**Performance analysis and optimization of a combined cooling and power (CCP) system for engine waste heat recovery**

**Abstract**

A combined cooling and power (CCP) system driven by exhaust gas and jacket water for an internal combustion engine is developed. A Brayton cycle is employed to absorb heat from the high-temperature exhaust gas directly to keep the stability of the system for avoiding the decomposition of the organic working fluid. A dual-pressure organic Rankine cycle (DORC) is designed to make fully use of the waste heat of the exhaust gas and the jacket water. An ejector refrigeration cycle driven by the jacket water energy is added to the system to fulfill the variable demand of customers. Thermodynamic and exergoeconomic analysis of the system are performed to evaluate six key parameters (compressor inlet temperature, compressor pressure ratio, BC turbine inlet temperature, high-pressure inlet temperature of the ORC turbine, low-pressure inlet temperature of the ORC turbine and the primary flow pressure). Single-objective optimization is carries out by means of genetic algorithm (GA) to reach the minimum average cost per unit of exergy product of the system.

A desirable exergy efficiency of the system is obtained with the net power of 577.5483 kW and the cooling capacity of 498.5947 kW.

Keyworks:

Combined cooing and power system

Internal combustion engine

Brayton cycle

Dual-pressure organic Rankine cycle

Waste heat recovery

**Nomenclature**

|  |  |  |  |
| --- | --- | --- | --- |
| A | area, m2 |  | Boltzmann’s constant |
| BC | Brayton cycle |  | thickness, m |
| Bo | boiling number |  | compressor pressure ratio |
| c | average cost per unit of exergy, $ (MWh)-1 | Subscript | |
| cp | specific heat, kJ (kg K)-1 | 1-16 | state points |
|  | cost rate, $ year-1 | g1-g4 | state points |
| CCP | combined cooling and power | bt | Brayton cycle |
| CRF | capital recovery factor | BM | bare module |
| CEPCI | chemical engineering plant cost index | cond | condenser |
| D | diameter, m | comp | compressor |
| e | exergy, kJ kg-1 | D | destruction |
|  | exergy flow rate, kJ s-1 | e | effective |
| y | exergy flow rate per year, kJ year-1 | elec | electricity |
| f | friction factor | ej / E | ejector refrigeration cycle |
| F | configuration factor | es | equivalent diameter |
| Fs | shape factor | ev | evaporation |
| G | mass flow rate, kg s-1 | ex | exergy |
| h | convection heat transfer coefficient, W m-2K-1 | F | fuel |
| hout | enthalpy, kJ kg-1 | g | geothermal |
| H | depth, m | he | heat exchanger |
| ieff | interest rate | hf | hot flow |
| l | length, m | in | inside |
|  | mass flow rate, kg s-1 | L | loss |
| n | lifetime, year | l | liquid |
| Nu | Nusselt number | M | material |
| P | space between pipe, m | m | mean |
| Pr | Prandtl number | mas | maximum |
| Pt | center distance between tubes, m | ORC | organic Rankine cycle |
| Pr | reduced pressure | ot | organic Rankine turbine |
|  | heat transfer rate, kW | out | outside |
| y | heat transfer rate per year, MWh year-1 | pump | pump |
| T | temperature, K | pump1 | pump 1 |
| U | overall heat transfer coefficient, W m-2K-1 | pump2 | pump 2 |
| v | velocity, m s-1 | s | single phase |
|  | power, kW | t | tube |
| y | power per year, MWh year-1 | turb | turbine |
| x | vapor quality | vg | vapor generator |
|  | annually levelized cost value, $ year-1 | w | tube wall/water |
| Greek symbol | | wbt | BC turbine power |  |
|  | heat load ratio | wot | ORC turbine power |
|  | efficiency, % |  |  |
|  | heat conductivity, W m-1K-1 |  |  |
|  | density, kg m3 |  |  |
|  | dynamic viscosity, m2 |  |  |

1. **Introduction**

Nowadays, internal combustion engines (ICEs) are the major motive power source for the society. Internal combustion engines are used widely in transport, construction, agriculture, etc. However, the environment problems caused by the fuel consumption in internal combustion engines are also severe. Nearly half of the total transportation fuel is consumed by internal combustion engines. Whereas, the thermal efficiency of the ICEs is only 30-45% [1], leading to energy waste and the environment pollution. As a result, finding sustainable ways to fully utilized the fuel energy in the ICEs has attracted more and more attention of the researchers.

Previous researchers mainly focused on recycling waste heat from the exhaust gas which is in high temperature state. Many works have been carried out to utilize the energy in exhaust gas with the help of organic Rankine cycle (ORC). In 2010, Vaja et al. [2] matched an organic Rankine cycle to the internal combustion engine to recovery the waste heat. Later, further studies were carried out to improve the performance of the electricity generation system. Shu et al. [3] designed an organic Rankine cycle to harvest the exhaust waste heat of engines. Rita et al. [4] optimized a shell and louvered fin heat exchanger in the organic Rankine cycle driven by the internal combustion engine gas. Beside the exhaust gas, the coolant of the ICEs also has the potential to be utilized, considering the large amount of the mass flow rate and the relatively high temperature.

Research works about utilizing the exhaust gas and jacket water have been developed by many researchers. Zhang et al. [5] modeled a novel system combining a dual loop ORC and a vehicular light-dual diesel engine. The exhaust gas, intake air and the coolant were analyzed. They compared the power output of the high-temperature loop and the low-temperature loop and found that the low-temperature loop can produce more power. To avoid the complexity of the conventional multi source ORC system, a confluent cascade expansion ORC system was introduced by Chen et al. [6]. Thermodynamic simulation method was employed to calculated the parameters of the system. By comparing the results, they concluded that the architecture of the novel system is simper, the net power output of the system is larger (generating 8% more net power) and the thermal efficiency of the novel system is higher (improved to 49.5% from 45.3%). A dual-loop organic Rankine cycle was developed by Ge et al. [7] to recover the waste heat of the ICEs. In that study, isobutane and isopentane are mixed as the working fluid for the low-temperature loop (LTL) and cyclopentane and benzene mixtures are selected as the working fluid for the high-temperature loop (HTL). Comparing between mixed organic working fluid and pure organic working fluid was carried out and they drew the conclusion that less exergy destruction rate was achieved in the mixture using system. Seyedali et al. [8] studied a two-parallel-step organic Rankine cycle driven by the waste heat of the IEC. Comprehensive thermodynamic performance analysis and the optimization of the system were carried out by analyzing the key design parameters of the system. The got the result that 468 kW electricity power was produced by the system with an exergy efficiency of 21 %.

