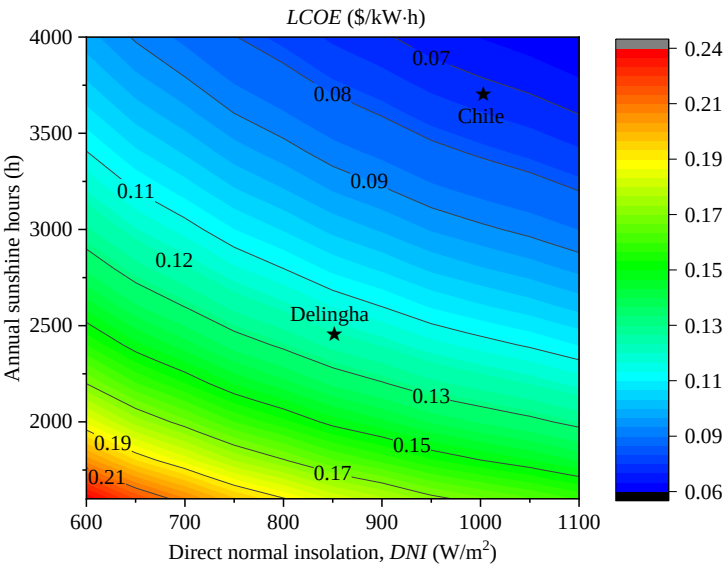
Of interest, for a fixed mirror aperture area of 306660 m², the SPT-RE-ORC system is always the optimal system with the lowest $LCOE$ for different solar conditions considered here, thereby the results of the SPT-RE-ORC system layout are presented. Fig. 22 shows the annual electricity generation increases with DNI and annual sunshine time. The SPT-RE-ORC plant can generate 261 GW·h of electricity annually when DNI and the annual sunshine time reach 1100 W/m² and 4000 hours, respectively. The variation of $LCOE$ in different solar conditions is shown in Fig. 23. Obviously, the plant has lower $LCOE$ in regions with high solar resources. With the increment of solar irradiance, the capital cost of the SPT plant can be reduced, especially for the solar field, due to the reduction of the number of heliostat mirrors. Moreover, longer sunshine hours enable the SPT plant to produce more electricity. For instance, the SPT plant located in Delingha has an $LCOE$ of 0.12 \$/kW·h, while the $LCOE$ of the SPT plant in Chile (3700 sunshine hours, 1000 W/(m²·h)) can

744 be as low as 0.07 $\$/\text{kW}\cdot\text{h}$. As shown in Fig. 24, the payback time of the SPT plant in Chile is less
 745 than six years. It is noted that an SPT plant with a RE-ORC power system can pay back in <10
 746 years when the annual direct normal solar irradiance exceeds 2000 $\text{kW}\cdot\text{h}/\text{m}^2/\text{year}$, which reveals
 747 that the SPT-RE-ORC system is attractive for many regions in the world.



748

749 **Fig. 28.** Annual electricity generation of SPT plants based on RE-ORC power systems, with a fixed
 750 mirror aperture area of 306660 m^2 which is the same as the GEMASOLAR power plant. The annual
 751 sunshine hour ranges from 1600 to 4000 h, and the DNI (at the summer solstice) ranges from 600 to
 752 1400 $\text{W}/(\text{m}^2\cdot\text{h})$, representing solar sources at most locations with high levels of DNI around the world.



753

754 **Fig. 29.** LCOE of 20 MWe SPT plants based on RE-ORC power systems, with a fixed mirror aperture
 755 area of 306660 m^2 which is the same as the GEMASOLAR power plant. The annual sunshine hour ranges
 756 from 1600 to 4000 h, and the design DNI (at the summer solstice) ranges from 600 to 1400 $\text{W}/(\text{m}^2\cdot\text{h})$,
 757 representing solar sources at most locations with high levels of DNI around the world.

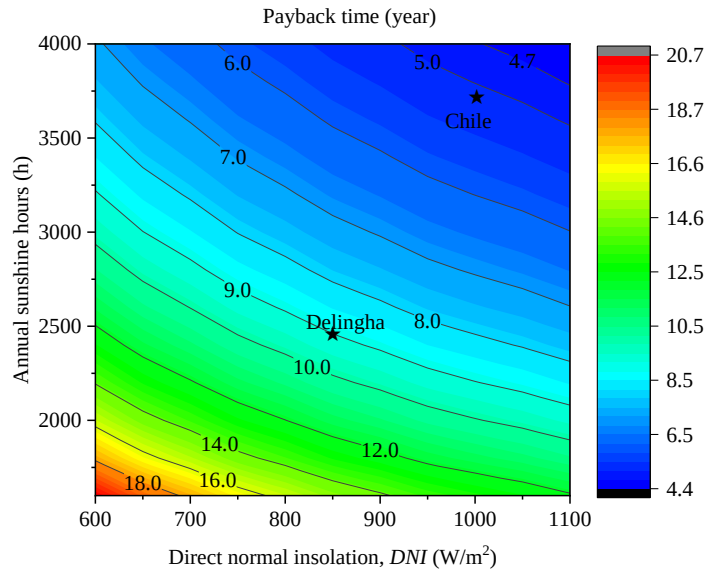


Fig. 30. Payback time of 20 MWe SPT plants based on RE-ORC power systems, with a fixed mirror aperture area of 306660 m² which is the same as the GEMASOLAR power plant. The annual sunshine hour ranges from 1600 to 4000 h, and the design *DNI* (at the summer solstice) ranges from 600 to 1400 W/(m²·h), representing solar sources at most locations with a high level of *DNI* around the world.

