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Abstract: This paper develops a combined cooling and power system, which consists of a carbon dioxide Brayton cycle, a dual-pressure organic Rankine cycle and an ejector refrigeration cycle, to recover waste heat from exhaust gas and jacket water in internal combustion engines. Thermodynamic models of the system are performed and exergoeconomic methods are used to calculate the levelized exergy cost of the component products. Effects of seven parameters, including Brayton cycle turbine inlet temperature and inlet pressure, organic Rankine cycle turbine high-pressure side and low-pressure side inlet temperature and ejector primary inlet pressure, are evaluated. Single-objective optimization is carried out by means of genetic algorithm to obtain the minimum levelized exergy cost of system product. Results show that the increase of pressure at Brayton cycle turbine inlet and high-pressure and low-pressure side of the organic Rankine cycle turbine inlet contributes to the decrease of levelized exergy cost of the system product. Optimization results show that minimum levelized exergy cost for system product is 53.25 \$ (MWh)⁻¹. When system product levelized exergy cost is minimum, system net power output, cooling capacity and exergy efficiency are 374.37 kW, 188.63 kW and 37.31%, respectively.

Dear Editor:

We are sending a manuscript entitled “Performance analysis and optimization of a combined cooling and power system using low boiling point working fluid driven by engine waste heat”, which we should like to submit for publication in Energy Conversion and Management. We investigate a combined cooling and power system driven by exhaust gas and jacket water from an internal combustion engine. The mathematical model of the system is established to simulate the cycles under steady-state conditions. A parametric analysis of seven key parameters is conducted to examine their effects on the thermodynamic and exergoeconomic performance of the system. An optimization is conducted by genetic algorithm to obtain better system performance.

We declare that the manuscript has not been previously published, is not currently submitted for review to any other journal and will not be submitted elsewhere before one decision is made. Its publication is approved by all authors. If accepted, it will not be published elsewhere in the same form, in English or in any other language.

We appreciate your consideration of our manuscript, and we look forward to receiving comments from the reviewers.

Sincerely,

Jiangfeng Wang (on behalf of the authors' team)

Institute of Turbomachinery

Cover letter

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Highlights

A combined cooling and power system driven by engine exhaust gas and jacket water is proposed.

Thermodynamic and exergoeconomic performance of the system are analyzed.

Optimization for the combined cooling and power system is conducted by genetic algorithm.

1 **Performance analysis and optimization of a combined**
2 **cooling and power system using low boiling point working**
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**Performance analysis and optimization of a combined
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Abstract

This paper develops a combined cooling and power system, which consists of a
carbon dioxide Brayton cycle, a dual-pressure organic Rankine cycle and an ejector
refrigeration cycle, to recover waste heat from exhaust gas and jacket water in internal
combustion engines. Thermodynamic models of the system are performed and
exergoeconomic methods are used to calculate the levelized exergy cost of the
component products. Effects of seven parameters, including temperature and pressure
at the Brayton cycle turbine inlet, temperature and pressure at the high-pressure and
low-pressure side of the organic Rankine cycle turbine inlet and pressure at the ejector
primary inlet, are evaluated. Single-objective optimization is carried out by means of
genetic algorithm to obtain the minimum levelized exergy cost of system product.
Results show that the increase of pressure at Brayton cycle turbine inlet and
high-pressure and low-pressure side of the organic Rankine cycle turbine inlet

38 contributes to the decrease of levelized exergy cost of the system product.
39 Optimization results show that minimum levelized exergy cost for system product is
40 53.25 \$ (MWh)⁻¹. When system product levelized exergy cost is minimum, system net
41 power output, cooling capacity and exergy efficiency are 374.37 kW, 188.63 kW and
42 37.31%, respectively.

43 **Nomenclature**

<i>Latin symbols</i>		ρ	density, kg m ⁻³
A	area, m ²	μ	dynamic viscosity, m ² s ⁻¹
B_o	boiling number	η	efficiency, %
c	levelized average cost, \$ (MWh) ⁻¹	δ	thickness, m
c_p	specific heat, kJ kg ⁻¹ K ⁻¹	<i>Subscripts</i>	
C	cost rate, \$ year ⁻¹	1-31	state points
D	diameter, m	g1-g3	state points
e	exergy, kJ kg ⁻¹	w1-w3	state points
E	exergy flow rate, kJ s ⁻¹	Bt	Brayton cycle turbine
E_y	exergy flow rate per year, kJ year ⁻¹	BM	bare module
F	multiplying factor	cond	condenser
f	friction factor	comp	compressor
G	mass flow rate, kg s ⁻¹	D	destruction
h	enthalpy, kJ kg ⁻¹	elec	electricity
H	depth, m	es	equivalent diameter
i_{eff}	interest rate	ev	evaporation/evaporator

l	length, m	ex	exergy
M	mass flow rate, kg s^{-1}	F	fuel
n	lifetime, year	g	exhaust gas
Nu	Nusselt number	gh	gas heater
P	pressure, MPa	he	heat exchanger
Pr	Prandtl number	L	loss
Pt	center distance between tubes, m	l	liquid
P_r	reduced pressure	M	material
Q	heat transfer rate, kW	Ot	ORC turbine
Q_{cool}	cooling capacity, kW	P	product
q_m	average imposed wall heat flux, W m^{-2}	p1	pump 1
r_f	enthalpy of vaporization, kJ kg^{-1}	p2	pump 2
T	temperature, K	p3	pump 3
U	overall heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$	p4	pump 4
W	power, kW	pf	primary flow
W_y	annually power, MWh year^{-1}	prec	precooler
x	vapor quality	preh	preheater
Z	annually levelized cost value, $\text{\$ year}^{-1}$	s	shell
z	capital cost, k\\$	t	tube
<i>Acronyms</i>		th	thermal
BC	Brayton cycle	turb	turbine
CBC	CO_2 Brayton cycle	vg	vapor generator

CCP	combined cooling and power	w	tube wall
CRF	capital recovery factor		
CEPCI	chemical engineering plant cost index		
DORC	dual-pressure organic Rankine cycle		
ERC	ejector refrigeration cycle		
GA	genetic algorithm		
TEG	thermoelectric generator		
AARC	ammonia absorption refrigeration cycle		
<i>Greek symbols</i>			
α	convection heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$		
λ	heat conductivity, $\text{W m}^{-1} \text{K}^{-1}$		

44 1. Introduction

45 Nowadays, internal combustion engine (ICE) is the major motive power source in
46 energy field, which are widely used in transport, construction, agriculture, etc. Over
47 50 % of the total transportation fuel is consumed by ICEs [1]. However, only 30-45%
48 of the fuel energy is converted into effective power output, while the remaining
49 energy is discharged to the environment via exhaust gas, jacket water and charge air,
50 causing a large amount of waste fuel energy [2]. Thus, technology for waste heat
51 recovery from ICEs has drawn much interest of researchers in the last decade. Much
52 effort has been devoted to the study of organic Rankine cycle (ORC) based ICE waste
53 heat recovery system for its advantages of high efficiency and simple structure [3].

There are two important pathways that will lead to the improvement of the ORC system for ICE waste heat recovery. One will be selecting organic working fluids which are suitable for the system under certain conditions. Another is to optimize the system configuration to make full use of the waste heat.

The work of selecting suitable organic working fluids for ORC was carried out by many researchers to improve the efficiency of the ICE waste heat recovery. Tian et al. [4] evaluated the performance of 20 different working fluids in an ORC system for ICE waste heat recovery. Rijpkema et al. [5] compared the performance of twelve working fluids in an ORC-based ICE waste heat recovery system to find the suitable candidate. Su et al. [6] developed a theoretical efficiency model about working fluids selecting for ORC-based ICE waste heat recovery system via strict mathematical derivation.

Configuration optimization in ORC-based ICE waste heat recovery system mainly focuses on reducing the system irreversible rate to fully utilize the engine waste heat. Vaja and Gambarotta [7] added a preheater and a recuperator separately to a simple ORC system to improve the performance for the ICE waste heat recovery. Kim et al. [8] proposed a novel single-loop ORC system to recovery engine waste heat. They employed two recuperators in series to heat the working fluid. Comparison showed that the net power output of the system was 35.6% more than simple ORC system. Because that the maximum power output of single-loop ORC is lower than that of the dual-loop ORC system [9], more attention has been focused on dual-loop ORC based ICE waste heat recovery system in recent years. Wang et al. [10] modeled a dual-loop

ORC system for engine waste heat recovery. The high-temperature loop absorbed heat from exhaust gas and its residual heat acted as heat source for the low-temperature loop. Wang et al. [11] investigated a dual-loop ORC system for ICE waste heat recovery. The high-temperature loop absorbed heat from exhaust gas for the first time. Then the low-temperature loop absorbed heat from the residual heat of the exhaust gas to realize the cascading utilization of the waste heat. Huang et al. [12] proposed a complex dual-loop ORC system for engine waste heat recovery. The high-temperature loop absorbed heat from the exhaust gas and residual heat from both the exhaust gas and the high-temperature loop provided heat for the low-temperature loop.

When referring to heat transfer in the high-temperature loop, thermal stability of organic working fluid is necessary to be considered. In previous studies, refrigerants were most selected as working fluids. The decomposition temperatures of refrigerants are relatively low (200-300 °C) [13], while the temperature of exhaust gas is above 450 °C [14]. Direct heat transfer between high-temperature exhaust gas and refrigerant caused the risk of working fluid decomposition. Though high decomposition temperature working fluids such as siloxanes and alkanes were adopted by some researchers, their flammability hindered their further applications [15]. Though placing a heat transfer oil intermediate loop between the exhaust gas and the ORC system could address this issue [16], it would cause a large amount of the high-temperature waste heat unharnessed. Therefore, some other high-temperature loops for waste heat recovery were employed by researchers to couple with the ORC. Miller et al. [17] introduced thermoelectric generator (TEG) technology.

High-temperature exhaust gas was first exploited by the TEG, then the cooled exhaust gas could drive the ORC safely. But the energy conversion capacity of TEG is low because of the material limitation. Steam Rankine for its high efficiency and stable operation attracted much attention of researchers. Shu et al. [18] placed a steam Rankine cycle between the ORC and the exhaust gas. Yu et al. [19] coupled a steam Rankine cycle with an ORC for the ICE waste heat recovery. However, the large bulk of the components in steam Rankine cycle limits further applications (such as application in vehicles) [20]. Considering the requirement of high thermal efficiency and compact configuration, Brayton cycle could be a compromise solution. Brayton cycle with CO₂ (carbon dioxide) as working fluid has the advantage of low environmental impact and good thermodynamic performance [21]. Few studies about ORC system coupled with CO₂ Brayton cycle (CBC) for ICE waste heat recovery have been published. Though Zhang et al. [20] carried out some relevant studies, their attention was focused on comparing the performance of CBC, TEG and steam Rankine cycle when coupled with the same bottom ORC. Detailed analysis of the CBC was not given and the energy in jacket water was not harnessed.

Jacket water, though containing large amounts of energy [22], obtained little attention in the previous studies. For its relatively low temperature, jacket water was mainly used to preheat the organic working fluid in the ORC system. In the ORC-based ICE waste heat recovery system designed by Zhang et al. [23] jacket water was used to preheat the organic working fluid. Then the organic working fluid was heat by the high-temperature exhaust gas to vapor state and expanded in the ORC

turbine. In Yang's [24] ICE waste heat recovery system, jacket water and secondary exhaust gas were used to preheat the organic working fluids in ORC. In the dual-loop ORC based ICE waste heat recovery system investigated by Song et al. [25], jacket water was used to preheat the low-temperature-loop. Yu et al. [26] calculated the energy recovery efficiency from an ORC-based ICE waste heat recovery system. 75% waste heat could be recovered from the exhaust gas, while only 9.5% waste heat was recovery from jacket water. The relatively low utilization rate of jacket water in the ORC system is caused by the mismatch of working fluid mass flow rate in the preheater and the evaporator. Thus, the utilization of energy in jacket water could be further explored.

Multigeneration system driven by waste heat has drawn increasing interest of researchers in light of the trend towards reducing emissions, increasing the efficiency of energy use and providing variable energy. Li et al. [27] modeled a combined cooling, heating and power system to highly utilize the waste heat. Yari et al. [28] proposed a waste heat recovery system to provide power, distilled water and heat. Bai et al. [29] investigated a cooling, heating and power system driven by exhaust gas to recovery the waste heat. Combined cooling and power (CCP) systems driven by ICE waste heat were also investigated by some researchers. Chen et al. [30] designed an ammonia-water combined cooling and power system using the waste heat from the ICEs. Ammonia-water was heated by exhaust gas and jacket water. One part of the ammonia-water vapor flew into the turbine to provide power and the other part flew into the evaporator to provide refrigeration. Salek et al. [31] coupled an ammonia

absorption refrigeration cycle and a bottoming Rankine cycle with internal combustion engine to produce power and cooling capacity.

Ammonia absorption refrigeration cycle (AARC) were widely used in the combined cooling and power system for its large refrigeration output. However, the complex cycle structure and high driven temperature requirement of AARC might sometimes limit its applications. On the contrary, ejector refrigeration cycle (ERC) exhibits the advantages of easy maintenance and high reliability [32] and it can be driven by low-temperature heat source such as the jacket water. Thus, ICE waste heat recovery system with ERC driven by jacket water not only simultaneously generate power and refrigeration but also fully utilized the jacket water waste heat.

In this study, a combined cooling and power system is developed, which comprises a CO₂ Brayton cycle, a dual-pressure organic Rankine (DORC) cycle and an ejector refrigeration cycle. The CO₂ Brayton cycle absorbs heat from the high-temperature exhaust gas directly to prevent the decomposition risk. The turbine exhaust in the CO₂ Brayton cycle and the engine exhaust gas after heat transfer are respectively regarded as the heat sources for the high-pressure side and low-pressure side of the dual-pressure ORC, realizing the cascading utilization of exhaust gas. Meanwhile, organic working fluids in high-pressure side and low-pressure side are both preheated by jacket water which increases the mass flow rate of the organic working fluid preheated by jacket water. What's more, the ejector refrigeration cycle is adopted to produce refrigeration and fully utilize waste heat in jacket water. Thermodynamic and exergoeconomic analysis is carried out to examine the effects of key parameters on

system performance. Then a system optimization is conducted to obtain the minimum levelized exergy cost for the system product by means of genetic algorithm (GA).

The innovative features of this paper are as follow:

- A CO₂ Brayton cycle is investigated to prevent the risk of decomposition of organic working fluid and provide power.
- A novel dual-pressure ORC system is developed to cascading utilize the waste heat in exhaust gas and jacket water and provide large amounts of power output.
- An ejector refrigeration cycle driven by jacket water is designed to provide refrigeration and fully utilize the jacket water waste heat.

2. System description

The combined cooling and power system is shown in Fig. 1. The system integrates a dual-pressure organic Rankine cycle with a CO₂ Brayton cycle and an ejector refrigeration, which can produce power and refrigeration simultaneously. High-temperature gas heat from the ICE enters the gas heater to provide heat for the CBC. In the CBC, compressor compresses the CO₂ to a supercritical state. The high-pressure CO₂ flows into the gas heater to absorb heat. Then CO₂ with high temperature and high pressure expands through the BC turbine to produce power.

After expanding in the BC turbine, the high-temperature exhaust CO₂ flows into vapor generator 2 to heat the organic working fluid. High-pressure side organic working fluid heated by the CO₂ then flows into the ORC turbine to produce power. Meanwhile low-pressure side organic working fluid absorbs heat from the secondary

185 engine exhaust gas in vapor generator 1 and then enters ORC turbine to produce
186 power.

187 Exhaust vapor from the ORC turbine is cooled by condenser 1 to liquid state and
188 pressured by pump 1. Jacket water with large mass flow rate is used to preheat the
189 organic working fluid in the preheater. The preheated organic working fluid then
190 separates. One part of the fluid is pumped by pump 3 to the vapor generator 1 to cycle
191 in the low-pressure side. The other part is pumped by pump 2 to the vapor generator 2
192 to cycle in the high-pressure side.

193 The jacket water then flows into vapor generator 3 to provide heat for the ejector
194 refrigeration cycle. After the condensation process in condenser 2, liquid working
195 fluid is divided into two separated parts. One part of the fluid is pumped to the vapor
196 generator 3 to absorb heat from jacket water and then becomes superheated vapor.
197 The other part of the working fluid flows through the throttle valve to become
198 low-pressure vapor-liquid mixture. The low-pressure mixture enters the evaporator to
199 produce cooling capacity when absorbing heat from the environment and become
200 low-pressure vapor. After that, the superheated vapor mixes with the low-pressure
201 vapor in the ejector. The mixed working fluid enters the condenser 2 to be condensed
202 to liquid.

203 R245fa is selected as the working fluid for the organic Rankine cycle and the
204 ejector refrigeration cycle because of the great thermodynamic performance [33] and
205 the low environment effects [34].

206 3. System model

207 Several assumptions are made to simplify the simulation of the system, which are: (1)
208 the system keeps a steady state; (2) the heat and friction in the system are not
209 considered; (3) the pressure losses in the vapor generators, preheater, evaporator,
210 condensers and pipes are neglected; (4) the gas temperature at the outlet of the vapor
211 generator 1 is higher than 110 °C [35], considering the low gas dew point temperature;
212 (5) the working fluids at the outlet of the condensers and the preheater are saturated
213 liquids, and the evaporator outlet state is saturated vapor; (6) the process through the
214 throttle valve is isenthalpic.

215 3.1. Energy model

216 The net power of the CO₂ Brayton cycle is expressed as:

$$217 \quad W_{BC} = W_{Bt} - W_{comp} \quad (1)$$

218 The net power of the DORC is given as:

$$219 \quad W_{ORC} = W_{Ot} - W_{p1} - W_{p2} - W_{p3} \quad (2)$$

220 The cooling capacity of the ERC is given as:

$$221 \quad Q_{cool} = M_{cool} \cdot (h_{21} - h_{20}) \quad (3)$$

222 The net power output of the whole system is calculated as:

$$223 \quad W_{net} = W_{ORC} + W_{BC} - W_{p4} \quad (4)$$

224 The thermal efficiency of the system is given as:

$$225 \quad \eta_{th} = \frac{W_{net} + Q_{cool}}{M_{g1} \cdot (h_{g1} - h_{g3}) + M_{w1} \cdot (h_{w1} - h_{w3})} \quad (5)$$

226 The detailed energy model equations of each component are listed in Table 1. Note

227 that there are two expanding processes in the ORC turbine. The high-pressure vapor
228 expands in the turbine and then mixes with the vapor from vapor generator 1. After
229 that, the mixed vapor expands in the turbine for the second time.

230 3.2.Exergy model

231 The energy model of the system is based on the first law of thermodynamics. From
232 the viewpoint of the first law, it is equivalent for work and heat. Nevertheless,
233 according to the second law of the thermodynamics, the irreversibility of work and
234 heat is different. The exergy is used to quantifies the difference between them. The
235 exergy model of the system is based on a dead state (the ambient condition in this
236 study). Definition of exergy is given as:

$$237 \quad e = (h - h_0) - T_0 \cdot (s - s_0) \quad (6)$$

238 where h_0 , T_0 and s_0 are the parameters under the ambient conditions.

239 The exergy flow rate in this study is given by:

$$240 \quad E = M \cdot e \quad (7)$$

241 In this study, all the components in the system are associated directly or indirectly
242 with fuel of other heat sources, such as exhaust as and jacket water. The heat sources
243 provide exergy for the components to operate. For each component, there is an exergy
244 balance equation, being expressed as [36]:

$$245 \quad E_F = E_p + E_D + E_L \quad (8)$$

246 where E_F , E_P , E_D , E_L donate the rate of exergy for the component fuel, the rate of
247 exergy for component product, the rate of component exergy destruction and the rate
248 of component exergy loss, respectively.

249 The details of the exergy balance equations for each component are listed in Table
250 1.

251 The exergy efficiency represents the degree of the utilization of the waste heat in
252 the system, being expressed as:

$$253 \quad \eta_{ex} = \frac{W_{net} + E_{cool}}{E_{g1} - E_{g3} + E_{w1} - E_{w3}} \quad (9)$$

254 where E_{cool} is the exergy rate of the cooling process, being expressed as:

$$255 \quad E_{cool} = E_{25} - E_{24} \quad (10)$$

256 3.3.Capital cost calculation

257 A method of modeling the capital costs of main components is used in this study.
258 According to Ref. [37], the bare module cost of the components is calculated as the
259 basic cost. The basic cost of the components includes the direct project cost (such as
260 component cost, material cost of the installation, etc.) and the indirect project cost
261 (like the taxes, insurance engineering expenses, etc.). The bare module cost of the
262 components is calculated under basic conditions. For deviations from the based
263 conditions, multiplying factors (the specific component type, the specific system
264 pressure and the specific material of construction) are added in the calculation to
265 correct the results. In the following text, equations from Eq. (11) to Eq. (21) are
266 proposed in Ref. [37].

267 Axial turbines (BC turbine and ORC turbine) are used in this study. The bare
268 module cost equation of the turbine is:

$$269 \log_{10} C_{\text{turb}}^0 = K_{1,\text{turb}} + K_{2,\text{turb}} \cdot \log_{10} W + K_{3,\text{turb}} \cdot (\log_{10} W)^2 \quad (11)$$

270 where $K_{i,\text{turb}}$ are constants corresponding to the turbine type; and W is the power
271 output of the turbine.

272 Turbines used in this study are made of carbon steel (CS) and operate under high
273 pressure. Thus, a multiplying factor is used to correct the result. The capital cost of
274 the turbine is given as:

$$275 C_{\text{turb}} = F_{\text{BM,turb}} \cdot C_{\text{turb}}^0 \quad (12)$$

276 where $F_{\text{BM,turb}}$ is the multiplying factor corresponding to the working conditions of the
277 turbine.

278 Reciprocating pumps are used in this study. The bare module cost equation of the
279 pumps is given as:

$$280 \log_{10} C_{\text{pump}}^0 = K_{1,\text{pump}} + K_{2,\text{pump}} \cdot \log_{10} W + K_{3,\text{pump}} \cdot (\log_{10} W)^2 \quad (13)$$

281 where $K_{i,\text{pump}}$ are the constants corresponding to the pump type; and W is the power
282 input of the pump.

283 Pumps used in this study are made of stainless steel (SS) and work under high
284 pressure. Thus, multiplying factors are used to correct the bare module cost. The
285 capital cost of the pump is given as:

$$286 C_{\text{pump}} = (B_{1,\text{pump}} + B_{2,\text{pump}} \cdot F_{\text{M,pump}} \cdot F_{\text{P,pump}}) \cdot C_{\text{pump}}^0 \quad (14)$$

where $B_{i, \text{pump}}$ are the constants corresponding to the type of the pump; $F_{M, \text{pump}}$ is the material factor of the pump and $F_{P, \text{pump}}$ is the pressure factor of the pump. The equation of the pressure factor is given as:

$$\log_{10} F_{P, \text{pump}} = C_{1, \text{pump}} + C_{2, \text{pump}} \cdot \log_{10} P_{\text{pump}} + C_{3, \text{pump}} \cdot (\log_{10} P_{\text{pump}})^2 \quad (15)$$

where $C_{i, \text{pump}}$ are the constants corresponding to the type of the pump; and P_{pump} is the pressure of the pump under working conditions.

Axial compressor is used in this study. The bare module cost equation of the compressor is given as:

$$\log_{10} C_{\text{comp}}^0 = K_{1, \text{comp}} + K_{2, \text{comp}} \cdot \log_{10} W + K_{3, \text{comp}} \cdot (\log_{10} W)^2 \quad (16)$$

where $K_{i, \text{comp}}$ are the constants corresponding to the type of the compressor; W is the power input of the compressor.

The compressor is made of carbon steel (CS) and works under high pressure.

Correction equation of the bare module cost is given as:

$$C_{\text{comp}} = F_{\text{BM, comp}} \cdot C_{\text{comp}}^0 \quad (17)$$

where $F_{\text{BM, comp}}$ is the constant corresponding to the type of the compressor.

Shell-and-tube heat exchangers (gas heater, vapor generators, precooler, preheater, evaporator and condensers) are used in this study. The bare module cost equation of the heat exchanger is given as:

$$\log_{10} C_{\text{he}}^0 = K_{1, \text{he}} + K_{2, \text{he}} \cdot \log_{10} A + K_{3, \text{he}} \cdot (\log_{10} A)^2 \quad (18)$$

where $K_{i, \text{he}}$ are the constants corresponding to the type of the heat exchanger; A is the heat transfer area of the heat exchanger. The calculation of the heat exchanger areas is presented in Appendix A.

Heat exchangers used in this study are made of carbon steel (CS) and work under different pressure. Multiplying factors are needed to correct the results, the equation is given as:

$$C_{he} = (B_{1,he} + B_{2,he} \cdot F_{M,he} \cdot F_{P,he}) \cdot C_{he}^0 \quad (19)$$

where $B_{i,he}$ are the constants corresponding to the type of the heat exchanger. $F_{M,he}$ and $F_{P,he}$ are the material factor and pressure factor, respectively. The pressure factor is obtained from the following equation:

$$\log_{10} F_{P,he} = C_{1,he} + C_{2,he} \cdot \log_{10} P_{he} + C_{3,he} \cdot (\log_{10} P_{he})^2 \quad (20)$$

where $C_{i,he}$ are the constants corresponding to the type of the heat exchanger; P_{he} is the designed working pressure for the heat exchanger.

The values of the constants mentioned above for the main components are listed in Appendix B.

The calculation of the bare module cost depends on past records or published correlations for price information. It is necessary to update the costs because of the inflation. This can be achieved by the following equation:

$$C_2 = C_1 \cdot \left(\frac{I_2}{I_1} \right)$$

(21)

where C is the purchased cost and I is the cost index. The subscript 1 refers to base time when cost is known and subscript 2 refers to time when cost is desired. The *CEPCI* (Chemical Engineering Plant Cost Index) is employed to calculate the inflation. The values of $CEPCI_{2016}$ and $CEPCI_{ref,2001}$ are 541.7 and 397, respectively [38].

331 3.4.Exergoeconomic model

332 Exergoeconomic is a branch of engineering which combines the thermodynamic
333 analysis and economic principles. Thermodynamic performance and economic cost of
334 the system are all taken into consideration.

335 To find the relationship between the present value of the expenditure and the
336 equivalent annually levelized costs, the capital recovery factor (CRF) is employed,
337 being expressed as [36]:

$$338 \quad Z_i = CRF \cdot C_i \quad (22)$$

$$339 \quad CRF = \frac{i_{\text{eff}} \cdot (1 + i_{\text{eff}})^n}{(1 + i_{\text{eff}})^n - 1} \quad (23)$$

340 where i_{eff} is the effective discount rate with a value of 0.05 [39]; and n is the lifetime
341 of the CCP system, being assumed as 30 years [40].

342 In order to calculate the equivalent annually levelized costs, the annual working
343 time of the system is assumed as 8000 h [41]. Then the annual exergy rates and annual
344 power output or consumption are obtained.

345 In a steady system, there are a number of entering and outing working fluid streams
346 and heat and work interactions with the surroundings. In exergoeconomic analysis,
347 each flowing stream is associated with a levelized exergy cost. The equations to
348 calculate the cost of the stream product are given as:

$$349 \quad C_{\text{in}} = c_{\text{in}} \cdot E_{\text{y,in}} \quad (24)$$

$$350 \quad C_{\text{out}} = c_{\text{out}} \cdot E_{\text{y,out}} \quad (25)$$

$$351 \quad C_{\text{work}} = c_{\text{work}} \cdot W_y \quad (26)$$

$$C_{\text{heat}} = c_{\text{heat}} \cdot E_{y,\text{heat}} \quad (27)$$

where c denotes levelized exergy cost of the streams; $E_{y,\text{in}}$ and $E_{y,\text{out}}$ are the exergy transfer rate of the stream flowing in and out of a component; W_y and $E_{y,\text{heat}}$ are the power and the heat transfer rate of the components considering the annual working time.

The cost balance equation applied to the k th system component is given as:

$$\sum_{\text{out}} C_{\text{out},k} + C_{w,k} = C_{\text{heat},k} + \sum_{\text{in}} C_{\text{in},k} + Z_k \quad (28)$$

Details of the cost balance equation are listed in Table 2.

