**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 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 shows 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.

**Nomenclature**

|  |  |  |  |
| --- | --- | --- | --- |
| Latin symbols | | ρ | density, kg m-3 |
| A | area, m2 | μ | dynamic viscosity, m2 s-1 |
| Bo | boiling number | η | efficiency, % |
| c | levelized average cost, $ (MWh)-1 | δ | thickness, m |
| cp | specific heat, kJ kg-1 K-1 | Subscribes | |
| 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 |
| Ey | 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 |
| ieff | 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 |
| Pr | reduced pressure | M | material |
| Q | heat transfer rate, kW | Ot | ORC turbine |
| Qcool | cooling capacity, kW | P | product |
| qm | average imposed wall heat flux, W m-2 | p1 | pump 1 |
| rf | enthalpy of vaporization, kJ kg-1 | p2 | pump 2 |
| T | temperature, K | p3 | pump 3 |
| U | overall heat transfer coefficient, W m-2 K-1 | p4 | pump 4 |
| W | power, kW | pf | primary flow |
| Wy | annually power, MWh year-1 | prec | precooler |
| x | vapor quality | preh | preheater |
| Z | annually levelized cost value, $ year-1 | s | shell |
| z | capital cost, k$ | t | tube |
| Acronyms | | th | thermal |
| BC | Brayton cycle | turb | turbine |
| CBC | CO2 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 |  |  |
| Greek symbols | |  |  |
| α | convection heat transfer coefficient, W m-2 K-1 |  |  |
| λ | heat conductivity, W m-1 K-1 |  |  |

## Introduction

Nowadays, internal combustion engines (ICEs) are the major motive power source in energy field, which are widely used in transport, construction, agriculture, etc. Over 49 % of the total transportation fuel is consumed by ICEs [1]. However, only 30-45% of the fuel energy is converted into effective power output, while the remaining energy is discharged to the environment via exhaust gas, jacket water and charge air [2]. A large amount of fuel energy is not harnessed. Thus, technology for waste heat recovery from ICEs has drawn much interest of researchers in the last decade. Compared to other waste heat methods such as steam Rankine cycle or Kalina cycle, organic Rankine cycle (ORC) is a promising technology with advantages of high efficiency and simple structure [3]. Consequently, much research effort has been devoted to the study of ORC-based ICE waste heat recovery system.

Traditionally, the organic working fluid in the ORC absorbs heat from the high-temperature exhaust gas in an evaporator. After that the vapor flows into the ORC turbine to expand and to produce power. More widespread use of ORC-based ICE waste heat recovery systems requires higher energy conversion efficiency and lower capital cost. There are two important pathways that will lead to the improvement of the ORC system. 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 fully use of the waste heat.

The work of selecting suitable organic working fluids was carried out by many researchers for its significant impacts on the thermal efficiency, design of the components and the capital cost of the power system. Rijpkema et al. [4] compared the thermodynamic performance of twelve working fluids in an ORC-based ICE waste heat recovery system to find the suitable candidate. Su et al. [5] developed a theoretical efficiency model about working fluids selecting for ORC-based ICE waste heat recovery system via strict mathematical derivation.

System configuration optimization mainly focuses on reducing the system irreversible rate to fully utilize the engine waste heat. Kim et al. [6] proposed a novel single-loop ORC system to recovery engine waste heat. They employed two recuperators to gradually heat the working fluid. Thus, temperature difference between exhaust gas and organic working fluid was relatively small. Comparison showed that the net power output of the system was 35.6% more than traditional ORC system. In general, the maximum power output of single-loop ORC is lower than that of the dual-loop ORC system [7]. Thus, more attention has been focused on dual-loop ORC based ICE waste heat recovery system in recent years. Huang et al. [8] modeled a dual-loop ORC system for engine waste heat recovery. Organic working fluid in the high-temperature loop absorbed heat from exhaust gas with a small temperature difference. After that, high-temperature exhaust vapor from the turbine provided heat for the low-temperature loop to further utilize the waste heat. Zhong et al. [9] proposed a dual-loop ORC system with zeotropic mixtures as working fluid in both loops. The temperatures of zeotropic mixtures are variable during the evaporation and condensation processes. Thus, in the dual-loop ORC system, temperature profiles between exhaust gas and working fluid match mutually, resulting in the increase of system thermal efficiency and power output. Wang et al. [10] investigated a complex dual-loop ORC system for ICE waste heat recovery. Exhaust gas provided heat for the high-temperature loop for the first time. Then the exhaust gas provided heat for the low-temperature loop for the second time to realize the cascading utilization of the waste heat.

