**Exergoeconomic and Exergoenvironmental analysis of a combined heating and power system**

**Abstract**

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This paper develops a geothermal driven combined heating and power system to realize the multi-generation. Organic Rankine Cycle is utilized to generate power and radiation floor heating system is employed to provide heat for the buildings. Comprehensive thermodynamic, exergoeconomic and exeergoenvironmental models of the system are performed and the capital cost and environmental impacts of the system components are analyzed. Superheat degree of the organic working fluid and inlet pressure of the turbine are selected as variables to assess the system performance. Results show that the increase of the superheat degree and the inlet pressure would cause the decrease of the turbine power output and the increase of the heat supplying. Condensation water holds the highest levelized exergy cost and heat exchanger 1 has the highest environmental impact reducing potential.

Keyworks:

Geothermal energy

Organic Rankine Cycle

Radiation floor heating

Exergoeconomic analysis

Exergoenvironmental analysis

**Nomenclature**

|  |  |  |  |
| --- | --- | --- | --- |
| A | area, m2 | Z | annually levelized cost value, $ year-1 |
| b | environmental impact per exergy unit | z | capital cost, K$ |
|  | environmental impact | Greek symbol | |
| Bo | boiling number | α | convection heat transfer coefficient, W m-2 K-1 |
| c | average cost per unit of exergy, $ (MWh)-1 | λ | heat conductivity, W m-1 K-1 |
| C | cost rate, $ year-1 | ρ | density, kg m-3 |
| CCP | combined cooling and power | μ | dynamic viscosity, m2 s-1 |
| CRF | capital recovery factor | η | efficiency, % |
| CEPCI | chemical engineering plant cost index | δ | thickness, m |
| D | diameter, m | Subscribe | |
| e | exergy, kJ kg-1 | 1-13 | state points |
| E | exergy flow rate, kJ s-1 | BM | bare module |
| Ey | exergy flow rate per year, kJ year-1 | 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 |
|  | mass flow rate, kg s-1 | F | fuel |
| n | lifetime, year | he | heat exchanger |
| Nu | Nusselt number | L | loss |
| P | pressure, MPa | l | liquid |
| Pr | Prandtl number | M | material |
| Pt | center distance between tubes, m | P | product |
| Pr | reduced pressure | p1 | pump 1 |
| Q | heat transfer rate, kW | p2 | pump 2 |
| qm | average imposed wall heat flux, W m-2 | s | shell |
| rf | enthalpy of vaporization, kJ kg-1 | t | tube |
| RFH | radiation floor heating system | th | thermal |
| T | temperature, K | turb | turbine |
| U | overall heat transfer coefficient, W m-2 K-1 | vg | vapor generator |
| W | power, kW | w | tube wall |
| Wy | annually power, MWh year-1 |  |  |
| x | vapor quality |  |  |

**Introduction**

The increasing development of the globe economy and industry raises the attention to energy problems. Nowadays, traditional fossil fuels such as coat, oil and gas take up the major part of the consumed resource and also cause the global environmental problems [1]. The increasing fossil fuel price and aggressive environmental problems have urged the using of alternative resources [2]. Renewable energy, environment friendly and naturally replenished, is introduced and developed by many researchers.

Geothermal energy is an important kind of renewable energy with huge reserve in internal earth. Theoretically calculating, there are approximately 3.61014 GWh geothermal energy in the earth’s crust [3]. The utilization of geothermal energy has been carried out around the world. Sun et al. [4] have assessed the benefits of deep geothermal energy utilization in China. Yousefi et al. [5] proposed a high-efficient plan for geothermal energy utilization in Iran.

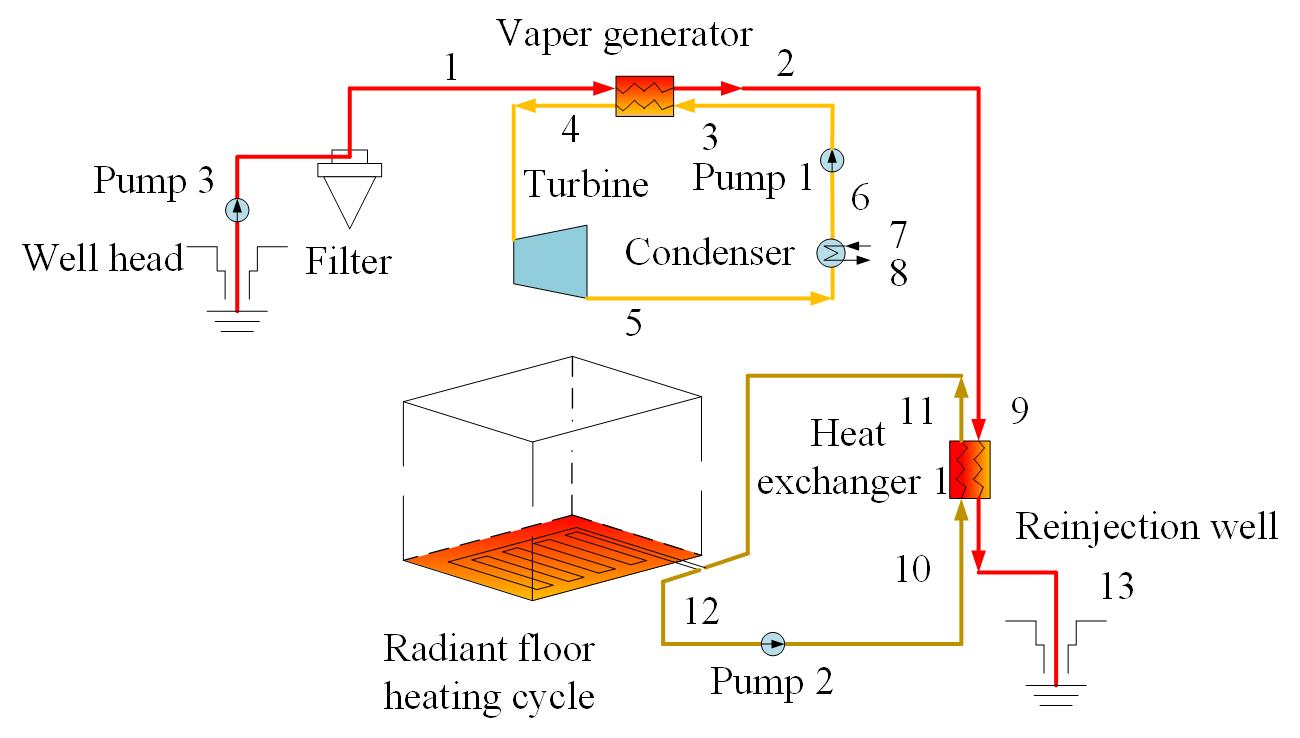
Thermal systems converting geothermal energy into electricity power have been widely researched. Most of the geothermal fields around the world are wet fields with outlet temperature between 80 ℃ and 150 ℃, which are suitable for low-grade energy power generation cycle such Organic Rankine Cycle (ORC) and Kalina Cycle (KC) [6]. Zhao et al. [7] designed a geothermal driven power generation system. With the 170 ℃ geothermal water temperature, they used a flashing device to separate the geothermal water into brine stream and steam stream, which were used to heat the organic working fluid vapor for the ORC and to drive a steam turbine respectively for power generation. The power generation system was optimized towards maximal thermal efficiency of 14.61% and obtained the power output 1091.6 kW and 271.2 kW for the steam turbine and the ORC turbine. Zare et al. [8] developed a geothermal power generation system coupled with Kalina cycle and thermoelectric generator. Ammonia-water was heated in an evaporator by the geothermal water into two-phase flow mixture. The ammonia-water vapor part was used to drive the turbine and the liquid part along with the exhaust turbine vapor was used to drive the thermoelectric generator. Performance assessment indicated a 7.3% power output, energy efficiency and exergy efficiency enhancement compared with the conventional Kalina cycle.