The levelized exergy cost for system product is chosen to indicate the exergoeconomic performance, being expressed as [42,43]:

$$c_{\text{product}} = c_{\text{capital}} + c_{\text{fuel}} \quad (29)$$

where c_{capital} is the capital-cost-related part of the levelized exergy cost for the system product, being expressed as:

$$c_{\text{capital}} = \frac{Z_{\text{total}}}{W_{\text{net}} + E_{\text{cool}}} \quad (30)$$

where c_{fuel} is the fuel-cost-related part of the levelized exergy cost for the system product, being expressed as:

$$c_{\text{fuel}} = \frac{c_{\text{Bt}} \cdot W_{y,\text{comp}} + c_{\text{Ot}} \cdot W_{y,\text{pump1}} + c_{\text{Ot}} \cdot W_{y,\text{pump2}} + c_{\text{Ot}} \cdot W_{y,\text{pump3}} + c_{\text{Ot}} \cdot W_{y,\text{pump4}}}{W_{\text{net}} + E_{\text{cool}}} \quad (31)$$

where c_{Bt} and c_{Ot} are the levelized exergy cost for the BC turbine power output and the ORC turbine power output, which are calculated in Table 3. Likewise, they can be expressed as the capital-cost-related part and the fuel-cost-related part, given by Eq. (32) and Eq. (33).

$$c_{Bt} = \frac{Z_{Bt}}{W_{y,Bt}} + \frac{c_3 \cdot (E_{y,3} - E_{y,4})}{W_{y,Bt}} \quad (32)$$

$$c_{Ot} = \frac{Z_{Ot}}{W_{y,Ot}} + \frac{c_{10} \cdot (E_{y,10} + E_{y,11} - E_{y,12})}{W_{y,Ot}} \quad (33)$$

In addition, the levelized exergy cost for the condensers and the precooler is equal to zero, being given by:

$$c_{26}=c_{28}=c_{30}=0 \quad (34)$$

The levelized exergy cost for the exhaust gas as well as the jacket water is zero, being expressed as:

$$c_{g1}=c_{w1}=0 \quad (35)$$

3.5.Internal combustion engine

In this study, the engine selected [7] is a 12-cylinder 4-stroke supercharged engine. The main designed parameters of the engine are listed in Table 3. The composition of the engine exhaust gas is presented in Table 4. The thermal load of the engine exhaust gas is about 1700 kW and 1000 kW can be obtained from the engine jacket water.

3.6.Simulation conditions for the system

The thermodynamic parameters of the working fluid are calculated under the environment of MATLAB with the help of REFPROP 9.1 [44]. The basic conditions of simulation for the CCP system are listed in Table 5.

Seven key parameters : BC turbine inlet temperature ($T_{Bt,in}$), BC turbine inlet pressure ($P_{Bt,in}$), inlet temperature at the high-pressure side of ORC turbine ($T_{Ot,in,h}$), inlet pressure at the high-pressure side of ORC turbine ($P_{Ot,in,h}$), inlet temperature at

the low-pressure side of ORC turbine ($T_{Ot, in, l}$), inlet pressure at the low-pressure side of ORC turbine ($P_{Ot, in, l}$) and the ejector primary inlet pressure ($P_{ej, in}$), are chosen to analyze the thermodynamic and exergoeconomic performance of the system. When one parameter is investigated to analyze the system performance, other parameters are maintained constants based on the conditions in Table 5.

In the thermodynamic aspect, the net power output of the CO₂ Brayton cycle (W_{BC}), net power output of the DORC (W_{ORC}), net power of the whole system (W_{net}), cooling capacity of the system (Q_{cool}) and the exergy efficiency of the system (η_{exergy}) are selected to reflect the system performance. Levelized exergy cost for the BC turbine power output (c_{Bt}), levelized exergy cost for the ORC turbine power output (c_{Ot}), levelized exergy cost the system product ($c_{product}$) and the system capital cost ($z_{capital}$) are chosen to represent the exergoeconomic performance.

4. Results and discussion

The influence of the BC turbine inlet temperature ($T_{Bt, in}$) on the output and the exergy efficiency of the system are shown in Fig. 2. The net power output of the CBC increases with the rise of $T_{Bt, in}$. That can be explained by the large decrease of the compressor power consumption. With the increase of the CO₂ temperature at the BC turbine inlet, the mass flow rate of CO₂ decreases, leading to the decrease of the compressor power consumption. Although the drop of CO₂ mass flow rate would cut down the BC turbine power output, the decrease quantity of compressor power consumption is larger than the decrease of the BC turbine power output. Thus, the

large decrease of the compressor power consumption determines the increase trend of the CBC net power output.

It is presented that the net power output of the DORC increases with the rise of the BC turbine inlet temperature. Since the residual heat in exhaust CO_2 acts as the heat source for the high-pressure side of DORC, the temperature rise of the exhaust CO_2 , caused by the rise of $T_{\text{Bt,in}}$, would offer more heat for the bottom cycle, which causes the increase of the mass flow rate of the organic working fluid in the high-pressure side of DORC. Hence, the power output of the ORC turbine increases, leading to the increase of the net power output of the DORC.

With the increase of $T_{\text{Bt,in}}$, the cooling capacity of the ERC decreases, as shown in Fig. 2. The increase of the organic working fluid mass flow rate in DORC would absorb more heat from jacket water, resulting in the decrease of energy available for the ERC. As a result, less secondary flow working fluid from the evaporator is entrained to the ejector, resulting the decrease of the cooling capacity of the CCP system.

The increase of the CBC net power output and the DORC net power output account for the increase of the net power output of the whole CCP system. Though the cooling capacity of the ERC is large, it produces only a small amount of exergy. The decrease of the exergy output caused by the refrigeration capacity decrease can be made up by the increase of the power exergy output. Thus, the exergy efficiency of the system increases.

The influences of the BC turbine inlet temperature on the levelized exergy cost and

the system capital cost of the system are shown in Fig. 3. The levelized exergy cost for the BC turbine power output (c_{Bt}) drops with the rise of the BC turbine inlet temperature ($T_{Bt,in}$). That can be explained by the decrease of the capital-cost-related part of c_{Bt} . The capital-cost-related part of c_{Bt} decreases with the decrease of cost of compressor, which is cut down by the drop of the compressor power consumption. The increase of the ORC turbine power output causes the decrease of both the capital-cost related part and the fuel-cost-related part of c_{Ot} , resulting in the decrease of c_{Ot} .

The system capital cost ($z_{capital}$) rises with the rise of $T_{Bt,in}$. The large increase of the ORC turbine power output increases the cost of the ORC turbine. Moreover, the increase of the mass flow rate of the organic working fluid in the DORC causes the increase of cost for the vapor generator 2 and the preheater. Though the cost of compressor decreases, it can't change the ascending trend of the total system capital.

It can be obtained in Fig. 3 that the levelized exergy cost for the system product ($c_{product}$) decreases with the rise of $T_{Bt,in}$. The decline in levelized exergy cost for the BC turbine and ORC turbine power output, according to Eq. (31), would cause the decrease of the fuel-cost related part of $c_{product}$. Though the increase of $z_{capital}$ would cut down the capital-cost-related part of $c_{product}$, the impact of levelized exergy cost for the BC turbine and ORC turbine is greater, which leads to the descending trend of $c_{product}$.

The influence of the BC turbine inlet pressure ($P_{Bt,in}$) on the output and the exergy efficiency of the system are shown in Fig. 4. The net power output of the CBC increases with the increase of $P_{Bt,in}$, which can be explained by the rise of enthalpy

drop of the CO₂ in the BC turbine. Though the rise of $P_{Bt, in}$ requires more compressor power consumption, the increase of the BC turbine power output is larger in quantity than the consumption, which leads to the increase of the CBC net power output.

The net power output of the DORC decreases with the rise of $P_{Bt, in}$. On the one hand, the temperature of the exhaust CO₂ at the BC turbine outlet decreases with the increase of $P_{Bt, in}$. Thus, less heat is offered to the high-pressure cycle of DORC, resulting in the decrease of the high-pressure cycle power output. On the other hand, the increase of $P_{Bt, in}$ causes the increase of the compressor power consumption, which results in the rise of the CO₂ temperature at the compressor outlet. Thus, less heat is released in the gas heater and more heat is provided to the low-pressure cycle of DORC, which leads to the increase of the low-pressure cycle power output. However, the increase of the power output in low-pressure side is smaller than the decrease of the power output in the high-pressure side. Thus, the net power of the DORC decreases slightly.

The cooling capacity of the system increases with the increase of $P_{Bt, in}$. Just like the variation of the power output, the decrease of the mass flow rate in the high-pressure side of DORC is larger than the increase of mass flow rate in the low-pressure side. Therefore, the total mass flow rate in the DORC decreases, resulting in the reduction of heat provided for the ejector refrigeration cycle. Thus, the cooling capacity of the ERC decreases.

The net power output of the whole system increases with the increase of the BC turbine inlet pressure. Though the net power output of the DORC decreases, the

increase of CBC net power output is much larger. Thus, the net power output of the CCP system increases with the increase of $P_{Bt, in}$. The exergy efficiency of the system likewise has the same rising trend.

The influences of the BC turbine inlet pressure ($P_{Bt, in}$) on the levelized exergy cost and the system capital cost of the system are depicted in Fig. 5. The levelized exergy cost for the BC turbine output c_{Bt} increases with the rise of the $P_{Bt, in}$, which can be explained by the variations of the capital-cost-related part and the fuel-cost-related part. The increase of $P_{Bt, in}$ causes the increase of cost for both the BC turbine and the compressor, which lead to the rise of the two related parts.

The levelized exergy cost for the ORC turbine product (c_{Ot}) increases with the rise of $P_{Bt, in}$. The decrease of the mass flow rate in the DORC causes that less exergy is produced in vapor generator 2, causing the increase of the fuel-related part of c_{Ot} . Therefore, the levelized exergy cost for the ORC turbine (c_{Ot}) increases.

The system capital cost ($z_{capital}$) increases with the rise of ($P_{Bt, in}$). The increase of the mass flow rate in the ERC causes the rise of capital cost for the evaporator and vapor generator 3, which combined with the rise of the BC turbine cost and compressor cost accounts for the system capital cost rise.

The levelized exergy cost for the system product decreases with the rise of $P_{Bt, in}$ as presented in Fig. 5. According to Eq. (31), the rise of the c_{Ot} , c_{Bt} would cause the rise of the fuel-cost-related part of $c_{product}$. However, because of the large increase of the system net power output, the capital-cost-related part and the fuel-cost-related part decrease actually, which determines the decrease of $c_{product}$.

The influence of inlet temperature at the high-pressure side of ORC turbine ($T_{Ot, in, h}$) on the output and the exergy efficiency of the system are shown in Fig. 6. The net power output of the CBC remains unchanged since thermal parameters in dual-pressure ORC are irrelevant to the thermodynamic performance of the CBC.

The net power output of the DORC decreases with the increase of $T_{Ot, in, h}$. Though the increase of the vapor temperature could lead to the rise of the enthalpy drop in the ORC turbine, it would also cause the decrease of the mass flow rate in the high-pressure side, whose impact is greater than that of the enthalpy drop. Therefore, the power output of the DORC decreases.

The cooling capacity of the ejector refrigeration cycle increases with the rise of $T_{Ot, in, h}$. More heat is provided for the ERC because of the decrease of the mass flow rate in the DORC, leading to the increase of the mass flow rate in vapor generator 3. Thus, more secondary flow from the evaporator is entrained into the ejector, resulting in the increase of the cooling capacity.

The net power output of the CCP system decreases with the rise of $T_{Ot, in, h}$. The unchanged CBC power output and the drop of the DORC power output determine the decrease of the net power output of the CCP system. The exergy efficiency of the system as well drops with the increase of the increase of $T_{Ot, in, h}$.

The influences of inlet temperature at the high-pressure side of ORC turbine on the levelized exergy cost and the system capital cost of the system are presented in Fig. 7. The levelized exergy cost for the ORC turbine output (c_{Ot}) increases with the increase of $T_{Ot, in, h}$. The reason is that the two related parts of c_{Ot} increase with the drop of the

ORC turbine power output.

The levelized exergy cost for the BC turbine power output (c_{Bt}) increases with the rise the $T_{Ot, in, h}$. Since the decrease of the mass flow rate in the high-pressure side of DORC, the exergy generated in the vapor generator 2 decreases, causing the increase of the levelized exergy cost of the vapor. Thus, the increase levelized exergy cost of the vapor, which is heated by the BC turbine residual heat, causes the increase of the levelized exergy cost for the exhaust CO_2 . According to Eq. (32), the fuel-cost-related part of c_{Bt} increases, leading to the increase of c_{Bt} .

The system capital cost ($z_{capital}$) decreases with the increase of $T_{Ot, in, h}$. The decrease of the DORC power output causes the drop of the ORC turbine cost, which leads to the descending trend of $z_{capital}$.

The levelized exergy cost for the system product ($c_{product}$) increases with the rise of $T_{Ot, in, h}$, as shown in Fig. 7. The increase of the levelized exergy cost for the BC turbine and ORC turbine power output cause the rise of fuel-cost-related part of $c_{product}$. Meanwhile, the large decrease of the net power of the CCP system causes the increase of the capital-cost-related part. The two increase parts determine the rise of $c_{product}$.

The influences of the inlet pressure at the high-pressure side of ORC turbine ($P_{Ot, in, h}$) on the output and exergy efficiency of the system are presented in Fig. 8. The net power of the CBC keeps unchanged because of the unchanged thermal parameters in the cycle.

The net power output of the DORC increase with the rise of $P_{Ot, in, h}$. The increase of the evaporation pressure cuts down the latent heat of the organic working fluid,

which causes the increase of the mass flow rate in the high-pressure side of DORC.

As a result, the net power output of the ORC turbine increases, leading to the increase of the net power output of the DORC.

Considering the increase of the DORC net power output and the unchanged CBC net power output, the net power output of the whole system increases. Also, the exergy efficiency of the system increases.

The increase of the mass flow rate in the DORC absorbs more heat from the jacket water in the preheater. Thus, less heat is released in the vapor generator 3, causing the decrease of the mass flow rate of the working fluid in the ERC. As a result, the cooling capacity of the system decreases.

The influence of the inlet pressure at the high-pressure side of the ORC turbine ($P_{Ot, in, h}$) on the levelized exergy cost and system capital cost of the system are presented in Fig. 9. The large increase of the ORC turbine power output accounts for the decrease of the levelized exergy cost for the ORC turbine power output (c_{Ot}). The increase of the mass flow rate in the high-pressure side of DORC means that more exergy in the vapor is generated by the vapor generator 2, which leads to the decrease of its levelized exergy cost. Thus, the levelized exergy cost for the BC turbine exhaust CO_2 , which provides heat for the vapor, decreases with the vapor levelized exergy cost. Moreover, the drop of the CO_2 levelized exergy cost causes the decrease of the fuel-cost-related part of c_{Bt} , which further results in the decrease of c_{Bt} .

The increase of the ORC turbine power output and the increase of mass flow rate in the DORC cause the increase of cost for the turbine and the vapor generator 2, leading

to the rise of the system capital cost.

The levelized exergy cost for the system product (c_{product}) decreases with the increase of $P_{\text{Ot, in, h}}$. The decrease of c_{Ot} and c_{Bt} account for the decrease of the fuel-cost-related part of the levelized exergy for the system product. The impact of c_{Ot} and c_{Bt} is greater than that of the system capital cost whose rise would result in the increase of the capital-cost-related part of c_{product} . Thus, the levelized exergy cost of the system product (c_{product}) shows a descending trend.

The influences of the inlet temperature at the low-pressure side of ORC turbine ($T_{\text{Ot, in, l}}$) on the output and the exergy efficiency of the system are presented in Fig. 10. Parameters changes in the DORC are irrelevant to the thermodynamic performance of the CBC. Thus, the net power of the CBC remains unchanged.

The net power output of the DORC decreases with the increase of $T_{\text{Ot, in, l}}$. The increase of the inlet temperature causes the decrease of the mass flow rate in the low-pressure side of the DORC, leading to the decrease of the DORC net power output.

Considering the decrease of the DORC net power output and the unchanged CBC net power output, the net power output of the whole system decreases. Likewise, the exergy efficiency of the system decreases.

The cooling capacity of the ejector refrigeration cycle increases with the increase of $T_{\text{Ot, in, l}}$. The decrease of the mass flow rate in the low-pressure side means that more heat is offered to the ERC. Thus, the mass flow rate of the working fluid in the vapor generator 3 increases and more working fluid is entrained to the ejector from the

evaporator, which leads to the increase of the refrigeration cycle.

The influence of inlet temperature at the low-pressure side of the ORC turbine ($T_{Ot, in, 1}$) on the levelized exergy cost and system capital cost of the system are presented in Fig. 11. The levelized cost for the BC turbine power output increase with the increase of $T_{Ot, in, 1}$. The decrease of the mass flow rate in the vapor generator 1 leads to the drop of the vapor exergy output, which results in the increase of the levelized exergy cost for the vapor. The levelized exergy cost for vapor in vapor generator 2, which is the equal to that of the vapor in vapor generator 1, increases as a result, causing the increase of the levelized exergy cost of the exhaust CO_2 after the BC turbine. Thus, the fuel-cost-related part of c_{Bt} increases, resulting in the rise of c_{Bt} .

The levelized exergy cost for the ORC turbine (c_{Ot}) increases with the increase of $T_{Ot, in, 1}$. That can be explained by the increase of the levelized exergy cost of the ORC low-pressure inlet vapor and the decrease of the power output of the ORC turbine power output. Both the fuel-cost-related part and the capital-cost-related part of c_{Ot} increases.

The decrease of the mass flow rate and the ORC turbine power output cause the decrease of the vapor generator 1 cost and the turbine cost. Thus, the capital cost of the system decreases.

The levelized exergy cost for the system product increases with the increase of $T_{Ot, in, 1}$. The increase of c_{Bt} and c_{Ot} cause the increase of the fuel-cost-related part of the levelized exergy cost for the system product. Though, the decrease of the system capital cost causes the decrease of the capital-cost-related part, its effect is less

important. Thus, the increase of the fuel-cost-related part determines the increase of the levelized exergy cost for the system product.

The influences of the inlet temperature of the low-pressure side of the ORC turbine ($P_{Ot, in, 1}$) on the output of the exergy efficiency of the system are shown in Fig. 12. The net power of the CBC keeps unchanged with the increase of the low evaporation pressure. The reason is that the thermodynamic of the CBC is irrelevant to the thermal parameters in DORC.

The net power output of the DORC increases with the rise of $P_{Ot, in, 1}$. The increase of enthalpy drop of the organic working fluid in the low-pressure side, which is caused by the rise of $P_{Ot, in, 1}$, results in the increase of the power output of the low-pressure side. Though mass flow rate in the low-pressure side would decrease, its impact is less important than that of the enthalpy drop. Thus, the net power output of DORC increases.

The unchanged CBC power output and the increase of the DORC power accounts for the increase of the system net power output and exergy efficiency of the system.

The cooling capacity increases slightly with the increase of $P_{Ot, in, 1}$. Because of the decrease of the mass flow rate in DORC, less heat is released in the preheater and more heat is provided in vapor generator 3. Thus, the mass flow rate of the working fluid in the ERC increases, resulting in the slight increase of the cooling capacity.

The influences of inlet pressure at the low-pressure side of the ORC turbine ($P_{Ot, in, 1}$) on the levelized exergy cost and system capital cost of the system are shown in Fig. 13. The levelized exergy cost for the ORC turbine power output decreases with the

increase of $P_{Ot, in, 1}$. That can be explained by the decrease of the vapor generator 1 cost, caused by the decrease of the mass flow rate in DORC, and the increase of the DORC power output. Both the capital-cost-related part and the fuel-cost-related part of c_{Ot} decrease.

The levelized exergy cost for the BC turbine power output decreases with of $P_{Ot, in, 1}$.

The decrease of the c_{Ot} causes the drop of levelized exergy cost for the vapor in vapor generator 2, which is heated by the residual heat in the BC turbine exhaust CO_2 . Thus, the levelized exergy cost of the exhaust CO_2 decreases, which further leads to the drop of the fuel-cost-related part of c_{Bt} . Therefore, the levelized exergy cost for the BC turbine power output (c_{Bt}) decreases, as shown in Fig. 13.

The increase of the ORC turbine power output causes the increase of the ORC turbine cost. Meanwhile, the increase of the cooling capacity causes the increase of the heat transfer area in the evaporator which requires the rise of the evaporator cost. Thus, the capacity cost of the system increases.

The levelized exergy cost for the system product decreases with the increase of $P_{Ot, in, 1}$. The decrease of the levelized exergy cost for the BC turbine power output and ORC turbine power cause the decrease of the fuel-cost-related part of the system levelized exergy cost, which determined the decrease of the levelized exergy cost for the system product.

The influence of ejector primary inlet pressure ($P_{ej, in}$) on the output and the exergy efficiency of the system are shown in Fig. 14. Thermal parameter changes in the ERC

can't affect the thermodynamic performance of the CBC and DORC. Thus, the net power output of the two cycles remain unchanged. With the increase of the ejector primary inlet pressure, the power consumption of pump 4 increases, leading to the slight decrease of the power output of the whole system.

The increase of the ejector primary inlet pressure causes the increase of the entrainment ratio of the ejector. Thus, more secondary flow is entrained to the ejector from the evaporation, leading to the increase of the cooling capacity.

With the increase of the ejector primary inlet pressure, the power consumption of pump 4 increases gradually. At first, the exergy loss in pump 4 is smaller than the exergy produced by the cooling capacity. Then, with the increase of the pump power consumption, the exergy loss in pump 4 becomes larger than the cooling exergy increase. Thus, the exergy efficiency for the system increases at first and then decreases with the increase of the ejector primary inlet pressure.

The influence of the ejector primary inlet pressure on the levelized exergy cost and the system capital cost of the system are presented in Fig. 15. The increase of the ejector primary inlet pressure can't affect the power output of the BC turbine and the ORC turbine. Thus, the levelized exergy cost for the BC turbine and the ORC power output remain unchanged.

The increase of the pump power consumption results in the increase of the pump 4 cost. The increase of the mass flow rate in the evaporator causes the increase of the evaporator cost. Thus, the system capital cost increases, which leads to the increase of the capital-cost-related part of c_{product} . As a result, the levelized exergy cost for the

system increases.

The parametric analysis reveals the potential of optimization for the CCP system. With the increase of the BC turbine inlet temperature, the net power output of the system increases while the cooling capacity decreases. With the increase of the inlet temperature at the high-pressure side of ORC turbine, the net power output of the system decreases while the cooling capacity increases. In this study, seven key parameters (BC turbine inlet temperature, BC turbine inlet pressure, inlet temperature at the high-pressure side of ORC turbine, inlet pressure at the high-pressure side of ORC turbine, inlet temperature at the low-pressure side of ORC turbine, inlet pressure at the low-pressure side of ORC turbine and the ejector primary inlet pressure) are chosen as the variables to optimize the system. The ranges of these parameters are listed in Table 6.

Considering that the levelized exergy cost reflects the thermodynamic and the exergoeconomic aspect of the system, the levelized exergy cost for the system product is selected as the objective function and genetic algorithm is selected to conduct the single-objective optimization.

Genetic algorithm (GA) is an optimization method based on the natural biological evaluation. [45] It simulates the natural genetic rules and searches the optimization result in all the generation. The control parameters of the GA are listed in Table 7.

The optimization results of GA are listed in Table 8. It can be obtained that the minimum levelized exergy cost for the system product c_{product} is $53.25 \text{ } \$(\text{MWh})^{-1}$. The net power output, exergy efficiency of the CCP system are 374.37 kW, 37.31%

respectively. The inlet pressure at the high-pressure side of ORC turbine is 1.85 MPa. Meanwhile, it can be evidenced from Fig. 8 and 9 that the highest output power (about 374.37 kW), exergy efficiency (about 37.31%) and the lowest levelized exergy cost (about 53.25 \$(MWh)⁻¹) at the highest inlet pressure at the high-pressure side ORC turbine (about 1.85 MPa). The results shown in Fig. 8 and 9 are close to the optimization results. The inlet pressure at the high-pressure side ORC turbine is varied while other six parameters are kept as constants in Fig. 8 and 9. Thus, inlet pressure at the high-pressure side ORC turbine plays a more important role than other six parameters when determining the performance of the system. When the inlet pressure at the high-pressure side ORC turbine is close to the highest permitted pressure, the system performance is close to the optimization performance.

Fig. 16 shows the exergy destruction of different components of the system under the optimization conditions. The largest exergy destruction takes place in the ORC turbine (41.26%), which is mainly caused by the mixing of the high-pressure vapor and the low-pressure vapor. Gas heater contributes 13.44% of the total exergy destruction. Three vapor generators take up 4.13%, 11.67% and 3.73% of the exergy destruction, respectively. The exergy destruction for the ejector is 5.61%, which is also caused by the working fluid mixing. For BC turbine, condenser 1, precooler and preheater, the exergy destruction are 3.31%, 4.64%, 3.69% and 3.65%, respectively. Other components contribute to the rest 4.87% of the exergy destruction.

5. Conclusion

In this paper, a combined cooling and power system is developed. Seven parameters are selected to analyze the thermodynamic and exergoeconomic performance of the system. Single-objective optimization is carried out with the help of GA. The conclusions of the study are presented as follows:

(1) In the CBC, the increase of $T_{Bt,in}$ and $P_{Bt,in}$ contribute to the increase of the system exergy efficiency and the decrease of the levelized exergy cost for the system product.

(2) In the DORC, the increase of $T_{Ot,in}$ and $T_{Ot,in,l}$ would cause the decrease of the system exergy efficiency and the increase of the levelized exergy cost for the system product. Meanwhile, the increase of $P_{Ot,in,h}$ and $P_{Ot,in,l}$ would result in the increase of the exergy efficiency and the decrease of the levelized exergy cost.

(3) In the ERC, the increase of $P_{ej,in}$ would cause the increase of the refrigeration capacity and the decrease of the system net power output.

(4) Single -objective optimization results show that the minimum levelized exergy cost for the system product is obtain as $53.25 \text{ } \$(\text{MWh})^{-1}$ with net power output of 374.37 kW, cooling capacity of 188.63 kW and system exergy efficiency of 37.31%.

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741 Appendix A

742 This section shows the calculation of the heat transfer area in the heat exchangers
743 used in this study.

744 All the heat exchangers used in this study are shell-and-tube heat exchanger. The
745 thermodynamic properties of the working fluid vary with the heat transfer process.
746 Thus, to calculate the heat transfer area actually, the heat transfer processes are
747 discretized into a lot of small sections. In each section, the heat transfer area is small
748 and the thermodynamic properties are assumed to be constant.

749 For each section the heat transfer area is calculated as:

$$750 \quad A_i = \frac{Q_i}{(\Delta T_i \cdot U_i)} \quad (B1)$$

751 where ΔT_i is the log-mean temperature difference (LMTD) and U_i is the overall heat
752 transfer coefficient.

$$753 \quad \frac{1}{U_i} = \frac{1}{\alpha_{t,i}} + \frac{\delta}{\lambda} + \frac{1}{\alpha_{s,i}} \quad (B2)$$

754 In Eq. (B2) δ and λ represent the thickness of the tube and the thermal conductivity
755 of the tube wall, respectively. $\alpha_{t,i}$ is the convection heat transfer coefficient in the tube
756 side and $\alpha_{s,i}$ is the convection heat transfer coefficient in the shell side.

757 For different heat transfer process, the convection heat transfer coefficient has
758 different format. We classify the heat transfer processes into single-phase heat transfer
759 process and two-phase heat transfer process. In gas heater, precooler and the preheater,
760 single-phase heat transfer process happens. In evaporator, two-phase heat transfer
761 process occurs. In vapor generators and the condensers, both the single-phase and the

762 two-phase heat transfer process happen.

763 In the single-phase heat transfer process, the convection heat transfer coefficient in
764 the tube side and the shell side are expressed as [46]:

$$765 \quad \alpha_{t,i} = \frac{\lambda \cdot Nu}{D_i} \quad (B3)$$

$$766 \quad \alpha_{s,i} = 0.36 \left(\frac{\lambda}{D_{es}} \right) \cdot \left(\frac{D_{es} \cdot G_s}{\mu} \right)^{0.55} \cdot Pr^{\frac{1}{3}} \cdot \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad (B4)$$

767 In Eq. (B3), the Nusselt number is calculated as [47,48]:

$$768 \quad Nu = \left[\frac{(f/8) \cdot Re \cdot Pr}{12.7(f/8)^{0.5} \cdot (Pr^{2/3} - 1) + 1.07} \right], \text{ for } Re < 10^4 \quad (B5)$$

$$769 \quad Nu = \left[\frac{(f/8) \cdot (Re - 1000) \cdot Pr}{12.7(f/8)^{0.5} \cdot (Pr^{2/3} - 1) + 1.07} \right], \text{ for } 10^4 < Re < 5 \times 10^6 \quad (B6)$$

770 where f is the Darcy friction factor, Re is the Reynolds and Pr is the Prandtl number.