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 were relatively low (200-300 ℃) [11], while the temperature of exhaust gas is above 450 ℃ [12]. 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 [13]. Thus, many researchers tried to optimize the configuration in the high-temperature loop. Traditionally, placing a heat transfer oil intermediate loop between the exhaust gas and the ORC system could address this issue [14]. But that would cause a large amount of the high-temperature waste heat unharnessed. Therefore, some other waste heat recovery systems were employed by researchers to couple with the ORC. Miller et al. [15] 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. Shu et al. [16] placed a steam Rankine cycle between the ORC and the exhaust gas. However, the system structure is complex because of the phase changing during the evaporation and condensation processes. 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 [17].

Few studies about ORC system coupled with CO2 Brayton cycle (CBC) have been published. Even Nader et al. [18] carried out some relevant studies, their attention was focused on the exhaust gas energy. The energy in jacket water was not harnessed at all in their system.

Typically, the jacket water cycles in the engine system with outlet temperature at about 90 ℃ and return temperature at about 80 ℃ [19]. There is a large amount energy in jacket water. For most ICEs (rated power between 500 kW and 200kW), thermal energy in jacket water is neatly the same as the energy in exhaust gas [20]. But the utilization rate of jacket water is low. Yu et al. [21] 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. In most ORC-based ICE waste heat recovery system, jacket water is mainly used to preheat the organic working fluids. However, the mismatch of mass flow rate of working fluid in the preheater and evaporator causes a great amount of energy in jacket water unharnessed. Thus, the utilization of energy in jacket water could be further explored.

To fulfill the variable demand of consumers, combined cooling and power systems which can generate power and refrigeration simultaneously with high efficiency are investigated by many researchers. Yin et al. [22] modeled a novel combined cooling and power system driven by low-grade waste heat. Fatih [23] developed a multigeneration system driven by solar energy. Yang et al. [24] designed a combined cooling and power system with biomass gasification.

Combined cooling and power systems driven by ICE waste heat were also investigated by some researchers. But in most of them, cooling capacity was provided by ammonia absorption refrigeration cycle (AARC) driven by high-temperature waste heat from exhaust gas. The complex cycle structure and high driven temperature requirement of AARC limit its further applications. On the contrary, ejector refrigeration cycle (ERC) exhibits the advantages of easy maintenance and high reliability [25]. Moreover, ERC can be driven by low-temperature heat source such as the jacket water. As mentioned ahead, a large amount in the jacket water can be further utilized. 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. Whereas, no combined cooling and power system with ERC driven by jacket water has been investigated before.

In this study, a combined cooling and power (CCP) system is developed, which comprises a CO2 Brayton cycle, a dual-pressure organic Rankine cycle and an ejector refrigeration cycle. The CO2 Brayton cycle absorbs heat from the high-temperature exhaust gas directly. 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 to utilize the waste heat. What’s more, the ejector refrigeration cycle is adopted to utilize rest waste heat in jacket water and produce refrigeration. 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.

## CCP 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.

Organic working fluid in the DORC is preheated by the jacket water. After that, the organic working fluid separates into two parts. One part is pumped to the vapor generator 1 to absorbs heat from the secondary engine exhaust gas in the low-pressure side cycle. Then this part of working fluid enters the low-pressure inlet of the ORC turbine to produce power. The other part of the organic working is pumped to the vapor generator 2 to absorb heat from the residual heat in exhaust CO2 after the BC turbine. Then this high-pressure part organic working fluid flows into the high-pressure inlet of the ORC turbine to produce power.

After the preheating, jacket water flows into the vapor generator 3 to provide heat for the ejector refrigeration cycle. There are two parts working fluid in the ERC. One part is pumped to the vapor generator 3 to absorb heat. After the vapor generator, the high-temperature and high-pressure vapor enters the primary inlet of the ejector, entraining a low-pressure part of working fluid into the secondary inlet of the ejector. The two parts of working fluid mixes with the low-pressure part in the ejector to obtain a relatively high average pressure. The mixed working fluid is condensed to liquid when flows out the ejector. After that, the liquid separates into two parts. One part of it flows through a throttle valve to provide refrigeration in the evaporator. The other part is pumped to the vapor generator to absorb heat again.