7. Conclusions

In the present study, bottoming ORC systems are coupled with different SCO₂ systems in order to assess the potential for further performance improvement in SPT plants. Four configurations of such SCO₂-ORC systems are presented: (1) recuperated SCO₂-ORC system (RE-ORC); (2) recompression SCO₂-ORC (RC-ORC); (3) intercooling SCO₂-ORC (IC-ORC); and (4) partial-cooling SCO₂-ORC (PC-ORC). A systematic comparison of these SPT-SCO₂-ORC system variants is conducted based on an integrative model, which includes the heliostat field, the solar receiver, the molten salt thermal energy storage system (TES), and the power cycle system.

Parameter sensitivity analysis is first conducted to evaluate the impact of power block characteristics and TES operating temperatures on the thermodynamic and economic performance of SPT plants. Gains in the power block efficiency contribute to the whole plant efficiency and reduce the heliostat number (i.e., solar field size) and heat storage capacity (i.e., storage system size), resulting in lower solar field and TES subsystem investment costs. On the other hand, various power cycle systems present different salt temperature differences (ΔT_{salt}) across the solar receiver. Larger ΔT_{salt} contribute to obtaining smaller salt storage tank sizes and reduced TES costs, which benefit the incorporation of power cycle systems with TES. Further work on novel SCO₂ power systems in SPT applications should aim to obtain higher cycle efficiency and larger ΔT_{salt} .

781 An annual system performance evaluation case study is undertaken based on the weather data in
782 Delingha, China. Levelised costs of electricity (*LCOE*) and payback time for different integrated
783 systems are calculated. The results show that adding a bottoming ORC system to SCO_2 systems
784 improves system thermal efficiency and reduces the cost of power blocks for SPT applications. Since
785 the RE-ORC system has a larger ΔT_{salt} and reduced flow rates of solar salt and SCO_2 , it achieves the
786 lowest *LCOE* and payback time. Specifically, the SPT-RE-ORC plant exhibits the greatest annual
787 electricity generation improvement (19%), and the most significant decrease in the *LCOE* (14%) and
788 payback time (20%) thanks to the largest amount of waste heat recovery from the SCO_2 cooling
789 process. In addition, this system has the simplest layout and highest technology readiness level (TRL)
790 among the four SPT- SCO_2 -ORC systems presented in this work. Therefore, the integration of ORC
791 bottoming cycles is recommended for solar power utilisation in these combined systems.

792 Finally, a generalised multi-climate analysis, conducted to provide insight into the techno-
793 economic feasibility of SPT plants over a range of different solar irradiance conditions, indicates that
794 *LCOEs* as low as 0.07 \$/kW·h and payback times as short as 6 years can be achieved in some regions
795 with an abundant solar resource (annual sunshine hours >3700, design *DNI* >1000 W/(m²h)).

796 In future work, the cost correlations will be further improved when more cost data is available to
797 enhance the reliability and accuracy of economic evaluation. More working fluids will be considered to
798 explore the optimal working fluid selection along with the optimal design of the ORC systems. Moreover,
799 a comparison with steam Rankine systems, condensation-based systems (e.g., with CO_2 mixtures as
800 working fluids for the topping CO_2 cycle systems) as well as other bottoming cycle system options will
801 be performed to address the potential of using the presented systems in practical applications.

802

803 Acknowledgement

804 This work was supported by the National Natural Science Foundation of China (Grant No. 51976145),
805 the UK Engineering and Physical Sciences Research Council (EPSRC) [grant numbers
806 EP/R045518/1 and EP/S032622/1], and the China Scholarship Council (CSC) for a Joint PhD
807 scholarship [grant number 202106280141] that supported Gang Fan's visit at Imperial College
808 London. Data supporting this publication can be obtained on request from cep-lab@imperial.ac.uk.
809 For the purpose of Open Access, the authors have applied a CC BY public copyright license to any
810 Author Accepted Manuscript version arising from this submission.

811

812 Appendix

813 The combined recompression SCO₂-ORC (RC-ORC) system is selected to describe the detailed
814 modelling process of the power blocks.

815 (1) Turbine modelling

816 Based on the energy balance equation, the output power of the SCO₂ turbine (ST) can be
817 calculated as:

$$818 \quad W_{\text{tur,SCO}_2} = m_4 \cdot (h_4 - h_5) \quad (\text{A1})$$

819 where m_4 is the turbine mass flow rate. h_4 and h_5 are the CO₂ enthalpies at the inlet and outlet of ST,
820 respectively.