771 In Eq. (B4), D_{es} is the equivalent diameter of the shell, being expressed as:

$$772 \quad D_{es} = \frac{1.10Pt^2}{D_{out,i}} - D_{out,i} \quad (B7)$$

773 where Pt is the center distance between the tubes.

774 Evaporation and condensation are two different two-phase heat transfer processes.

775 In this study, the cold organic working fluid flows in the tubes of the heat exchangers.

776 The convection heat transfer coefficient of evaporation and condensation are
777 expressed as [49,50]:

$$778 \quad \alpha_{ev,i} = 0.023 \left[\frac{G(1-x)}{\mu_l} \right]^{0.8} \cdot Pr_l^{0.4} \cdot \frac{\lambda_l}{d} \cdot \left[1 + 3000Bo^{0.86} + 1.12 \left(\frac{x}{1-x} \right)^{0.75} \cdot \left(\frac{\rho_l}{\rho_v} \right)^{0.41} \right] \quad (B8)$$

$$779 \quad \alpha_{cond,i} = 0.023 \left[\frac{G(1-x)}{\mu_l} \right]^{0.8} \cdot Pr_l^{0.4} \cdot \frac{\lambda_l}{d} \cdot \left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)0.04}{P_r^{0.38}} \right] \quad (B9)$$

In Eq. (B9), P_r is the reduced pressure. In Eq. (B8) Bo is the boiling number, being expressed as:

$$Bo = \frac{q_m}{G \cdot r_f} \quad (B10)$$

Appendix B

The constants for component capital cost calculation are list in Table B1.

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923

924 **Figure captions**

925 **Fig. 1.** Schematic diagram of the CCP system

926 **Fig. 2.** Influences of BC turbine inlet temperature on the output and the exergy
927 efficiency of the system.

928 **Fig. 3.** Influences of BC turbine inlet temperature on the levelized exergy cost and the
929 system capital cost of the system.

930 **Fig. 4.** Influences of BC turbine inlet pressure on the output and the exergy efficiency
931 of the system.

932 **Fig. 5.** Influences of BC turbine inlet pressure on the levelized exergy cost and the
933 system capital cost of the system.

934 **Fig. 6.** Influences of inlet temperature at the high-pressure side of ORC turbine on the
935 output and the exergy efficiency of the system.

936 **Fig. 7.** Influences of inlet temperature at the high-pressure side of ORC turbine on the
937 levelized exergy cost and the system capital cost of the system.

938 **Fig. 8.** Influences of inlet pressure at the high-pressure side of ORC turbine on the
939 output and the exergy efficiency of the system.

940 **Fig. 9.** Influences of inlet pressure at the high-pressure side of ORC turbine on the
941 levelized exergy cost and the system capital cost of the system.

942 **Fig. 10.** Influences of inlet temperature at the low-pressure side of ORC turbine on the
943 output and the exergy efficiency of the system.

944 **Fig. 11.** Influences of inlet temperature at the low-pressure side of ORC turbine on the
945 levelized exergy cost and system capital cost of the system.

946 **Fig. 12.** Influences of inlet pressure at the low-pressure side of ORC turbine on the
947 output and the exergy efficiency of the system.

948 **Fig. 13.** Influences of inlet pressure at the low-pressure side of ORC turbine on the
949 levelized exergy cost and system capital cost of the system.

950 **Fig. 14.** Influences of ejector primary inlet pressure on the output and the exergy
951 efficiency of the system.

952 **Fig. 15.** Influences of ejector primary inlet pressure on the levelized exergy cost and
953 the system capital cost of the system.

954 **Fig. 16.** Exergy destruction of different components
955

Component	Energy equation	E_F	E_P	E_D	E_L
Gas heater	$M_{g1} \cdot (h_{g1} - h_{g2}) = M_2 \cdot (h_3 - h_2)$	$E_{g1} - E_{g2}$	$E_3 - E_2$	$E_{g1} + E_2 - E_3 - E_{g2}$	/
BC turbine	$W_{Bt} = M_3 \cdot (h_3 - h_4) = M_3 \cdot (h_3 - h_{4s}) \cdot \eta_{Bt}$	$E_3 - E_4$	W_{Bt}	$E_3 - E_4 - W_{Bt}$	/
Vapor generator 2	$M_4 \cdot (h_4 - h_5) = M_9 \cdot (h_{10} - h_9)$	$E_4 - E_5$	$E_{10} - E_9$	$E_4 + E_9 - E_5 - E_{10}$	/
Precooler	$M_1 \cdot (h_5 - h_1) = M_{26} \cdot (h_{27} - h_{26})$	/	/	$E_5 + E_{26} - E_1 - E_{27}$	$E_{27} - E_{26}$
Compressor	$W_{comp} = M_1 \cdot (h_2 - h_1) = M_1 \cdot (h_{2s} - h_1) / \eta_{comp}$	W_{comp}	$E_2 - E_1$	$E_1 - E_2 + W_{comp}$	/
Vapor generator 1	$M_{g2} \cdot (h_{g2} - h_{g3}) = M_8 \cdot (h_{11} - h_8)$	$E_{g2} - E_{g1}$	$E_{11} - E_8$	$E_{g2} + E_8 - E_{11} - E_{g3}$	/
ORC turbine	$W_{Ot} = M_{10} \cdot (h_{10} - h_{12}) + M_{11} \cdot (h_{11} - h_{12})$	$E_{10} + E_{11} - E_{12}$	W_{Ot}	$E_{10} + E_{11} - E_{12} - W_{Ot}$	/
Condenser 1	$M_{12} \cdot (h_{12} - h_{13}) = M_{28} \cdot (h_{29} - h_{28})$	/	/	$E_{12} + E_{28} - E_{13} - E_{29}$	$E_{29} - E_{28}$
Pump 1	$W_{p1} = M_{13} \cdot (h_{14} - h_{13}) = M_{13} \cdot (h_{14s} - h_{13}) / \eta_{p1}$	W_{p1}	$E_{14} - E_{13}$	$E_{13} - E_{14} + W_{p1}$	/
Preheater	$M_{15} \cdot (h_{15} - h_{14}) = M_{w1} \cdot (h_{w1} - h_{w2})$	$E_{w1} - E_{w2}$	$E_{15} - E_{14}$	$E_{w1} + E_{14} - E_{15} - E_{w2}$	/
Pump 2	$W_{p2} = M_7 \cdot (h_9 - h_7) = M_7 \cdot (h_{9s} - h_7) / \eta_{p2}$	W_{p2}	$E_9 - E_7$	$E_7 - E_9 + W_{p2}$	/
Pump 3	$W_{p3} = M_6 \cdot (h_8 - h_6) = M_6 \cdot (h_{8s} - h_6) / \eta_{p3}$	W_{p3}	$E_8 - E_6$	$E_6 - E_8 + W_{p3}$	/
Vapor generator 3	$M_{23} \cdot (h_{23} - h_{22}) = M_{w2} \cdot (h_{w2} - h_{w3})$	$E_{w2} - E_{w3}$	$E_{23} - E_{22}$	$E_{w2} + E_{22} - E_{23} - E_{w3}$	/
Condenser 2	$M_{16} \cdot (h_{16} - h_{17}) = M_{30} \cdot (h_{31} - h_{30})$	/	/	$E_{16} + E_{30} - E_{17} - E_{31}$	$E_{31} - E_{30}$
Valve	$h_{19} = h_{20}$	/	/	$E_{19} - E_{20}$	/
Pump 4	$W_{p4} = M_{22} \cdot (h_{22} - h_{18}) = M_{22} \cdot (h_{22s} - h_{18}) / \eta_{p4}$	W_{p4}	$E_{22} - E_{18}$	$E_{18} - E_{22} + W_{p4}$	/
Ejector	$M_{16} \cdot h_{16} = M_{23} \cdot h_{23} + M_{21} \cdot h_{21}$	$E_{23} + E_{21}$	E_{16}	$E_{23} + E_{21} - E_{16}$	/
Evaporator	$M_{20} \cdot (h_{21} - h_{20}) = M_{24} \cdot (h_{24} - h_{25})$	$E_{20} - E_{21}$	$E_{25} - E_{24}$	$E_{20} + E_{24} - E_{21} - E_{25}$	/

Component	Cost balance	Auxiliary relation
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Gas heater	$c_{g2} \cdot E_{y,g2} + c_3 \cdot E_{y,3} = c_{g1} \cdot E_{y,g1} + c_2 \cdot E_{y,2} + Z_{gh}$	$c_{g1}=c_{g2}=0$
Vapor generator 2	$c_5 \cdot E_{y,5} + c_{10} \cdot E_{y,10} = c_4 \cdot E_{y,4} + c_9 \cdot E_{y,9} + Z_{vg,2}$	$c_4=c_5$
BC turbine	$c_4 \cdot E_{y,4} + c_{Bt} \cdot W_{y,Bt} = c_3 \cdot E_{y,3} + Z_{Bt}$	$c_4=c_3$
Precooler	$c_1 \cdot E_{y,1} + c_{26} \cdot E_{y,26} = c_5 \cdot E_{y,5} + c_{27} \cdot E_{y,27} + Z_{prec}$	$c_1=c_5$
Compressor	$c_2 \cdot E_{y,2} = c_1 \cdot E_{y,1} + c_{elec,1} \cdot W_{y,comp} + Z_{comp}$	$c_{elec,1}=c_{Bt}$
Vapor generator 1	$c_{g3} \cdot E_{y,g3} + c_{11} \cdot E_{y,11} = c_{g2} \cdot E_{y,g2} + c_8 \cdot E_{y,8} + Z_{vg,1}$	$c_{g2}=c_{g3}$
ORC turbine	$c_{12} \cdot E_{y,12} + c_{Ot} \cdot W_{y,Ot} = c_{11} \cdot E_{y,11} + c_{10} \cdot E_{y,10} + Z_{Ot}$	$c_{10}=c_{11}=c_{12}$
Pump 1	$c_{14} \cdot E_{y,14} = c_{13} \cdot E_{y,13} + c_{elec,3} \cdot W_{y,pump1} + Z_{pump1}$	$c_{elec,3}=c_{Ot}$
Condenser 1	$c_{13} \cdot E_{y,13} + c_{29} \cdot E_{y,29} = c_{28} \cdot E_{y,28} + c_{12} \cdot E_{y,12} + Z_{cond1}$	$c_{13}=c_{12}$
Preheater	$c_{w2} \cdot E_{y,w2} + c_{15} \cdot E_{y,15} = c_{w1} \cdot E_{y,w1} + c_{14} \cdot E_{y,14} + Z_{preh}$	$c_{w1}=c_{w2}=0$
Pump 2	$c_9 \cdot E_{y,9} = c_7 \cdot E_{y,7} + c_{elec,2} \cdot W_{y,pump2} + Z_{pump2}$	$c_{elec,2}=c_{Ot}$
Pump 3	$c_8 \cdot E_{y,8} = c_6 \cdot E_{y,6} + c_{elec,3} \cdot W_{y,pump3} + Z_{pump3}$	$c_{elec,3}=c_{Ot}$
Vapor generator 3	$c_{w3} \cdot E_{y,w3} + c_{23} \cdot E_{y,23} = c_{w2} \cdot E_{y,w2} + c_{22} \cdot E_{y,22} + Z_{vg,3}$	$c_{w3}=c_{w2}$
Valve	/	$c_{19}=c_{20}$
Pump 4	$c_{22} \cdot E_{y,22} = c_{18} \cdot E_{y,18} + c_{elec,3} \cdot W_{y,pump4} + Z_{pump4}$	$c_{elec,4}=c_{Ot}$
Condenser 2	$c_{17} \cdot E_{y,17} + c_{31} \cdot E_{y,31} = c_{30} \cdot E_{y,30} + c_{16} \cdot E_{y,16} + Z_{cond2}$	$c_{16}=c_{17}$
Ejector	$c_{16} \cdot E_{y,16} = c_{23} \cdot E_{y,23} + c_{21} \cdot E_{y,21}$	/
Evaporator	$c_{21} \cdot E_{y,21} + c_{25} \cdot E_{y,25} = c_{20} \cdot E_{y,20} + c_{24} \cdot E_{y,24} + Z_{ev}$	$c_{20}=c_{21}$

958 **Table 3** Main parameters of the engine [7]

Parameters	Value
Power output (kW)	2928
Rotation (r(min) ⁻¹)	1000

Exhaust gas temperature (°C)	470
Exhaust gas mass flow rate (kg s ⁻¹)	4.35
Temperature of jacket water (°C)	90/79
Mass flow rate of jacket water (kg s ⁻¹)	25

959 **Table 4** Composition of the exhaust gas [7]

Composition	Molecular (g(mol) ⁻¹)	Fraction (%)
O ₂	32.00	9.3
CO ₂	44.00	9.1
H ₂ O	18.01	7.4
N ₂	28.01	74.2

960 **Table 5** Condition of simulation for the CCP system

Parameter	Value
Ambient temperature (°C)	20
Ambient pressure (MPa)	0.101
Compressor inlet temperature (°C)	35
BC turbine inlet temperature (°C)	400
BC turbine inlet pressure (MPa)	18
BC turbine outlet pressure (MPa)	8
Inlet temperature at the high-pressure side of ORC turbine (°C)	150
Inlet pressure at the high-pressure side of ORC turbine (MPa)	1.6
Inlet temperature at the low-pressure side of ORC turbine (°C)	100
Inlet pressure at the low-pressure side of ORC turbine (MPa)	1.0

Outlet pressure of pump 1 (MPa)	0.9
Ejector primary inlet pressure (MPa)	0.4
Terminal temperature difference at gas heater outlet (°C)	100
Pinch point temperature difference in vapor generator 1 (°C)	30
Pinch point temperature difference in vapor generator 2 (°C)	30
Pinch point temperature difference in vapor generator 3 (°C)	25
Condensation temperature of condenser 1 (°C)	30
Condensation temperature of condenser 2 (°C)	30
Evaporation temperature of evaporator (°C)	5
Isentropic efficiency of BC turbine (%)	80
Isentropic efficiency of ORC turbine (%)	80
Isentropic efficiency of compressor (%)	80
Isentropic efficiency of pump 1 (%)	75
Isentropic efficiency of pump 2 (%)	75
Isentropic efficiency of pump 3 (%)	75
Inlet temperature of cooling water (°C)	20

961 **Table 6** Parameters for GA

Parameter	Operation range
BC turbine inlet temperature (°C)	330-440
BC turbine inlet pressure (MPa)	15-20
Inlet temperature at the high-pressure side of ORC turbine (°C)	130-180
Inlet pressure at the high-pressure side of ORC turbine (MPa)	1.4-2

Inlet temperature at the low-pressure side of ORC turbine (°C)	90-150
Inlet pressure at the low-pressure side of ORC turbine (MPa)	0.9-1.3
Ejector primary inlet pressure (MPa)	0.3-1

962 **Table 7** Control parameters of GA

Tuning parameters	Value
Population size	20
Mutation probability	0.01
Crossover probability	0.8
Stop generation	200

963 **Table 8** Single-objective optimization results

Parameter	Value
BC turbine inlet temperature (°C)	425.46
BC turbine inlet pressure (MPa)	20.00
Inlet temperature at the high-pressure side of ORC turbine (°C)	144.32
Inlet pressure at the high-pressure side of ORC turbine (MPa)	1.85
Inlet temperature at the low-pressure side of ORC turbine (°C)	100.03
Inlet pressure at the low-pressure side of ORC turbine (MPa)	1.26
Ejector primary inlet pressure (MPa)	0.54
Net power output (kW)	374.37
Cooling capacity (kW)	188.63
Exergy efficiency (%)	37.31
Levelized exergy cost (\$ (MWh) ⁻¹)	53.25

964 **Table B1** Constants for component costs [37]

Constant	Value	Constant	Value	Constant	Value
$B_{1,he}$	1.63	$K_{3,pump}$	0.1538	$C_{3,he}$	0.08183
$B_{2,he}$	1.66	$K_{1,turb}$	2.7051	$C_{1,pump}$	-0.3635
$B_{1,pump}$	1.89	$K_{2,turb}$	1.4398	$C_{2,pump}$	0.3957
$B_{2,pump}$	1.35	$K_{3,turb}$	-0.1776	$C_{3,pump}$	-0.0026
$K_{1,he}$	4.3247	$K_{1,comp}$	2.2897	$F_{M,he}$	1.0
$K_{2,he}$	-0.3030	$K_{2,comp}$	1.3604	$F_{BM,turb}$	3.5
$K_{3,he}$	0.1634	$K_{3,comp}$	-0.1027	$F_{BM,comp}$	2.7
$K_{1,pump}$	3.3892	$C_{1,he}$	0.03881	$F_{M,pump}$	2.2
$K_{2,pump}$	0.0536	$C_{2,he}$	-0.11272		

965

966

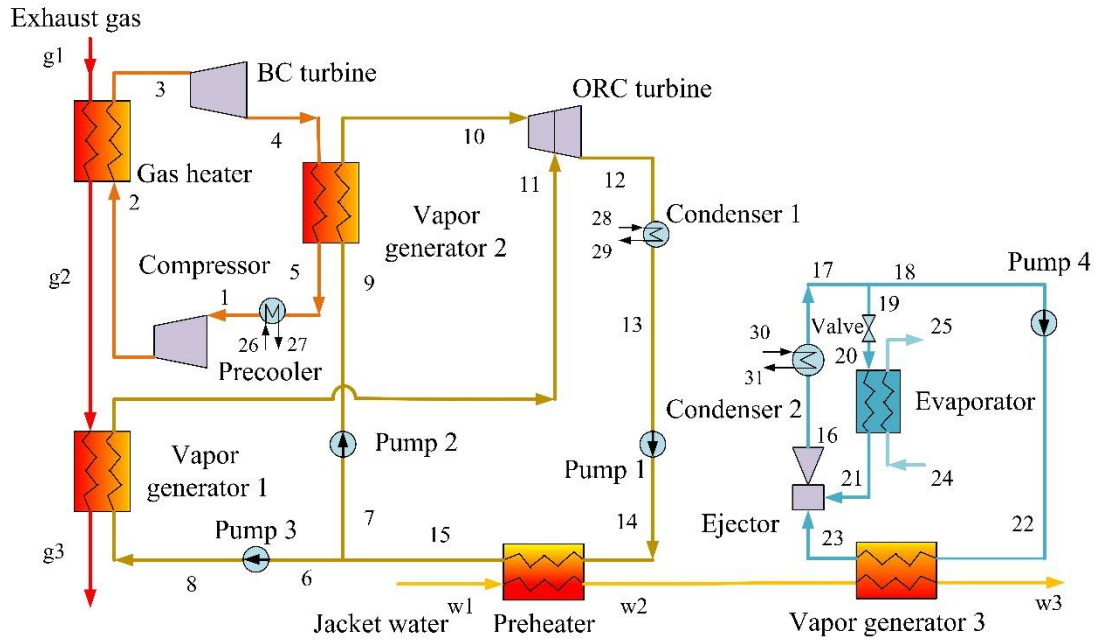


Fig. 1. Schematic diagram of the CCP system

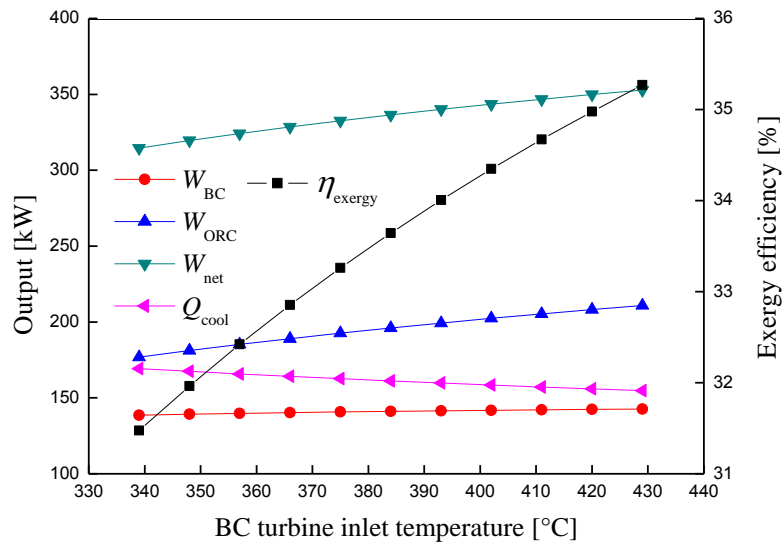


Fig. 2. Influences of BC turbine inlet temperature on the output and the exergy efficiency of the system.

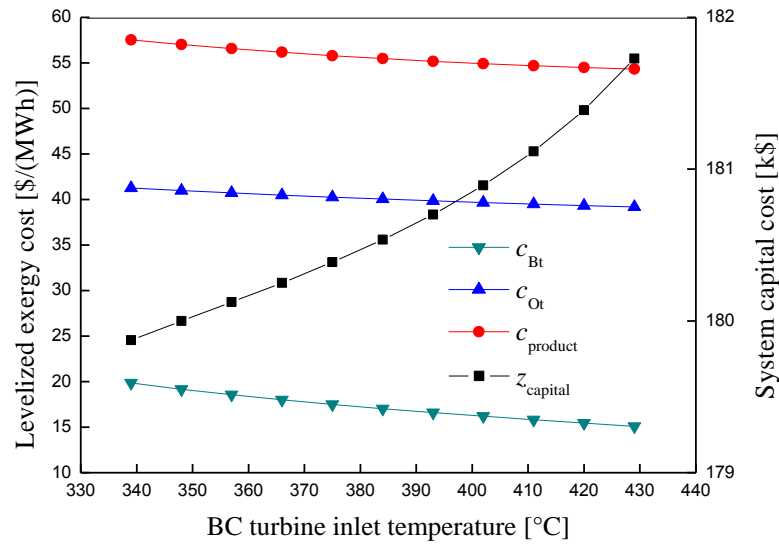


Fig. 3. Influences of BC turbine inlet temperature on the levelized exergy cost and the system capital cost of the system.

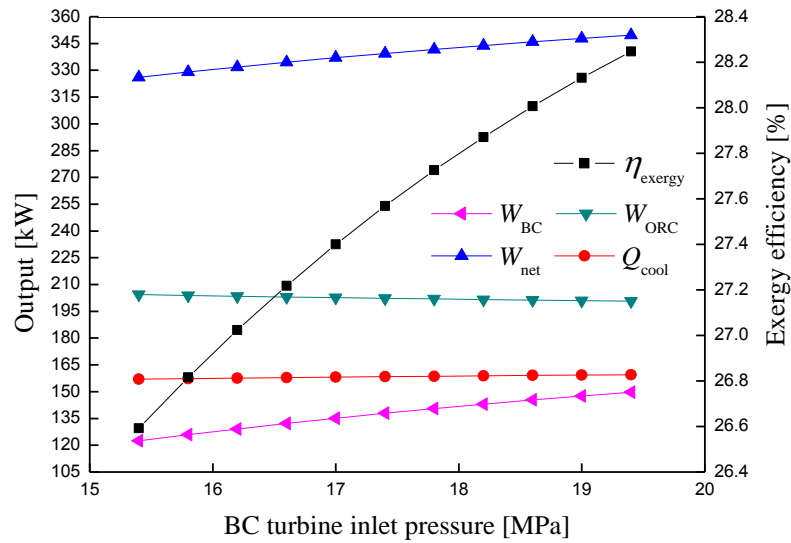


Fig. 4. Influences of BC turbine inlet pressure on the output and the exergy efficiency of the system.

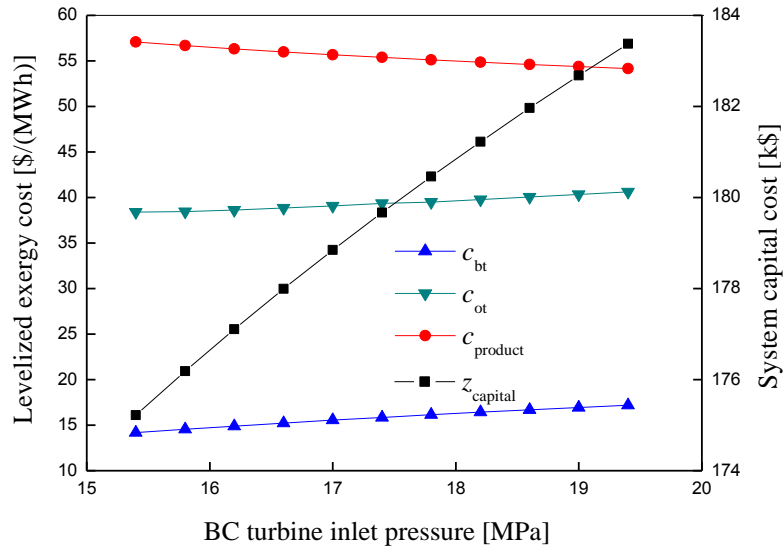


Fig. 5. Influences of BC turbine inlet pressure on the levelized exergy cost and the system capital cost of the system.

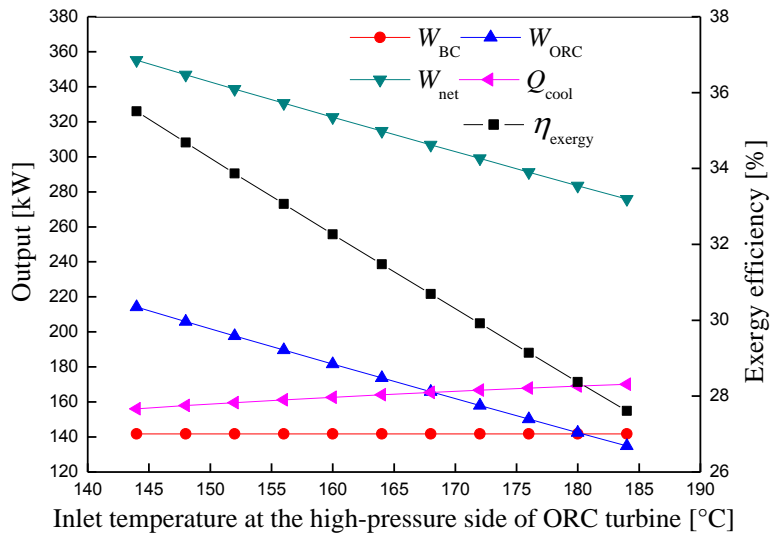


Fig. 6. Influences of inlet temperature at the high-pressure side of ORC turbine on the output and the exergy efficiency of the system.

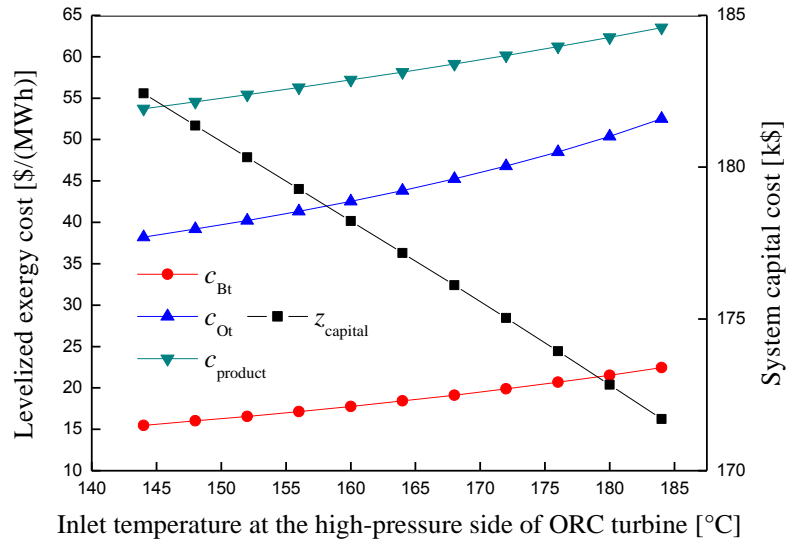


Fig. 7. Influences of inlet temperature at the high-pressure side of ORC turbine on the levelized exergy cost and the system capital cost of the system.