R245fa is selected as the working fluid for the organic Rankine cycle and the ejector refrigeration cycle because of the great thermodynamic performance [26] and the low environment effects [27].

## System model

Several assumptions are made to simplify the simulation of the system, which are: (1) the system keeps a steady state; (2) the heat and frication 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 ℃ [28], 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.

### Energy model

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

 (1)

The net power of the DORC is given as:

 (2)

The cooling capacity of the ERC is given as:

 (3)

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

 (4)

The thermal efficiency of the system is given as:

 (5)

The detailed energy model equations of each component are list 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.

### 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:

 (6)

where h0, T0 and s0 are the parameters under the ambient conditions.

The exergy flow rate in this study is given by:

 (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 [29]:

 (8)

where EF, EP, ED­, EL donate the rate of exergyfor the component fuel, the rate of exergyfor 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:

 (9)

where Ecool is the exergy rate of the cooling process, being expressed as:

 (10)

### Capital cost calculation

A method of modeling the capital costs of main components is used in this study. According to Ref. [30], 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. [30].

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

 (11)

where Ki, turb are constants corresponding to the turbine type; and W is the power output of the turbine.

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

 (12)

where FBM,turb is the multiplying factor corresponding to the working conditions of the turbine.

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

 (13)

where Ki, pump are the constants corresponding to the pump type; and W is the power input of the pump.

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

 (14)

where Bi, pump are the constants corresponding to the type of the pump; FM,pump is the material factor of the pump and FP,pump is the pressure factor of the pump. The equation of the pressure factor is given as:

 (15)

where Ci, pump are the constants corresponding to the type of the pump; and Ppump 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:

 (16)

where Ki,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:

 (17)

where FBM,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:

 (18)

where Ki,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:

 (19)

where Bi,he are the constants correspond to the type of the heat exchanger. FM,he and FP,he are the material factor and pressure factor, respectively. The pressure factor is obtained from the following equation:

 (20)

where Ci,he are the constants corresponding to the type of the heat exchanger; Phe 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:

 (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 CEPCI2016 and CEPCIref,2001 are 541.7 and 397, respectively [31].

### Exergoeconomic model

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

To find the relationship between the present value of the expenditure and the equivalent annually levelized costs, the capital recovery factor (CRF) is employed, being expressed as [29]:

 (22)

 (23)

where ieff is the effective discount rate with a value of 0.05 [32]; and n is the lifetime of the CCP system, being assumed as 30 years [33].

In order to calculate the equivalent annually levelized costs, the annual working time of the system is assumed as 8000 h [34]. Then the annual exergy rates and annual power output or consumption are obtained.

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:

 (24)

 (25)

 (26)

 (27)

where c denotes levelized exergy cost of the streams; Ey,in and Ey,out are the exergy transfer rate of the stream flowing in and out of a component; Wy and Ey,heat are the power and the heat transfer rate of the components considering the annual working time.

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

 (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 [35,36]:

 (29)

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

 (30)

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

 (31)

where cBt and cOt 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).

 (32)

 (33)

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

c26=c28=c30=0 (34)

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

cg1=cw1=0 (35)

### Internal combustion engine

In this study, the engine selected [37] 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.

### 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 [38]. The basic conditions of simulation for the CCP system are listed in Table 5.

Seven key parameters : BC turbine inlet temperature (TBt,in), BC turbine inlet pressure (PBt, in), inlet temperature at the high-pressure side of ORC turbine (TOt, in, h), inlet pressure at the high-pressure side of ORC turbine (POt, in, h), inlet temperature at the low-pressure side of ORC turbine (TOt, in, l), inlet pressure at the low-pressure side of ORC turbine (POt, in, l) and the ejector primary inlet pressure (Pej, in), are chosen to analyze the thermodynamic and exergoeconomic performance of the system. When one parameter is investigated to analyze the system performance, other parameter 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 (Qcool) and the exergy efficiency of the system () are selected to reflect the system performance. Levelized exergy cost for the BC turbine power output (cBt), levelized exergy cost for the ORC turbine power output (cOt), levelized exergy cost the system product (cproduct) and the system capital cost (zcapital) are chosen to represent the exergoeconomic performance.