With the variation of the customers’ demand, more products other than electricity power were required. A number of studies began to pay attention to the combined systems. The design of combined cooling and power (CCP) system as a result was carried out by many researchers [9-11]. Chen et al. [12] investigated an ammonia-water combined cooling and power system using the waste heat from the ICEs. A gas engine with the power output of 300 kW was selected as the data source. By calculating the thermodynamic performance of the system, they concluded that the equivalent power output of the system is 92.86 kW and the exergy efficiency of the combined cooling and power system is 33.69%. In order to gain a large cooling capacity, ammonia-absorption cooling cycle was utilized by many researchers. Whereas, the capital investment of the system components and the operation cost was relatively high. Ejector refrigeration systems which were low in capital cost and simple in operation therefore were combined with the electricity generation cycles. And the economic analysis of a system has attracted the attention of studiers in recent years.

To better evaluate the thermodynamic and economic performance of a system, exergoeconomic (thermoeconomic) analysis methods were established. It provided a new aspect to design and operate the energy systems. YD Lee et al. [13] evaluated an SOFC-Engine hybrid power generation system. Exergoeconomic analysis methods were employed to analyzed the economic performance as well as the thermodynamic performance of the system. They found that the internal combustion engine accounted for the largest exergy destruction and followed the heat exchanger and the SOFC stack. An ejector refrigeration system driven by homogeneous charge compression ignition (HCCI) engine was designed by Mohsen et al. [14]. Exergoeconomic and thermodynamic performance of the system was calculated in MATLAB software. Multi-objective optimization was carried out with the objective function of exergy efficiency and the product unit cost of the system. Conclusion was obtained that in the highest exergy efficiency and the lowest product unit cost, the generator, condenser and the evaporator should work at temperature of 94.54℃, 33.44, and 0.03, respectively. A combined cooling, heating and power (CCHP) system was analyzed by Wang et al. [15] using exergoeconomic methods. Energy costs of products in the system were calculated considering the natural gas price. They found that the cost of electricity increased from 0.537 to 1.077 Yuan/kWh with the change of the power output range (100-20%).

In most of the study, the exhaust gas was utilized directly by working fluid in organic Rankine cycle without considering the chemical stability. In general, the decomposition temperature is about 200 to 300 ℃ for many kinds of the organic working fluid. However, the exhaust gas temperature can get as high as 600 ℃, causing the potential of the decomposition for the working fluid during the long working process. As a result, there is contradiction between the thermal stability of the system and the maximum utilization of the thermal energy when combining the organic Rankine cycle with the ICEs.

To solve the problems mentioned above. A Brayton cycle is employed to utilized the high temperature of the exhaust gas in this study. To fully make use of the thermal energy of the exhaust gas, an organic Rankine cycle is employed which is driven by the exhaust gas after the Brayton cycle. Considering the high temperature of the exhaust gas in the Brayton, another organic Rankine cycle is added. Therefore, a dual-pressure organic Rankine cycle with the exhaust gas as the high-pressure heat source and the secondary exhaust gas as the low-pressure heat source is introduced. Because of the large mass flow rate and relative high temperature of the jacket water, it is utilized to preheat the organic working fluid before the separating of the working fluid in the dual-pressure ORC cycle. Besides, an ejector refrigeration cycle is added to the system for the cascade utilization of the jacket water. To consider the system comprehensively, thermodynamic and exergoeconomic analysis of the system is employed. Optimization for the system based on the exergoeconomic analysis is obtained by means of genetic algorithm.

**2. System description**



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

The combined cooling and power (CCP) is shown in Fig 1. The system begins with a CO2 Brayton cycle. The exhaust gas from the internal consumption engines (ICEs) enters the gas heater and releases heat to the CO2. Absorbing heat from the gas heater, the CO2 enters the Brayton cycle turbine with high temperature and high pressure. After expanding in the turbine, the exhaust CO2 is still in a high temperature state. The thermal energy of the exhaust gas is utilized in the vapor generator 2 to heat the organic working fluid. After the precooler, the cooling CO2 enters the compressor. In the compressor the CO2 is compressed to supercritical state to continue the cycle.

A dual-pressure organic Rankine cycle is employed to utilized the thermal energy from the exhaust gas from the gas heater as well as the exhaust CO2 from the vapor generator 2. The organic working fluid separates at 6 state point into two parts. One part of the working fluid exchanges heat with the exhaust gas and then becomes low-pressure vapor. The other part of the working fluid is pumped into a relatively high pressure by pump 1. Then the high-pressure working fluid absorbs heat in the vapor generator 2 to get to a high temperature state. The high-pressure working fluid along with the low-pressure working fluid enters the organic Rankine cycle (ORC) turbine to generate electricity. After the condensation process, all the working fluid is pumped to a preheater in which the jacket water provides heat source to heat the working fluid.

The jacket water then flows into vapor generator 3 to release heat for the ejector refrigeration cycle. After the condensation process in condenser 2, liquid working fluid is divided into two separated parts. One part of the working fluid flows through the throttle valve where the working fluid is vaporized to vapor state. The vapor enters the evaporator to produce cooling capacity when absorbing heat from the environment. The other part of the fluid is pumped to the vapor generator 3 to absorb heat. The superheated vapor mixes with the cooling fluid in the ejector to rise the total pressure. The mixed working fluid enters the condenser 2 to be condensed to liquid to continue the cycle.

R245fa is selected as the working fluid for the organic Rankine cycle because of the great thermodynamic performance and the low environment effect. [16] [17] Also, the working fluid in the ejector cooling cycle is selected as R245fa.

**3.System modeling**

3.1. Thermodynamic analysis

3.1.1. Energy analysis



**Fig. 2.** Temperature profiles in vapor generator

The process about generating vapor in this study is divided into three regions as shown in Fig. 2. The heat balance equation in the vapor generator about the three different regions are given as follows:

Superheated region

 (1)

Evaporation region

 (2)

Sub-cooled region

 (3)

The pinch point temperature different can be defined as the lowest temperature difference between exhaust gas temperature and the saturation temperature corresponding to the evaporation pressure, being expressed as:

 (4)

As a result, the mass follow rate of the working fluid can be obtained based on the energy balance as:

 (5)

The structure of the ejector is shown in Fig 3. In the ejector the primary fluid flows into the ejector in a high-pressure state, while the secondary fluid is in a relatively low pressure. [18] The velocities of the steams at the inlet and the outlet of the ejector can are negligible. In the nozzle section, the primary fluid accelerates to a high velocity at the expense of decreasing pressure. Then the secondary fluid enters the ejector and mixes with the primary fluid. With the exchange of momentum, the mixed fluid gains a relative high velocity. In the diffuser section, the pressure of the mixed fluid increases accompanied with the drop of the velocity.