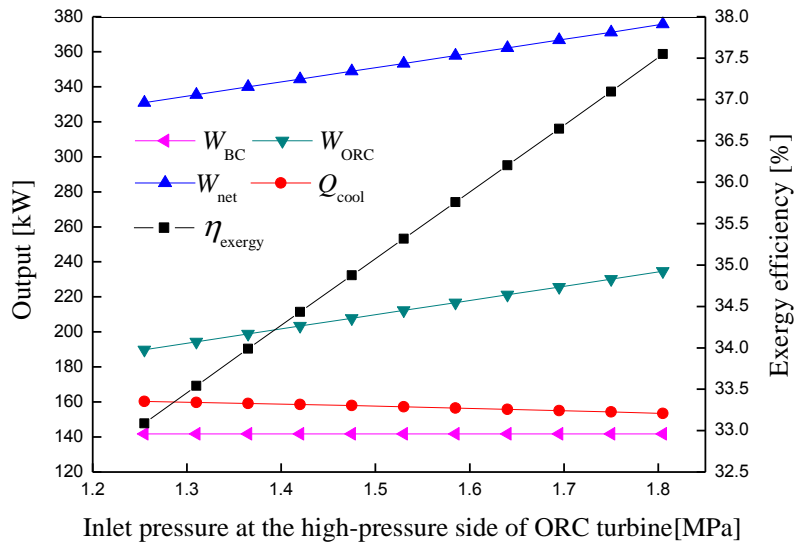


Fig. 8. Influences of inlet pressure at the high-pressure side of ORC turbine on the output and the exergy efficiency of the system.

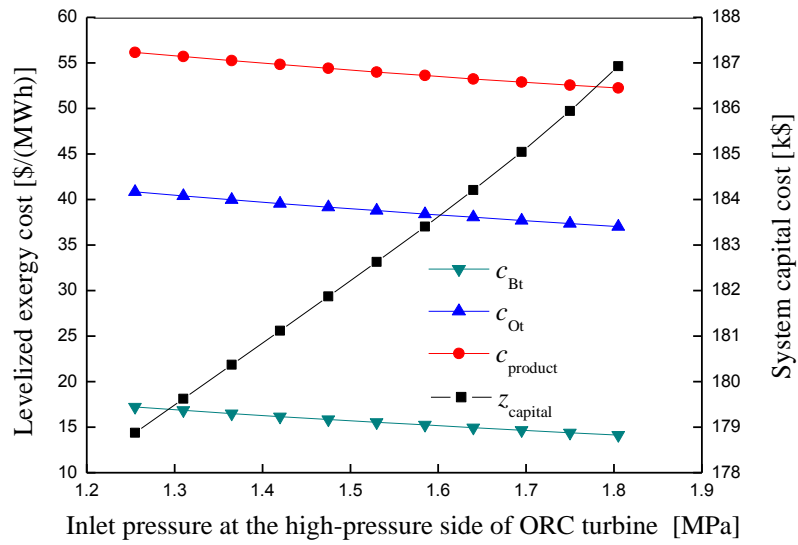


Fig. 9. Influences of inlet pressure at the high-pressure side of ORC turbine on the levelized exergy cost and the system capital cost of the system.

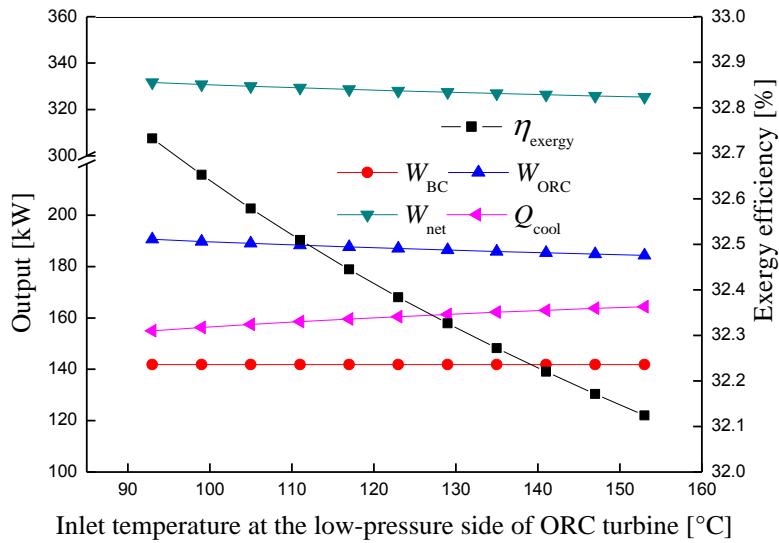


Fig. 10. Influences of inlet temperature at the low-pressure side of ORC turbine on the output and the exergy efficiency of the system.

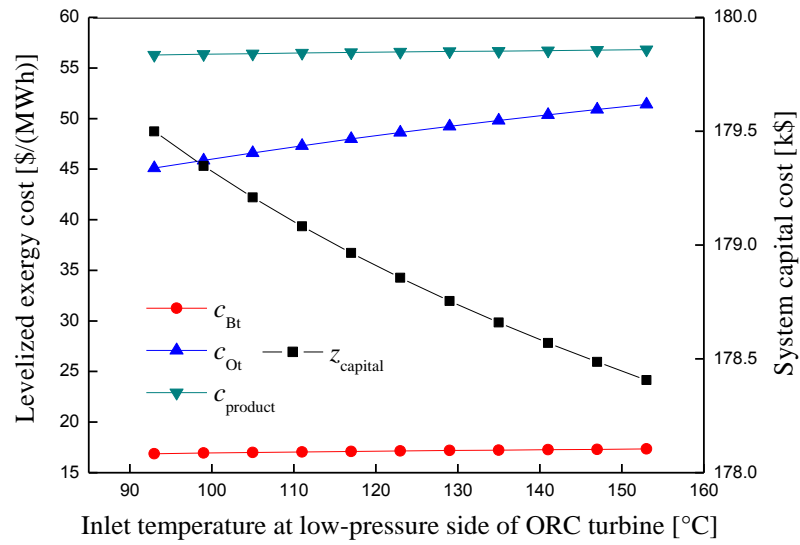


Fig. 11. Influences of inlet temperature at the low-pressure side of ORC turbine on the levelized exergy cost and system capital cost of the system.

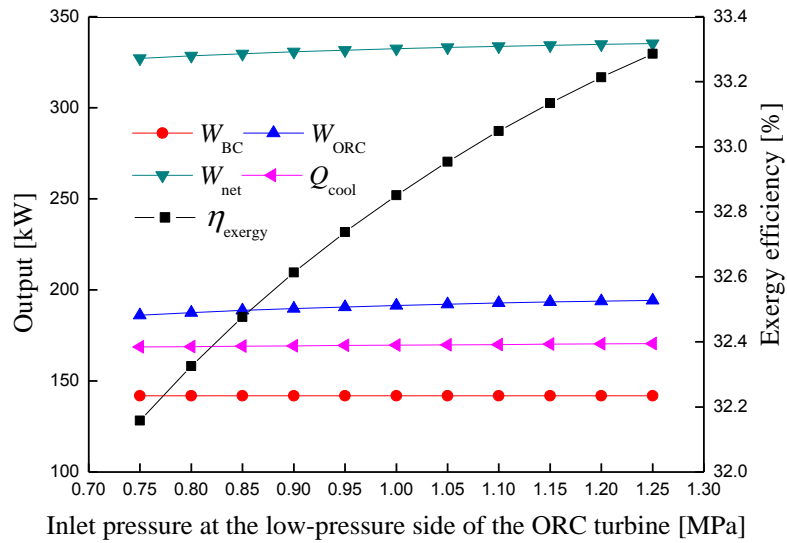


Fig. 12. Influences of inlet pressure at the low-pressure side of ORC turbine on the output and the exergy efficiency of the system.

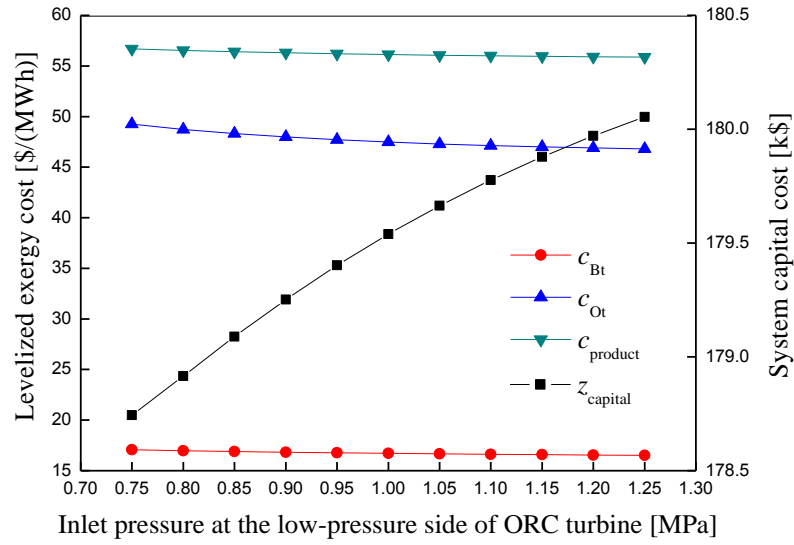


Fig. 13. Influences of the inlet pressure at the low-pressure side of ORC turbine on the levelized exergy cost and system capital cost of the system.

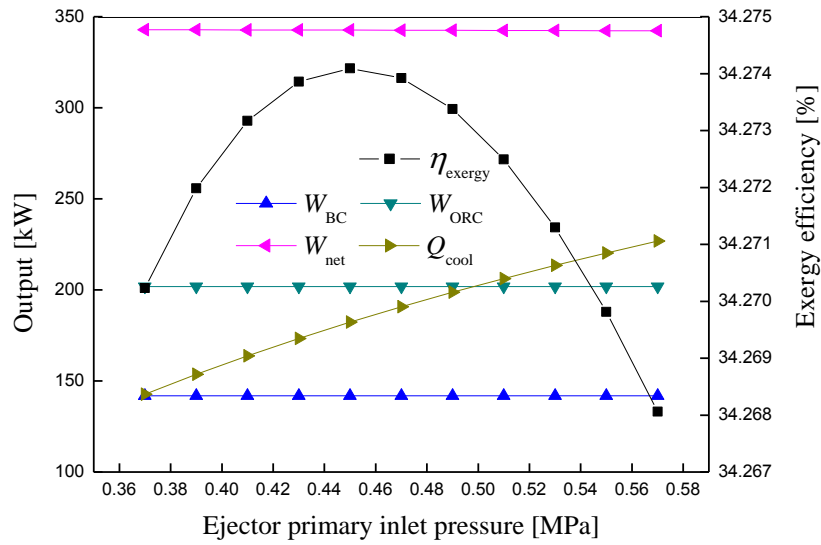


Fig. 14. Influences of ejector primary inlet pressure on the output and the exergy efficiency of the system.

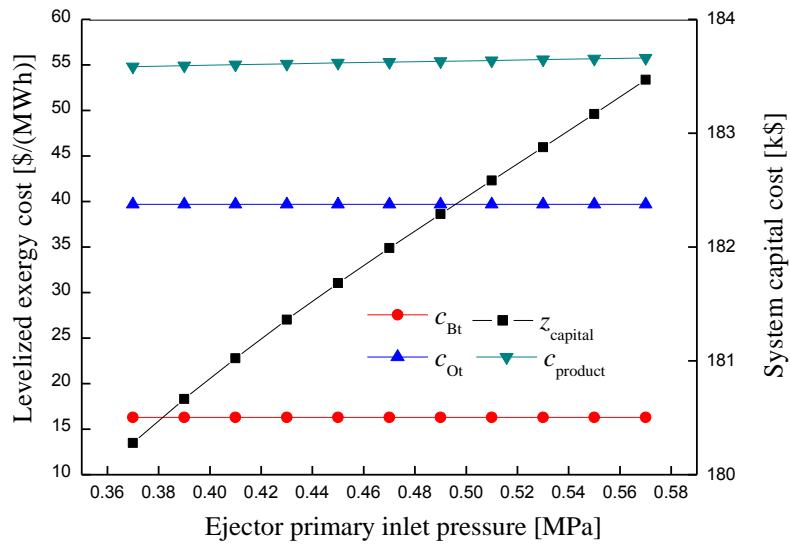


Fig. 15. Influences of ejector primary inlet pressure on the levelized exergy cost and the system capital cost of the system.

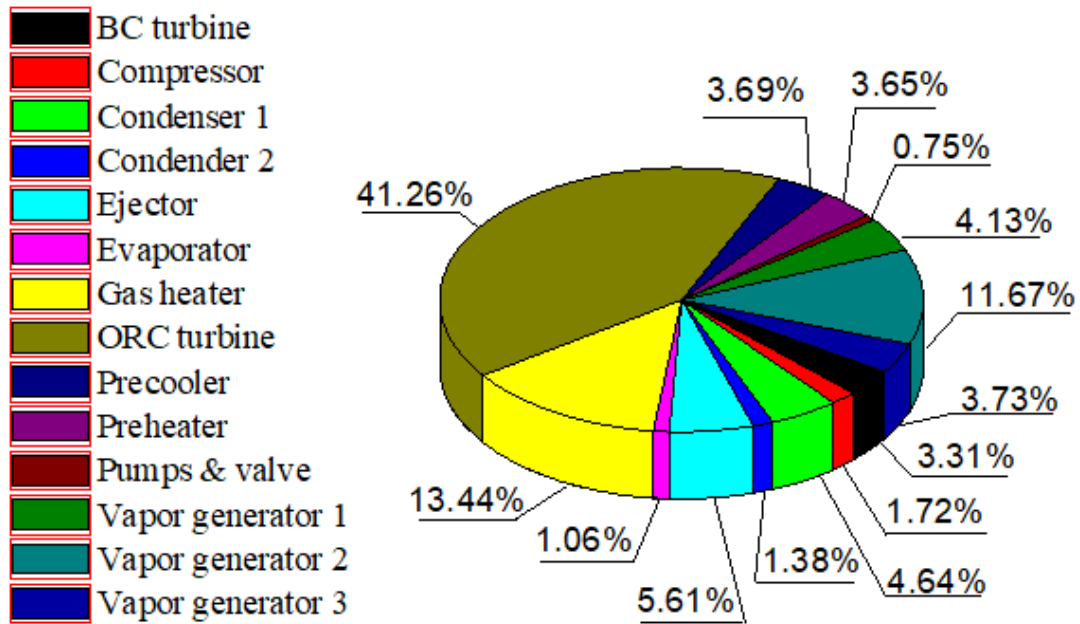


Fig. 16. Exergy destruction of different components

Response to Reviewers

Paper No. ECM-D-18-05037, Energy Conversion and Management

Title: Performance analysis and optimization of a combined cooling and power system using low boiling point working fluid driven by engine waste heat

Dear editor & reviewers,

Thank you very much for your review of our paper entitled “Performance analysis and optimization of a combined cooling and power system using low boiling working fluid driven by engine waste heat”, and for your comments and recommendations. These comments and recommendations help us to make better modifications and improve the quality of the paper. We have modified the manuscript accordingly in the revised manuscript. Please find below our response and explanations for your comments and questions.

Editor Comments:

1. Avoid lumping references as in [2, 3] and all other. Instead summarize the main contribution of each referenced paper in a separate sentence. How each paper is related to the work presented in the manuscript? What is being challenged or extended?

We thank the editor for the suggestion. We have avoided the reference lumping in the revised manuscript. The lumping parts have been removed in the previous manuscript such as “waste heat recovery technology [2,3]”, “by many researchers

[13-15]”, “~~avoid the decomposition [17,18]~~”, “~~simple layout [27,28]~~”.

Research work in references were summarized to describe the present research and existing questions. Ref. [1] described the widely use of internal combustion engines. Ref. [2] described the low fuel utilization efficiency of internal combustion engines. Ref. [3] described the advantages of organic Rankine cycle system. Ref. [4], [5] and [6] described the work of selecting suitable organic working fluids for organic Rankine cycle system by three different research groups. Ref. [7] and [8] described the configuration improving work in single-loop organic Rankine cycle system for internal combustion engine waste heat recovery. The maximum power output of single-loop organic Rankine cycle was challenged. Ref. [9] pointed out that maximum power output of a single-loop organic Rankine cycle system was lower than that of a dual-loop organic Rankine cycle. Ref. [10], [11] and [12] described the configuration improving work in dual-loop organic Rankine cycle system by three different research groups. But thermal stability of the organic working fluids in these systems was neglected. To avoid this issue, Ref. [17], [18] and [19] developed thermoelectric generator and steam Rankine cycle to address the decomposition issue of the high temperature heat transfer. But the energy conversion efficiency of the thermoelectric generator and component size of steam Rankine cycle in the references were challenged. Low thermal efficiency and large component bulk might limit their application. Thus, we provided our solution: a carbon dioxide Brayton cycle. For Ref. [20], the authors didn't analysis the parameter variation of carbon dioxide Brayton cycle and they didn't utilized the energy in the jacket water. Ref. [23], [24] and [25]

described the utilization of jacket water in the organic Rankine cycle system for internal combustion engine waste heat recovery. But the utilization efficiency of jacket water is low. Thus, we designed a dual-pressure organic Rankine cycle to increase the mass flow rate of the working fluid preheated by jacket water. Ref. [27], [28] and [29] described the recent research work about multigeneration system driven by waste heat. They expressed the importance of the multigeneration. Ref. [30] and [31] described the combined cooling and power system driven by internal combustion engine. The authors used ammonia absorption refrigeration cycle in the systems to provide refrigeration. But the structure of the ammonia absorption refrigeration cycle is complex and it requires a relatively high driven temperature. Thus, we employed the ejector refrigeration cycle to provide refrigeration and further utilize the jacket water energy.

2. Please avoid having heading after heading with nothing in between, either merge your headings or provide a small paragraph in between.

We thank the editor for this suggestion. Heading after heading was avoided in the revised manuscript.

3. Avoid using abbreviations and acronyms in title, abstract, headings and highlights.

The first time you use a chemical formula in the text, please write the full compound name and the formula in parenthesis. Do not use chemical formula in the title, abstract, chapter headings and highlights.

We thank the editor for this suggestion. Abbreviations and acronyms in abstract and headings were removed and rewritten in the revised manuscript. “(carbon dioxide)”

was used to explain CO₂ the first time it appeared in the text in Line 105: “Brayton cycle with CO₂ (carbon dioxide) as working fluid”

4. The first time you use an acronym in the text, please write the full name and the acronym in parenthesis. Do not use acronyms in the title, abstract, chapter headings and highlights.

We thank the editor for this suggestion. We have written a full name of the acronym in parenthesis the first time it appeared in the text in the revised manuscript.

The use of acronym in the text were “internal combustion engine (ICE)” in Line 44; “study of organic Rankine cycle (ORC)” in Line 51; “introduced thermoelectric generator (TEG) technology” in Line 96; “coupled with CO₂ Brayton cycle (CBC)” in Line 107; “Combined cooling and power (CCP) systems” in Line 133; “Ammonia absorption refrigeration cycle (AARC)” in Line 141; “ejector refrigeration cycle (ERC)” in Line 144; “dual-pressure organic Rankine (DORC)” in Line 150.

5. The introduction should include problem context, literature review and the hypothesis based on the gap analysis of the previously published research.

We thank the editor for this suggestion. Introduction was thoroughly rewritten in the revised manuscript. Problem context, literature review and hypothesis were included in the introduction.

6. The originality of the paper needs to be further clarified.

We thank the editor for this suggestion. Originality of the paper was further clarified in the revised manuscript.

There are three innovative features in this paper:

(1) We investigated a CO₂ Brayton cycle to prevent the risk of the decomposition of organic working fluid and provide power with high efficiency.

Organic Rankine cycles are used by many researchers to recover waste heat from internal combustion engines. Refrigerants are widely used as working in organic Rankine cycles but their decomposition temperatures are relatively low (200-300 °C). When the high-temperature (above 450 °C) engine exhaust gas transfers heat with these organic working fluids, there is risk of decomposition of the organic working fluids. Though organic working fluids with high decomposition temperature were investigated researchers, their flammability hindered their further applications. Some high-temperature loops for waste heat recovery were proposed by some researchers such as thermoelectric generator, steam Rankine cycle. But the thermal efficiency of the thermoelectric generator is low and the bulk of components in steam Rankine cycle is large, which might limit their application. Thus, we use a CO₂ Brayton cycle work as a high-temperature loop in the system to prevent the decomposition risks and provide power with high efficiency and compact structure.

(2) We developed a dual-pressure organic Rankine cycle to increase the utilization efficiency of the jacket water energy.

The previous internal combustion engine waste heat recovery systems, energy in jacket water was not fully utilized. In some studies, energy in jacket water was not utilized as all. Jacket water was mainly used to preheat the organic working fluid in the organic Rankine cycle based internal combustion engine waste heat recovery system. But there is mismatch of the mass flow rate of the organic working fluid in

the preheater and evaporator. Thus, only a small part of the energy in jacket water was harnessed. We developed a dual-pressure organic Rankine cycle as a bottom cycle after the CO₂ Brayton cycle. Dual-pressure organic Rankine cycle can provide a large amount of power output. Moreover, organic working fluid in both high-pressure and low-pressure are preheated by the jacket water, which increases the utilization rate of the jacket water.

(3) We developed an ejector refrigeration cycle to fully utilize the jacket water waste heat and provide refrigeration.

Combined cooling and power system driven by engine waste heat shows the advantages of high efficiency and multiple energy supply. Ammonia absorption refrigeration cycle are widely used for refrigeration in these systems. But the configuration of it is complex and it requires a high driven temperature. We coupled a ejector refrigeration cycle with the organic Rankine cycle system instead of ammonia absorption refrigeration cycle to provide refrigeration. The ejector refrigeration cycle has a simple structure and requires relatively low driven temperature. Thus, jacket water is used to drive the ejector refrigeration cycle to provide refrigeration and fully utilize the waste heat.

Reviewer #1:

This manuscript introduces the performance analysis and optimization of a combined cooling and power system using low boiling point working fluid driven by engine waste. The novelty and the impact of the present manuscript is limited and the relation to previous works is poor.

Thank the reviewer for pointing out the shortcomings of the paper. We have thoroughly rewritten the introduction section to further clarify the novelty and the relation to previous work of the paper.

There are three innovative features in this paper:

(1) We investigated a CO₂ Brayton cycle to prevent the risk of the decomposition of organic working fluid and provide power with high efficiency.

Organic Rankine cycles are used by many researchers to recover waste heat from internal combustion engines. Refrigerants are widely used as working in organic Rankine cycles but their decomposition temperatures are relatively low (200-300 °C). When the high-temperature (above 450 °C) engine exhaust gas transfers heat with these organic working fluids, there are risks of decomposition of the organic working fluids. Though organic working fluids with high decomposition temperature were investigated researchers, their flammability hindered their further applications. Some high-temperature loops for waste heat recovery were proposed by some researchers such as thermoelectric generator, steam Rankine cycle. But the thermal efficiency of the thermoelectric generator is low and the bulk of components in steam Rankine cycle is large, which might limit their application. Thus, we use a CO₂ Brayton cycle work as a high-temperature loop in the system to prevent the decomposition risks and provide power with high efficiency and compact structure.

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About the impact of this paper:

There are two pathways to improve the performance of the organic Rankine cycle system for engine waste heat recovery. One will be selecting organic working fluids which are suitable for the system under certain conditions. Another is to optimize the system configuration to make full use of the waste heat. In this paper, we designed a combined cooling and power system with a novel configuration which prevents the decomposition risk of organic working fluid, fully utilizes the jacket water energy and provides power and refrigeration simultaneously.

About the relation of this paper to previous work:

We have thoroughly rewritten the introduction section of this paper. More recently published papers were added to describe research conditions and existing problems.

We carried out our work based on the analysis of previous work.

Reviewer #2:

Corrections mainly involve formatting points, such as English grammar, citation of references, etc.; and insufficient information regarding some equipment used in the research. Concerning the formatting and methodology the points are as follows:

1. In Nomenclature, present separately the Latin symbols, Greek symbols, Acronyms, Subscripts and Superscripts, because of the form that is presented, it is confused; include the acronym TEG - Thermoelectric generator and GA - Genetic Algorithm; if the "K" of K\$ is "kilo", must be represented by "k".

Thank the reviewer for the kind suggestion. The Nomenclature section was separated as Latin symbols, Acronyms, Greek symbols and Subscripts in the revised manuscript. TEG and GA were added to the list and "K\$" was changed to "k\$".

2. In lines 72, 96, 97 and 203, separate the temperature values of its units, for example, 90 °C (line 72).

In the revised manuscript, temperature values were separated with their units.

In Line 87: “low (200-300 °C) [13], while the temperature of exhaust gas is above 450 °C”; in Line

3. In line 104 change "to the utilize" by other word.

We have thoroughly rewritten the introduction section and the error was avoided in the revised manuscript.

4. In line 126 put comma between gas and while, and after "But" in line 127.

We have thoroughly rewritten the introduction section and the error was avoided in the revised manuscript.

5. In line 139 and 140 change the word "flew" by "flows".

We have thoroughly rewritten the introduction section and the error was avoided in the revised manuscript.

6. In lines 148, 149 and 150 put "2" of "CO₂" as subscript.

We have thoroughly rewritten the introduction section and the error was avoided in the revised manuscript.

7. In line 166 change the word "consumption" by "combustion".

We have thoroughly rewritten the introduction section and the error was avoided in the revised manuscript.

8. In line 174 put comma after "Meanwhile".

We have thoroughly rewritten the introduction section and the error was avoided in

the revised manuscript.

9. From line 197 to 206 write "Several assumptions are made to simplify the simulation of the system, which are: (1) the system keeps a steady state; (2) the heat and friction ...are not considered; and so on.

We thank the reviewer for the suggestion. The system assumptions part was rewritten based on the suggestion in the revised manuscript from Line 369 to Line 337 "Several assumptions are made to simplify the simulation of the system, which are: (1) the system keeps a steady state; (2) the heat and friction in the system are not considered; (3) the pressure losses in the vapor generators, preheater, evaporator, condensers and pipes are neglected; (4) the gas temperature at the outlet of the vapor generator 1 is higher than 110 °C [35], considering the low gas dew point temperature; (5) the working fluids at the outlet of the condensers and the preheater are saturated liquids, and the evaporator outlet state is saturated vapor; (6) the process through the throttle valve is isenthalpic."

10. In line 228 put a comma after "thermodynamic".

We have rewritten that paragraph in the revised manuscript from line 221 to line 226. The error was avoided.

11. Begin the sentence of the line 236 by "In this study, all components in the system ..." and remove "in this study" of the end of the sentence.

The sentence was rewritten in the revised manuscript in Line 235 as following:

"In this study, all the components in the system are associated directly or indirectly with fuel of other heat sources, such as exhaust as and jacket water."

12. In line 257 put comma between "expenses" and "etc."

A comma was put between "expenses" and "etc." in the revised manuscript in line 250 as following:

(like the taxes, insurance engineering expenses, etc.)

13. Indicate that equations from 11 to 21 are proposed in [33].

We have indicated in the revised manuscript as following:

Line 360: "In the following text, equations from Eq. (11) to Eq. (21) are proposed in Ref. [37]."

14. In line 265 write "where $K_{i,turb}$ are the constants corresponding to the turbine type; and W is the power"; and in similar manner in lines 276, 282 and 286.

We have rewritten the sentences in the revised manuscript in Line 259 " $K_{i,turb}$ are constants corresponding to", Line 270 " $K_{i,pump}$ are the constants corresponding to", Line 276 " $B_{i,pump}$ are the constants corresponding to", Line 280 " $C_{i,pump}$ are the constants corresponding to", Line 285 " $K_{i,comp}$ are the constants corresponding to", Line 295 " $K_{i,he}$ are the constants corresponding to", Line 302 " $B_{i,he}$ are the constants corresponding to" and Line 306 " $C_{i,he}$ are the constants corresponding to".

15. In lines 297 and 784 I suggest to change the word "Tube-and-shell heat exchangers" to "Shell-and-tube heat exchangers".

It was corrected in the revised manuscript in lines 291 and 702 as following:

16. Verify in the equation (23) if the exponent of the term in brackets of the numerator is " $n - 1$ ".

Thank the reviewer for the kind suggestion. We have verified the equation. It was

"n".

17. In line 336 add the word "years" after 30.

It was corrected in the revised manuscript in line 328 as following: “being assumed as 30 years”

18. In lines 340 and 348 change the word "steams" by "streams"; and in lines 342, 343 and 349 change the word "steam" by "stream".

We have changed the expression in the revised manuscript in lines 332, 334, 335, 340 and 341 as following:

“In a steady system, there are a number of entering and outing working fluid streams and heat and work interactions with the surroundings. In exergoeconomic analysis, each flowing stream is associated with a levelized exergy cost. The equations to calculate the cost of the stream product are given as:”

“where c denotes levelized exergy cost of the streams; $E_{y,in}$ and $E_{y,out}$ are the exergy transfer rate of the stream flowing in and out of a component;”

19. In line 348 remove the space before "where" and verify if the variable "c" is the "levelized exergy cost of the system" or is the "average cost per unit of exergy" according to the Nomenclature.

Thank the reviewer for the kind suggestion. "c" is the "levelized exergy cost of the system". We have corrected the error in the Nomenclature and remove the space in line 340.