For the high thermal efficiency, geothermal energy is often used to provide heat [9]. Combined power and heating systems driven by geothermal water are conducted by many researchers. He et al. [10] designed a combined power and water system driven by geothermal energy. In a flashing evaporator, geothermal water was separated into vapor and liquid. The geothermal vapor flowed into a turbine to generate power and the rest geothermal water was used to the seawater, which was sprayed into the humidifier to rise the temperature and humidity of the ambient air. Simulating results showed that the maximal power output of the steam turbine was 157 kW with 1379.56 kW of the spraying heat supply system. Erdeweghe et al. [11] proposed a combined heat and power system and compared the performance of different configurations. Geothermal water was used to provide heat for an ORC and a district heating (DH) system. Comparison results indicated that the configuration with geothermal water heating the organic working fluid in an evaporator first and then separating in parallel to heat the DH system and a preheater had the best performance.

Besides the thermodynamic performance of the system, the economic performance and the environmental impact should also be taken into consideration when designing a thermal system. Exergoeconomic analysis and exergoenvironmental analysis were introduced by researchers to address the issues. Exergoeconomic analysis is a thermodynamic analysis combining the principles of exergy and economics to assess the exergy cost of the system [12]. Capital costs of system components are considered and levelized exergy cost is calculated as an assessment indicator. Exergoenvironmental analysis likewise is an analysis coupling the principles of exergy and environmics to assess the environmental impacts of the system [13]. In exergoenvironmental analysis, system components are considered by life cycle assessment (LCA) with five phases [14]: materials, production processes, transport processes, energy generation processes and disposal scenarios. Environmental impacts of system components are considered and environmental impact rate per exergy unit is calculated. Thermal systems assessed by exergoeconomic analysis or exergoenvironmental analysis can be found in past publication. Bina et al. [15] compared the performance of a geothermal power plant with single flash or double flash cycle by exergoeconomic model. Results shows that the produce cost of the double flash cycle is higher than that of the single flash cycle (4.39 $/GJ vs. 4.104 $/GJ). Ghaebi et al. [16] developed a geothermal heat source Kalina cycle system. Exergoeconomic analysis was used to assess the system performance and product cost was selected for the system optimization. Kecebas et al. [17] proposed a geothermal district heating system and exergoenvironmental analysis was used to guide the system design.

In this paper, a combined heating and power system driven by geothermal water is developed. Organic Rankine cycle is used to utilize the high temperature geothermal water to generate power. After the ORC, geothermal water is still in a relative high temperature state and can be used to provide heat. In the past studies, geothermal water is primarily utilized directly as hot water. Some studies would use geothermal water to drive a heat system to provide heat for buildings with conventional heat exchangers. Radiant floor heating systems, which require lower driven temperature and provide more comfortable heat supplying, are always ignored. In this paper, we design a radiant floor heating system to further utilize the geothermal water energy and provide heat for buildings. Though exergoeconomic analysis and exergoenvironmental analysis are used by past researchers to assess the system performance and guide the system design, most of them used one of the two analysis: either the exergoeconomic or exergoenvironmental analysis. Few systems were analyzed by the two methods at the same time. In this paper, we used both exergoeconomic and exergoenvironmental analysis to assess the system performance.

**System description**



**Fig. 1.** Schematic diagram of the combined heating and power system

The diagram of the combined heating and power system is shown in Figure 1. The system is composed of an organic Rankine cycle (ORC), and a radiant floor heating cycle (RFC). As presented in Figure 1, geothermal water is pumped from the geothermal water well, after separating sand, to drive the whole system. In the vapor generator, the geothermal water provides heat for the organic working fluid, which finally becomes the superheated vapor to flow into the turbine to generate power. The exhaust organic working fluid, after expanding in the turbine, is cooled to saturated liquid in the condenser and then pumped to the vapor generation to complete the cycle. After releasing heat for the ORC, geothermal water is still in a high temperature state and provide heat for the radiant floor heating cycle. Water absorbs heat from the geothermal water in heat exchanger 1 and flows into the building to provide heat. Then the cool water is pumped to heat exchanger 1 to absorb heat and complete the heating supply cycle. Finally, the geothermal water flows into the reinjection well.

R245fa is selected as the working fluid for the organic Rankine cycle because of the great thermodynamic performance and the low environment effect [18] [19].

**System model**

The following assumptions are made to simulate the system:(1) the system operates in a steady state; (2) friction and heat losses are not taken into account; (3) the pressure losses in the heat exchanger 1, vapor generator and condenser are not considered. (4) the working fluid out of the condenser is saturated liquid; (5) the throttle valve process is isenthalpic.

Energy model

The net power of the ORC is expressed as:

 (1)

The heat load of the RFC is given as:

 (2)

The energy model equations for each component are list in Table 1.

**Table 1**

Energy analysis for each component in the CHP system

|  |  |
| --- | --- |
| Component | Energy equation |
| ORC turbine |  |
| Vapor generator |  |
| Condenser |  |
| Pump 1 |  |
| Heat exchanger 1 |  |
| Pump 2 |  |

Exergy model

Differing from the energy model of the system which is based on the first law of thermodynamics, exergy model of the system is based on the second lay of thermodynamics and focuses on the evaluation the quality of the thermal energy. The exergy of a system is the maximum part of useful work in the equilibrium process of the system. The expression of the exergy equation for unit working fluid is given as:

 (3)

where the subscript 0 expresses the ambient condition.