**Fig. 3.** The structure of the ejector

The energy balance of the ejector can be expressed as:

 (6)

where h A,B,C is the enthalpy of the three different fluid and x is the entrainment ratio of the ejector.

Several assumptions are made to simplify the simulation of the system.

1. The system is in a steady state.
2. The heat and friction losses in the system are not taken into account.
3. The pressure losses in the vapor generator, heat exchangers, evaporator and condenser are not taken into account.
4. The process through the throttle valve is isenthalpic.
5. The working fluid out of the condenser and preheater is saturated liquid and the state at the outlet of the evaporator is saturated vapor.
6. Considering the low gas acid dew point temperature , the gas temperature outlet the vapor generator 1 is 110℃ as confined. [19]

The energy analysis for each component is based on the first law of thermodynamic. The energy balance equations of each component are listed in Table 1

**Table 1**

Energy analysis for each component in CCP system

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

The net power of the Brayton cycle is given as:

 (7)

The net power of the dual-pressure organic Rankine cycle is expressed as:

 (8)

The cooling capacity of the ejector refrigeration cycle is calculated by:

 (9)

The net power of the system is obtained as:

 (10)

The thermal efficiency of the system is given in the expression: (11)

3.1.2. Exergy analysis

Exergy represents the useful work potential of the system at the specified state. The total exergy of a system includes the kinetic exergy and the potential exergy which are no taken into account in this study. The exergy for unit weight working fluid can be defined as:

 (12)

where h0, T0 and s0 are the parameter of the ambient state.

The exergy flow rate in this study is given by the expression:

 (13)

Thermal systems are always connected with exergy inputs associated directly or indirectly with fuels or other energy resources, such as exhaust gas thermal energy and jacket water thermal energy in this study. As a result, the destructions and losses of exergy represent the waste of the energy resources. The exergetic efficiency provides a useful measure of the performance of an energy system. The exergy rate balance for components in the system is given as: [20]

 (14)

where  and  represent the rates of exergy for product and fuel;  and  denote the rates of exergy destruction and exergy loss, respectively.

The analysis on the exergy rate for each component is listed in Table 2.

**Table 2**

Exergy analysis for each component for CCP system

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| Component |  |  |  |  |
| Gas heater |  |  |  | / |
| BC turbine |  |  |  | / |
| Vapor generator 2 |  |  |  | / |
| Precooler | / | / |  |  |
| Compressor |  |  |  | / |
| Vapor generator 1 |  |  |  | / |
| ORC turbine |  |  |  | / |
| Condenser 1 | / | / |  |  |
| Pump 2 |  |  |  | / |
| Preheater |  |  |  | / |
| Pump 1 |  |  |  | / |
| Vapor generator 3 |  |  |  | / |
| Condenser 2 | / | / |  |  |
| Valve 4 | / | / |  | / |
| Pump 3 |  |  |  | / |
| Ejector |  |  |  | / |
| Evaporator |  |  |  | / |

The exergy of the ejector refrigeration cycle is given as:

(15)

The exergy efficiency of the whole system is given by the expression:

(16)

3.2. The area of the heat exchanger

Because of the high heat transfer efficiency and the low price, all the heat exchangers in this study are shell-and-tube type. To calculate the area of the heat exchanger conveniently, the process in the heat exchanger is divided into two different processes which are single-phase heat transfer process and two-phase heat transfer process respectively. The two-phase heat transfer process takes place in the vapor generators and the condensers. The single-phase heat exchange process occurs in the gas heater, precooler and the preheater.

For the ununiform thermodynamic properties of the working fluid in the heat transfer process, the processes are discretized in to a lot of subsection during which the thermodynamic properties of the working fluid are assumed to be uniform and constant. The following is the discussion of the heat transfer processes of single-phase flow and the two-phase flow.

3.2.1. Single-phase heat transfer process

For each subsection, the heat transfer rate is given by:

(17)

whereis the overall hear transfer coefficient in each subsection, is the heat transfer area and is the log-mean temperature difference (LMTD).

The overall heat exchanger coefficient is given by:

(18)

where is the convection heat transfer coefficient of the hot fluid and is the convection heat transfer coefficient of the cold fluid. λ and δ denote the thickness of the tube and the thermal conductivity of the tube wall, respectively.

The log-mean temperature difference is calculated as:

(19)

The convection heat transfer coefficient for the tube side is given by:

(20)

The Nusselt number is expressed as: [21][22]

(21)

(22)

where the Darcy frication factor can be determined by: [23]

(23)

The Reynolds number can be calculated as:

(24)

where is the mass velocity of the fluid in the tube-side and is given by:

(25)

where N is the number of the tubes inside the shell.

The Prandtl number can be determined as:

(26)

The convection heat transfer coefficient for the shell side is given by: [24]

(27)

where is the equivalent diameter in the shell side, being expressed as:

(28)

is the mass velocity of the fluid in the shell side and is given by:

(29)

where is the maximum area where the fluid flows and can be determined as:

(30)

3.2.2. Two-phase heat transfer process

The two-phase heat transfer process takes place in the vapor generator and the condenser. The fluid in this process will have the gas phase as well as the liquid phase which increase the difficult to analyze the heat transfer process. Just as the same as the single-phase process, the region in the heat exchanger is discretized to many subsection. As a result, the thermodynamic properties are assumed to be uniform and constant. We divide the two-phase heat transfer process into evaporation and condensation.

For the evaporation heat transfer process, the convection heat transfer coefficient can be expressed by: [25]

(31)

where is the boiling number, being determined as:

(32)

For the condensation heat transfer process, the convection heat transfer coefficient can be expressed by: [26]

(33)

where is the reduced pressure which is the ratio of state point pressure to critical pressure of the fluid.

As a result, the area of each subsection in the heat exchanger can be calculated with the necessary data. By adding up the area of each subsection, the total area of the heat exchanger can be obtained.

3.3. Capital costs of the system

To assess the feasibility of a system, the cost analysis associated with the components should be estimated. In this study, a method of modeling the capital costs of the main component is employed [27]. This method considers the direct project expenses such as equipment cost, material for installation labor, and indirect project expenses including taxes, insurance, etc. The costing technique relates the costs back to the purchased cost of equipment evaluated for some base conditions. For conditions which deviate from the base conditions, multiplying factors (the specific equipment type, the specific system pressure and the specific material of construction) are used to solve the problem.

When one depends on past records or published correlations for price information, it is necessary to update the costs to take changing economic conditions (inflation) into consideration. This can be achieved by the following expression:

(36)

where is the purchased cost and 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 and is 638.1 and 397, respectively [28] [29].