821 The efficiency of ST is defined as:

$$822 \quad \eta_{\text{tur,SCO}_2} = (h_4 - h_5) / (h_4 - h_{5s}) \quad (\text{A2})$$

823 where h_{5s} is the CO₂ enthalpy at the ST outlet for an isentropic expansion process.

824 The output power and efficiency of the turbine in the bottoming ORC system can also be
825 calculated as follows:

$$826 \quad W_{\text{tur,ORC}} = m_{04} \cdot (h_{04} - h_{05}) \quad (\text{A3})$$

$$827 \quad \eta_{\text{tur,ORC}} = (h_{04} - h_{05}) / (h_{04} - h_{05s}) \quad (\text{A4})$$

828 (2) Compressor modelling

829 The isentropic efficiencies and power consumptions of the SCO₂ compressors are expressed as
830 follows:

$$831 \quad \eta_{\text{MC}} = (h_{2s} - h_1) / (h_2 - h_1) \quad (\text{A5})$$

$$832 \quad W_{\text{MC}} = m_1 \cdot (h_2 - h_1) \quad (\text{A6})$$

$$833 \quad \eta_{\text{RC}} = (h_{3s} - h_8) / (h_3 - h_8) \quad (\text{A7})$$

$$834 \quad W_{\text{RC}} = m_{8b} \cdot (h_{3b} - h_{8b}) \quad (\text{A8})$$

835 where subscripts “MC” and “RC” represent the main compressor and recompression compressor in
836 the SCO₂ system, respectively.

837 (3) Pump modelling

838 The isentropic efficiency and output power of the ORC pump can be calculated as follows:

$$\eta_{\text{pump}} = (h_{02s} - h_{01}) / (h_{02} - h_{01}) \quad (\text{A9})$$

$$W_{\text{pump}} = m_{01} \cdot (h_{02} - h_{01}) \quad (\text{A10})$$

where h_{02s} is the r245fa enthalpy at the pump outlet for an isentropic expansion process. h_{01s} is the r245fa enthalpy at the pump inlet.

(4) Heat exchanger modelling

In the heater, the hot salt transfers the heat to the CO₂ working fluid of the power block. The energy balance equation for the heater can be expressed as:

$$m_{\text{salt,b}} \cdot (T_{\text{salt,hot}} - T_{\text{salt,cold}}) = m_4 \cdot (h_5 - h_4) \quad (\text{A11})$$

where $m_{\text{salt,b}}$ denotes the mass flow rate of salt entering the heater, $T_{\text{salt,in}}$ and $T_{\text{salt,out}}$ are salt inlet and outlet temperatures, respectively.

There are two recuperators in the recompression SCO₂ cycle: high-temperature recuperator (HTR) and low-temperature recuperator (LTR), the energy conservations for them are given by,

$$h_6 - h_7 = h_4 - h_3 \quad (\text{A12})$$

$$h_7 - h_8 = (1 - x)(h_{3a} - h_2) \quad (\text{A13})$$

The effectiveness of recuperators is defined as:

$$\varepsilon = (T_{\text{lp,in}} - T_{\text{lp,out}}) / (T_{\text{lp,in}} - T_{\text{hp,in}}) \quad (\text{A13})$$

where $T_{\text{lp,in}}$, $T_{\text{lp,out}}$ are the inlet and outlet temperatures of the low-pressure CO₂ working fluid. $T_{\text{hp,in}}$ denotes the inlet temperature of the high-pressure CO₂ working fluid in the recuperator.

The topping SCO₂ cycle and the bottoming ORC cycle are coupled by a heat exchanger called an “evaporator” in this work. The R245fa working fluid absorbs the waste heat from the CO₂ stream and it turns into vapour. The energy conservation equation of this process is expressed as:

$$m_9 \cdot (h_{8a} - h_9) = m_{02} \cdot (h_{03} - h_{02}) \quad (\text{A15})$$

The SCO₂-air cooler works as the heat sink in the SCO₂ cycle. The energy balance equation for the cooler is given by:

$$m_1 \cdot (h_9 - h_1) = m_{10} \cdot (h_{11} - h_{10}) \quad (\text{A16})$$

The power consumption of the cooling fan is calculated as:

$$W_{\text{fan,cooler}} = 0.5 \cdot m_{10} \cdot \Delta P_{\text{air}} \cdot (\rho_{10} - \rho_{11}) / \eta_{\text{fan}} \quad (\text{A17})$$

where m_{10} denotes the mass flow rate of the air in the cooler, ΔP_{air} represents the pressure drop of air

867 across the cooler. ρ_{10} and ρ_{11} are the densities at the cooler inlet and outlet. η_{fan} is the fan efficiency.

868 The exhausted R245fa vapour changes into liquid at the condenser, and the energy conservation
869 equation of the condensation process and the power consumption of the fan are expressed as:

870
$$m_{01} \cdot (h_{04} - h_{01}) = m_{05} \cdot (h_{06} - h_{05}) \quad (\text{A18})$$

871
$$W_{\text{fan,cond}} = 0.5 \cdot m_{05} \cdot \Delta P_{\text{air}} \cdot (\rho_{05} - \rho_{06}) / \eta_{\text{fan}} \quad (\text{A19})$$

872

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