Thus, the exergy flow rate in the *k*th point of the system can be calculated as:

 (4)

For each component, there is an exergy balance equation considering the exergy generation and exergy destruction, being expressed as [20]:

 (5)

where the  is the exergy provided for the component as fuel;  is the exergy generated by the component as product;  is exergy destruction during the operation of the component and  is the exergy is the exergy loss to the outside of the system.

The exergy efficiency () and the exergy destruction ratio () is calculated as:

 (6)

 (7)

The exergy model equations for each component are listed in Table 2.

**Table 2**

Exergy analysis for each component in CHP system

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| Component |  |  |  |  |
| ORC turbine |  |  |  | / |
| Vapor generator |  |  |  | / |
| Condenser | / | / |  |  |
| Pump 1 |  |  |  | / |
| Heat exchanger 1 |  |  |  | / |
| Pump 2 |  |  |  | / |

Heat exchanger model

Considering the high efficiency and low price, all the heat exchangers in this study are chosen as shell and tube type. We classify the heat exchangers, based on the heat transfer process, into the single-phase type (the heat exchanger 1) and the two-phase type (the vapor generator and the condenser).

To calculate the heat transfer area accurately, we discretized the heat transfer process into many subsections in which the thermodynamic properties of the working fluid are assumed changed and uniform.

Single-phase heat transfer type

The heat transfer equation in the *i*th subsection is given as:

 (8)

where *Q* is the heat transferred in the subsection; *U* is the overall heat transfer coefficient; *A* is the heat transfer area and *T* is the log-mean temperature difference (LMTD).

The equation to obtain the overall heat transfer coefficient is:

 (9)

where *α* is the convection heat transfer coefficient of the working fluid and subscript h and c represent the hot and cool working fluid, respectively; *λ* and *δ* are the thickness and the thermal conductivity of the tube wall.

The equation to get the log-mean temperature difference is:

 (10)

The equation to obtain the convection heat transfer coefficient of the tube side is:

** (11)

The equation of the Nusselt number is expressed as [21] [22]:

** (13)

** (14)

where *f* is the Darcy frication factor, being expressed as [23]:

** (15)

The equation of the Reynolds number is:

** (16)

where *G*in is the velocity of the fluid in the tube, being expressed as:

** (17)

where *N* is the number of the tubes in the heat exchanger.

The equation of the Prandtl number is:

** (18)

The equation of the convection heat transfer coefficient for the shell side is [24]:

** (19)

where *D*es is the equivalent diameter of the shell, being expressed as:

** (20)

Two-phase heat transfer type

There is phase change in the two-phase heat transfer process: liquid working fluid is heated to vapor or exhaust working vapor is condensed to saturate liquid.

The equation of the convection heat transfer coefficient in the evaporation process is expressed as [25]:

** (21)

where *Pr* is the ratio of the state-point pressure to the critical pressure of the working fluid; and *Bo* is the boiling number:

** (22)

The equation of the convection heat transfer coefficient in the condensation process is given as [26]:

** (23)

Radiant floor heating model

In the cold winter, a heating system can maintain the inside temperature at a comfortable level for living. Traditional heating system, which provides heat for the building through a small liquid-air heat exchanger, would cause an ununiform temperature field in the building: the closer to the heat exchanger, the warmer it is. The radiant floor heating system greatly addressed this issue that the whole floor is a large heat exchanger. The radiant floor heating system provides heat from the lower floor up to the air, causing a more comfortable temperature field. Moreover, the driven temperature of the hot water for the radiant floor heating system is lower than that of the traditional system.

Heat balance of the building

Maintaining the stable temperature of the house requires the heat loss of the building be equal to heat provided by the heat source. There are mainly two ways of heat losing: the heat conduction through the wall and the air convection through doors and windows.

Heat loss through the wall

The heat transfer equation of the wall is given as:

 (24)

The overall heat transfer coefficient of the building is expressed as:

 (25)

where *α* is the convection heat transfer coefficient of the air and subscript in and out represent inside and outside the building, respectively; *λ* and *δ* are the thickness and the thermal conductivity of the materials of the wall. The specific thickness and the thermal conductivity of the wall materials are listed in Table 3.

**Table 3**

Thermal conductivity of the wall materials

|  |  |  |
| --- | --- | --- |
| Material of layer | Thickness/mm | Thermal conductivity/Wm-1K-1 |
| 1:3 cement mortar of outer wall | 12 | 0.93 |
| Insulation mortar of outer wall | 60 | 0.29 |
| Moisture proof mortar of outer wall | 17 | 0.93 |
| Brick of outer wall | 370 | 0.81 |
| 1:2.5 cement mortar of outer wall | 6 | 0.93 |
| 1:2.5 cement mortar of inner wall | 6 | 0.93 |
| Moisture proof mortar of inner wall | 17 | 0.93 |
| Brick of inner wall | 240 | 0.81 |
| 1:2.5 cement mortar of inner wall | 6 | 0.93 |
| Insulation of roof | 100 | 0.04 |
| Moisture proof of roof | 20 | 0.1 |

Heat loss through the inflow of clod air

The heat loss of the inflow air is given as:

 (26)

where *c*p is the specific heat of the air at a mean temperature of the inside and outside; is the average mass of the air flowing into the building.

Heat supply of the building

The heat transfer to the room by means of radiation is expressed as:

 (27)

The heat transfer equation of the convection is expressed as [27]:

 (28)

The average temperature of the floor surface is determined by:

 (29)

where *Q*e is the effective heat load to the building

Exergoeconomic model

Exergoeconomic analysis considers the thermodynamic performance as well the economic performance of the system. Besides the thermal efficiency, the capital costs of the system components should also be taken into account, for it greatly determines the feasibility of the system [28]. Under the basic conditions, the costs of components can be obtained counting the price of material, cost of construction, installation labor and indirect project expenses such as taxes. For components operating under specific conditions, multiplying factors including the specific equipment type, specific system pressure and specific materials are used to amend the costs.

Because of the inflation, it is necessary to consider the changes of the basic cost of the components, for it is based on the past records and published information. The equation for the correction is given as:

 (30)

where *I* is the cost index and *C* is the component cost; subscript 1 represents the basic time and subscript 2 represents the present time.

The cost of the heat exchanger is calculated as:

 (31)

where *B*i,he is the constant corresponding to the type of the heat exchanger; *F*M,he and *F*P,he are the material factor and pressure factor for correction; the *CEPCI* (chemical engineering plant cost index) is used as the cost index and the values of *CEPCI*2016 and *CEPCI*ref are 541.7 and 397, respectively [29].

The material factor is selected based on the materials and the pressure factor is given as [28]:

 (32)

where *C*i,he is the constant corresponding to the type of the heat exchanger and *P*he is the operation pressure of the heat exchanger.