The heat exchanger used in this study is shell-and-tube which is made from carbon steel (CS). The cost of the heat exchanger considering the inflation is given as:

(37)

where and is the constants corresponded to the type of the heat exchanger. and are the material factor and pressure factor, respectively. For carbon steel as material, is 1. And the pressure factor is obtained from the following equation:

(38)

where , and are constants corresponded to the type of the heat exchanger. is the pressure for the heat exchanger.

The basic cost for the heat exchanger is given as:

(39)

where is the constant corresponded to the type of the heat exchanger. is the capacity or size parameter for the equipment. For heat exchanger, is the heat exchanger area.

The turbine used in this study is axial type which is made from carbon steel (CS). The cost of the turbine considering the inflation is given as:

(40)

where is the model factor of the turbine. The basic cost for the turbine is given as:

(41)

where is the constant corresponded to the type of the turbine. is the power output of the turbine.

The pumps used in this study are reciprocating which are made from stainless steel (SS). The cost of the pumps considering the inflation is given as:

(42)

where and is the constants corresponded to the type of the pump. and are the material factor and pressure factor, respectively. And the pressure factor is obtained from the following equation:

(43)

where , and are constants corresponded to the type of the pump. is the design pressure for the pump.

The basic cost for the heat exchanger is given as:

(44)

where is the constant corresponded to the type of the pump. is the power consumption of the pump.

The compressor used in this study is axial type which is made from carbon steel (CS). The cost of the compressor considering the inflation is given as:

(45)

where is the model factor of the compressor. The basic cost for the compressor is given as:

(46)

where is the constant corresponded to the type of the compressor. is the power consumption of the compressor.

The values of the constants mentioned above for the main components are listed in Table 5

**Table 5**

Constants for component costs [27]

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Constant | Value | Constant | Value | Constant | Value |
|  | 1.63 |  | 1.4398 |  | -0.3935 |
|  | 1.66 |  | -0.1776 |  | 0.3957 |
|  | 1.89 |  | 2.2897 |  | -0.00226 |
|  | 1.35 |  | 1.3604 |  | 0 |
|  | 4.3247 |  | -0.1027 |  | 0 |
|  | -0.3030 |  | 0.03881 |  | 0 |
|  | 0.1634 |  | -0.11272 |  | 1.0 |
|  | 3.3892 |  | 0.08183 |  | 3.5 |
|  | 0.0536 |  | 0 |  | 2.7 |
|  | 0.1538 |  | 0 |  | 2.2 |
|  | 2.7051 |  | 0 |  |  |

3.4. Exergoeconomic analysis

The term exergoeconomic combines the exergy analysis and economic principles. 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: [30]

(47)

where is the effective discount rate with a value of 0.05 [31]. n is the lifetime of the CHP system being assumed as 30 [32].

The equivalent annually levelized costs is given as:

(48)

In order to calculate the equivalent annually levelized costs, the annual working hours of the system is assumed as 8000 h [33]. Then the annual exergy rates and annual power output or consumption are obtained.

For a system operating at steady state, there may be a number of entering and exiting material streams as well as both heat and work interactions with the surroundings. In exergy costing, a cost is associated with each exergy stream. The entering and exiting streams of matter with associated rates of exergy transfer and , power , and the exergy transfer rate associate with heat transfer are given, respectively:

(49)

(50)

(51)

(52)

here , , and denote average costs per unit of exergy in dollars per joule

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

(53)

note that when a component receives power (as in a compressor or a pump) the term can be moved to the right side with positive sign. The term can appear in positive sign on the left side if there is a heat transfer from the component. Details of the cost balance equation are listed in Table 6.

**Table 6**

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

|  |  |  |
| --- | --- | --- |
| Component | Cost balance | Auxiliary relation |
| Gas heater |  |  |
| Vapor generator 2 |  |  |
| BC turbine |  |  |
| Precooler |  |  |
| Compressor |  |  |
| Vapor generator 1 |  |  |
| ORC turbine |  |  |
| Pump 2 |  |  |
| Condenser 1 |  |  |
| Preheater |  |  |
| Pump 1 |  |  |
| Vapor generator 3 |  |  |
| Valve 4 | / |  |
| Pump 3 |  |  |
| Condenser 2 |  |  |
| Ejector |  | / |
| Evaporator |  |  |

The average cost per unit of exergy product is chose to represent the exergoeconomic performance, being expressed as:

(54)

where is the fictitious oust rate [34] associated with the use of dissipative component, being expressed as:

(55)

(56)

(57)

In addition, the average cost per unit of exergy of the condensers and the precooler is equal to zero, being given by:

(58)

The average cost per unit of exergy of the exhaust gas as well as the jacket water is zero, being expressed as:

(59)

3.5. Model validation

Published data is used to validate the model in this study. The results about the validation for the Brayton cycle are listed in Table 7 [35]. The dual-pressure organic Rankine cycle and the ejector cycle are validated by Ref. [36] and Ref. [37]. The results are listed in Table 8 and Table 9, respectively. It can be seen that the data in present study are in good agreement with the data in the published literature.

**Table 7**

Validation for Brayton cycle

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Term | Present | Ref. [35] | Term | Present | Ref. [35] |
| Fluid |  |  |  | 2000 | 2000 |
| (℃) | 20 | 20 |  | 2000 | 2000 |
| (℃) | 114.55 | 114.6 |  | 573 | 573 |
| (℃) | 600 | 600 |  | 1 | 1 |
| (℃) | 449.1 | 449 | (kW) | 169.85 | 169.9 |
|  | 573 | 573 | (kW) | 50.5 | 50.4 |

**Table 8**

Validation for dual-pressure organic Rankine cycle

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Term | Present | Ref. [36] | Term | Present | Ref. [36] |
| Fluid | Isopentane | Isopentane |  | 485.97 | 486.0 |
| (℃) | 120.0 | 120.0 |  | 431.99 | 432.1 |
| (℃) | 80.5 | 80.5 |  | 392.27 | 396.0 |
| (℃) | 57.7 | 57.7 |  | 1.312 | 1.312 |
|  | 1086.4 | 1086.4 |  | 1.2941 | 1.294 |
|  | 300 | 300 |  | 1.289 | 1.299 |
|  | 109.2 | 109.2 |  |  |  |

**Table 9**

Validation for ejector cycle

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Term | Present | Ref. [37] | Term | Present | Ref. [37] |
| Fluid | R245fa | R245fa |  | 232.26 | 232.3 |
| (℃) | 67.85 | 67.85 |  | 409.48 | 410.1 |
| (℃) | 49.76 | 50.05 |  | 1.806 | 1.809 |
| (℃) | 6.85 | 6.85 |  | 1.823 | 1.830 |
| (℃) | 6.85 | 6.85 |  | 1.115 | 1.114 |
|  | 401 | 401 |  | 1.749 | 1.751 |
|  | 147.43 | 147 |  | 1 | 1 |
|  | 71.75 | 72 |  | 1.318 | 1.318 |
|  | 71.75 | 72 |  | 0.318 | 0.318 |
|  | 458.13 | 459.1 |  | 0.318 | 0.318 |
|  | 446.39 | 446.9 |  |  |  |

**4. Results and discussion**

4.1. Conditions of the system for simulation.

In this study, the exhaust gas and the jacket water come from a commercial cogeneration engine. The exhaust gas has the composition of , , and with the mass fraction of 9.1%, 7.4%, 9.3% and 74.2%, respectively. The main parameters about the engine are listed in Table 9.