20. In line 349 add "of" after out.

We have corrected the error in the revised manuscript in line 341 as following:

“flowing in and out of a component”

21. In line 360 and 363 add the word "where" before "cfuel" and "cBt", respectively; and remove the initial space.

We have corrected the error in the revised manuscript in lines 352 and 355.

22. In line 365 add "... and the fuel-cost-related part, given by Eq. (32) and (33).

We have corrected the error in the revised manuscript in lines 357 and 358.

“capital-cost-related part and the fuel-cost-related part, given by Eq. (32) and Eq. (33).”

23. I suggest that sections 4.1.1, 4.1.2 and 4.2 are inserted at the end of section 3, because they represent materials and methods and not results and discussion.

Thank the reviewer for the kind suggestion. We have put sections 4.1.1, 4.1.2 and 4.2 to the end of section 3.

24. In line 380 remove the word "gas" and maintain only "... supercharger engine."; in line 382 replace "The heat load capacity" by "The thermal load of the ..." and deleted "when cooled down to the acid dew temperature".

We have rewritten the sentences in the revised manuscript in line 370.

“In this study, the engine selected [7] is a 12-cylinder 4-stroke supercharged engine. The main designed parameters of the engine are listed in Table 3. The composition of the engine exhaust gas is presented in Table 4. The thermal load of the engine exhaust gas is about 1700 kW and 1000 kW can be obtained from the engine jacket water.”

25. In line 386 express the seven key parameters by its symbols.

We have added the symbols of the key parameters in the revised manuscript from

line 376 to 380 as following:

“Seven key parameters : BC turbine inlet temperature ($T_{\text{Bt, in}}$), BC turbine inlet pressure ($P_{\text{Bt, in}}$), inlet temperature at the high-pressure side of ORC turbine ($T_{\text{Ot, in, h}}$), inlet pressure at the high-pressure side of ORC turbine ($P_{\text{Ot, in, h}}$), inlet temperature at the low-pressure side of ORC turbine ($T_{\text{Ot, in, l}}$), inlet pressure at the low-pressure side of ORC turbine ($P_{\text{Ot, in, l}}$) and the ejector primary inlet pressure ($P_{\text{ej, in}}$),”

26. In lines 392, 393, 395 and 396 express the variables "W", "Q" and "c" in italics.

We have corrected the error in the revised manuscript from line 384 to 389.

“In the thermodynamic aspect, the net power output of the CO₂ Brayton cycle (W_{BC}), net power output of the DORC (W_{ORC}), net power of the whole system (W_{net}), cooling capacity of the system (Q_{cool}) and the exergy efficiency of the system (η_{exergy}) are selected to reflect the system performance. Levelized exergy cost for the BC turbine power output (c_{Bt}), levelized exergy cost for the ORC turbine power output (c_{Ot}), levelized exergy cost the system product (c_{product}) and the system capital cost (z_{capital}) are chosen to represent the exergoeconomic performance.”

27. Start the Results and discussion section from section 4.2.1.

We have put that section at the beginning of Results and discussion.

28. In all results shown from Fig. 2 to Fig. 15 only one parameter at a time was varied, while the others were maintained constants? Clear this in the text.

We have explained this in the revised manuscript from line 381 to 383 as following:

“When one parameter is investigated to analyze the system performance, other parameters are maintained constants based on the conditions in Table 5.”

29. In line 552 change "The can be explained ..." by "This can be explained ...".

It was corrected in the revised manuscript.

30. Seems incomplete to me the sentence of the line 612 "Thus, the capita-cost-related ...".

We have rewritten the sentence in the revised manuscript.

31. In line 641 remove the word "vapor" after "... vapor generator 2".

We have rewritten the sentence in the revised manuscript in line 575.

32. In line 658 add "of" after "Though".

We have rewritten the sentence in the revised manuscript.

33. In the sentence of line 676 I suggest to write "The cooling capacity (Q_{cool}) increases slightly with the ..." because by Figure 12 the increase is very small and, hence, should also be corrected at the end of line 680.

We have rewritten the sentence in the revised manuscript in lines 604 and 607.

34. In line 709 add a "t" after "can".

The error was corrected in the revised manuscript in line 631.

35. Rewrite the two sentences from line 713 to 715.

We have rewritten the two sentences in the revised manuscript from line 635 to 637 as following:

"The increase of the ejector primary inlet pressure causes the increase of the entrainment ratio of the ejector. Thus, more secondary flow is entrained to the ejector from the evaporation, leading to the increase of the cooling capacity."

36. In the paragraph from line 751 to line 753 I suggest to refer to Fig. 8 and 9 where

are evidenced the highest output power, exergy efficiency and the lowest levelized exergy cost (but not the cooling capacity) at the highest inlet pressure at the high-pressure side ORC turbine.

We thank the reviewer for this kind suggestion. We compared the results of the genetic algorithm optimization results with the parameter trend in Fig. 8 and 9. The value of net power output, exergy efficiency, levelized exergy cost and the inlet pressure at the high-pressure side ORC turbine in the two parts were nearly the same. The inlet pressure at the high-pressure side ORC turbine is varied while other six parameters are kept as constants in Fig. 8 and 9. Thus, the inlet pressure at the high-pressure side ORC turbine plays a more important other six parameters. When the inlet pressure at the high-pressure side ORC turbine is close to the highest permitted pressure, the system performance is close to the optimization performance. Note that: When the inlet pressure at the high-pressure side ORC turbine increases, the pinch point temperature difference in vapor generator 2 decreases. Thus, there would be temperature cross in the vapor generator when inlet pressure at the high-pressure side ORC turbine is larger than critical value. That's why the optimization results shows that the value of the inlet pressure at the high-pressure side ORC turbine is 1.85 MPa in stead of the maximum value 2 MPa.

We have rewritten the paragraph from Line to Line as following:

“The optimization results of GA are listed in Table 8. It can be obtained that the minimum levelized exergy cost for the system product c_{product} is $53.25 \text{ } \$(\text{MWh})^{-1}$. The net power output, exergy efficiency of the CCP system are 374.37 kW, 37.31%

respectively. The inlet pressure at the high-pressure side of ORC turbine is 1.85 MPa. Meanwhile, it can be evidenced from Fig. 8 and 9 that the highest output power (about 374.37 kW), exergy efficiency (about 37.31%) and the lowest levelized exergy cost (about 53.25 \$(MWh)⁻¹) at the highest inlet pressure at the high-pressure side ORC turbine (about 1.85 MPa). The results shown in Fig. 8 and 9 are close to the optimization results. The inlet pressure at the high-pressure side ORC turbine is varied while other six parameters are kept as constants in Fig. 8 and 9. Thus, inlet pressure at the high-pressure side ORC turbine plays a more important role than other six parameters when determining the performance of the system. When the inlet pressure at the high-pressure side ORC turbine is close to the highest permitted pressure, the system performance is close to the optimization performance.”

37. The sentence of conclusion (2) is very extensive and I suggest ending it in "... for the system product.", beginning the next sentence such as, "Meanwhile, the increase of the ORC ...".

We have rewritten the sentence in the revised manuscript in line 689 as following: “In the DORC, the increase of $T_{Ot, in}$ and $T_{Ot, in, l}$ would cause the decrease of the system exergy efficiency and the increase of the levelized exergy cost for the system product. Meanwhile, the increase of $P_{Ot, in, h}$ and $P_{Ot, in, l}$ would result in the increase of the exergy efficiency and the decrease of the levelized exergy cost.”

38. In line 810 replace the word "frication" by "friction".

The error was corrected in the revised manuscript in line 728 as following: “the Darcy friction factor,”

39. In the "References" use the abbreviation names of the Journals, such as, Energy Convers Manage and the number of pages as 201-14, instead of using 201-214, for example; in the reference [14] line 863 change the number of the pages to 215-32 to differentiate from reference [12];

Thank the reviewer for the suggestion. We have rewritten the abbreviation names of Journals and the page number in the revised manuscript.

40. in reference [16] correct the names of authors to Rajabloo T, Bonalumi D, Lora P; in reference [17] add at the end of the author names, the author Zhu W; in reference [31] add at the end of the author names, the authors Liu H, Wang E, Yao B; in reference [32] correct the author names to Bejan A, Tsatsaronis G, Moran M and the first name of the publisher to John; in reference [33] add other authors, i.e., Turton R, Bailie RC, Whiting WB, Shaeiwitz JA;

We have corrected the errors in the revised manuscript in lines 816, 819, 823, 824 and 825 as following:

[34] Shu G, Zhao M, Tian H, Huo Y, Zhu W. Experimental comparison of R123 and R245fa as working fluids for waste heat recovery from heavy-duty diesel engine. Energy 2016; 115:756-69.

[35] Zhang J, Zhang H, Yang K, Yang F, Wang Z, Zhao G, Liu H, Wang E, Yao B. Performance analysis of regenerative organic Rankine cycle (RORC) using the pure working fluid and the zeotropic mixture over the whole operating range of a diesel engine. Energy Convers Manage 2014; 84:282-94.

[36] Bejan A, Tsatsaronis G, Moran M. Thermal design and optimization. New York:

John Wiley & Sons; 1996.

41. What is the reference [35]?

That reference was a website for the $CEPCI_{ref,2001}$. We have replaced it with reference [38]. In Ref. [38] $CEPCI_{ref,2001}$ is 398.

42. In table 3 replace the word "Term" by "Parameter" and the same in Table 8; in this table I suggest that the values presented be limited in two digits after comma to standardize all.

We have replaced the expression and limited the digits of the value in Table 8 in the revised manuscript.

43. In Table 6 replace "Ranges of the decision variables" by only "Parameters or Variables" and add "Operation" before "Range".

We have replaced the expression in Table 6 in the revised manuscript.

44. In Table B1 the source is [33] and not [32].

Thank the reviewer for helping us find the error. It was corrected in the revised manuscript.

Reviewer #3:

The authors have conducted a study that covers a topic of great interest: "Performance analysis and optimization of a combined cooling and power system using low boiling point working fluid driven by engine waste heat". This very important topic deserved a great deal of attention. However, many shortcomings can be identified. Therefore, I recommend that these shortcomings, as listed in the following, should be addressed before it can be considered for publication;

1. Abstract section, present in more detail and clarity

Thanks for the reviewer's kind suggestion. The abstract section was thoroughly rewritten in the revised manuscript. More details were added in the abstract.

“This paper develops a combined cooling and power system, which consists of a carbon dioxide Brayton cycle, a dual-pressure organic Rankine cycle and an ejector refrigeration cycle, to recover waste heat from exhaust gas and jacket water in internal combustion engines. Thermodynamic models of the system are performed and exergoeconomic methods are used to calculate the levelized exergy cost of the component products. Effects of seven parameters, including temperature and pressure at the Brayton cycle turbine inlet, temperature and pressure at the high-pressure and low-pressure side of the organic Rankine cycle turbine inlet and pressure at the ejector primary inlet, are evaluated. Single-objective optimization is carried out by means of genetic algorithm to obtain the minimum levelized exergy cost of system product. Results show that the increase of pressure at Brayton cycle turbine inlet and high-pressure and low-pressure side of the organic Rankine cycle turbine inlet contributes to the decrease of levelized exergy cost of the system product. Optimization results show that minimum levelized exergy cost for system product is $53.25 \$ (MWh)^{-1}$. When levelized exergy cost is minimum, system net power output, cooling capacity and exergy efficiency are 374.37 kW, 188.63 kW and 37.31%, respectively.”

2. Please, Give more numerical results about the study results in the abstract section.

Thank the reviewer for the suggestion. The abstract section was rewritten in the

revised manuscript and more numerical results were put in the abstract section.

3. Analysis of the state of the art in the introduction is insufficient, which undermines novelty of this work. An updated and complete literature review should be conducted.

The introduction section was thoroughly rewritten in the revised manuscript. More recent references were discussed and some old references were removed. The novelty of this work was further clarified.

4. Literature section should be given current papers after 2017.

We thank the reviewer for the kind suggestion. References in 2017 and 2018 were put in the introduction sections and some old references were removed in the revised manuscript.

[5] Rijpkema J, Munch K, Andersson S. Thermodynamic potential of twelve working fluids in Rankine and flash cycles for waste heat recovery in heavy duty diesel engines. *Energy* 2018; 160:996-1007.

[6] Su X, Shedd T A. Towards working fluid properties and selection of Rankine cycle based waste heat recovery (WHR) systems for internal combustion engines – A fundamental analysis. *Appl Therm Eng* 2018; 142:502-10.

[10] Wang X, Shu G, Tian H, Liu P, Jing D, Li X. Dynamic analysis of the dual-loop Organic Rankine Cycle for waste heat recovery of a natural gas engine. *Energy Convers Manage* 2017; 148:724-736.

[11] Wang E, Yu Z, Zhang H, Yang F. A regenerative supercritical dual-loop organic Rankine cycle system for energy recovery from the waste heat of internal combustion engines. *Appl Energy* 2017; 190:574-90.

[12] Huang H, Zhu J, Deng W, Ouyang T, Yan B, Yang X. Influence of exhaust heat distribution on the performance of dual-loop organic Rankine Cycle (DORC) for waste heat recovery. *Energy* 2018; 151:54-65.

[13] Rajabloo T, Bonalumi D, Lora P. Effect of a partial thermal decomposition of the working fluid on the performances of ORC power plants. *Energy* 2017; 133:1013-26.

[14] Shi L, Shu G, Tian H, Deng S. A review of modified Organic Rankine cycles (ORCs) for internal combustion engine waste heat recovery (ICE-WHR). *Renew Sustain Energy Rev* 2018; 92:95-110.

5. How were reference conditions (environmental pressure and temperature) considered?

In this paper, environment temperature is 20 °C and environment pressure is 101.3 kPa. We referenced it from the following reference:

Yang X, Zheng N, Zhao L, Deng S, Li H, Yu Z. Analysis of a novel combined power and ejector refrigeration cycle. *Energy Convers Manage*; 2016; 108:266-74.

6. The level of English throughout the manuscript does not meet the journal's desired standard. There are a number of grammatical errors.

Thank the reviewer for the kind suggestion. We have checked the grammatical errors and thoroughly rewrote the paper in the revised manuscript.

7. Introduction part needs to be extended by some of the recently published papers to show the importance of multigeneration systems in high-quality journals

We thank the reviewer for the kind suggestion. We have expressed the importance of the multigeneration systems with three recently published paper in the introduction

section of the revised manuscript.

[27] Li Fan, Sun Bo, Zhang C, Zhang L. Operation optimization for combined cooling, heating, and power system with condensation heat recovery. *Appl Energy* 2018; 230:305-16.

[28] Yari Mortaza, Ariyanfar Leyli, Aghdam EA. Analysis and performance assessment of a novel ORC based multigeneration system for power, distilled water and heat. *Renew Energy* 2018; 119:262-81.

[29] Bai Z, Liu T, Liu Q, Lei J, Gong L, Jin H. Performance investigation of a new cooling, heating and power system with methanol decomposition based chemical recuperation process. *Appl Energy* 2018; 229: 1152-63.

8. Originality of the paper should be emphasized clearly. How this study differs from related published papers?

The introduction section was rewritten in the revised manuscript. We analyzed the gaps existing in the published papers and provided our original solutions. The innovative features of the paper were summarized at the end of the introduction section.

There are three innovative features in this paper:

(1) We investigated a CO₂ Brayton cycle to prevent the risk of the decomposition of organic working fluid and provide power with high efficiency.

Organic Rankine cycles are used by many researchers to recover waste heat from internal combustion engines. Refrigerants are widely used as working in organic Rankine cycles but their decomposition temperatures are relatively low (200-300 °C).

When the high-temperature (above 450 °C) engine exhaust gas transfers heat with these organic working fluids, there are risks of decomposition of the organic working fluids. Though organic working fluids with high decomposition temperature were investigated by researchers, their flammability hindered their further applications. Some high-temperature loops for waste heat recovery were proposed by some researchers such as thermoelectric generator, steam Rankine cycle. But the thermal efficiency of the thermoelectric generator is low and the bulk of components in steam Rankine cycle is large, which might limit their application. Thus, we use a CO₂ Brayton cycle work as a high-temperature loop in the system to prevent the decomposition risks and provide power with high efficiency and compact structure.

(2) We developed a dual-pressure organic Rankine cycle to increase the utilization efficiency of the jacket water energy.

The previous internal combustion engine waste heat recovery systems, energy in jacket water was not fully utilized. In some studies, energy in jacket water was not utilized as all. Jacket water was mainly used to preheat the organic working fluid in the organic Rankine cycle based internal combustion engine waste heat recovery system. But there is mismatch of the mass flow rate of the organic working fluid in the preheater and evaporator. Thus, only a small part of the energy in jacket water was harnessed. We developed a dual-pressure organic Rankine cycle as a bottom cycle after the CO₂ Brayton cycle. Dual-pressure organic Rankine cycle can provide a large amount of power output. Moreover, organic working fluid in both high-pressure and low-pressure are preheated by the jacket water, which increases the utilization rate of

the jacket water.

(3) We developed an ejector refrigeration cycle to fully utilize the jacket water waste heat and provide refrigeration.

Combined cooling and power system driven by engine waste heat shows the advantages of high efficiency and multiple energy supply. Ammonia absorption refrigeration cycle are widely used for refrigeration in these systems. But the configuration of it is complex and it requires a high driven temperature. We coupled an ejector refrigeration cycle with the organic Rankine cycle system instead of ammonia absorption refrigeration cycle to provide refrigeration. The ejector refrigeration cycle has a simple structure and requires relatively low driven temperature. Thus, jacket water is used to drive the ejector refrigeration cycle to provide refrigeration and fully utilize the waste heat.

This paper differs from the previous paper mainly in the following four point:

(1) We used a CO₂ Brayton cycle to prevent the risk of decomposition of the organic working fluid and comprehensively analyzed its performance.

(2) We designed a dual-pressure organic Rankine cycle as a bottom cycle to increase the utilization efficiency of the jacket water.

(3) We used jacket water to drive the ejector refrigeration cycle to fully utilize the jacket water energy and provide refrigeration.

(4) We used exergoeconomic method to analyze the system performance.

9. Discuss and elaborate more on the exergy destruction rates of system and sub-systems. They were not written in the text.

Exergy destruction rates of the components in the system were calculated and presented in the revised manuscript from line 670 to 678 as following:

“Fig. 16 shows the exergy destruction of different components of the system under the optimization conditions. The largest exergy destruction takes place in the ORC turbine (41.26%), which is mainly caused by the mixing of the high-pressure working fluid and the low-pressure working fluid. Gas heater contributes 13.44% of the total exergy destruction. Three vapor generators take up 4.13%, 11.67% and 3.73% of the exergy destruction, respectively. The exergy destruction for the ejector is 5.61%, which is also caused by the working fluid mixing. For BC turbine, condenser 1, precoolers and preheaters, the exergy destruction are 3.31%, 4.64%, 3.69% and 3.65%, respectively. Other components contribute to the rest 4.87% of the exergy destruction.”

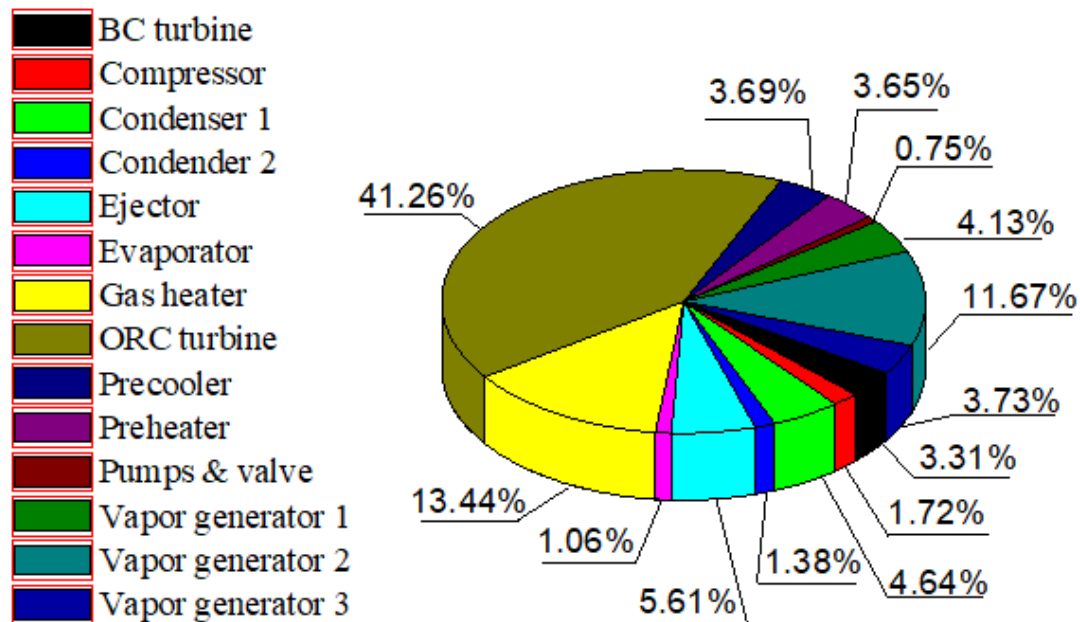


Fig. 16. Exergy destruction of different components

Performance analysis and optimization of a combined cooling and power system using low boiling point working fluid driven by engine waste heat

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Abstract

This paper develops a combined cooling and power system, which consists of a carbon dioxide Brayton cycle, a dual-pressure organic Rankine cycle and an ejector refrigeration cycle, to recover waste heat from exhaust gas and jacket water in internal combustion engines. Thermodynamic models of the system are performed and exergoeconomic methods are used to calculate the levelized exergy cost of the component products. Effects of seven parameters, including Brayton cycle turbine inlet temperature and inlet pressure, organic Rankine cycle turbine high-pressure side and low-pressure side inlet temperature and ejector primary inlet pressure, are evaluated. Single-objective optimization is carried out by means of genetic algorithm to obtain the minimum levelized exergy cost of system product. Results show that the increase of pressure at Brayton cycle turbine inlet and high-pressure and low-pressure side of the organic Rankine cycle turbine inlet contributes to the decrease of levelized

38 exergy cost of the system product. Optimization results show that minimum levelized
39 exergy cost for system product is 53.25 \$ (MWh)⁻¹. When system product levelized
40 exergy cost is minimum, system net power output, cooling capacity and exergy
41 efficiency are 374.37 kW, 188.63 kW and 37.31 %, respectively.

42 Nomenclature

<i>Latin symbols</i>		ρ	density, kg m ⁻³
A	area, m ²	μ	dynamic viscosity, m ² s ⁻¹
B_o	boiling number	η	efficiency, %
c	levelized average cost, \$(MWh) ⁻¹	δ	thickness, m
c_p	specific heat, kJ kg ⁻¹ K ⁻¹	<i>Subscriptes</i>	
C	cost rate, \$ year ⁻¹	1-31	state points
D	diameter, m	g1-g3	state points
e	exergy, kJ kg ⁻¹	w1-w3	state points
E	exergy flow rate, kJ s ⁻¹	Bt	Brayton cycle turbine
E_y	exergy flow rate per year, kJ year ⁻¹	BM	bare module
F	multiplying factor	cond	condenser
f	friction factor	comp	compressor
G	mass flow rate, kg s ⁻¹	D	destruction
h	enthalpy, kJ kg ⁻¹	elec	electricity
H	depth, m	es	equivalent diameter
i_{eff}	interest rate	ev	evaporation/evaporator
l	length, m	ex	exergy

M	mass flow rate, kg s^{-1}	F	fuel
n	lifetime, year	g	exhaust gas
Nu	Nusselt number	gh	gas heater
P	pressure, MPa	he	heat exchanger
Pr	Prandtl number	L	loss
P_t	center distance between tubes, m	l	liquid
P_r	reduced pressure	M	material
Q	heat transfer rate, kW	Ot	ORC turbine
Q_{cool}	cooling capacity, kW	P	product
q_m	average imposed wall heat flux, W m^{-2}	p1	pump 1
r_f	enthalpy of vaporization, kJ kg^{-1}	p2	pump 2
T	temperature, K	p3	pump 3
U	overall heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$	p4	pump 4
W	power, kW	pf	primary flow
W_y	annually power, MWh year^{-1}	prec	precooler
x	vapor quality	preh	preheater
Z	annually levelized cost value, $\text{\$ year}^{-1}$	s	shell
z	capital cost, k\$	t	tube
<i>Acronyms</i>		th	thermal
BC	Brayton cycle	turb	turbine
CBC	CO_2 Brayton cycle	vg	vapor generator
CCP	combined cooling and power	w	tube wall

<i>CRF</i>	capital recovery factor
<i>CEPCI</i>	chemical engineering plant cost index
DORC	dual-pressure organic Rankine cycle
ERC	ejector refrigeration cycle
GA	genetic algorithm
TEG	thermoelectric generator
AARC	ammonia absorption refrigeration cycle

Greek symbols

α	convection heat transfer coefficient, $\text{W m}^{-2} \text{K}^{-1}$
λ	heat conductivity, $\text{W m}^{-1} \text{K}^{-1}$

43 **1. Introduction**

44 Nowadays, internal combustion engine (ICE) is the major motive power source in
45 energy field, which are widely used in transport, construction, agriculture, etc. Over
46 50 % of the total transportation fuel is consumed by ICEs [1]. However, only 30-45 %
47 of the fuel energy is converted into effective power output, while the remaining
48 energy is discharged to the environment via exhaust gas, jacket water and charge air,
49 causing a large amount of waste fuel energy [2]. Thus, technology for waste heat
50 recovery from ICEs has drawn much interest of researchers in the last decade. Much
51 effort has been devoted to the study of organic Rankine cycle (ORC) based ICE waste
52 heat recovery system for its advantages of high efficiency and simple structure [3].

53 There are two important pathways that will lead to the improvement of the ORC

system for ICE waste heat recovery. One will be selecting organic working fluids which are suitable for the system under certain conditions. Another is to optimize the system configuration to make full use of the waste heat.

The work of selecting suitable organic working fluids for ORC was carried out by many researchers to improve the efficiency of the ICE waste heat recovery. Tian et al. [4] evaluated the performance of 20 different working fluids in an ORC system for ICE waste heat recovery. Rijpkema et al. [5] compared the performance of twelve working fluids in an ORC-based ICE waste heat recovery system to find the suitable candidate. Su et al. [6] developed a theoretical efficiency model about working fluids selecting for ORC-based ICE waste heat recovery system via strict mathematical derivation.

Configuration optimization in ORC-based ICE waste heat recovery system mainly focuses on reducing the system irreversible rate to fully utilize the engine waste heat. Vaja and Gambarotta [7] added a preheater and a recuperator separately to a simple ORC system to improve the performance for the ICE waste heat recovery. Kim et al. [8] proposed a novel single-loop ORC system to recovery engine waste heat. They employed two recuperators in series to heat the working fluid. Comparison showed that the net power output of the system was 35.6 % more than simple ORC system. Because that the maximum power output of single-loop ORC is lower than that of the dual-loop ORC system [9], more attention has been focused on dual-loop ORC based ICE waste heat recovery system in recent years. Wang et al. [10] modeled a dual-loop ORC system for engine waste heat recovery. The high-temperature loop absorbed heat

from exhaust gas and its residual heat acted as heat source for the low-temperature loop. Wang et al. [11] investigated a dual-loop ORC system for ICE waste heat recovery. The high-temperature loop absorbed heat from exhaust gas for the first time. Then the low-temperature loop absorbed heat from the residual heat of the exhaust gas to realize the cascading utilization of the waste heat. Huang et al. [12] proposed a complex dual-loop ORC system for engine waste heat recovery. The high-temperature loop absorbed heat from the exhaust gas and residual heat from both the exhaust gas and the high-temperature loop provided heat for the low-temperature loop.