The basic cost of the heat exchanger is calculated as [28]:

 (33)

where *K*i,he is the constant corresponding to the type of the heat exchanger; *A* is the heat transfer area of the heat exchanger.

The cost of the turbine is calculated as [28]:

 (34)

where *F*BM,turb is the constant corresponding to the type of the turbine.

The basic cost of the turbine is given as [28]:

 (35)

where *K*i, turb is the constant corresponding to the type of the turbine and  is the power output of the turbine.

The cost of the pump is calculated as [28]:

 (36)

where *B*i,pump is the constant corresponding to the type of the pump; *F*M,pump and *F*P,pump are the material factor and pressure factor for correction;

The material factor is selected based on the materials and the pressure factor is given as [28]:

 (37)

where *C*i,pump is the constant corresponding to the type of the pump and *P*pump is the designed pressure for the pump.

The basic cost for the pump is given as [28]:

 (38)

where *K*i,pump is the constant corresponding to the type of the pump and  is the power consumption of the pump.

The values of the constants (*B*, *C*, *F*, *K*) are listed in Table 4 [28].

**Table 4**

Constants for component costs [28].

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

Considering the operation life time of the components, the capital cost of the component should be translated into annually levelized cost, which is calculated as [30]:

 (39)

where *CRF* is the capital recovery factor, being expressed as [30]:

 (40)

where *i*eff is the effective discount rate and *n* is the operation life time system, the value of *i*eff and *n* are selected as 0.05 and 30 [31].

The flowing of the working fluids in the system component is accompanied with the production or destruction of the exergy, so is the increase or the decrease of the exergy cost. When the working fluid operates in a component, the capital cost of the component is passed to the working fluid as exergy cost. The products of the working fluid, as a result, obtain the capital cost meaning. The equations for the calculation of the working fluid product are expressed as:

 (41)

 (42)

 (43)

 (44)

where *c* is the levelized exergy cost of the working fluid.

The levelized exergy cost balance equation of the system component is given as:

 (45)

The levelized exergy cost balance equations for each component are listed in Table 5

**Table 5**

Cost balance and auxiliary relation for each component of the system

|  |  |  |
| --- | --- | --- |
| Component | Cost balance | Auxiliary relation |
| Geothermal resource |  | / |
| ORC turbine |  | c4=c5 |
| Vapor generator |  | c1=c2 |
| Pump 1 |  | celec=cot |
| Condenser |  | c5=c6 |
| Heat exchanger 1 |  | c9=c13 |
| Pump 2 |  | celec=cot |

Erergoenvironmental model

Exergoenvironmental analysis is a method connecting thermodynamics and environmics by assessing environmental impacts with thermodynamic concepts. Environmental impacts should be taken into account when designing a thermal system considering the aggravating of the environmental problems. There are three steps for the exergoenvironmental analysis [32]: first, the exergy analysis for the overall system; second, calculating the environmental impacts by means of life cycle assessment (LCA) method; finally, connecting the environmental impacts with exergy streams.

The life cycle assessment is used to evaluated the environmental impacts of a product over its lifetime. The equation of the component-related environmental impact for component is given as:

 (46)

where  is the construction environmental impact of a component;  is the operation and maintenance environmental impact of a component and  is the disposal environmental impact of a component.

The environmental impacts for the system components are list in table 6 [33].

**Table 6**

Environmental impacts of the system components [33]

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Component | Material composition | material/ mPts/kg | Process mPts/kg | Disposal mPts/kg | Total mPts/kg |
| Turbine | Steel 25%  Steel high alloy 75% | 202 | 11.7 | -70 | 143.7 |
| Heat exchanger | Steel 67%  Copper 33% | 519 | 12.1 | -70 | 461.1 |
| Condenser | Steel 100% | 86 | 12.1 | -70 | 28.1 |
| Pump | Steel 35%  Cast iron 65% | 186 | 16.9 | -70 | 132.9 |

The weight equation of the ORC turbine is given as [33]:

 (47)

The weight equation for the heat exchanger is given as [33]:

 (48)

The weight equation for the condenser is given as [33]:

 (49)

The weight equation for the pump is given as [33]:

 (50)

The Eco-indicator 99, the value of which represents one thousandth of the yearly environmental impact for one average European inhabitant, is used to assess the system [32]. Analogous to the exergoeconomic, the environmental impacts of components are passed to the working fluid as environmental impact rate. The equation of the environmental impact balance is given as:

 (51)

 (52)

 (53)

where  is the environmental impact rate and *b* is the environmental impact per exergy unit.

To assess the environmental impacts of the components, relative difference (*r*b) and exergoenvironmental factor (*f*b) are selected [32].

 (54)

 (55)

where  is the environmental impact rate related to the exergy destruction, being expressed as:

 (56)

Relative difference of a component indicates the environmental impact reducing potential. In general, the environmental impact of a component with a high relative difference is easier to be reduced than that with a lower relative difference. Exergoenvironmental factor indicates the ratio of component-related environmental impact to the total environmental impact of a component. When the exergoenvironmental factor is higher than 0.7 the component-related environmental impact of a component is more important than the effect of the exergy destruction. When the exergoenvironmental factor is lower than 0.3, the environmental impact is dominated by the exergy destruction.

The environmental impact balance equations of the system components are list in Table 7

**Table 7**

Environmental impact balance and auxiliary relation for each component of the system

|  |  |  |
| --- | --- | --- |
| Component | Environmental impact balance | Auxiliary relation |
| Geothermal resource |  | / |
| ORC turbine |  | b4=b5 |
| Vapor generator |  | b1=b2 |
| Pump 1 |  | belec=bot |
| Condenser |  | b5=b6 |
| Heat exchanger 1 |  | b9=b13 |
| Pump 2 |  | belec=bot |

Simulation conditions for the system

The exergoeconomic analysis and exergoenvironmental analysis of the system is calculated by means of REFPROP 9.1 [34] under the environmental of MATLAB. The basic simulation conditions of the system are listed in Table 8.