## Results and discussion

The influence of the BC turbine inlet temperature (TBt,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 TBt,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 CO2 acts as the heat source for the high-pressure side of DORC, the temperature rise of the exhaust CO2, caused by the rise of TBt,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 TBt,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 (cBt) drops with the rise of the BC turbine inlet temperature (TBt,in). That can be explained by the decrease of the capital-cost-related part of cBt. The capital-cost-related part of cBt decreases with the decrease of cost of compressor, which is cut down by the drop of the compressor power consumption. The levelized exergy cost for the ORC turbine (cOt) decreases with the rise of TBt,in. 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 cOt, resulting in the decrease of cOt.

The system capital cost (zcapital) rises with the rise of TBt,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 (cproduct) decreases with the rise of TBt,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 cproduct. Though the increase of zcapital would cut down the capital-cost-related part of cproduct, the impact of levelized exergy cost for the BC turbine and ORC turbine is greater, which leads to the descending trend of cproduct.

The influence of the BC turbine inlet pressure (PBt, 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 PBt, in, which can be explained by the rise of enthalpy drop of the CO2 in the BC turbine. Though the rise of PBt, 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 PBt, in. On the one hand, the temperature of the exhaust CO2 at the BC turbine outlet decreases with the increase of PBt, 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 PBt, 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 PBt, 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.

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 PBt, in. The exergy efficiency of the system likewise has the same rising trend.

The influences of the BC turbine inlet pressure (PBt, 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 cBt increases with the rise of the PBt, in, which can be explained by the variations of the capital-cost-related part and the fuel-cost-related part. The increase of PBt, 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 (cOt) increases with the rise of PBt,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 cOt. Therefore, the levelized exergy cost for the ORC turbine (cOt) increases.

The system capital cost (zcapital) increases with the rise of (PBt, 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 PBt, in as presented in Fig. 5. According to Eq. (31), the rise of the cOt, cBt would cause the rise of the fuel-cost-related part of cproduct. 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 cproduct.

The influence of inlet temperature at the high-pressure side of ORC turbine (TOt, 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 TOt, 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 TOt, 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 TOt, 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 TOt, 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 (cOt) increases with the increase of TOt, in, h. The reason is that the two related parts of cOt increase with the drop of the ORC turbine power output.

The levelized exergy cost for the BC turbine power output (cBt) increases with the rise the TOt, 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 cBt increases, leading to the increase of cBt.

The system capital cost (zcapital) decreases with the increase of TOt, in, h. The decrease of the DORC power output causes the drop of the ORC turbine cost, which leads to the descending trend of zcapital.

The levelized exergy cost for the system product (cproduct) increases with the rise of TOt, 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 cproduct. 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 cproduct.

The influences of the inlet pressure at the high-pressure side of ORC turbine (POt, 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 POt, 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 (POt, 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 (cOt). 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 CO2, which provides heat for the vapor, decreases with the vapor levelized exergy cost. Moreover, the drop of the CO2 levelized exergy cost causes the decrease of the fuel-cost-related part of cBt, which further results in the decrease of cBt.

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 (cproduct) decreases with the increase of POt, in, h. The decrease of cOt and cBt account for the decrease of the fuel-cost-related part of the levelized exergy for the system product. The impact of cOt and cBt is greater than that of the system capital cost whose rise would result in the increase of the capital-cost-related part of cproduct. Thus, the levelized exergy cost of the system product (cproduct) shows a descending trend.

The influences of the inlet temperature at the low-pressure side of ORC turbine (TOt, 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 TOt, 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 deceases. Likewise, the exergy efficiency of the system decreases.

The cooling capacity of the ejector refrigeration cycle increases with the increase of TOt, 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 (TOt, in, l) 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 TOt, in, l. 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 CO2 after the BC turbine. Thus, the fuel-cost-related part of cBt increases, resulting in the rise of cBt.