When referring to heat transfer in the high-temperature loop, thermal stability of organic working fluid is necessary to be considered. In previous studies, refrigerants were most selected as working fluids. The decomposition temperatures of refrigerants are relatively low (200-300 °C) [13], while the temperature of exhaust gas is above 450 °C [14]. Direct heat transfer between high-temperature exhaust gas and refrigerant caused the risk of working fluid decomposition. Though high decomposition temperature working fluids such as siloxanes and alkanes were adopted by some researchers, their flammability hindered their further applications [15]. Though placing a heat transfer oil intermediate loop between the exhaust gas and the ORC system could address this issue [16], it would cause a large amount of the high-temperature waste heat unharnessed. Therefore, some other high-temperature loops for waste heat recovery were employed by researchers to couple with the ORC. Miller et al. [17] introduced thermoelectric generator (TEG) technology. High-temperature exhaust gas was first exploited by the TEG, then the cooled exhaust

gas could drive the ORC safely. But the energy conversion capacity of TEG is low because of the material limitation. Steam Rankine for its high efficiency and stable operation attracted much attention of researchers. Shu et al. [18] placed a steam Rankine cycle between the ORC and the exhaust gas. Yu et al. [19] coupled a steam Rankine cycle with an ORC for the ICE waste heat recovery. However, the large bulk of the components in steam Rankine cycle limits further applications (such as application in vehicles) [20]. Considering the requirement of high thermal efficiency and compact configuration, Brayton cycle could be a compromise solution. Brayton cycle with CO₂ (carbon dioxide) as working fluid has the advantage of low environmental impact and good thermodynamic performance [21]. Few studies about ORC system coupled with CO₂ Brayton cycle (CBC) for ICE waste heat recovery have been published. Though Zhang et al. [20] carried out some relevant studies, their attention was focused on comparing the performance of CBC, TEG and steam Rankine cycle when coupled with the same bottom ORC. Detailed analysis of the CBC was not given and the energy in jacket water was not harnessed.

Jacket water, though containing large amounts of energy [22], obtained little attention in the previous studies. For its relatively low temperature, jacket water was mainly used to preheat the organic working fluid in the ORC system. In the ORC-based ICE waste heat recovery system designed by Zhang et al. [23] jacket water was used to preheat the organic working fluid. Then the organic working fluid was heat by the high-temperature exhaust gas to vapor state and expanded in the ORC turbine. In Yang's [24] ICE waste heat recovery system, jacket water and secondary

exhaust gas were used to preheat the organic working fluids in ORC. In the dual-loop ORC based ICE waste heat recovery system investigated by Song et al. [25], jacket water was used to preheat the low-temperature-loop. Yu et al. [26] calculated the energy recovery efficiency from an ORC-based ICE waste heat recovery system. 75 % waste heat could be recovered from the exhaust gas, while only 9.5 % waste heat was recovery from jacket water. The relatively low utilization rate of jacket water energy in the ORC system is caused by the mismatch of working fluid mass flow rate in the preheater and the evaporator. Thus, the utilization of energy in jacket water could be further explored.

Multigeneration system driven by waste heat has drawn increasing interest of researchers in light of the trend towards reducing emissions, increasing the efficiency of energy use and providing variable energy. Li et al. [27] modeled a combined cooling, heating and power system to highly utilize the waste heat. Yari et al. [28] proposed a waste heat recovery system to provide power, distilled water and heat. Bai et al. [29] investigated a cooling, heating and power system driven by exhaust gas to recovery the waste heat. Combined cooling and power (CCP) systems driven by ICE waste heat were also investigated by some researchers. Chen et al. [30] designed an ammonia-water combined cooling and power system using the waste heat from the ICEs. Ammonia-water was heated by exhaust gas and jacket water. One part of the ammonia-water vapor flew into the turbine to provide power and the other part flew into the evaporator to provide refrigeration. Salek et al. [31] coupled an ammonia absorption refrigeration cycle and a bottoming Rankine cycle with internal

combustion engine to produce power and cooling capacity.

Ammonia absorption refrigeration cycle (AARC) were widely used in the combined cooling and power system for its large refrigeration output. However, the complex cycle structure and high driven temperature requirement of AARC might sometimes limit its applications. On the contrary, ejector refrigeration cycle (ERC) exhibits the advantages of easy maintenance and high reliability [32] and it can be driven by low-temperature heat source such as the jacket water. Thus, ICE waste heat recovery system with ERC driven by jacket water not only simultaneously generate power and refrigeration but also fully utilized the jacket water waste heat.

In this study, a combined cooling and power system is developed, which comprises a CO₂ Brayton cycle, a dual-pressure organic Rankine (DORC) cycle and an ejector refrigeration cycle. The CO₂ Brayton cycle absorbs heat from the high-temperature exhaust gas directly to prevent the decomposition risk. The turbine exhaust in the CO₂ Brayton cycle and the engine exhaust gas after heat transfer are respectively regarded as the heat sources for the high-pressure side and low-pressure side of the dual-pressure ORC, realizing the cascading utilization of exhaust gas. Meanwhile, organic working fluids in high-pressure side and low-pressure side are both preheated by jacket water which increases the mass flow rate of the organic working fluid preheated by jacket water. What's more, the ejector refrigeration cycle is adopted to produce refrigeration and fully utilize waste heat in jacket water. Thermodynamic and exergoeconomic analysis is carried out to examine the effects of key parameters on system performance. Then a system optimization is conducted to obtain the minimum

levelized exergy cost for the system product by means of genetic algorithm (GA).

The innovative features of this paper are as follow:

- A CO₂ Brayton cycle is investigated to prevent the risk of decomposition of organic working fluid and provide power.
- A novel dual-pressure ORC system is developed to cascading utilize the waste heat in exhaust gas and jacket water and provide large amounts of power output.
- An ejector refrigeration cycle driven by jacket water is designed to provide refrigeration and fully utilize the jacket water waste heat.

2. System description

The combined cooling and power system is shown in Fig. 1. The system integrates a dual-pressure organic Rankine cycle with a CO₂ Brayton cycle and an ejector refrigeration, which can produce power and refrigeration simultaneously. High-temperature gas heat from the ICE enters the gas heater to provide heat for the CBC. In the CBC, compressor compresses the CO₂ to a supercritical state. The high-pressure CO₂ flows into the gas heater to absorb heat. Then CO₂ with high temperature and high pressure expands through the BC turbine to produce power.

After expanding in the BC turbine, the high-temperature exhaust CO₂ flows into vapor generator 2 to heat the organic working fluid. High-pressure side organic working fluid heated by the CO₂ then flows into the ORC turbine to produce power. Meanwhile low-pressure side organic working fluid absorbs heat from the secondary engine exhaust gas in vapor generator 1 and then enters ORC turbine to produce

185 power.

186 Exhaust vapor from the ORC turbine is cooled by condenser 1 to liquid state and
187 pressured by pump 1. Jacket water with large mass flow rate is used to preheat the
188 organic working fluid in the preheater. The preheated organic working fluid then
189 separates. One part of the fluid is pumped by pump 3 to the vapor generator 1 to cycle
190 in the low-pressure side. The other part is pumped by pump 2 to the vapor generator 2
191 to cycle in the high-pressure side.

192 The jacket water then flows into vapor generator 3 to provide heat for the ejector
193 refrigeration cycle. After the condensation process in condenser 2, liquid working
194 fluid is divided into two separated parts. One part of the fluid is pumped to the vapor
195 generator 3 to absorb heat from jacket water and then becomes superheated vapor.
196 The other part of the working fluid flows through the throttle valve to become
197 low-pressure vapor-liquid mixture. The low-pressure mixture enters the evaporator to
198 produce cooling capacity when absorbing heat from the environment and become
199 low-pressure vapor. After that, the superheated vapor mixes with the low-pressure
200 vapor in the ejector. The mixed working fluid enters the condenser 2 to be condensed
201 to liquid.

202 R245fa is selected as the working fluid for the organic Rankine cycle and the
203 ejector refrigeration cycle because of the great thermodynamic performance [33] and
204 the low environment effects [34].

205 3. System model

206 Several assumptions are made to simplify the simulation of the system, which are: (1)
207 the system keeps a steady state; (2) the heat and friction in the system are not
208 considered; (3) the pressure losses in the vapor generators, preheater, evaporator,
209 condensers and pipes are neglected; (4) the gas temperature at the outlet of the vapor
210 generator 1 is higher than 110 °C [35], considering the low gas dew point temperature;
211 (5) the working fluids at the outlet of the condensers and the preheater are saturated
212 liquids, and the evaporator outlet state is saturated vapor; (6) the process through the
213 throttle valve is isenthalpic.

214 3.1. Energy model

215 The net power of the CO₂ Brayton cycle is expressed as:

$$216 W_{BC} = W_{Bt} - W_{comp} \quad (1)$$

217 The net power of the DORC is given as:

$$218 W_{ORC} = W_{Ot} - W_{p1} - W_{p2} - W_{p3} \quad (2)$$

219 The cooling capacity of the ERC is given as:

$$220 Q_{cool} = M_{cool} \cdot (h_{21} - h_{20}) \quad (3)$$

221 The net power output of the whole system is calculated as:

$$222 W_{net} = W_{ORC} + W_{BC} - W_{p4} \quad (4)$$

223 The thermal efficiency of the system is given as:

$$224 \eta_{th} = \frac{W_{net} + Q_{cool}}{M_{g1} \cdot (h_{g1} - h_{g3}) + M_{w1} \cdot (h_{w1} - h_{w3})} \quad (5)$$

225 The detailed energy model equations of each component are listed in Table 1. Note

that there are two expanding processes in the ORC turbine. The high-pressure vapor expands in the turbine and then mixes with the vapor from vapor generator 1. After that, the mixed vapor expands in the turbine for the second time.

3.2.Exergy model

The energy model of the system is based on the first law of thermodynamics. From the viewpoint of the first law, it is equivalent for work and heat. Nevertheless, according to the second law of the thermodynamics, the irreversibility of work and heat is different. The exergy is used to quantifies the difference between them. The exergy model of the system is based on a dead state (the ambient condition in this study). Definition of exergy is given as:

$$e = (h - h_0) - T_0 \cdot (s - s_0) \quad (6)$$

where h_0 , T_0 and s_0 are the parameters under the ambient conditions.

The exergy flow rate in this study is given by:

$$E = M \cdot e \quad (7)$$

In this study, all the components in the system are associated directly or indirectly with fuel of other heat sources, such as exhaust as and jacket water. The heat sources provide exergy for the components to operate. For each component, there is an exergy balance equation, being expressed as [36]:

$$E_F = E_p + E_d + E_L \quad (8)$$

where E_F , E_P , E_D , E_L donate the rate of exergy for the component fuel, the rate of exergy for component product, the rate of component exergy destruction and the rate of component exergy loss, respectively.

The details of the exergy balance equations for each component are listed in Table 1.

The exergy efficiency represents the degree of the utilization of the waste heat in the system, being expressed as:

$$\eta_{ex} = \frac{W_{net} + E_{cool}}{E_{g1} - E_{g3} + E_{w1} - E_{w3}} \quad (9)$$

where E_{cool} is the exergy rate of the cooling process, being expressed as:

$$E_{cool} = E_{25} - E_{24} \quad (10)$$

3.3.Capital cost calculation

A method of modeling the capital costs of main components is used in this study. According to Ref. [37], the bare module cost of the components is calculated as the basic cost. The basic cost of the components includes the direct project cost (such as component cost, material cost of the installation, etc.) and the indirect project cost (like the taxes, insurance engineering expenses, etc.). The bare module cost of the components is calculated under basic conditions. For deviations from the based conditions, multiplying factors (the specific component type, the specific system pressure and the specific material of construction) are added in the calculation to correct the results. In the following text, equations from Eq. (11) to Eq. (21) are proposed in Ref. [37].

266 Axial turbines (BC turbine and ORC turbine) are used in this study. The bare
267 module cost equation of the turbine is:

$$268 \log_{10} C_{\text{turb}}^0 = K_{1,\text{turb}} + K_{2,\text{turb}} \cdot \log_{10} W + K_{3,\text{turb}} \cdot (\log_{10} W)^2 \quad (11)$$

269 where $K_{i,\text{turb}}$ are constants corresponding to the turbine type; and W is the power
270 output of the turbine.

271 Turbines used in this study are made of carbon steel (CS) and operate under high
272 pressure. Thus, a multiplying factor is used to correct the result. The capital cost of
273 the turbine is given as:

$$274 C_{\text{turb}} = F_{\text{BM,turb}} \cdot C_{\text{turb}}^0 \quad (12)$$

275 where $F_{\text{BM,turb}}$ is the multiplying factor corresponding to the working conditions of the
276 turbine.

277 Reciprocating pumps are used in this study. The bare module cost equation of the
278 pumps is given as:

$$279 \log_{10} C_{\text{pump}}^0 = K_{1,\text{pump}} + K_{2,\text{pump}} \cdot \log_{10} W + K_{3,\text{pump}} \cdot (\log_{10} W)^2 \quad (13)$$

280 where $K_{i,\text{pump}}$ are the constants corresponding to the pump type; and W is the power
281 input of the pump.

282 Pumps used in this study are made of stainless steel (SS) and work under high
283 pressure. Thus, multiplying factors are used to correct the bare module cost. The
284 capital cost of the pump is given as:

$$285 C_{\text{pump}} = (B_{1,\text{pump}} + B_{2,\text{pump}} \cdot F_{\text{M,pump}} \cdot F_{\text{P,pump}}) \cdot C_{\text{pump}}^0 \quad (14)$$

where $B_{i, \text{pump}}$ are the constants corresponding to the type of the pump; $F_{M, \text{pump}}$ is the material factor of the pump and $F_{P, \text{pump}}$ is the pressure factor of the pump. The equation of the pressure factor is given as:

$$\log_{10} F_{P, \text{pump}} = C_{1, \text{pump}} + C_{2, \text{pump}} \cdot \log_{10} P_{\text{pump}} + C_{3, \text{pump}} \cdot (\log_{10} P_{\text{pump}})^2 \quad (15)$$

where $C_{i, \text{pump}}$ are the constants corresponding to the type of the pump; and P_{pump} is the pressure of the pump under working conditions.

Axial compressor is used in this study. The bare module cost equation of the compressor is given as:

$$\log_{10} C_{\text{comp}}^0 = K_{1, \text{comp}} + K_{2, \text{comp}} \cdot \log_{10} W + K_{3, \text{comp}} \cdot (\log_{10} W)^2 \quad (16)$$

where $K_{i, \text{comp}}$ are the constants corresponding to the type of the compressor; W is the power input of the compressor.

The compressor is made of carbon steel (CS) and works under high pressure.

Correction equation of the bare module cost is given as:

$$C_{\text{comp}} = F_{\text{BM, comp}} \cdot C_{\text{comp}}^0 \quad (17)$$

where $F_{\text{BM, comp}}$ is the constant corresponding to the type of the compressor.

Shell-and-tube heat exchangers (gas heater, vapor generators, precooler, preheater, evaporator and condensers) are used in this study. The bare module cost equation of the heat exchanger is given as:

$$\log_{10} C_{\text{he}}^0 = K_{1, \text{he}} + K_{2, \text{he}} \cdot \log_{10} A + K_{3, \text{he}} \cdot (\log_{10} A)^2 \quad (18)$$

where $K_{i, \text{he}}$ are the constants corresponding to the type of the heat exchanger; A is the heat transfer area of the heat exchanger. The calculation of the heat exchanger areas is presented in Appendix A.

Heat exchangers used in this study are made of carbon steel (CS) and work under different pressure. Multiplying factors are needed to correct the results, the equation is given as:

$$C_{he} = (B_{1,he} + B_{2,he} \cdot F_{M,he} \cdot F_{P,he}) \cdot C_{he}^0 \quad (19)$$

where $B_{i,he}$ are the constants corresponding to the type of the heat exchanger. $F_{M,he}$ and $F_{P,he}$ are the material factor and pressure factor, respectively. The pressure factor is obtained from the following equation:

$$\log_{10} F_{P,he} = C_{1,he} + C_{2,he} \cdot \log_{10} P_{he} + C_{3,he} \cdot (\log_{10} P_{he})^2 \quad (20)$$

where $C_{i,he}$ are the constants corresponding to the type of the heat exchanger; P_{he} is the designed working pressure for the heat exchanger.

The values of the constants mentioned above for the main components are listed in Appendix B.

The calculation of the bare module cost depends on past records or published correlations for price information. It is necessary to update the costs because of the inflation. This can be achieved by the following equation:

$$C_2 = C_1 \cdot \left(\frac{I_2}{I_1} \right) \quad (21)$$

where C is the purchased cost and I is the cost index. The subscript 1 refers to base time when cost is known and subscript 2 refers to time when cost is desired. The *CEPCI* (Chemical Engineering Plant Cost Index) is employed to calculate the inflation. The values of $CEPCI_{2016}$ and $CEPCI_{ref,2001}$ are 541.7 and 397, respectively [38].

329 3.4.Exergoeconomic model

330 Exergoeconomic is a branch of engineering which combines the thermodynamic
331 analysis and economic principles. Thermodynamic performance and economic cost of
332 the system are all taken into consideration.

333 To find the relationship between the present value of the expenditure and the
334 equivalent annually levelized costs, the capital recovery factor (CRF) is employed,
335 being expressed as [36]:

$$336 \quad Z_i = CRF \cdot C_i \quad (22)$$

$$337 \quad CRF = \frac{i_{\text{eff}} \cdot (1 + i_{\text{eff}})^n}{(1 + i_{\text{eff}})^n - 1} \quad (23)$$

338 where i_{eff} is the effective discount rate with a value of 0.05 [39]; and n is the lifetime
339 of the CCP system, being assumed as 30 years [40].

340 In order to calculate the equivalent annually levelized costs, the annual working
341 time of the system is assumed as 8000 h [41]. Then the annual exergy rates and annual
342 power output or consumption are obtained.

343 In a steady system, there are a number of entering and outing working fluid streams
344 and heat and work interactions with the surroundings. In exergoeconomic analysis,
345 each flowing stream is associated with a levelized exergy cost. The equations to
346 calculate the cost of the stream product are given as:

$$347 \quad C_{\text{in}} = c_{\text{in}} \cdot E_{\text{y,in}} \quad (24)$$

$$348 \quad C_{\text{out}} = c_{\text{out}} \cdot E_{\text{y,out}} \quad (25)$$

$$349 \quad C_{\text{work}} = c_{\text{work}} \cdot W_y \quad (26)$$

$$C_{\text{heat}} = c_{\text{heat}} \cdot E_{y,\text{heat}} \quad (27)$$

where c denotes levelized exergy cost of the streams; $E_{y,\text{in}}$ and $E_{y,\text{out}}$ are the exergy transfer rate of the stream flowing in and out of a component; W_y and $E_{y,\text{heat}}$ are the power and the heat transfer rate of the components considering the annual working time.

The cost balance equation applied to the k th system component is given as:

$$\sum_{\text{out}} C_{\text{out},k} + C_{w,k} = C_{\text{heat},k} + \sum_{\text{in}} C_{\text{in},k} + Z_k \quad (28)$$

Details of the cost balance equation are listed in Table 2.

The levelized exergy cost for system product is chosen to indicate the exergoeconomic performance, being expressed as [42,43]:

$$c_{\text{product}} = c_{\text{capital}} + c_{\text{fuel}} \quad (29)$$

where c_{capital} is the capital-cost-related part of the levelized exergy cost for the system product, being expressed as:

$$c_{\text{capital}} = \frac{Z_{\text{total}}}{W_{\text{net}} + E_{\text{cool}}} \quad (30)$$

where c_{fuel} is the fuel-cost-related part of the levelized exergy cost for the system product, being expressed as:

$$c_{\text{fuel}} = \frac{c_{\text{Bt}} \cdot W_{y,\text{comp}} + c_{\text{Ot}} \cdot W_{y,\text{pump1}} + c_{\text{Ot}} \cdot W_{y,\text{pump2}} + c_{\text{Ot}} \cdot W_{y,\text{pump3}} + c_{\text{Ot}} \cdot W_{y,\text{pump4}}}{W_{\text{net}} + E_{\text{cool}}} \quad (31)$$

where c_{Bt} and c_{Ot} are the levelized exergy cost for the BC turbine power output and the ORC turbine power output, which are calculated in Table 3. Likewise, they can be expressed as the capital-cost-related part and the fuel-cost-related part, given by Eq. (32) and Eq. (33).

$$c_{Bt} = \frac{Z_{Bt}}{W_{y,Bt}} + \frac{c_3 \cdot (E_{y,3} - E_{y,4})}{W_{y,Bt}} \quad (32)$$

$$c_{Ot} = \frac{Z_{Ot}}{W_{y,Ot}} + \frac{c_{10} \cdot (E_{y,10} + E_{y,11} - E_{y,12})}{W_{y,Ot}} \quad (33)$$

In addition, the levelized exergy cost for the condensers and the precooler is equal to zero, being given by:

$$c_{26}=c_{28}=c_{30}=0 \quad (34)$$

The levelized exergy cost for the exhaust gas as well as the jacket water is zero, being expressed as:

$$c_{g1}=c_{w1}=0 \quad (35)$$

3.5.Internal combustion engine

In this study, the engine selected [7] is a 12-cylinder 4-stroke supercharged engine. The main designed parameters of the engine are listed in Table 3. The composition of the engine exhaust gas is presented in Table 4. The thermal load of the engine exhaust gas is about 1700 kW and 1000 kW can be obtained from the engine jacket water.

3.6.Simulation conditions for the system

The thermodynamic parameters of the working fluid are calculated under the environment of MATLAB with the help of REFPROP 9.1 [44]. The basic conditions of simulation for the CCP system are listed in Table 5.

Seven key parameters : BC turbine inlet temperature ($T_{Bt,in}$), BC turbine inlet pressure ($P_{Bt,in}$), inlet temperature at the high-pressure side of ORC turbine ($T_{Ot,in,h}$), inlet pressure at the high-pressure side of ORC turbine ($P_{Ot,in,h}$), inlet temperature at

the low-pressure side of ORC turbine ($T_{Ot, in, l}$), inlet pressure at the low-pressure side of ORC turbine ($P_{Ot, in, l}$) and the ejector primary inlet pressure ($P_{ej, in}$), are chosen to analyze the thermodynamic and exergoeconomic performance of the system. When one parameter is investigated to analyze the system performance, other parameters are maintained constants based on the conditions in Table 5.

In the thermodynamic aspect, the net power output of the CO₂ Brayton cycle (W_{BC}), net power output of the DORC (W_{ORC}), net power of the whole system (W_{net}), cooling capacity of the system (Q_{cool}) and the exergy efficiency of the system (η_{exergy}) are selected to reflect the system performance. Levelized exergy cost for the BC turbine power output (c_{Bt}), levelized exergy cost for the ORC turbine power output (c_{Ot}), levelized exergy cost the system product ($c_{product}$) and the system capital cost ($z_{capital}$) are chosen to represent the exergoeconomic performance.

4. Results and discussion

The influence of the BC turbine inlet temperature ($T_{Bt, in}$) on the output and the exergy efficiency of the system are shown in Fig. 2. The net power output of the CBC increases with the rise of $T_{Bt, in}$. That can be explained by the large decrease of the compressor power consumption. With the increase of the CO₂ temperature at the BC turbine inlet, the mass flow rate of CO₂ decreases, leading to the decrease of the compressor power consumption. Although the drop of CO₂ mass flow rate would cut down the BC turbine power output, the decrease quantity of compressor power consumption is larger than the decrease of the BC turbine power output. Thus, the

large decrease of the compressor power consumption determines the increase trend of the CBC net power output.

It is presented that the net power output of the DORC increases with the rise of the BC turbine inlet temperature. Since the residual heat in exhaust CO_2 acts as the heat source for the high-pressure side of DORC, the temperature rise of the exhaust CO_2 , caused by the rise of $T_{\text{Bt,in}}$, would offer more heat for the bottom cycle, which causes the increase of the mass flow rate of the organic working fluid in the high-pressure side of DORC. Hence, the power output of the ORC turbine increases, leading to the increase of the net power output of the DORC.

With the increase of $T_{\text{Bt,in}}$, the cooling capacity of the ERC decreases, as shown in Fig. 2. The increase of the organic working fluid mass flow rate in DORC would absorb more heat from jacket water, resulting in the decrease of energy available for the ERC. As a result, less secondary flow working fluid from the evaporator is entrained to the ejector, resulting the decrease of the cooling capacity of the CCP system.

The increase of the CBC net power output and the DORC net power output account for the increase of the net power output of the whole CCP system. Though the cooling capacity of the ERC is large, it produces only a small amount of exergy. The decrease of the exergy output caused by the cooling capacity drop can be made up by the increase of the power exergy output. Thus, the exergy efficiency of the system increases.

The influences of the BC turbine inlet temperature on the levelized exergy cost and

the system capital cost of the system are shown in Fig. 3. The levelized exergy cost for the BC turbine power output (c_{Bt}) drops with the rise of the BC turbine inlet temperature ($T_{Bt,in}$). That can be explained by the decrease of the capital-cost-related part of c_{Bt} . The capital-cost-related part of c_{Bt} decreases with the decrease of cost of compressor, which is cut down by the drop of the compressor power consumption. The increase of the ORC turbine power output causes the decrease of both the capital-cost related part and the fuel-cost-related part of c_{Ot} , resulting in the decrease of c_{Ot} .

The system capital cost ($z_{capital}$) rises with the rise of $T_{Bt,in}$. The large increase of the ORC turbine power output increases the cost of the ORC turbine. Moreover, the increase of the mass flow rate of the organic working fluid in the DORC causes the increase of cost for the vapor generator 2 and the preheater. Though the cost of compressor decreases, it can't change the ascending trend of the total system capital.

It can be obtained in Fig. 3 that the levelized exergy cost for the system product ($c_{product}$) decreases with the rise of $T_{Bt,in}$. The decline in levelized exergy cost for the BC turbine and ORC turbine power output, according to Eq. (31), would cause the decrease of the fuel-cost related part of $c_{product}$. Though the increase of $z_{capital}$ would cut down the capital-cost-related part of $c_{product}$, the impact of levelized exergy cost for the BC turbine and ORC turbine is greater, which leads to the descending trend of $c_{product}$.

The influence of the BC turbine inlet pressure ($P_{Bt,in}$) on the output and the exergy efficiency of the system are shown in Fig. 4. The net power output of the CBC increases with the increase of $P_{Bt,in}$, which can be explained by the rise of enthalpy

drop of the CO₂ in the BC turbine. Though the rise of $P_{Bt, in}$ requires more compressor power consumption, the increase of the BC turbine power output is larger in quantity than the consumption, which leads to the increase of the CBC net power output.

The net power output of the DORC decreases with the rise of $P_{Bt, in}$. On the one hand, the temperature of the exhaust CO₂ at the BC turbine outlet decreases with the increase of $P_{Bt, in}$. Thus, less heat is offered to the high-pressure cycle of DORC, resulting in the decrease of the high-pressure cycle power output. On the other hand, the increase of $P_{Bt, in}$ causes the increase of the compressor power consumption, which results in the rise of the CO₂ temperature at the compressor outlet. Thus, less heat is released in the gas heater and more heat is provided to the low-pressure cycle of DORC, which leads to the increase of the low-pressure cycle power output. However, the increase of the power output in low-pressure side is smaller than the decrease of the power output in the high-pressure side. Thus, the net power of the DORC decreases slightly.

The cooling capacity of the system increases with the increase of $P_{Bt, in}$. Just like the variation of the power output, the decrease of the mass flow rate in the high-pressure side of DORC is larger than the increase of mass flow rate in the low-pressure side.

Therefore, the total mass flow rate in the DORC decreases, resulting in the reduction of heat provided for the ejector refrigeration cycle. Thus, the cooling capacity of the ERC decreases.

The net power output of the whole system increases with the increase of the BC turbine inlet pressure. Though the net power output of the DORC decreases, the

increase of CBC net power output is much larger. Thus, the net power output of the CCP system increases with the increase of $P_{Bt, in}$. The exergy efficiency of the system likewise has the same rising trend.

The influences of the BC turbine inlet pressure ($P_{Bt, in}$) on the levelized exergy cost and the system capital cost of the system are depicted in Fig. 5. The levelized exergy cost for the BC turbine output c_{Bt} increases with the rise of the $P_{Bt, in}$, which can be explained by the variations of the capital-cost-related part and the fuel-cost-related part. The increase of $P_{Bt, in}$ causes the increase of cost for both the BC turbine and the compressor, which lead to the rise of the two related parts.

The levelized exergy cost for the ORC turbine product (c_{Ot}) increases with the rise of $P_{Bt, in}$. The decrease of the mass flow rate in the DORC causes that less exergy is produced in vapor generator 2, causing the increase of the fuel-related part of c_{Ot} . Therefore, the levelized exergy cost for the ORC turbine (c_{Ot}) increases.

The system capital cost ($z_{capital}$) increases with the rise of ($P_{Bt, in}$). The increase of the mass flow rate in the ERC causes the rise of capital cost for the evaporator and vapor generator 3, which combined with the rise of the BC turbine cost and compressor cost accounts for the system capital cost rise.

The levelized exergy cost for the system product decreases with the rise of $P_{Bt, in}$ as presented in Fig. 5. According to Eq. (31), the rise of the c_{Ot} , c_{Bt} would cause the rise of the fuel-cost-related part of $c_{product}$. However, because of the large increase of the system net power output, the capital-cost-related part and the fuel-cost-related part decrease actually, which determines the decrease of $c_{product}$.