**Table 8**

Simulation conditions of the system

|  |  |
| --- | --- |
| Term | Value |
| Ambient temperature (℃) | 5 |
| Ambient pressure (kPa) | 101.3 |
| Mass flow rate of geothermal water (kg/s) | 36.11 |
| Geothermal water temperature (℃) | 130 |
| Geothermal water pressure (kPa) | 500 |
| Pinch point temperature difference in vapor generator 1 (℃) | 10 |
| Superheat degree of organic working fluid (℃) | 10 |
| ORC turbine inlet pressure of (kPa) | 1000 |
| Condensation temperature of condenser 1 (℃) | 30 |
| Condenser outlet temperature of organic working fluid (℃) | 10 |
| Isentropic efficiency of ORC turbine (%) | 76 |
| Isentropic efficiency of pump 1 (%) | 70 |
| Isentropic efficiency of pump 2 (%) | 70 |
| Isentropic efficiency of pump 3 (%) | 70 |
| Inlet temperature of cooling water (℃) | 5 |
| Outlet temperature of cooling water in condenser 1 (℃) | 10 |
| Inlet temperature of radiant floor heating cycle (℃) | 45 |
| Outlet temperature of radiant floor heating cycle (℃) | 35 |
| Temperature of reinjection water (℃) | 45 |

Results and discussion

Energy and exergy analysis are conducted on the combined heating and power system and the simulation results of the whole system is shown in Table 9. Temperature of the geothermal water after the filter drops from 130 ℃ to 78.3 ℃, providing 2063.3 kW thermal energy for the organic working fluid in the vapor generator. The 10.5 ℃ liquid organic working fluid is heated to 99.7 ℃ superheated vapor. Then the vapor expands in the ORC turbine to generate power and its temperature simultaneously drops to 45.5 ℃. Then in the condenser, the organic working fluid is cooled by water and flows out of the condenser as 10 ℃ saturated liquid. After the ORC, geothermal water again provides for the radiant floor heating cycle with temperature dropping from 78.3 ℃ to 45 ℃. In the heat exchanger 1, water is heated from 35 ℃ to 45 ℃ to provide heat in the building.

**Table 9**

Energy and exergy values of the system station

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| Station | *T* (℃) | *P* (kPa) | (kg/s) | (kW) |
| 1 | 130 | 500 | 36.11 | 3322.6 |
| 2 | 78.3 | 500 | 36.11 | 1259.3 |
| 3 | 10.5 | 1000 | 29.56 | 21.1 |
| 4 | 99.7 | 1000 | 29.56 | 1555.2 |
| 5 | 45.4 | 82.04 | 29.56 | 176.3 |
| 6 | 10 | 82.04 | 29.56 | 1.3 |
| 7 | 5 | 101.3 | 325.47 | 0 |
| 8 | 10 | 101.3 | 325.47 | 60.7 |
| 10 | 35 | 400 | 120.44 | 796.6 |
| 11 | 45 | 400 | 120.44 | 1358.8 |
| 12 | 34.9 | 100 | 120.44 | 746.7 |
| 13 | 45 | 500 | 36.11 | 410.9 |



**Fig. 2.** Effects of the organic working fluid superheat degree on the performance of the ORC turbine power output, heat providing and ORC efficiency.

The effects of the organic working fluid superheat degree on the performance of the ORC turbine power output, heat providing and ORC efficiency are shown in Fig. 2.

As presented in Fig. 2, heat provided by the geothermal water in the vapor generator decreases with the increase of the organic working fluid superheat degree. When the operation pressure of the vapor generator keeps unchanged, the increase of the superheat degree of the organic working fluid means the rise of organic working fluid’s outlet temperature. Considering the unchanged temperature of the unsaturated liquid, two-phase part and the temperature increase of the superheat vapor, the average temperature of the organic working fluid increases. As a result, the temperature difference between the geothermal water and the organic working fluid decreases, causing the decrease of the heat released in the vapor generator.

The ORC turbine power output decreases with the increase of the organic working fluid superheat degree. The increase of the organic working fluid superheat degree also causes the increase of the fluid enthalpy, which means the increase of the average organic working fluid enthalpy. Combined with the decrease of the heat transferred in the vapor generator, the mass flow rate of the organic working fluid must decrease. The effect of the decrease of the mass fluid rate is greater than that of the enthalpy increase of the organic working fluid, which determines the decrease of the power output of the ORC turbine.

The heat provided in the heat exchanger 1 increases with the increase of the superheat degree of the organic working fluid. With the decrease of the heat released in the vapor generator, the temperature of the outflow geothermal water increases. Thus, more energy is released in the heat exchanger 1 for the radiant floor heating cycle. It can be seen that the thermal efficiency of the ORC increases first and then decreases with the increase of the superheat degree of the organic working fluid. This can be explained that the decease of the ORC turbine power output is slower than the decrease of the heat provided in the vapor generator first and then becomes faster.



**Fig. 3.** Effects of the ORC turbine inlet pressure on the performance of the ORC turbine power output, heat providing and ORC efficiency.

The effects of the ORC turbine inlet pressure on the performance of the ORC turbine power output, heat providing and ORC efficiency are shown in Fig. 3.

As presented in Fig. 3, the heat provided by the geothermal water in the vapor generator decreases with the increase of the ORC turbine inlet pressure. The increase of the ORC turbine inlet pressure means the increase of the saturated pressure of the organic working fluid in the vapor generator, which simultaneously causes the increase of the temperature of the saturated vapor. As the superheat degree keeps unchanged, the average temperature of the organic working fluid increases, resulting in the decrease of the temperature difference between the geothermal water and the organic working fluid in the vapor generator. Thus, less heat is released in the vapor generator.

The power output of the ORC turbine decreases with the increase of the ORC turbine inlet pressure. The increase of the average temperature of the organic working fluid causes the decrease of the mass flow rate. Though the enthalpy of the organic working fluid increases with the temperature rise, the effect of the mass flow rate decrease is more important, which determines the decrease of the ORC turbine power output.

As heat provided in the vapor generator deceases, more heat is released in the heat exchanger 1. The thermal efficiency of the ORC increases with the increase of the ORC turbine inlet pressure. This is because the decrease of the heat provided in the vapor generator is much faster than the decrease of the ORC turbine power output.

Exergoeconomic analysis of the system is conducted and the exergoeconomic data of the system components are listed in Table 10. Due to the exergy destruction and exergy loss, the fuel exergy is larger than the product exergy. For the vapor generator, fuel is the geothermal water and product is the organic working fluid vapor. Then for the ORC turbine, the organic working fluid vapor drives the turbine as fuel and electricity power is what obtained as product. The electricity power drives pump 1 as fuel and the organic working fluid pumped by pump 1 is the product. Geothermal water provides heat in the heat exchanger 1 as fuel and the hot water is obtained as product. The condenser is a exergy destruction component with no fuel and product.

**Table 10**

Exergoeconomic data for components of the system

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Component | (GJ/year) | (GJ/year) | cf ($/GJ) | cp ($/GJ) | ($/year) |
| ORC turbine | 39713 | 30994 | 0.37 | 3.7 | 100290 |
| Vapor generator | 59425 | 44182 | 0.05 | 0.24 | 13000 |
| Pump 1 | 809 | 571 | 3.7 | 0.24 | 3131 |
| Condenser | / | / | 0.37 | 20 | 31580 |
| Heat exchanger 1 | 24431 | 16190 | 0.05 | 0.6 | 7554 |



**Fig. 4.** Effects of the organic working fluid superheat degree on the performance of levelized exergy cost of the system.