**Table 9**

Main engine characters [2]

|  |  |
| --- | --- |
| term | Value |
| Electrical power output (kW) | 2928 |
| Fuel consumption (kW) | 7002 |
| Rated electricity efficiency | 41.8 |
| Engine speed (r/min) | 1000 |
| Exhaust gas temperature (℃) | 470 |
| Exhaust mass flow (kg/h) | 15673 |
| Engine jacket temperature (℃) | 79/90 |
| Engine jacket flow (kg/h) | 90 |

The thermodynamic parameters of the working fluid are calculated under the environment of MATLAB with the help of REFPROP 9.1 [38]. The conditions of simulation for the CCP system are listed in Table 10

**Table 9**

Condition of simulation for the CCP system

|  |  |
| --- | --- |
| term | Value |
| Ambient temperature (℃) | 20 |
| Ambient pressure (kPa) | 101.3 |
| Compressor inlet temperature (℃) | 35 |
| Compressor pressure ratio | 3 |
| BC turbine inlet temperature (℃) | 420 |
| BC turbine inlet pressure (kPa) | 1200 |
| High-pressure inlet temperature of ORC turbine (℃) | 146.85 |
| High-pressure inlet pressure of ORC turbine (kPa) | 3000 |
| Low-pressure inlet temperature of ORC turbine (℃) | 126.85 |
| Low-pressure inlet pressure of ORC turbine (kPa) | 1200 |
| Ejector primary flow inlet pressure (kPa) | 600 |
| Terminal temperature difference at gas heater outlet (℃) | 15 |
| Terminal temperature difference at preheater inlet (℃) | 13.15 |
| Pinch point temperature difference in vapor generator 1 (℃) | 10 |
| Pinch point temperature difference in vapor generator 2 (℃) | 60 |
| Pinch point temperature difference in vapor generator 3 (℃) | 5 |
| Ejector entrainment ratio | 0.46 |
| 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 low-pressure 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 |
| Outlet temperature of cooling water in precooler (℃) | 30 |
| Outlet temperature of cooling water in condenser 1 (℃) | 30 |
| Outlet temperature of cooling water in condenser 2 (℃) | 30 |

4.2. Thermodynamic and exergeconomic analysis



**Fig. 4.** Effect of the compressor inlet temperature on the thermodynamic performance of the system.



**Fig. 5.** Effect of the compressor inlet temperature on the exergoeconomic performance of the system.

The effect of the compressor inlet pressure on the thermodynamic performance is illustrated in Fig. 4. With the increase of the inlet temperature of the compressor, the power output of the Brayton cycle decreases. That can be explained that the BC turbine outlet temperature increases with the rise of the compressor inlet temperature. But the inlet temperature of the BC turbine is unchanged. Thus, the enthalpy drop of the in the turbine decreases, leading to the decrease of the power output of the BC turbine. The power output of the Brayton cycle WBC decreases.

The exhaust temperature increases which brings more thermal energy to the vapor generator 2. As a result, the mass flow rate of the working fluid increases, leading to the increase of the power output of the ORC turbine. The power output of the organic Rankine cycle WORC increases.

The decrease of the power output of the Brayton cycle is greater than that of the organic Rankine cycle. The total net power of the CCP system consists the power output of the Brayton cycle and the organic Rankine cycle. Thus, the net power of the CCP system W decrease slightly.

The increase of the mass flow rate in the organic Rankine cycle causes the drop of the jacket water temperature at the outlet of the preheater. Hence, the less thermal energy of the jacket water is provided in the vapor generator 3, resulting in the decrease of the mass flow rate in the ejector cycle. Thus, the cooling capacity of the ejector cycle decreases.

As for the decrease of the exergy efficiency of the CCP system, it can be explained that the exergy provided by the exhaust gas along with the jacket water remains unchanged. While the both the net power output of the system and the cooling capacity decreases. Thus, the exergy of the CCP system decreases.

The effect of the compressor inlet temperature on the exergoeconomic performance of the system is shown in Fig. 5. The average cost per unit of exergy of the power output of the Brayton can be separated to two parts. One is the exergy-fuel-related part and the other one is the equipment-cost-related part. The average cost per unit of exergy of the BC turbine (c3 which is equal to c4) decreases as shown in the figure. The decrease of the BC turbine power output cuts down the cost of the turbine. As a result, the equipment-cost-per part of the BC turbine power decreases. Combing the two parts mentioned above, the average cost per unit of exergy product of the Brayton cycle (cbt) decreases.

The exergy-fuel-related part of the power output of the organic Rankine cycle (c12) decreases. It can be explained that with the increase of the compressor inlet temperature, the log-mean temperature difference in the vapor generator 2 increases. As a result, the heat transfer area in the vapor generator decreases, cutting down the cost of the equipment. Thus, the average cost per unit of exergy fuel (c12) decreases. At the same, c12 is the exergy-fuel-related part of the average cost per unit of exergy product of the organic Rankine cycle. The equipment-cost-related part of the ORC turbine cycle increases as more power is provided by the turbine. However, the exergy-fuel-related part is more important. Thus, the average cost per unit of exergy product for the organic Rankine cycle (cot) decreases.

As mentioned before, the jacket water temperature decreases at the outlet of the preheater. Thus, the log-mean temperature difference in the vapor generator 3 decreases, resulting in the increase of the heat transfer area in the vapor generator. So, the cost of the vapor generator increases, leading to the rise of the average cost per unit of exergy fuel at the outlet of the ejector (c13). The c13 is the exergy-fuel-related part of the cooling product. The decrease of the equipment-cost-related part of the evaporator is less important when compared with the fuel part. As a result, the average cost per unit of exergy for the cooling increases.

With the increase of the mass flow rate in the organic Rankine cycle, the fictitious cost of the condenser 1 (ccond1) increases. Meanwhile, the fictitious cost of the condenser 2 (ccond2) in the ejector cycle decreases with the decrease of the mass flow rate. The value of the ccond2 is greater than that of the condenser 1. Compared with the fictitious cost of the condenser 1 and condenser 2, the fictitious cost of the preheater can be neglected. Thus, the fictitious cost part of the system product (ccproduct) decreases. The decrease of the Brayton cycle part (cbt) and the organic Rankine cycle part (cot) is greater than the increase of the cooling capacity part. Finally, the average cost per unit of exergy for the product of the whole system decreases.