The levelized exergy cost for the ORC turbine (cOt) increases with the increase of TOt, in, l. 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 cOt 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 TOt, in, l. The increase of cBt and cOt cause the increase of the fuel-cost-related part of the levelized exergy cost for the system product. Though, the decease 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 (POt, in, l) 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 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 POt, in, l. The increase of enthalpy drop of the organic working fluid in the low-pressure side, which is caused by the rise of POt, in, l, 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 POt, in, l. 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 (POt, in, l) 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 POt, in, l. 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 cOt decrease.

The levelized exergy cost for the BC turbine power output decreases with of POt, in, l. The decrease of the cOt 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 CO2. Thus, the levelized exergy cost of the exhaust CO2 decreases, which further leads to the drop of the fuel-cost-related part of cBt. Therefore, the levelized exergy cost for the BC turbine power output (cBt) 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 POt, in, l. 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 (Pej, 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 cproduct. As a result, the levelized exergy cost for the system increases.

The parametric analysis reveals that a high output power, exergy efficiency and the low levelized exergy cost are achieved at the highest inlet pressure at the high-pressure side ORC turbine evidenced from Fig. 8 and 9 when inlet pressure at the high-pressure side ORC turbine is varied and others are kept as constants. While the continue increasing of the inlet pressure at the high-pressure side ORC turbine would causes the temperature cross in vapor generator 2. Also, the variation of other six parameters can also affect the system power output and levelized exergy cost. To obtain the optimal performance of the system, genetic algorithm (GA) [39] is employed to conducted the optimization with the variation of all the seven parameters (TBt,in, PBt, in, TOt, in, h, POt, in, h, TOt, in, l, POt, in, l and Pej, in). The ranges of these parameters are listed in Table 6.

The levelized exergy cost reflects both the thermodynamic and the exergoeconomic aspects of the system. Thus, it is selected as the objective function for the single-objective optimization. The control parameters of 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 cproduct is 53.25 $(MWh)-1. The exergy efficiency of the CCP system is 37.31% which is also desirable.

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, 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.

## 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 TBt,in and PBt, 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 TOt, in and TOt, 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 POt, in, h and POt, 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 Pej, 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 $(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 used in this study.

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

For each section the heat transfer area is calculated as:

 (B1)

where Ti is the log-mean temperature difference (LMTD) and Ui is the overall heat transfer coefficient.

 (B2)

In Eq. (B2) δ and λ represent the thickness of the tube and the thermal conductivity of the tube wall, respectively. αt,i is the convection heat transfer coefficient in the tube side and αs,i is the convection heat transfer coefficient in the shell side.

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

In the single-phase heat transfer process, the convection heat transfer coefficient in the tube side and the shell side are expressed as [40]:

 (B3)

 (B4)

In Eq. (B3), the Nusselt number is calculated as [41,42]:

 (B5)

 (B6)

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

In Eq. (B4), Des is the equivalent diameter of the shell, being expressed as:

 (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 [43,44]:

 (B8)

 (B9)

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

 (B10)

**Appendix B**

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

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**Figure captions**

**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 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

**Table 1** Energy and exergy models for each component in CCP system

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Component | Energy equation | EF | EP | ED | EL |
| Gas heater |  |  |  |  | / |
| BC turbine |  |  |  |  | / |
| Vapor generator 2 |  |  |  |  | / |
| Precooler |  | / | / |  |  |
| Compressor |  |  |  |  | / |
| Vapor generator 1 |  |  |  |  | / |
| ORC turbine |  |  |  |  | / |
| Condenser 1 |  | / | / |  |  |
| Pump 1 |  |  |  |  | / |
| Preheater |  |  |  |  | / |
| Pump 2 |  |  |  |  | / |
| Pump 3 |  |  |  |  | / |
| Vapor generator 3 |  |  |  |  | / |
| Condenser 2 |  | / | / |  |  |
| Valve |  | / | / |  | / |
| Pump 4 |  |  |  |  | / |
| Ejector |  |  |  |  | / |
| Evaporator |  |  |  |  | / |