Effects of the organic working fluid superheat degree on the performance of levelized exergy cost of the system are shown in Fig. 4.

As presented in Fig. 4, levelized cost of station 3 *c*3 and station 4 *c*4 increase with the increase of the superheat degree of the organic working fluid. This can be explained by the increase of the vapor generator cost. As mentioned before, the heat provided by the geothermal water in the vapor generator decreases, whereas the heat transfer area of the vapor generator on the contrary increases. With the increase of the superheat degree, the heat transfer area of the superheat vapor increases while the heat transfer area of the liquid and two-phase part decreases. Because of the bad heat transfer ability of the superheat vapor, the increase of the heat transfer area for the superheat vapor in larger than the decrease of the heat transfer area for the liquid and two-phase part, which as a result causes the increase of the total heat transfer area. More cost is transferred to the organic working fluid from the vapor generator, leading to the increase of the levelized exergy cost of station 3 and station 4.

The levelized exergy cost for station 8 *c*8 decreases with the increase of the superheat degree of the organic working fluid first and then slightly increases. This can be explained by the decrease of the cost of the condenser and the decrease of the exergy of the organic working fluid. As the superheat degree increase, the mass flow rate of the organic working fluid decreases, causing the decease of the heat transfer area in the condenser. Thus, less cost of the condenser is passed to the organic working fluid, leading to the decrease of the levelized exergy cost for station 8. The cost of the condenser decreases fast when the heat transfer is large and drops down slowly when the heat transfer area is small. So, when the superheat degree is large the continuous decrease of the mass flow rate of the organic working fluid leads to a small heat transfer area in the condenser. At this time, the decrease of the exergy of the organic working fluid dominates the increases of the levelized exergy cost of the station 8.

The levelized exergy cost for station 10 *c*10 decreases with the increase of the superheat degree of the organic working fluid. With the decrease of the mass flow rate of the organic working fluid in the ORC, the temperature of the geothermal water flowing out the vapor generator increases. Thus, more heat is provided in heat exchanger 1 and the exergy of the hot water in the radiant floor heating cycle increases consequently, which results in the decrease of the levelized exergy cost for station 10.

The levelized exergy cost for the ORC turbine power output *c*Ot increases with the increase of the superheat degree of organic working fluid. This can be explained by the decrease of the ORC turbine power output. As is shown in Fig. 2, the ORC turbine power output decreases with the increase of the superheat degree of the organic working fluid. Thus, the exergy of the electricity power decreases, leading to the increase of the levelized exergy cost for the ORC turbine power output.



**Fig. 5.** Effects of the ORC turbine inlet pressure on the performance of the levelized exergy cost of the system.

Effects of the ORC turbine inlet pressure on the performance of the levelized exergy cost of the system are shown in Fig. 5.

As presented in Fig. 5, the levelized exergy cost of the station 3 decreases with the increase of the ORC turbine inlet pressure. The two-phase part of the organic working fluid in the vapor generator decreases with the increase of the inlet pressure. Thus, more heat is transferred to the organic working fluid unsaturated part from the geothermal water. Compared with the organic working fluid vapor, the liquid organic working fluid has a better heat transfer ability, causing the decrease of the heat transfer area in the vapor generator. Thus, the capital cost of the vapor generator decreases, dominating the decrease of the levelized exergy cost of the station 3.

The levelized exergy cost of the station 4 increases slightly with the increase of the ORC turbine inlet pressure. The change of levelized exergy cost of the station 4 is decided by the change of the capital cost of the vapor generator and the exergy of the organic working fluid. Though the decrease of the vapor generator could result in the decrease of the levelized exergy cost, the effect of the exergy drop caused by the decrease of the mass flow rate is more important. As a result, the levelized exergy cost of the station 4 increases.

The levelized exergy cost of the station 8 *c*8 increases with the increase of the ORC turbine inlet pressure. As mentioned before, the mass flow rate of the organic working fluid decreases with the increase of the ORC turbine inlet pressure. Thus, the exergy of the organic working fluid decreases rapidly, causing the increase of the levelized exergy cost of the station 8.

The levelized exergy cost of the station 10 *c*10 decreases with the increase of the ORC turbine inlet pressure. With the increase of the inlet pressure, less heat is transferred to the ORC from the geothermal water in the vapor generator, causing the increase temperature of the geothermal water when flowing out the vapor generator. Thus, more heat is provided in the heat exchanger 1 and more exergy is obtained by the radiant floor heating cycle hot water, causing the decrease of the levelized exergy cost of the station 10.

The levelized exergy cost of the ORC turbine power output increases with the increase of the ORC turbine inlet pressure. This can be explained by the ORC turbine power output decrease. The increase of the ORC turbine power output causes the decrease of the organic working fluid mass flow rate, which further causes the decrease of the ORC turbine power output. Thus, the exergy generated by the turbine decreases, resulting in the increase of the levelized exergy cost of the ORC turbine power output.

Exergoenvironmental analysis of the system is conducted and the exergoenvironmental data of the system components are listed in Table 11. environmental impact rate per exergy unit *b* of the system components increases from fuel (*b*f) to product (*b*p). For example, the environmental impact rate per exergy unit for the ORC turbine fuel increases from 7.1 mPts/GJ to 10.29 mPts/GJ the environmental impact rate per exergy unit for the ORC turbine product. Relative difference (*r*) and exergoenvironmental factor (*f*) are selected to assess the environmental impacts of the components. The relative difference of the vapor generator, condenser and heat exchanger 1 are 332.93%, 193.66% and 626.22%, respectively, which are larger than 1. Thus, there is a great potential of reducing the environmental impact for these components. The exergoenvironmental factor of the vapor generator and heat exchanger 1 are 89.35% and 91.88%, which are both larger than 70%. So, the environmental impacts of the vapor generator and heat exchanger 1 are primarily determined by the component-related environmental impact. The exergoenvironmental factor of the pump 1 and condenser are 1% and 2.99%, which are smaller than 30%. Thus, the environmental impacts of pump 1 and condenser are mainly decided by the exergy destruction. The exergoenvironmental factor of the ORC turbine is 37.48%, which means both the component-related environmental impact and the exergy destruction have an influence on the environmental impact of the component.

**Table 11**

Exergoenvironmental data for components of the system

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Component | bf (mPts/GJ) | bp (mPts/GJ) | r (%) | f (%) | (mPts/h) |
| ORC turbine | 7.10 | 10.29 | 44.93 | 37.48 | 4.64 |
| Vapor generator | 1.64 | 7.10 | 332.93 | 89.35 | 26.17 |
| Pump 1 | 10.29 | 18.23 | 77.16 | 1 | 0.31 |
| Condenser | 7.10 | 20.85 | 193.66 | 2.99 | 0.09 |
| Heat exchanger 1 | 1.64 | 11.91 | 626.22 | 91.88 | 19.11 |



**Fig. 6.** Effects of the organic working fluid superheat degree on the performance of the environmental impact per exergy unit of the system.