The influence of inlet temperature at the high-pressure side of ORC turbine ($T_{Ot, in, h}$) on the output and the exergy efficiency of the system are shown in Fig. 6. The net power output of the CBC remains unchanged since thermal parameters in dual-pressure ORC are irrelevant to the thermodynamic performance of the CBC.

The net power output of the DORC decreases with the increase of $T_{Ot, in, h}$. Though the increase of the vapor temperature could lead to the rise of the enthalpy drop in the ORC turbine, it would also cause the decrease of the mass flow rate in the high-pressure side, whose impact is greater than that of the enthalpy drop. Therefore, the power output of the DORC decreases.

The cooling capacity of the ejector refrigeration cycle increases with the rise of $T_{Ot, in, h}$. More heat is provided for the ERC because of the decrease of the mass flow rate in the DORC, leading to the increase of the mass flow rate in vapor generator 3. Thus, more secondary flow from the evaporator is entrained into the ejector, resulting in the increase of the cooling capacity.

The net power output of the CCP system decreases with the rise of $T_{Ot, in, h}$. The unchanged CBC power output and the drop of the DORC power output determine the decrease of the net power output of the CCP system. The exergy efficiency of the system as well drops with the increase of the increase of $T_{Ot, in, h}$.

The influences of inlet temperature at the high-pressure side of ORC turbine on the levelized exergy cost and the system capital cost of the system are presented in Fig. 7. The levelized exergy cost for the ORC turbine output (c_{Ot}) increases with the increase of $T_{Ot, in, h}$. The reason is that the two related parts of c_{Ot} increase with the drop of the

ORC turbine power output.

The levelized exergy cost for the BC turbine power output (c_{Bt}) increases with the rise the $T_{Ot, in, h}$. Since the decrease of the mass flow rate in the high-pressure side of DORC, the exergy generated in the vapor generator 2 decreases, causing the increase of the levelized exergy cost of the vapor. Thus, the increase levelized exergy cost of the vapor, which is heated by the BC turbine residual heat, causes the increase of the levelized exergy cost for the exhaust CO₂. According to Eq. (32), the fuel-cost-related part of c_{Bt} increases, leading to the increase of c_{Bt} .

The system capital cost ($z_{capital}$) decreases with the increase of $T_{Ot, in, h}$. The decrease of the DORC power output causes the drop of the ORC turbine cost, which leads to the descending trend of $z_{capital}$.

The levelized exergy cost for the system product ($c_{product}$) increases with the rise of $T_{Ot, in, h}$, as shown in Fig. 7. The increase of the levelized exergy cost for the BC turbine and ORC turbine power output cause the rise of fuel-cost-related part of $c_{product}$. Meanwhile, the large decrease of the net power of the CCP system causes the increase of the capital-cost-related part. The two increase parts determine the rise of $c_{product}$.

The influences of the inlet pressure at the high-pressure side of ORC turbine ($P_{Ot, in, h}$) on the output and exergy efficiency of the system are presented in Fig. 8. The net power of the CBC keeps unchanged because of the unchanged thermal parameters in the cycle.

The net power output of the DORC increase with the rise of $P_{Ot, in, h}$. The increase of the evaporation pressure cuts down the latent heat of the organic working fluid,

which causes the increase of the mass flow rate in the high-pressure side of DORC.

As a result, the net power output of the ORC turbine increases, leading to the increase of the net power output of the DORC.

Considering the increase of the DORC net power output and the unchanged CBC net power output, the net power output of the whole system increases. Also, the exergy efficiency of the system increases.

The increase of the mass flow rate in the DORC absorbs more heat from the jacket water in the preheater. Thus, less heat is released in the vapor generator 3, causing the decrease of the mass flow rate of the working fluid in the ERC. As a result, the cooling capacity of the system decreases.

The influence of the inlet pressure at the high-pressure side of the ORC turbine ($P_{Ot, in, h}$) on the levelized exergy cost and system capital cost of the system are presented in Fig. 9. The large increase of the ORC turbine power output accounts for the decrease of the levelized exergy cost for the ORC turbine power output (c_{Ot}). The increase of the mass flow rate in the high-pressure side of DORC means that more exergy in the vapor is generated by the vapor generator 2, which leads to the decrease of its levelized exergy cost. Thus, the levelized exergy cost for the BC turbine exhaust CO_2 , which provides heat for the vapor, decreases with the vapor levelized exergy cost. Moreover, the drop of the CO_2 levelized exergy cost causes the decrease of the fuel-cost-related part of c_{Bt} , which further results in the decrease of c_{Bt} .

The increase of the ORC turbine power output and the increase of mass flow rate in the DORC cause the increase of cost for the turbine and the vapor generator 2, leading

to the rise of the system capital cost.

The levelized exergy cost for the system product (c_{product}) decreases with the increase of $P_{\text{Ot, in, h}}$. The decrease of c_{Ot} and c_{Bt} account for the decrease of the fuel-cost-related part of the levelized exergy for the system product. The impact of c_{Ot} and c_{Bt} is greater than that of the system capital cost whose rise would result in the increase of the capital-cost-related part of c_{product} . Thus, the levelized exergy cost of the system product (c_{product}) shows a descending trend.

The influences of the inlet temperature at the low-pressure side of ORC turbine ($T_{\text{Ot, in, l}}$) on the output and the exergy efficiency of the system are presented in Fig. 10. Parameters changes in the DORC are irrelevant to the thermodynamic performance of the CBC. Thus, the net power of the CBC remains unchanged.

The net power output of the DORC decreases with the increase of $T_{\text{Ot, in, l}}$. The increase of the inlet temperature causes the decrease of the mass flow rate in the low-pressure side of the DORC, leading to the decrease of the DORC net power output.

Considering the decrease of the DORC net power output and the unchanged CBC net power output, the net power output of the whole system decreases. Likewise, the exergy efficiency of the system decreases.

The cooling capacity of the ejector refrigeration cycle increases with the increase of $T_{\text{Ot, in, l}}$. The decrease of the mass flow rate in the low-pressure side means that more heat is offered to the ERC. Thus, the mass flow rate of the working fluid in the vapor generator 3 increases and more working fluid is entrained to the ejector from the

evaporator, which leads to the increase of the refrigeration cycle.

The influence of inlet temperature at the low-pressure side of the ORC turbine ($T_{Ot, in, 1}$) on the levelized exergy cost and system capital cost of the system are presented in Fig. 11. The levelized cost for the BC turbine power output increase with the increase of $T_{Ot, in, 1}$. The decrease of the mass flow rate in the vapor generator 1 leads to the drop of the vapor exergy output, which results in the increase of the levelized exergy cost for the vapor. The levelized exergy cost for vapor in vapor generator 2, which is the equal to that of the vapor in vapor generator 1, increases as a result, causing the increase of the levelized exergy cost of the exhaust CO_2 after the BC turbine. Thus, the fuel-cost-related part of c_{Bt} increases, resulting in the rise of c_{Bt} .

The levelized exergy cost for the ORC turbine (c_{Ot}) increases with the increase of $T_{Ot, in, 1}$. That can be explained by the increase of the levelized exergy cost of the ORC low-pressure inlet vapor and the decrease of the power output of the ORC turbine power output. Both the fuel-cost-related part and the capital-cost-related part of c_{Ot} increases.

The decrease of the mass flow rate and the ORC turbine power output cause the decrease of the vapor generator 1 cost and the turbine cost. Thus, the capital cost of the system decreases.

The levelized exergy cost for the system product increases with the increase of $T_{Ot, in, 1}$. The increase of c_{Bt} and c_{Ot} cause the increase of the fuel-cost-related part of the levelized exergy cost for the system product. Though, the decrease of the system capital cost causes the decrease of the capital-cost-related part, its effect is less

important. Thus, the increase of the fuel-cost-related part determines the increase of the levelized exergy cost for the system product.

The influences of the inlet temperature of the low-pressure side of the ORC turbine ($P_{Ot, in, 1}$) on the output of the exergy efficiency of the system are shown in Fig. 12. The net power of the CBC keeps unchanged with the increase of the low evaporation pressure. The reason is that the thermodynamic of the CBC is irrelevant to the thermal parameters in DORC.

The net power output of the DORC increases with the rise of $P_{Ot, in, 1}$. The increase of enthalpy drop of the organic working fluid in the low-pressure side, which is caused by the rise of $P_{Ot, in, 1}$, results in the increase of the power output of the low-pressure side. Though mass flow rate in the low-pressure side would decrease, its impact is less important than that of the enthalpy drop. Thus, the net power output of DORC increases.

The unchanged CBC power output and the increase of the DORC power accounts for the increase of the system net power output and exergy efficiency of the system.

The cooling capacity increases **slightly** with the increase of $P_{Ot, in, 1}$. Because of the decrease of the mass flow rate in DORC, less heat is released in the preheater and more heat is provided in vapor generator 3. Thus, the mass flow rate of the working fluid in the ERC increases, resulting in the **slight** increase of the cooling capacity.

The influences of inlet pressure at the low-pressure side of the ORC turbine ($P_{Ot, in, 1}$) on the levelized exergy cost and system capital cost of the system are shown in Fig. 13. The levelized exergy cost for the ORC turbine power output decreases with the

increase of $P_{Ot, in, 1}$. That can be explained by the decrease of the vapor generator 1 cost, caused by the decrease of the mass flow rate in DORC, and the increase of the DORC power output. Both the capital-cost-related part and the fuel-cost-related part of c_{Ot} decrease.

The levelized exergy cost for the BC turbine power output decreases with of $P_{Ot, in, 1}$. The decrease of the c_{Ot} causes the drop of levelized exergy cost for the vapor in vapor generator 2, which is heated by the residual heat in the BC turbine exhaust CO_2 . Thus, the levelized exergy cost of the exhaust CO_2 decreases, which further leads to the drop of the fuel-cost-related part of c_{Bt} . Therefore, the levelized exergy cost for the BC turbine power output (c_{Bt}) decreases, as shown in Fig. 13.

The increase of the ORC turbine power output causes the increase of the ORC turbine cost. Meanwhile, the increase of the cooling capacity causes the increase of the heat transfer area in the evaporator which requires the rise of the evaporator cost. Thus, the capacity cost of the system increases.

The levelized exergy cost for the system product decreases with the increase of $P_{Ot, in, 1}$. The decrease of the levelized exergy cost for the BC turbine power output and ORC turbine power cause the decrease of the fuel-cost-related part of the system levelized exergy cost, which determined the decrease of the levelized exergy cost for the system product.

The influence of ejector primary inlet pressure ($P_{ej, in}$) on the output and the exergy efficiency of the system are shown in Fig. 14. Thermal parameter changes in the ERC can't affect the thermodynamic performance of the CBC and DORC. Thus, the net

power output of the two cycles remain unchanged. With the increase of the ejector primary inlet pressure, the power consumption of pump 4 increases, leading to the slight decrease of the power output of the whole system.

The increase of the ejector primary inlet pressure causes the increase of the entrainment ratio of the ejector. Thus, more secondary flow is entrained to the ejector from the evaporation, leading to the increase of the cooling capacity.

With the increase of the ejector primary inlet pressure, the power consumption of pump 4 increases gradually. At first, the exergy loss in pump 4 is smaller than the exergy produced by the cooling capacity. Then, with the increase of the pump power consumption, the exergy loss in pump 4 becomes larger than the cooling exergy increase. Thus, the exergy efficiency for the system increases at first and then decreases with the increase of the ejector primary inlet pressure.

The influence of the ejector primary inlet pressure on the levelized exergy cost and the system capital cost of the system are presented in Fig. 15. The increase of the ejector primary inlet pressure can't affect the power output of the BC turbine and the ORC turbine. Thus, the levelized exergy cost for the BC turbine and the ORC power output remain unchanged.

The increase of the pump power consumption results in the increase of the pump 4 cost. The increase of the mass flow rate in the evaporator causes the increase of the evaporator cost. Thus, the system capital cost increases, which leads to the increase of the capital-cost-related part of c_{product} . As a result, the levelized exergy cost for the system increases.

The parametric analysis reveals the potential of optimization for the CCP system. With the increase of the BC turbine inlet temperature, the net power output of the system increases while the cooling capacity decreases. With the increase of the inlet temperature at the high-pressure side of ORC turbine, the net power output of the system decreases while the cooling capacity increases. In this study, seven key parameters (BC turbine inlet temperature, BC turbine inlet pressure, inlet temperature at the high-pressure side of ORC turbine, inlet pressure at the high-pressure side of ORC turbine, inlet temperature at the low-pressure side of ORC turbine, inlet pressure at the low-pressure side of ORC turbine and the ejector primary inlet pressure) are chosen as the variables to optimize the system. The ranges of these parameters are listed in Table 6.

Considering that the levelized exergy cost reflects the thermodynamic and the exergoeconomic aspect of the system, the levelized exergy cost for the system product is selected as the objective function and genetic algorithm is selected to conduct the single-objective optimization.

Genetic algorithm (GA) is an optimization method based on the natural biological evaluation. [45] It simulates the natural genetic rules and searches the optimization result in all the generation. The control parameters of the GA are listed in Table 7.

The optimization results of GA are listed in Table 8. It can be obtained that the minimum levelized exergy cost for the system product c_{product} is $53.25 \text{ } \$(\text{MWh})^{-1}$. The net power output, exergy efficiency of the CCP system are 374.37 kW, 37.31 % respectively. The inlet pressure at the high-pressure side of ORC turbine is 1.85 MPa.

Meanwhile, it can be evidenced from Fig. 8 and 9 that the highest output power (about 374.37 kW), exergy efficiency (about 37.31 %) and the lowest levelized exergy cost (about 53.25 \$(MWh)⁻¹) at the highest inlet pressure at the high-pressure side ORC turbine (about 1.85 MPa). The results shown in Fig. 8 and 9 are close to the optimization results. The inlet pressure at the high-pressure side ORC turbine is varied while other six parameters are kept as constants in Fig. 8 and 9. Thus, inlet pressure at the high-pressure side ORC turbine plays a more important role than other six parameters when determining the performance of the system. When the inlet pressure at the high-pressure side ORC turbine is close to the highest permitted pressure, the system performance is close to the optimization performance.

Fig. 16 shows the exergy destruction of different components of the system under the optimization conditions. The largest exergy destruction takes place in the ORC turbine (41.26 %), which is mainly caused by the mixing of the high-pressure vapor and the low-pressure vapor. Gas heater contributes 13.44 % of the total exergy destruction. Three vapor generators take up 4.13 %, 11.67 % and 3.73 % of the exergy destruction, respectively. The exergy destruction for the ejector is 5.61 %, which is also caused by the working fluid mixing. For BC turbine, condenser 1, precoolers and preheater, the exergy destruction are 3.31 %, 4.64 %, 3.69 % and 3.65 %, respectively. Other components contribute to the rest 4.87 % of the exergy destruction.

5. Conclusion

In this paper, a combined cooling and power system is developed. Seven

parameters are selected to analyze the thermodynamic and exergoeconomic performance of the system. Single-objective optimization is carried out with the help of GA. The conclusions of the study are presented as follows:

(1) In the CBC, the increase of $T_{Bt,in}$ and $P_{Bt,in}$ contribute to the increase of the system exergy efficiency and the decrease of the levelized exergy cost for the system product.

(2) In the DORC, the increase of $T_{Ot,in}$ and $T_{Ot,in,l}$ would cause the decrease of the system exergy efficiency and the increase of the levelized exergy cost for the system product. Meanwhile, the increase of $P_{Ot,in,h}$ and $P_{Ot,in,l}$ would result in the increase of the exergy efficiency and the decrease of the levelized exergy cost.

(3) In the ERC, the increase of $P_{ej,in}$ would cause the increase of the refrigeration capacity and the decrease of the system net power output.

(4) Single -objective optimization results show that the minimum levelized exergy cost for the system product is obtain as $53.25 \text{ } \$(\text{MWh})^{-1}$ with net power output of 374.37 kW, cooling capacity of 188.63 kW and system exergy efficiency of 37.31 %.

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Appendix A

This section shows the calculation of the heat transfer area in the heat exchangers

740 used in this study.

741 All the heat exchangers used in this study are **shell-and-tube** heat exchanger. The
742 thermodynamic properties of the working fluid vary with the heat transfer process.
743 Thus, to calculate the heat transfer area actually, the heat transfer processes are
744 discretized into a lot of small sections. In each section, the heat transfer area is small
745 and the thermodynamic properties are assumed to be constant.

746 For each section the heat transfer area is calculated as:

$$747 \quad A_i = \frac{Q_i}{(\Delta T_i \cdot U_i)} \quad (B1)$$

748 where ΔT_i is the log-mean temperature difference (LMTD) and U_i is the overall heat
749 transfer coefficient.

$$750 \quad \frac{1}{U_i} = \frac{1}{\alpha_{t,i}} + \frac{\delta}{\lambda} + \frac{1}{\alpha_{s,i}} \quad (B2)$$

751 In Eq. (B2) δ and λ represent the thickness of the tube and the thermal conductivity
752 of the tube wall, respectively. $\alpha_{t,i}$ is the convection heat transfer coefficient in the tube
753 side and $\alpha_{s,i}$ is the convection heat transfer coefficient in the shell side.

754 For different heat transfer process, the convection heat transfer coefficient has
755 different format. We classify the heat transfer processes into single-phase heat transfer
756 process and two-phase heat transfer process. In gas heater, precoolers and the preheaters,
757 single-phase heat transfer process happens. In evaporators, two-phase heat transfer
758 process occurs. In vapor generators and the condensers, both the single-phase and the
759 two-phase heat transfer process happen.

760 In the single-phase heat transfer process, the convection heat transfer coefficient in

the tube side and the shell side are expressed as [46]:

$$\alpha_{ti} = \frac{\lambda \cdot Nu}{D_i} \quad (B3)$$

$$\alpha_{si} = 0.36 \left(\frac{\lambda}{D_{es}} \right) \cdot \left(\frac{D_{es} \cdot G_s}{\mu} \right)^{0.55} \cdot Pr^{\frac{1}{3}} \cdot \left(\frac{\mu}{\mu_w} \right)^{0.14} \quad (B4)$$

In Eq. (B3), the Nusselt number is calculated as [47,48]:

$$Nu = \left[\frac{(f/8) \cdot Re \cdot Pr}{12.7(f/8)^{0.5} \cdot (Pr^{2/3} - 1) + 1.07} \right], \text{ for } Re < 10^4 \quad (B5)$$

$$Nu = \left[\frac{(f/8) \cdot (Re - 1000) \cdot Pr}{12.7(f/8)^{0.5} \cdot (Pr^{2/3} - 1) + 1.07} \right], \text{ for } 10^4 < Re < 5 \times 10^6 \quad (B6)$$

where f is the Darcy friction factor, Re is the Reynolds and Pr is the Prandtl number.

In Eq. (B4), D_{es} is the equivalent diameter of the shell, being expressed as:

$$D_{es} = \frac{1.10Pt^2}{D_{out,i}} - D_{out,i} \quad (B7)$$

where Pt is the center distance between the tubes.

Evaporation and condensation are two different two-phase heat transfer processes.

In this study, the cold organic working fluid flows in the tubes of the heat exchangers.

The convection heat transfer coefficient of evaporation and condensation are

expressed as [49,50]:

$$\alpha_{ev,i} = 0.023 \left[\frac{G(1-x)}{\mu_l} \right]^{0.8} \cdot Pr_l^{0.4} \cdot \frac{\lambda_l}{d} \cdot \left[1 + 3000Bo^{0.86} + 1.12 \left(\frac{x}{1-x} \right)^{0.75} \cdot \left(\frac{\rho_l}{\rho_v} \right)^{0.41} \right] \quad (B8)$$

$$\alpha_{cond,i} = 0.023 \left[\frac{G(1-x)}{\mu_l} \right]^{0.8} \cdot Pr_l^{0.4} \cdot \frac{\lambda_l}{d} \cdot \left[(1-x)^{0.8} + \frac{3.8x^{0.76}(1-x)^{0.04}}{P_r^{0.38}} \right] \quad (B9)$$

In Eq. (B9), P_r is the reduced pressure. In Eq. (B8) Bo is the boiling number, being

expressed as:

$$Bo = \frac{q_m}{G \cdot r_f} \quad (B10)$$

Appendix B

The constants for component capital cost calculation are list in Table B1.

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920

921 **Figure captions**

922 **Fig. 1.** Schematic diagram of the CCP system

923 **Fig. 2.** Influences of BC turbine inlet temperature on the output and the exergy
924 efficiency of the system.

925 **Fig. 3.** Influences of BC turbine inlet temperature on the levelized exergy cost and the
926 system capital cost of the system.

927 **Fig. 4.** Influences of BC turbine inlet pressure on the output and the exergy efficiency
928 of the system.

929 **Fig. 5.** Influences of BC turbine inlet pressure on the levelized exergy cost and the
930 system capital cost of the system.

931 **Fig. 6.** Influences of inlet temperature at the high-pressure side of ORC turbine on the
932 output and the exergy efficiency of the system.

933 **Fig. 7.** Influences of inlet temperature at the high-pressure side of ORC turbine on the
934 levelized exergy cost and the system capital cost of the system.

935 **Fig. 8.** Influences of inlet pressure at the high-pressure side of ORC turbine on the
936 output and the exergy efficiency of the system.

937 **Fig. 9.** Influences of inlet pressure at the high-pressure side of ORC turbine on the
938 levelized exergy cost and the system capital cost of the system.

939 **Fig. 10.** Influences of inlet temperature at the low-pressure side of ORC turbine on the
940 output and the exergy efficiency of the system.

941 **Fig. 11.** Influences of inlet temperature at the low-pressure side of ORC turbine on the
942 levelized exergy cost and system capital cost of the system.

943 **Fig. 12.** Influences of inlet pressure at the low-pressure side of ORC turbine on the
944 output and the exergy efficiency of the system.

945 **Fig. 13.** Influences of inlet pressure at the low-pressure side of ORC turbine on the
946 levelized exergy cost and system capital cost of the system.

947 **Fig. 14.** Influences of ejector primary inlet pressure on the output and the exergy
948 efficiency of the system.

949 **Fig. 15.** Influences of ejector primary inlet pressure on the levelized exergy cost and
950 the system capital cost of the system.

951 **Fig. 16.** Exergy destruction of different components

952

Component	Energy equation	E_F	E_P	E_D	E_L
Gas heater	$M_{g1} \cdot (h_{g1} - h_{g2}) = M_2 \cdot (h_3 - h_2)$	$E_{g1} - E_{g2}$	$E_3 - E_2$	$E_{g1} + E_2 - E_3 - E_{g2}$	/
BC turbine	$W_{Bt} = M_3 \cdot (h_3 - h_4) = M_3 \cdot (h_3 - h_{4s}) \cdot \eta_{Bt}$	$E_3 - E_4$	W_{Bt}	$E_3 - E_4 - W_{Bt}$	/
Vapor generator 2	$M_4 \cdot (h_4 - h_5) = M_9 \cdot (h_{10} - h_9)$	$E_4 - E_5$	$E_{10} - E_9$	$E_4 + E_9 - E_5 - E_{10}$	/
Precooler	$M_1 \cdot (h_5 - h_1) = M_{26} \cdot (h_{27} - h_{26})$	/	/	$E_5 + E_{26} - E_1 - E_{27}$	$E_{27} - E_{26}$
Compressor	$W_{comp} = M_1 \cdot (h_2 - h_1) = M_1 \cdot (h_{2s} - h_1) / \eta_{comp}$	W_{comp}	$E_2 - E_1$	$E_1 - E_2 + W_{comp}$	/
Vapor generator 1	$M_{g2} \cdot (h_{g2} - h_{g3}) = M_8 \cdot (h_{11} - h_8)$	$E_{g2} - E_{g1}$	$E_{11} - E_8$	$E_{g2} + E_8 - E_{11} - E_{g3}$	/
ORC turbine	$W_{Ot} = M_{10} \cdot (h_{10} - h_{12}) + M_{11} \cdot (h_{11} - h_{12})$	$E_{10} + E_{11} - E_{12}$	W_{Ot}	$E_{10} + E_{11} - E_{12} - W_{Ot}$	/
Condenser 1	$M_{12} \cdot (h_{12} - h_{13}) = M_{28} \cdot (h_{29} - h_{28})$	/	/	$E_{12} + E_{28} - E_{13} - E_{29}$	$E_{29} - E_{28}$
Pump 1	$W_{p1} = M_{13} \cdot (h_{14} - h_{13}) = M_{13} \cdot (h_{14s} - h_{13}) / \eta_{p1}$	W_{p1}	$E_{14} - E_{13}$	$E_{13} - E_{14} + W_{p1}$	/
Preheater	$M_{15} \cdot (h_{15} - h_{14}) = M_{w1} \cdot (h_{w1} - h_{w2})$	$E_{w1} - E_{w2}$	$E_{15} - E_{14}$	$E_{w1} + E_{14} - E_{15} - E_{w2}$	/
Pump 2	$W_{p2} = M_7 \cdot (h_9 - h_7) = M_7 \cdot (h_{9s} - h_7) / \eta_{p2}$	W_{p2}	$E_9 - E_7$	$E_7 - E_9 + W_{p2}$	/
Pump 3	$W_{p3} = M_6 \cdot (h_8 - h_6) = M_6 \cdot (h_{8s} - h_6) / \eta_{p3}$	W_{p3}	$E_8 - E_6$	$E_6 - E_8 + W_{p3}$	/
Vapor generator 3	$M_{23} \cdot (h_{23} - h_{22}) = M_{w2} \cdot (h_{w2} - h_{w3})$	$E_{w2} - E_{w3}$	$E_{23} - E_{22}$	$E_{w2} + E_{22} - E_{23} - E_{w3}$	/
Condenser 2	$M_{16} \cdot (h_{16} - h_{17}) = M_{30} \cdot (h_{31} - h_{30})$	/	/	$E_{16} + E_{30} - E_{17} - E_{31}$	$E_{31} - E_{30}$
Valve	$h_{19} = h_{20}$	/	/	$E_{19} - E_{20}$	/
Pump 4	$W_{p4} = M_{22} \cdot (h_{22} - h_{18}) = M_{22} \cdot (h_{22s} - h_{18}) / \eta_{p4}$	W_{p4}	$E_{22} - E_{18}$	$E_{18} - E_{22} + W_{p4}$	/
Ejector	$M_{16} \cdot h_{16} = M_{23} \cdot h_{23} + M_{21} \cdot h_{21}$	$E_{23} + E_{21}$	E_{16}	$E_{23} + E_{21} - E_{16}$	/
Evaporator	$M_{20} \cdot (h_{21} - h_{20}) = M_{24} \cdot (h_{24} - h_{25})$	$E_{20} - E_{21}$	$E_{25} - E_{24}$	$E_{20} + E_{24} - E_{21} - E_{25}$	/