**Fig. 6.** Effect of the pressure ratio on the thermodynamic performance of the system.

Fig. 6 explains the pressure ratio’s effect on the thermodynamic performance of the CCP system. With the increase of the pressure, the net power output of the Brayton cycle decreases. That can be explained that though the rise of the pressure causes the output of the BC turbine to increase, the power consumption of the compressor is much greater than the increase. As a result, the net power output of the Baryton cycle (WBC) decreases.

The increase of the pressure ratio enables the pressure of the at the outlet of the BC turbine to drop. Thus, the less heat is transfer at the vapor generator 2 and the mass flow rate of the working fluid in the high-pressure organic Rankine cycle decreases. The power output of the organic Rankine cycle decreases, as a result. The net power output of the whole CCP system consists of the power output of the Brayton cycle along with the organic Rankine cycle. The net power of the CCP system decreases.

Moreover, the decrease of the mass flow rate of the working fluid in the organic Rankine cycle causes the rise of the temperature of the jacket water at the preheater outlet. Thus, more thermal energy is utilized in the vapor generator 3. Hence, the mass flow rate of the primary flow fluid increases, leading to the increase of the mass flow rate of the fluid at the ejector outlet. Consequently, the cooling capacity of the evaporator increases.

The exergy provided by the exhaust gas and the jacket water remains unchanged while the net power of the CCP system decreases. The exergy of the cooling capacity accounts for only a little part of the whole system. Thus, the increase of the cooling capacity of the ejector can not change the drop the exergy efficiency of the system.

That less power is generated by the CCP system causes the decrease of the exergy efficiency of the system.



**Fig. 7.** Effect of the pressure ratio on the exergoeconomic performance of the system.

The effect of the pressure ratio on the exergoeconomic part of the system is shown in Fig. 7. The average cost per unit of exergy fuel (c4) decreases with the rise of the pressure ratio. That can be explained by the increase of the power provided by the BC turbine. And the average cost per unit of exergy of the power generated by the BC turbine decreases.

The average cost per unit of exergy fuel of the ORC turbine power (c12) decreases with the rise of the pressure ratio as shown in the Fig. Thus, the exergy-fuel-related part of the organic Rankine cycle power decreases, leading to the drop of the average cost per unit of exergy of the power output (cot).

With the rise of the inlet temperature of the jacket water at the vapor generator 3, the log-mean temperature difference of the vapor generator increases. As a result, the heat transfer area in the vapor generator decreases, leading to the decrease of the equipment-cost-related part of the fuel at the outlet of the ejector. The exergy-fuel-related part of the evaporator cooling capacity decreases, consequently. As for the equipment-cost-related part of the evaporator, the mass flow rate in the evaporator is just a small part of the total mass flow rate in the ejector. Thus, the increase of the evaporator cost is small. The decrease of the exergy-fuel-related part is greater than the equipment-cost-related part. As a result, the average cost per unit of exergy product of the cooling capacity (ccool) decreases.

As for the system product, the drop of the average cost per unit of exergy of the product (cproduct) can be divided into two parts. For the power output part, the decrease of the average cost per unit of exergy for the organic Rankine cycle power output and the cooling capacity counteracts with the increase of the Brayton cycle power output. On the fictitious cost part, the decrease of the mass flow rate in the organic Rankine cycle results in the drop of the rate of the cooing water in the condenser. Thus, the fictitious cost for the condenser 1 (ccond1) decrease. Compared with the fictitious cost for condenser 1, the increase of the fictitious for preheater (cpreh) and the condenser 2 (ccond2) is smaller. Considering the two parts mentioned above, the average cost per unit of exergy of the system product decreases.



**Fig. 8.** Effect of the BC turbine inlet temperature on the thermodynamic performance of the system.

Fig. 8 illustrates the effect of the BC turbine inlet temperature increase on the thermodynamic performance of the system. The power of the Brayton cycle increases with the increase of the inlet temperature. The increase of the BC turbine inlet temperature lifts the enthalpy drop of the in the expanding process. Hence, the power output of the BC turbine increases, leading to the increase of the net power output of the Brayton cycle.

The rise of the inlet temperature of the BC turbine at the same time lifts the outlet temperature, resulting in the rise of the temperature at the inlet of the vapor generator 2. Thus, the log-mean temperature difference in the vapor generator 2 increase, leading to the increase of the rise of the mass flow rate of the working fluid in the organic Rankine cycle. Hence, the power output of the organic Rankine cycle increases.

The rise of the Brayton cycle power output as well as the organic Rankine cycle power output lifts the power of the whole CCP power output.

The increase of the mass flow rate in the organic Rankine cycle causes the increase of the heat transfer in the preheater, which causes the drop of the jacket water temperature at the inlet of the vapor generator 3. Less thermal energy input in the vapor generator 3 results in the decrease of the mass flow rate in the ejector cycle, leading to the decrease of the cooling capacity.

Though the cooling capacity decreases with the increase of the BC turbine inlet temperature, the exergy of the cooling water provided by the evaporator is far less than that of the electricity power generated by the two cycles. Thus, the increase of the system net power brings about the rapid increase of the exergy efficiency of the CCP system.



**Fig. 9.** Effect of the BC turbine inlet temperature on the exergoeconomic performance of the system.

The change of the exergoeconomic performance with the increase of BC turbine inlet temperature is presented in Fig. 9. With the rise of the inlet temperature of the BC turbine, the cost of the gas heater increases because of the increase of the heat transfer area. Thus, the average cost per unit of exergy fuel (c3 which is equal to c4) increases, leading to the increase of the average cost per unit of exergy for the power of the Brayton cycle (cbt).

With the increase of the BC turbine inlet temperature, the outlet temperature of the BC turbine increases as well, resulting in the increase of the log-mean temperature difference in the vapor generator 2. Thus, the heat transfer area in the vapor generator decreases, cutting down the equipment cost. The average cost per unit of exergy fuel at the high-pressure inlet of ORC turbine (c12) decreases. Moreover, the average cost per unit of exergy for the ORC turbine power (cot) decreases.

As the temperature decrease of the jacket water in the vapor generator 3, the heat transfer area required in the vapor generator increases, causing the increase of the cost of the equipment. Thus, the average cost per unit of exergy fuel for the evaporator (c13) increase. As a result, the average cost per unit of exergy of the cooling capacity (ccool) increases.