Effects of the organic working fluid superheat degree on the performance of the environmental impact per exergy unit of the system are shown in Fig. 6.

As presented in Fig. 6, the environmental impact per exergy unit of station 3 and station 4 increase with the increase of the organic working fluid superheat degree. As mentioned before, the heat transferred in the vapor generator decreases with the increase of the superheat degree of the organic working fluid while the heat transfer area on the contrary increases. Thus, the environmental impact of the vapor generator increases and the exergy of the organic working fluid decreases, which both causes the increase of the environmental impact per exergy unit of the station 3 and station 4.

The environmental impact per exergy unit for station 8 increases with the increase of the organic working fluid superheat degree. This can be explained by the decrease of the mass flow rate of the organic working fluid. The decrease of the mass flow rate causing the decrease of the organic working fluid exergy, whose effect is more important than the decrease of the component-related environmental impact of the condenser.

The environmental impact per exergy unit for station 10 decreases with the increase of the organic working fluid superheat degree. As mentioned before, less heat is provided in the vapor generator and more heat is released in the heat exchanger 1 from the geothermal water. Thus, the exergy of the radiant floor heating cycle hot water increases, causing the decrease of the environmental impact per exergy unit.

The environmental impact per exergy unit for the ORC turbine power output decreases with the increase of the organic working fluid superheat degree. The increase of the superheat degree causes the decrease of the ORC turbine power output, which causes the decrease of the turbine component-related environmental impact and the decrease of the exergy of the electricity power. The decrease of the component environmental impact is more important than that of the decrease of the exergy. Thus, the environmental impact per exergy unit of the ORC turbine power output decreases.



**Fig. 7.** Effects of the ORC turbine inlet pressure on the performance of the environmental impact per exergy unit of the system.

Effects of the ORC turbine inlet pressure on the performance of the environmental impact per exergy unit of the system are shown in Fig. 7.

As presented in Fig.7, both the environmental impact per exergy unit for station 3 and station 4 decrease with the increase of the ORC turbine inlet pressure first and then increase. The increase of the ORC turbine inlet pressure causes the decrease of the mass flow rate of the organic working fluid, which on the one hand causes the decrease of the component-related environmental impact and on the other hand causes the decrease of the exergy of the organic working fluid. The decrease of the component-related environmental impact causes the first decrease of the environmental impact per exergy unit and the decrease of the organic working fluid exergy causes the final increase of the environmental impact per exergy unit.

The environmental impact per exergy unit for station 8 increases with the increase of the ORC turbine inlet pressure. This can be explained by the decrease of the mass flow rate of the organic working fluid. The decrease of the mass flow rate causing the decrease of the organic working fluid exergy, whose effect is more important than the decrease of the component-related environmental impact of the condenser. Thus, the environmental impact per exergy unit for station 8 increases.

The environmental impact per exergy unit for station 10 decreases with the increase of the ORC turbine inlet pressure. As mentioned before, less heat is provided in the vapor generator and more heat is released in the heat exchanger 1 from the geothermal water. Thus, the exergy of the radiant floor heating cycle hot water increases, resulting in the decrease of the environmental impact per exergy unit for station 10.

The environmental impact per exergy unit for the ORC turbine power output in decreases with the increase of the ORC turbine inlet pressure first and then increases. The increase of the ORC turbine inlet pressure causes the decrease of the mass flow rate of the organic working fluid, which on the one hand causes the decrease of the component-related environmental impact of the ORC turbine and on the other hand causes the decrease of the exergy of the electricity power. The decrease of the turbine component-related environmental impact causes the first decrease of the environmental impact per exergy unit and the decrease of the electricity power exergy causes the final increase of the environmental impact per exergy unit.

**Conclusions**

A geothermal heat source combined heating and power system with ORC and RFH is introduced and exergoeconomic analysis and exergoenvironmental analysis are used to assess the system performance in view of thermodynamics, economics and environics. The conclusions of this paper are presented as follows:

* The ORC turbine power output decreases while the RFH heat supply increases with the increase of the superheat degree of the organic working fluid and the ORC turbine inlet pressure.
* The condensation water holds the highest levelized exergy cost. ORC turbine power output holds the second highest levelized exergy cost and increases with the rise of the organic working fluid superheat degree and ORC turbine inlet pressure.
* The heat exchanger 1, vapor generator and condenser have great environmental impact reducing potential and condensation water has the highest environment impact per exergy unit value.

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The thermodynamic and economic performance of the system changes with the ORC turbine inlet pressure (varies from 650 kPa to 1150 kPa) and superheat degree of the organic working fluid (varies from 7 ℃ to 27 ℃). To obtain the highest power output and heating supply with lowest economic cost and lowest environmental impact, performance optimization of system is necessary. Multi-objective optimization is performed on the system to obtain the highest ORC turbine power output, highest heat supply, lowest exergy cost and lowest environmental impact cost. Genetic algorithm which bases on the natural biological evaluation [35] is chosen to realize the single-objective optimization. The control parameters of the genetic algorithm are presented in Table 12.

The optimization results of the system are presented in Table 13. The net power output of the whole system is 0.92 MW and the heat supply of the radiation floor heating is 6.4 MW. The levelized exergy cost of the system is 0.59 $/GJ and the levelized environmental impact cost is 2.64 mpts/GJ.

Power output by the ORC turbine is what the system provides. Levelized exergy cost of the system obtained by Eq. (47) represents the exergoeconomic performance of the system and levelized environmental cost of the system obtained by Eq. (58) represents the exergoenvironmental performance of the system. These three system results vary with the ORC turbine inlet pressure (varies from 650 kPa to 1150 kPa) and superheat degree of the organic working fluid (varies from 7 ℃ to 27 ℃). Thus, it is necessary to carry out multi-objective optimization to get the highest ORC turbine power output, lowest exergy cost and lowest environmental impact cost.

Multi-objective optimization is aimed to find an optimal result corresponding to every objective function [35]. There is a mapping between the multi-dimensional solution vector and the multi-dimensional objective vector. The optimal solution of the multi-objective optimization is call Pareto optimal solutions in which the decrease of one objective function is bound to result in the increase another objective function.

Genetic algorithm which bases on the natural biological evaluation is chosen to realize the multi-objective optimization. It begins from a population which contains the potential solutions to a problem. In the population, individuals are programmed by their genes which determine the outside expression of the individuals. Solutions are generated in the process of population generation evolution with the crossover and mutation of the genetic operators. The control parameters of the genetic algorithm are presented in Table 12.