**Table 2** Cost balance and auxiliary relation [30] for each component of CCP system

|  |  |  |
| --- | --- | --- |
| Component | Cost balance | Auxiliary relation |
| Gas heater |  | cg1=cg2=0 |
| Vapor generator 2 |  | c4=c5 |
| BC turbine |  | c4=c3 |
| Precooler |  | c1=c5 |
| Compressor |  | celec,1=cBt |
| Vapor generator 1 |  | cg2=cg3 |
| ORC turbine |  | c10=c11=c12 |
| Pump 1 |  | celec,3=cOt |
| Condenser 1 |  | c13=c12 |
| Preheater |  | cw1=cw2=0 |
| Pump 2 |  | celec,2=cOt |
| Pump 3 |  | celec,3=cOt |
| Vapor generator 3 |  | cw3=cw2 |
| Valve | / | c19=c20 |
| Pump 4 |  | celec,4=cOt |
| Condenser 2 |  | c16=c17 |
| Ejector |  | / |
| Evaporator |  | c20=c21 |

**Table 3** Main parameters of the engine [37]

|  |  |
| --- | --- |
| Parameters | Value |
| Power output (kW) | 2928 |
| Rotation (r(min)-1) | 1000 |
| Exhaust gas temperature (℃ ) | 470 |
| Exhaust gas mass flow rate (kg s-1) | 4.35 |
| Temperature of jacket water (℃ ) | 90/79 |
| Mass flow rate of jacket water (kg s-1) | 25 |

**Table 4** Composition of the exhaust gas [37]

|  |  |  |
| --- | --- | --- |
| Composition | Molecular (g(mol)-1) | Fraction (%) |
| O2 | 32.00 | 9.3 |
| CO2 | 44.00 | 9.1 |
| H2O | 18.01 | 7.4 |
| N2 | 28.01 | 74.2 |

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

|  |  |
| --- | --- |
| Parameter | Value |
| Ambient temperature (℃ ) | 20 |
| Ambient pressure (MPa) | 0.101 |
| Compressor inlet temperature (℃ ) | 35 |
| BC turbine inlet temperature (℃ ) | 400 |
| BC turbine inlet pressure (MPa) | 18 |
| BC turbine outlet pressure (MPa) | 8 |
| Inlet temperature at the high-pressure side of ORC turbine (℃ ) | 150 |
| Inlet pressure at the high-pressure side of ORC turbine (MPa) | 1.6 |
| Inlet temperature at the low-pressure side of ORC turbine (℃ ) | 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 (℃ ) | 100 |
| Pinch point temperature difference in vapor generator 1 (℃ ) | 30 |
| Pinch point temperature difference in vapor generator 2 (℃ ) | 30 |
| Pinch point temperature difference in vapor generator 3 (℃ ) | 25 |
| Condensation temperature of condenser 1 (℃ ) | 30 |
| Condensation temperature of condenser 2 (℃ ) | 30 |
| Evaporation temperature of evaporator (℃ ) | 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 (℃ ) | 20 |

**Table 6** Parameters for GA

|  |  |
| --- | --- |
| Parameter | Operation range |
| BC turbine inlet temperature (℃ ) | 330-440 |
| BC turbine inlet pressure (MPa) | 15-20 |
| Inlet temperature at the high-pressure side of ORC turbine (℃ ) | 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 (℃ ) | 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 |