The increase of the BC turbine inlet temperature requires more cooling water in the precooler, leading to the increase of the fictitious cost of the precooler (cprec). Likewise, the increase of the mass flow rate in the organic Rankine cycle needs more cooling water in the condenser 1, resulting in the increase of the fictitious cost of the condenser 1 (ccond1). At the same time, the increase of the average cost per unit of exergy for the BC turbine power and the cooling capacity counteracts the slightly decrease of the average cost per unit of exergy of the ORC turbine power. Thus, the average cost per unit of exergy of the system product (cproduct) increases.



**Fig. 10.** Effect of the low-pressure ORC turbine inlet temperature on the thermodynamic performance of the system.

Fig. 10 presents the effect of the increase of the low-pressure ORC turbine inlet temperature on the thermodynamic part of the system. It can be seen in the figure that the power output of the Brayton cycle remains unchanged. That’s because the Brayton cycle is the upper class of the organic cycle, the thermodynamic performance is not relevant to the organic Rankine cycle.

The increase of the low-pressure ORC turbine inlet temperature increases the power of the low-pressure part of the organic Rankine cycle. However, the rise of the inlet temperature causes the increase of the outlet temperature of the exhaust steam of the ORC turbine. While the working fluid of the high-pressure part shares the same ORC turbine with the working fluid in the low-pressure of the part. As a result, the outlet temperature of the exhaust steam rises while the inlet temperature of the working fluid keeps the same, leading to the decrease of the temperature difference for the working fluid in the expanding process. The enthalpy drop of the working fluid decreases, resulting the decrease of the power output of the high-pressure part. The mass flow rate of the working fluid in the high-pressure part of the organic Rankine cycle is far more than that in the low-pressure part. Thus, the net power output of the organic Rankine cycle decreases.

The drop of the net power output of the organic Rankine cycle combines with the unchanged Brayton cycle explains the drop of the net power output of the whole CCP system.

The increase of the temperature at the outlet of the ORC turbine means that less heat is absorbed in the preheater by the working fluid of the organic Rankine cycle. Thus, heat is released in the vapor generator 3, bringing about the increase of the mass flow rate of the primary fluid. Consequently, the cooling capacity of the ejector increases.

With the drop of the net power of the CCP system which is far more than the exergy of the cooling water provided by the evaporator, the exergy efficiency of the whole CCP system drops as a result.



**Fig. 11.** Effect of the low-pressure ORC turbine inlet temperature on the exergoeconomic performance of the system.

It can be seen in Fig. 11 that the average cost per unit of exergy for the steam at the high-pressure inlet of the ORC turbine (c12) increases. With the increase of the low-pressure inlet temperature of the ORC turbine, the log-mean temperature difference in the vapor generator 1 decreases. The heat transfer area required exchange heat increases, causing the increase of the cost of the average cost per unit of exergy for the steam at the vapor generator outlet (c7 which is equal to c12).

The increase of average cost per unit of exergy fuel (c12) for the ORC turbine results in the increase of the average cost per unit of exergy of the power generated by the ORC turbine.

The rise of the average cost per unit of exergy of the steam at the high-pressure inlet of the ORC turbine causes the increase of the average cost per unit of exergy for the steam at the outlet of the BC turbine (c4). The exergy-fuel-related part of the BC turbine (c3 which is equal to c4) increases, leading to the increase of the average cost per unit of exergy for the power of the BC turbine.

The increase of the temperature of the jacket water at the inlet of the vapor generator 3 means the increase of the log-mean temperature difference in the vapor generator. Thus, the required heat transfer area decreases, leading to the decrease of the cost of the vapor generator. The average cost per unit of exergy for the fluid at the outlet of the ejector decreases. Hence, the exergy-fuel-related part of the cooling capacity decreases. The average cost per unit of exergy of the cooling capacity decreases.

The average cost per unit of exergy for the power produced by the ORC turbine accounts for a large part of the system product. The increase of cot determines the rise of the system product.



**Fig. 12.** Effect of the high-pressure ORC turbine inlet temperature on the thermodynamic performance of the system.

The effect of the high-pressure ORC turbine inlet temperature on the thermodynamic performance of the system is illustrated in Fig. 12. The net power output of the Brayton cycle remains the same, because of the upper class of the cycle.

With the increase of the high-pressure inlet temperature of the ORC turbine, the outlet temperature of the exhaust steam increases. However, the enthalpy drop of the working fluid in the increases, leading to the increase of the net power output of the high-pressure part. Meanwhile, the net power output in the low-pressure part decreases. Whereas, the mass flow rate in the low-pressure is far less than that in the high-pressure part. The decrease of the low-pressure power can not offset the increase of the power output in the high-pressure part. Thus, the net power of the whole organic Rankine cycle increases.

Consequently, the net power of the whole CCP system increases with the increase of the high-pressure inlet temperature of the ORC turbine.

Because of the rise of the temperature of the working fluid at the outlet of the ORC turbine, less heat is released in the preheater. As a result, the temperature of the jacket water at the inlet of the vapor generator 3 increases. The mass flow rate of working fluid in the ejector cooling cycle increases, explaining the increase of the cooling capacity.

The increase of net power of the CCP system along with the cooling capacity brings about the increase of the exergy efficiency of the whole system.



**Fig. 13.** Effect of the high-pressure ORC turbine inlet temperature on the exergoeconomic performance of the system.

Fig. 13 shows the effect of the high-pressure inlet temperature of the ORC turbine on the exergoeconomic performance of the system. The average cost per unit of exergy fuel for the ORC turbine (c12) decreases with the increase of the inlet temperature. The can be explained by the increase of the net power output of the organic Rankine cycle. Thus, the exergy-fuel-related part of the ORC turbine decreases, leading to the decrease of the average cost per unit of the power provided by the ORC turbine.

The decrease of the average cost per unit of exergy of the steam at the inlet of the high-pressure inlet of the ORC turbine causes the decrease of the average cost per unit of the fluid (c4) in the vapor generator 2. Thus, the average cost per unit of exergy fuel for the BC turbine (c3 which is equal to c4) decreases, resulting in the decrease of the average cost per unit of exergy for the power generated by the BC turbine.

The increase of the high-pressure inlet temperature of the ORC turbine rises the exhaust steam temperature. Thus, the temperature increases at the inlet of the vapor generator 3. The cost of the equipment decreases as less heat transfer area is needed, leading to the decrease of the average cost per unit of exergy for the cooling capacity.

The fictitious cost of the precooler keeps unchanged because of the unchanged parameters of the Brayton cycle. The increase of the outlet temperature of the ORC turbine require more cooling water in the condenser 1, leading to the increase of the fictitious cost. However, the average cost per unit of exergy of the BC turbine, the ORC turbine and the cooling capacity decreases, counteracting the increase of the increase of the fictitious. As a result, the average cost per unit of exergy for the system decreases.