Component	Cost balance	Auxiliary relation
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Gas heater	$c_{g2} \cdot E_{y,g2} + c_3 \cdot E_{y,3} = c_{g1} \cdot E_{y,g1} + c_2 \cdot E_{y,2} + Z_{gh}$	$c_{g1}=c_{g2}=0$
Vapor generator 2	$c_5 \cdot E_{y,5} + c_{10} \cdot E_{y,10} = c_4 \cdot E_{y,4} + c_9 \cdot E_{y,9} + Z_{vg,2}$	$c_4=c_5$
BC turbine	$c_4 \cdot E_{y,4} + c_{Bt} \cdot W_{y,Bt} = c_3 \cdot E_{y,3} + Z_{Bt}$	$c_4=c_3$
Precooler	$c_1 \cdot E_{y,1} + c_{26} \cdot E_{y,26} = c_5 \cdot E_{y,5} + c_{27} \cdot E_{y,27} + Z_{prec}$	$c_1=c_5$
Compressor	$c_2 \cdot E_{y,2} = c_1 \cdot E_{y,1} + c_{elec,1} \cdot W_{y,comp} + Z_{comp}$	$c_{elec,1}=c_{Bt}$
Vapor generator 1	$c_{g3} \cdot E_{y,g3} + c_{11} \cdot E_{y,11} = c_{g2} \cdot E_{y,g2} + c_8 \cdot E_{y,8} + Z_{vg,1}$	$c_{g2}=c_{g3}$
ORC turbine	$c_{12} \cdot E_{y,12} + c_{Ot} \cdot W_{y,Ot} = c_{11} \cdot E_{y,11} + c_{10} \cdot E_{y,10} + Z_{Ot}$	$c_{10}=c_{11}=c_{12}$
Pump 1	$c_{14} \cdot E_{y,14} = c_{13} \cdot E_{y,13} + c_{elec,3} \cdot W_{y,pump1} + Z_{pump1}$	$c_{elec,3}=c_{Ot}$
Condenser 1	$c_{13} \cdot E_{y,13} + c_{29} \cdot E_{y,29} = c_{28} \cdot E_{y,28} + c_{12} \cdot E_{y,12} + Z_{cond1}$	$c_{13}=c_{12}$
Preheater	$c_{w2} \cdot E_{y,w2} + c_{15} \cdot E_{y,15} = c_{w1} \cdot E_{y,w1} + c_{14} \cdot E_{y,14} + Z_{preh}$	$c_{w1}=c_{w2}=0$
Pump 2	$c_9 \cdot E_{y,9} = c_7 \cdot E_{y,7} + c_{elec,2} \cdot W_{y,pump2} + Z_{pump2}$	$c_{elec,2}=c_{Ot}$
Pump 3	$c_8 \cdot E_{y,8} = c_6 \cdot E_{y,6} + c_{elec,3} \cdot W_{y,pump3} + Z_{pump3}$	$c_{elec,3}=c_{Ot}$
Vapor generator 3	$c_{w3} \cdot E_{y,w3} + c_{23} \cdot E_{y,23} = c_{w2} \cdot E_{y,w2} + c_{22} \cdot E_{y,22} + Z_{vg,3}$	$c_{w3}=c_{w2}$
Valve	/	$c_{19}=c_{20}$
Pump 4	$c_{22} \cdot E_{y,22} = c_{18} \cdot E_{y,18} + c_{elec,3} \cdot W_{y,pump4} + Z_{pump4}$	$c_{elec,4}=c_{Ot}$
Condenser 2	$c_{17} \cdot E_{y,17} + c_{31} \cdot E_{y,31} = c_{30} \cdot E_{y,30} + c_{16} \cdot E_{y,16} + Z_{cond2}$	$c_{16}=c_{17}$
Ejector	$c_{16} \cdot E_{y,16} = c_{23} \cdot E_{y,23} + c_{21} \cdot E_{y,21}$	/
Evaporator	$c_{21} \cdot E_{y,21} + c_{25} \cdot E_{y,25} = c_{20} \cdot E_{y,20} + c_{24} \cdot E_{y,24} + Z_{ev}$	$c_{20}=c_{21}$

955 **Table 3** Main parameters of the engine [7]

Parameters	Value
Power output (kW)	2928
Rotation (r(min) ⁻¹)	1000

Exhaust gas temperature (°C)	470
Exhaust gas mass flow rate (kg s ⁻¹)	4.35
Temperature of jacket water (°C)	90/79
Mass flow rate of jacket water (kg s ⁻¹)	25

956 **Table 4** Composition of the exhaust gas [7]

Composition	Molecular (g(mol) ⁻¹)	Fraction (%)
O ₂	32.00	9.3
CO ₂	44.00	9.1
H ₂ O	18.01	7.4
N ₂	28.01	74.2

957 **Table 5** Condition of simulation for the CCP system

Parameter	Value
Ambient temperature (°C)	20
Ambient pressure (MPa)	0.101
Compressor inlet temperature (°C)	35
BC turbine inlet temperature (°C)	400
BC turbine inlet pressure (MPa)	18
BC turbine outlet pressure (MPa)	8
Inlet temperature at the high-pressure side of ORC turbine (°C)	150
Inlet pressure at the high-pressure side of ORC turbine (MPa)	1.6
Inlet temperature at the low-pressure side of ORC turbine (°C)	100
Inlet pressure at the low-pressure side of ORC turbine (MPa)	1.0

Outlet pressure of pump 1 (MPa)	0.9
Ejector primary inlet pressure (MPa)	0.4
Terminal temperature difference at gas heater outlet (°C)	100
Pinch point temperature difference in vapor generator 1 (°C)	30
Pinch point temperature difference in vapor generator 2 (°C)	30
Pinch point temperature difference in vapor generator 3 (°C)	25
Condensation temperature of condenser 1 (°C)	30
Condensation temperature of condenser 2 (°C)	30
Evaporation temperature of evaporator (°C)	5
Isentropic efficiency of BC turbine (%)	80
Isentropic efficiency of ORC turbine (%)	80
Isentropic efficiency of compressor (%)	80
Isentropic efficiency of pump 1 (%)	75
Isentropic efficiency of pump 2 (%)	75
Isentropic efficiency of pump 3 (%)	75
Inlet temperature of cooling water (°C)	20

958 **Table 6** Parameters for GA

Parameter	Operation range
BC turbine inlet temperature (°C)	330-440
BC turbine inlet pressure (MPa)	15-20
Inlet temperature at the high-pressure side of ORC turbine (°C)	130-180
Inlet pressure at the high-pressure side of ORC turbine (MPa)	1.4-2

Inlet temperature at the low-pressure side of ORC turbine (°C)	90-150
Inlet pressure at the low-pressure side of ORC turbine (MPa)	0.9-1.3
Ejector primary inlet pressure (MPa)	0.3-1

959 **Table 7** Control parameters of GA

Tuning parameters	Value
Population size	20
Mutation probability	0.01
Crossover probability	0.8
Stop generation	200

960 **Table 8** Single-objective optimization results

Parameter	Value
BC turbine inlet temperature (°C)	425.46
BC turbine inlet pressure (MPa)	20.00
Inlet temperature at the high-pressure side of ORC turbine (°C)	144.32
Inlet pressure at the high-pressure side of ORC turbine (MPa)	1.85
Inlet temperature at the low-pressure side of ORC turbine (°C)	100.03
Inlet pressure at the low-pressure side of ORC turbine (MPa)	1.26
Ejector primary inlet pressure (MPa)	0.54
Net power output (kW)	374.37
Cooling capacity (kW)	188.63
Exergy efficiency (%)	37.31
Levelized exergy cost (\$ (MWh) ⁻¹)	53.25

961 **Table B1** Constants for component costs [37]

Constant	Value	Constant	Value	Constant	Value
$B_{1,he}$	1.63	$K_{3,pump}$	0.1538	$C_{3,he}$	0.08183
$B_{2,he}$	1.66	$K_{1,turb}$	2.7051	$C_{1,pump}$	-0.3635
$B_{1,pump}$	1.89	$K_{2,turb}$	1.4398	$C_{2,pump}$	0.3957
$B_{2,pump}$	1.35	$K_{3,turb}$	-0.1776	$C_{3,pump}$	-0.0026
$K_{1,he}$	4.3247	$K_{1,comp}$	2.2897	$F_{M,he}$	1.0
$K_{2,he}$	-0.3030	$K_{2,comp}$	1.3604	$F_{BM,turb}$	3.5
$K_{3,he}$	0.1634	$K_{3,comp}$	-0.1027	$F_{BM,comp}$	2.7
$K_{1,pump}$	3.3892	$C_{1,he}$	0.03881	$F_{M,pump}$	2.2
$K_{2,pump}$	0.0536	$C_{2,he}$	-0.11272		

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efficiency of the system.

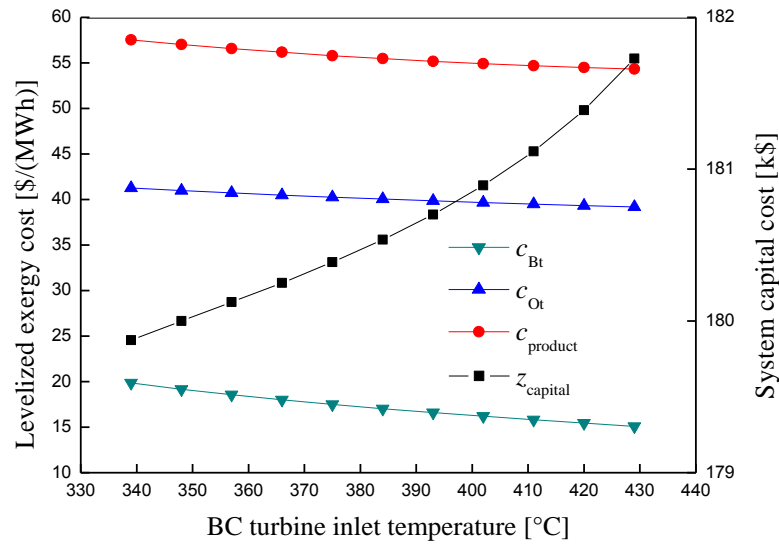


Fig. 3. Influences of BC turbine inlet temperature on the levelized exergy cost and the system capital cost of the system.

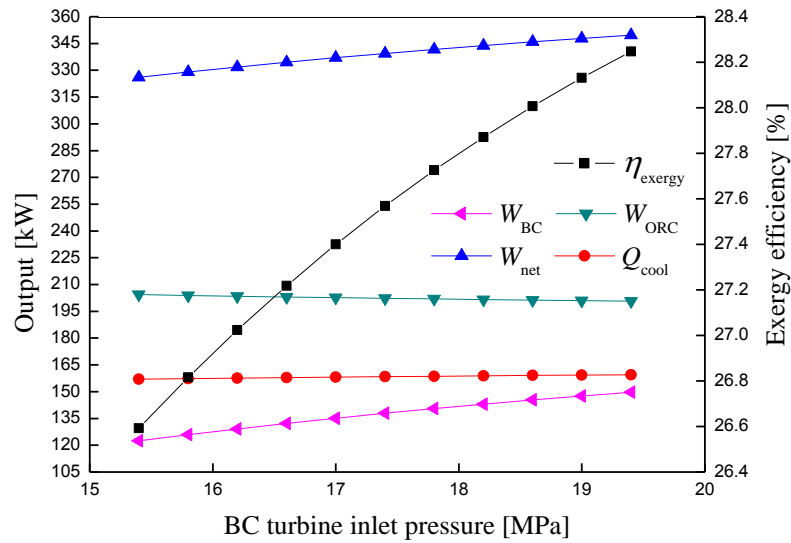


Fig. 4. Influences of BC turbine inlet pressure on the output and the exergy efficiency of the system.

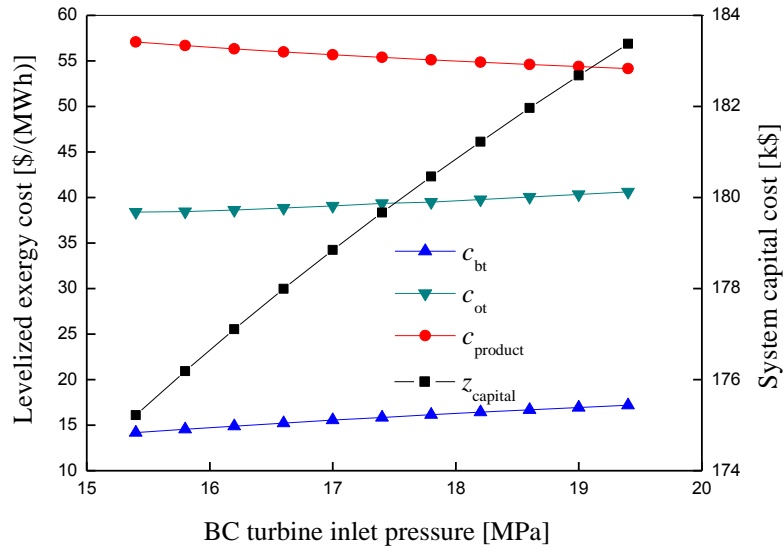


Fig. 5. Influences of BC turbine inlet pressure on the levelized exergy cost and the system capital cost of the system.

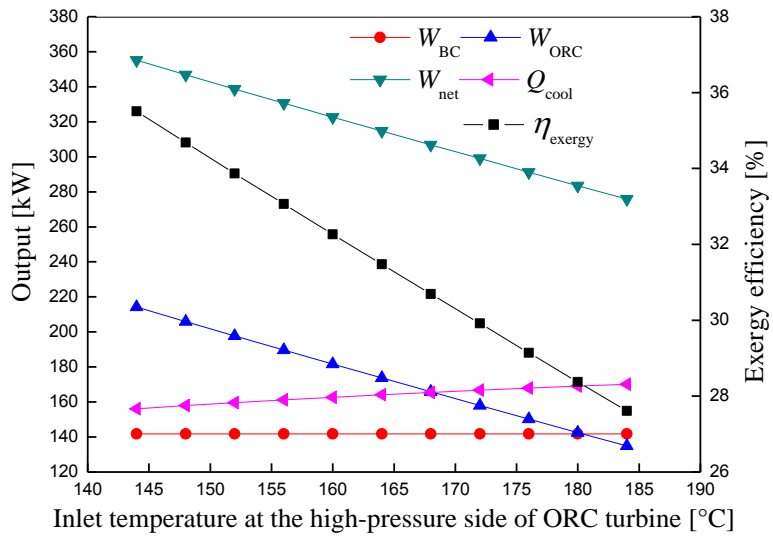


Fig. 6. Influences of inlet temperature at the high-pressure side of ORC turbine on the output and the exergy efficiency of the system.

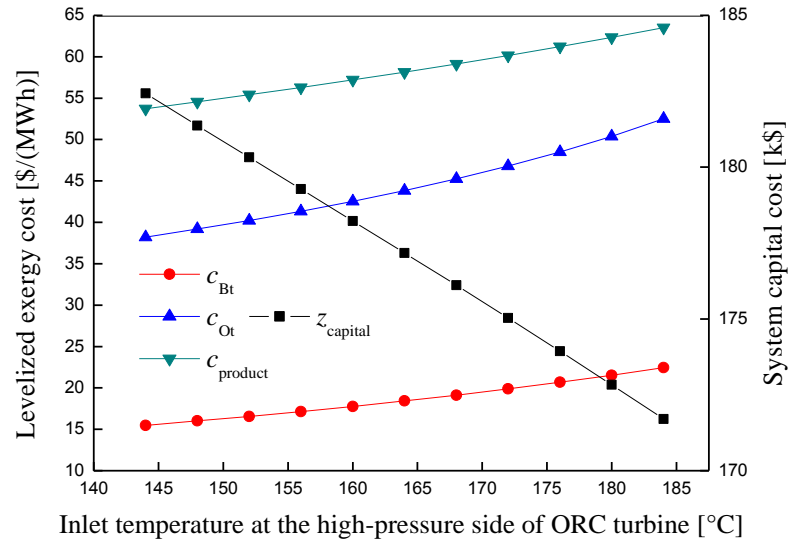


Fig. 7. Influences of inlet temperature at the high-pressure side of ORC turbine on the levelized exergy cost and the system capital cost of the system.

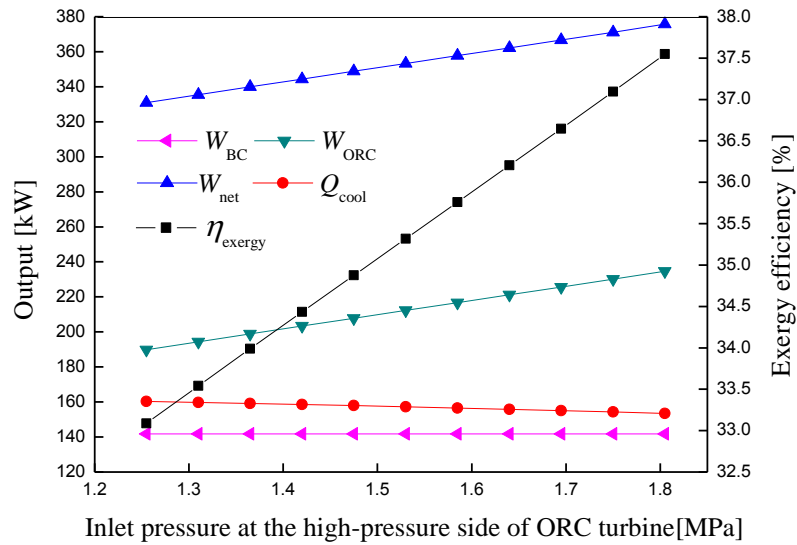


Fig. 8. Influences of inlet pressure at the high-pressure side of ORC turbine on the output and the exergy efficiency of the system.

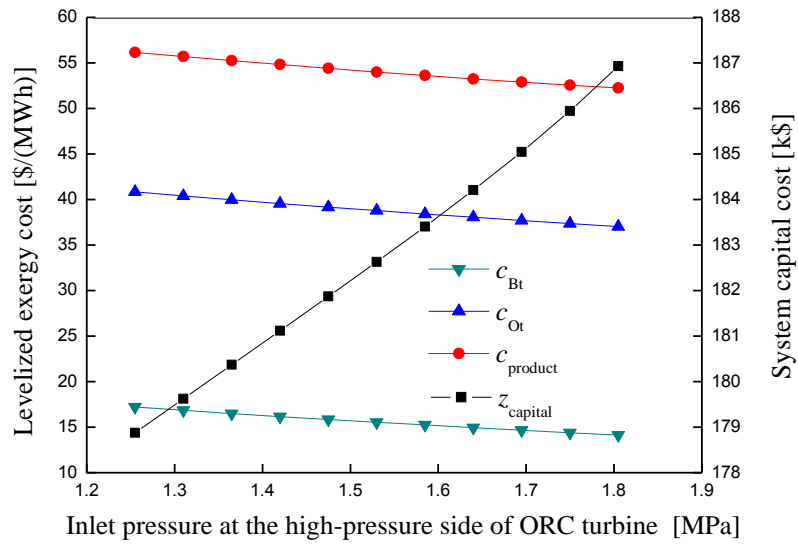


Fig. 9. Influences of inlet pressure at the high-pressure side of ORC turbine on the levelized exergy cost and the system capital cost of the system.

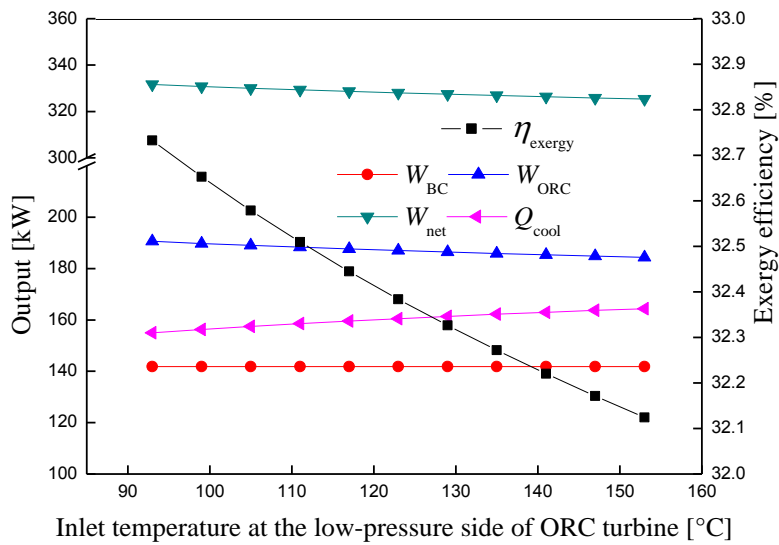


Fig. 10. Influences of inlet temperature at the low-pressure side of ORC turbine on the output and the exergy efficiency of the system.

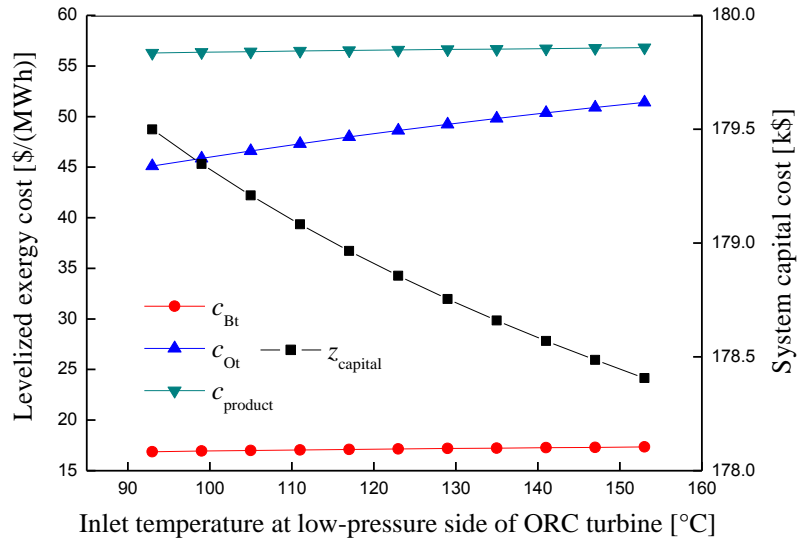


Fig. 11. Influences of inlet temperature at the low-pressure side of ORC turbine on the levelized exergy cost and system capital cost of the system.

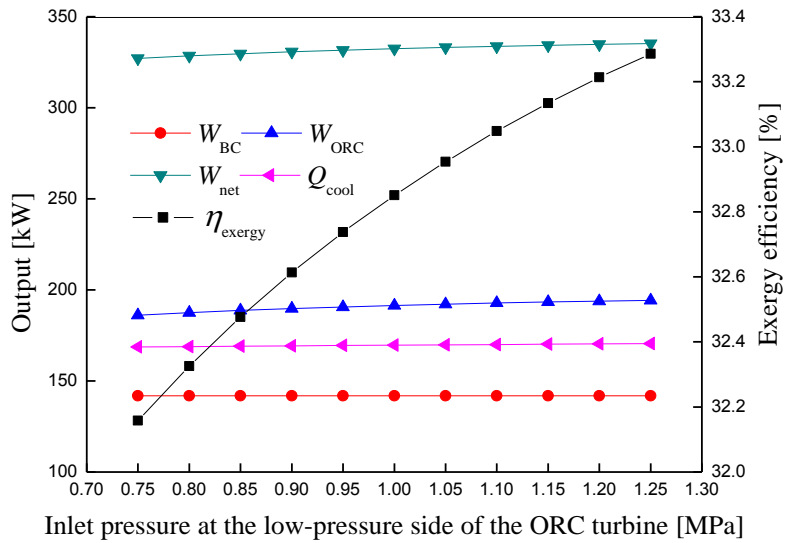


Fig. 12. Influences of inlet pressure at the low-pressure side of ORC turbine on the output and the exergy efficiency of the system.

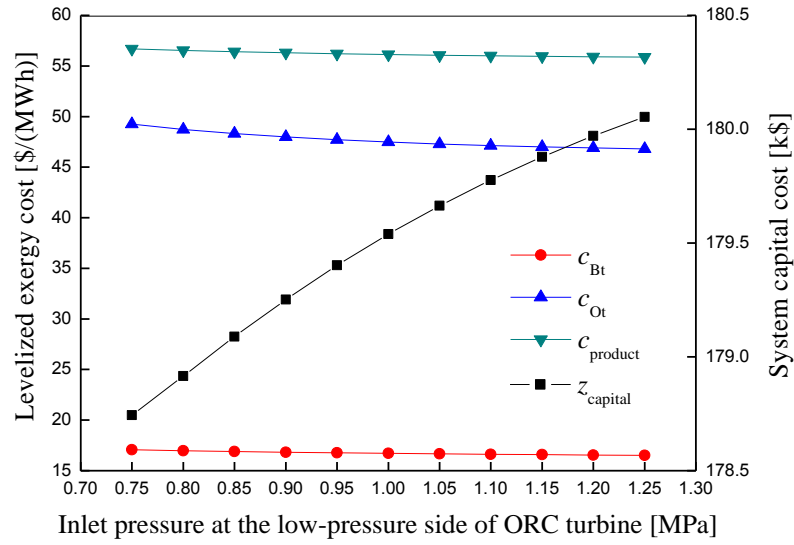


Fig. 13. Influences of the inlet pressure at the low-pressure side of ORC turbine on the levelized exergy cost and system capital cost of the system.

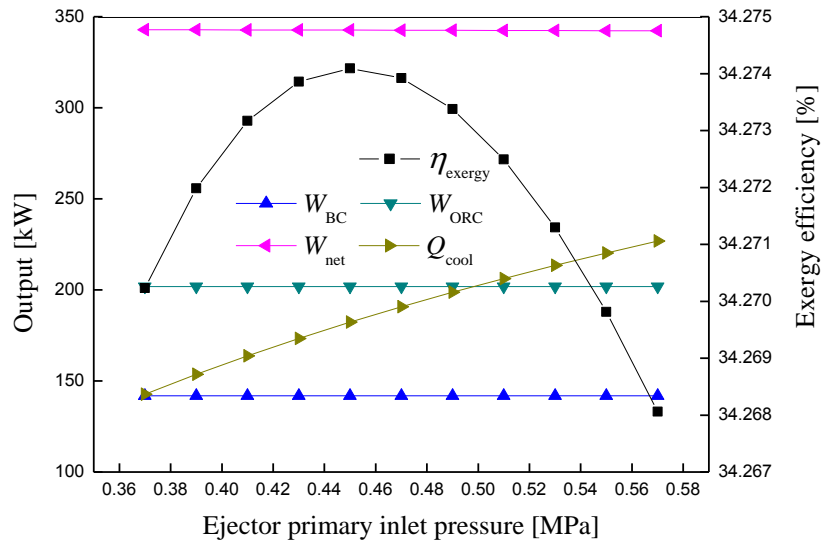


Fig. 14. Influences of ejector primary inlet pressure on the output and the exergy efficiency of the system.

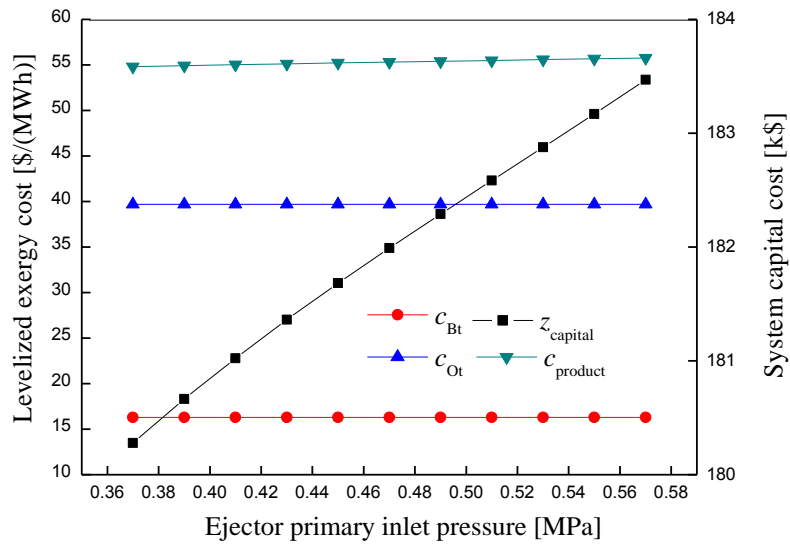


Fig. 15. Influences of ejector primary inlet pressure on the levelized exergy cost and the system capital cost of the system.

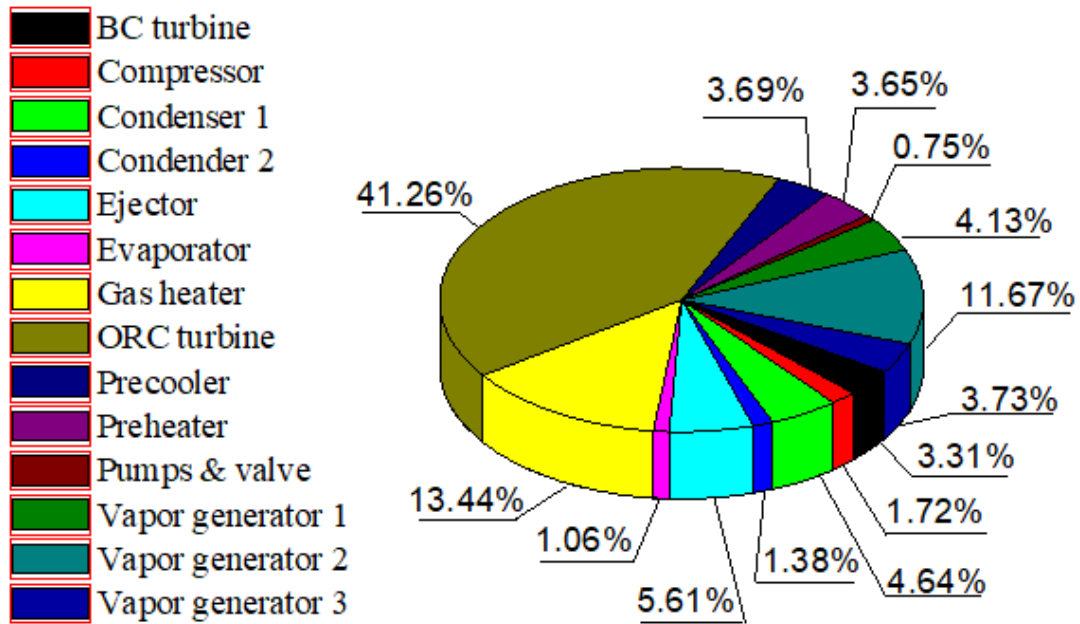


Fig. 16. Exergy destruction of different components