The result of two-objective optimization for the system power output and levelized cost per exergy unit is shown in Fig. 8. In the figure, each point is a potential optimum solution to obtain the highest power output and lowest levelized cost per unit exergy. The point with the highest power output has the highest levelized cost per exergy unit while the point with the lowest power output has the lowest levelized cost per exergy unit. We can find a pointO with the highest power output and the lowest levelized cost per exergy unit simultaneously. The potential optimum solution point which is the closest point to O is the final solution [36]. Consequently, the point with 1.17 MW power output and 1.13 $/GJ levelized cost per exergy unit is the optimum solution. The superheat degree of the organic working fluid is 11.3 ℃ and ORC turbine inlet pressure is 867.8 kPa for this solution.

The result of two-objective optimization for the system power output and levelized environmental impact per exergy unit is shown in Fig. 9. In this figure, each point is a potential optimum solution to obtain the highest power output and the lowest levelized environmental impact per exergy unit. The point with the highest power output has the highest levelized environmental impact per exergy unit while the point with the lowest power output has the lowest levelized environmental impact per exergy unit. We can find a point O with the highest power output and the lowest levelized environmental impact per exergy unit. The optimum solution is the point which is the closest to point O. The point has 1.19 MW power output and 4.06 mpts/GJ levelized environmental impact per exergy unit. The superheat degree of the organic working fluid is 12.8 ℃ and the ORC turbine inlet pressure is 818.1 kPa for this solution.

The two-objective optimization for the two cases (power output with levelized cost per exergy cost and power output with levelized environmental impact per exergy unit) have different optimum solutions. Thus, it is necessary to carry out the three-objective optimization for the highest power out, lowest levelized cost per exergy and lowest levelized environmental impact per exergy unit. The result of the three-objective optimization is shown in Fig. 10. Each point in this figure is a potential optimum solution. The point with the highest power output has the highest levelized cost per exergy unit and the highest levelized environmental impact per exergy unit while the point with the lowest power output has the lowest levelized cost per exergy unit and the lowest levelized environmental impact per exergy unit. We can find a point O with the highest power output and lowest levelized cost per exergy unit and lowest environmental impact per exergy unit. The solution point which is the closest to O is the optimum solution. This point has 1.19 MW power output, 1.2 $/GJ levelized cost per exergy unit and 3.99 mpts/GJ levelized environmental impact per exergy unit. The superheat degree of the organic working fluid is 11 ℃ and the ORC turbine inlet pressure is 833.4 kPa for this solution.

4.4. Multi-objective optimization of the system

ORC turbine power output and radiant floor heating supply can represent the energy and exergy performance of the system. Levelized exergy cost of the system obtained by Eq. (47) represents the exergoeconomic performance of the system and levelized environmental cost of the system obtained by Eq. (58) represents the exergoenvironmental performance of the system. These four system results vary with the ORC turbine inlet pressure (varies from 650 kPa to 1150 kPa) and superheat degree of the organic working fluid (varies from 7 ℃ to 27 ℃). Thus, it is necessary to carry out multi-objective optimization to get the highest ORC turbine power output, highest heat supply, lowest exergy cost and lowest environmental impact cost.

Multi-objective optimization is aimed to find an optimal result corresponding to every objective function [35]. There is a mapping between the multi-dimensional solution vector and the multi-dimensional objective vector. The optimal solution of the multi-objective optimization is call Pareto optimal solutions in which the decrease of one objective function is bound to result in the increase another objective function.

Genetic algorithm which bases on the natural biological evaluation [36] is chosen to realize the multi-objective optimization. It begins from a population which contains the potential solutions to a problem. In the population, individuals are programmed by their genes which determine the outside expression of the individuals. Solutions are generated in the process of population generation evolution with the crossover and mutation of the genetic operators. The control parameters of the genetic algorithm are presented in Table 12.

The optimization results of the system are presented in Table 13. The net power output of the whole system is 0.92 MW and the heat supply of the radiation floor heating is 6.4 MW. The levelized exergy cost of the system is 0.59 $/GJ and the levelized environmental impact cost is 2.64 mpts/GJ.



**图3.**有机透平进口压力变化对系统功率输出，热量供应以及有机朗肯循环效率的影响规律



**图4.**有机工质过热度变化对系统单位火用成本的影响规律



**图5.**有机透平进口压力变化对系统单位火用成本的影响规律



**图6.**有机工质过热度变化对系统单位火用环境影响性的影响规律



**图7.**有机透平进口压力变化对系统单位火用环境影响性的影响规律



**图8.**系统供电量和单位火用成本的基于遗传算法的双目标优化



**图9.**系统供电量和单位火用环境影响性的基于遗传算法的双目标优化



**图10.**系统供电量、单位火用成本和单位火用环境影响性的基于遗传算法的三目标优化

This paper develops a geothermal driven combined heating and power system to obtain multi-generation, in which power is generated by Organic Rankine Cycle and heating is supplied by radiant floor heating system. Comprehensive thermodynamic, exergoeconomic and exergoenvironmental models of the system are performed and the capital cost and environmental impacts of the system components are analyzed. Superheat degree of the organic working fluid and inlet pressure of the ORC turbine are selected as variables to assess the system performance. Multi-objective optimization is employed to obtain the maximum power output, minimum levelized exergy cost and minimum levelized environmental impact of the system. Results show that the increase of the working fluid superheat degree and the ORC turbine inlet pressure would cause the decrease of the ORC turbine power output and the increase of the heat supplying. Cooling water holds the highest levelized exergy cost and heat exchanger for heating has the highest environmental impact reducing potential. Three-objective optimization shows that the ORC turbine power output is 1.19 MW, the levelized exergy cost per exergy unit is 1.20 $/GJ and the levelized environmental impact per exergy unit is 3.99 mpts/GJ.

This paper designs a combined heating and power system consisting of organic Rankine cycle and radiant floor heating system driven by geothermal source. Exergy analysis, exergoeconomic analysis and exergoenvironmental analysis are employed to assess the performance of the system. Superheat degree of the organic working fluid and inlet pressure of the ORC turbine are selected as to key variables to analyze the increase and decrease of system power output, heat supply, levelized cost per exergy unit and levelized environmental impact per exergy unit. System power output, levelized cost per exergy unit of the system product and levelized environmental impact of system product are selected as objective functions to carry out multi-objective optimization. System analysis results show that both the increase of the working fluid superheat degree and inlet pressure of ORC turbine cause the decrease of the ORC turbine power output and the increase of the heat supplying. Optimization results show that the ORC turbine power output is 1.19 MW, the levelized exergy cost per exergy unit is 1.20 $/GJ and the levelized environmental impact per exergy unit is 3.99 mpts/GJ.