**Fig. 14.** Effect of the ejector primary flow pressure on the thermodynamic performance of the system.



**Fig. 15.** Effect of the ejector primary flow pressure on the exergoeconomic performance of the system.

Fig. 14 presents the effect of the ejector primary flow pressure on the thermodynamic performance of the CCP system. The power output of the Brayton cycle and the organic Rankine cycle remain the same. That is because the increase of the parameters in the ejector cycle can not affect the performance in the Brayton cycle and the organic Rankine cycle.

The net power output of the whole system increases slightly. The net power output of the system consists of the net power of the Brayton cycle, the net power output of the organic Rankine cycle and the consumption of the pump 3 in the ejector cooling cycle. The increase of the primary flow pressure decreases the pressure ratio of the pump 3, resulting the decrease of the pump consumption. Thus, the net power output of the system increases slightly.

The cooling capacity of the ejector cooling cycle decreases rapidly. With the increase of the primary flow pressure, the superheated steam produced by the vapor generator decreases rapidly. Thus, the mass flow rate of the fluid in the ejector cycle decreases, leading to the decrease of the cooling capacity.

Less mass flow rate of the working fluid in the ejector cooling cycle means the increase of the outlet temperature of the jacket water in the vapor generator 3. The exergy efficiency of the ejector cooling cycle is much smaller than the exergy efficiency of the electricity generation cycles. Thus, while the cooling capacity decrease, the exergy efficiency of the whole CCP system increases.

As seen in Fig. 15, the exergoeconomic performance is affected by the increase of the primary flow pressure. The primary flow pressure can not affect the performance of the Brayton cycle and the organic Rankine cycle. As a result, c4, c12, cbt and cot remain the same.

The increase of the primary pressure the causes the increase of the exergy-fuel-related part (c13). Therefore, the average cost per unit of exergy for the cooling capacity increases.

As for the average cost per unit of exergy for the system product, the change of the fictitious cost of the condenser 1 should be taken into consideration. The decrease of the mass flow rate in the ejector cycle permits the decrease of the cooling water in the condenser, resulting in the decrease of the fictitious cost of the condenser. The fictitious cost of the condenser offsets the increase of the average cost per unit of exergy of the cooling capacity. Considering the unchanged part of the product, the average cost per unit of exergy for the system product decreases slightly.

4.3. System optimization

The performance of the system changes with variation of the selected parameters. The net power of the system increases with the increase of the high-pressure inlet temperature of the ORC turbine while decrease with the increase of the low-pressure inlet temperature. Moreover, the average cost per unit of exergy for the system fluctuates with the change of the parameters. Thus, to gain the highest power output and the cooling capacity at the lowest expense, optimization of the system is needed.

Considering that the average cost per unit of exergy reflects the thermodynamic and the economic aspect of the system, the average cost per unit of exergy for the system product is selected as the objective function and genetic algorithm (GA) is selected to realize the single-objective optimization.

GA is an optimization method based on the natural biological evaluation. [39] It simulates the natural genetic rules and searches the optimization result in all the generation. For the optimization, five parameters are chosen as the variate for GA and the ranges of these parameters are listed in Table 3.

**Table 3**

|  |  |
| --- | --- |
| Ranges of the decision variables | Range |
| Compressor pressure ratio | 2.8-3.5 |
| Compressor inlet temperature (℃) | 30-37 |
| BC turbine inlet temperature (℃) | 400-450 |
| Low-pressure ORC turbine inlet temperature (℃) | 360-420 |
| High-pressure ORC turbine inlet temperature (℃) | 400-500 |

**Table 4**

Single-objective optimization results

|  |  |
| --- | --- |
| Term | Value |
| Compressor pressure ratio | 3.268 |
| Compressor inlet temperature (℃) | 37 |
| BC turbine inlet temperature (℃) | 410.182 |
| Low-pressure ORC turbine inlet temperature (℃) | 97.657 |
| High-pressure ORC turbine inlet temperature (℃) | 195.646 |
| Preheater outlet temperature (℃) | 76.85 |
| Ejector primary flow pressure (kPa) | 600 |
| Net power output (kW) | 577.5483 |
| Cooling capacity (kW) | 498.5947 |
| Exergy efficiency (%) | 51.09 |
| Average cost per unit of exergy product ($/MWh) | 36.4684 |

**Table 5**

Optimization results for the components in the system

|  |  |  |
| --- | --- | --- |
| Term | Average cost per unit of exergy($/MWh) | Power (kW) |
| BC turbine power | 19.5213 | 604.40 |
| ORC turbine power | 27.2183 | 518.21 |
| Compressor power | / | 531.07 |
| Pump 1 power | / | 7.16 |
| Pump 2 power | / | 5.32 |
| Pump 3 power | / | 1.81 |

The optimization results of GA are listed in Table 4. It can be obtained that the minimum average cost per unit of exergy for the system product cproduct is 36.4684 $/MWh. The exergy efficiency of the CCP system is 51.09% which is also desirable.

The operation conditions of the fluid machineries are listed in Table 5. The power provided by the BC turbine is 604.4 kW and the power generated by the ORC turbine is 518.21 kW. The consumption of the compressor is 531.07 kW which accounts for a large proportion of the BC turbine. The average cost per unit of exergy for the power of the BC turbine and the ORC turbine are 19.5213 $/MWh and 27.2183 $/MWh, respectively. The cost of the electricity from the national grid is 133$/MWh which is much more expensive.

**5. Conclusion**

In this paper, a combined cooling and power system is developed. Six parameters (compressor inlet temperature, compressor pressure ratio, BC turbine inlet temperature, high-pressure inlet temperature of the ORC turbine, low-pressure inlet temperature of the ORC turbine and the primary flow pressure) are selected to analyze the thermodynamic performance and the exergoeconomic performance of the system. Single-optimization is carried out with the help of GA. The conclusions of this study are presented as follows:

(1) The increase of the BC turbine inlet temperature and the high-pressure inlet temperature of the ORC turbine contribute to the increase of the net power of the CCP system while the increase of the low-pressure inlet temperature of the ORC turbine and the primary flow pressure in the ejector cycle obstruct the increase of the net power.

(2) The average cost per unit of exergy of the ORC turbine power accounts for a large part of the average cost per unit of exergy of the system while the average cost per unit of exergy of the cooling capacity is less important. The fictitious cost of the system also has a great influence of the average cost per unit of exergy for the system product.

(3) By the single-optimization, the minimum average cost per unit of exergy for the system product is obtained as 36.4684 $/MWh. The net power of the CCP system is gotten as 577.5483 kW. The cooling capacity of the CCP system is harvested as 498.5947 kW.

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