**Table 7** Control parameters of GA

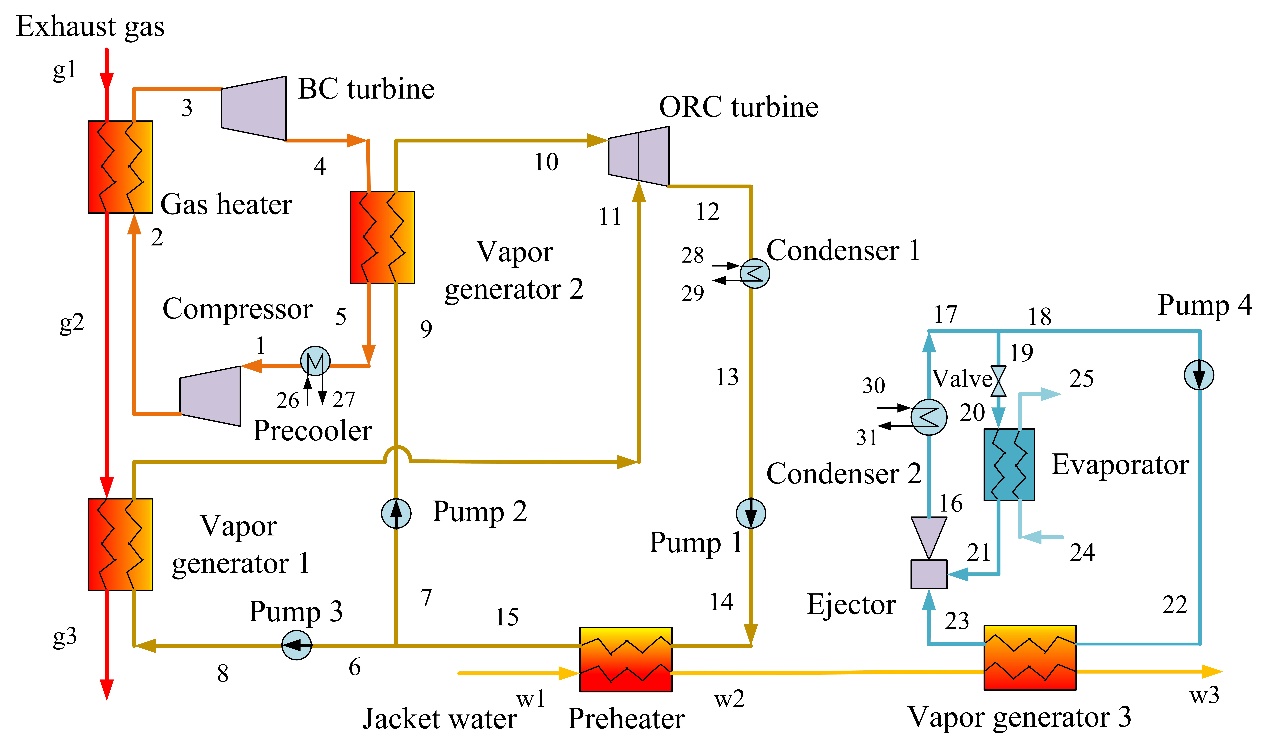
|  |  |
| --- | --- |
| Tuning parameters | Value |
| Population size | 20 |
| Mutation probability | 0.01 |
| Crossover probability | 0.8 |
| Stop generation | 200 |

**Table 8** Single-objective optimization results

|  |  |
| --- | --- |
| Parameter | Value |
| BC turbine inlet temperature (℃ ) | 425.46 |
| BC turbine inlet pressure (MPa) | 20.00 |
| Inlet temperature at the high-pressure side of ORC turbine (℃ ) | 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 (℃ ) | 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 |

**Table B1** Constants for component costs [30]

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Constant | Value | Constant | Value | Constant | Value |
| B1,he | 1.63 | K3,pump | 0.1538 | C3,he | 0.08183 |
| B2,he | 1.66 | K1,turb | 2.7051 | C1,pump | -0.3635 |
| B1,pump | 1.89 | K2,turb | 1.4398 | C2,pump | 0.3957 |
| B2,pump | 1.35 | K3,turb | -0.1776 | C3,pump | -0.0026 |
| K1,he | 4.3247 | K1,comp | 2.2897 | FM,he | 1.0 |
| K2,he | -0.3030 | K2,comp | 1.3604 | FBM,turb | 3.5 |
| K3,he | 0.1634 | K3,comp | -0.1027 | FBM,comp | 2.7 |
| K1,pump | 3.3892 | C1,he | 0.03881 | FM,pump | 2.2 |
| K2,pump | 0.0536 | C2,he | -0.11272 |  |  |



**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.



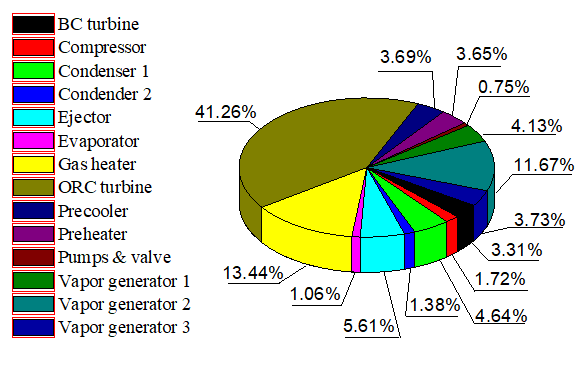
**Fig. 13.** Influences of the inlet pressure at the low-pressure side of ORC turbine on the levelized exergy cost and system capital cost of the system.



**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