**Thermo-economic analysis and performance optimization of a combined heating and power (CHP) system for cascaded utilization of geothermal water**

Abstract

A combined heating and power (CHP) system driven by geothermal water is investigated. An organic Rankine cycle is employed to utilize the high temperature part of the geothermal water and the radiant floor heating system is established to make use of the low-grade thermal energy. To fully utilize the exhausted geothermal water energy and to fulfill the increase demand of the heat load in winter, an improved CHP system is developed. In that system, a heat pump cycle is added to the basic CHP system. Mathematical models of the two systems are built. Inlet pressure of the ORC turbine, pinch point temperature difference, outlet of geothermal water temperature, geothermal water mass flow ratio and compressor pressure ratio are chosen as the parameters to analysis the thermodynamic and economic performance. Average cost per unit of exergy is introduced to reflect the thermo-economic performance of the two systems. To obtain the optimization of the systems genetic algorithm (GA) is used to obtain the optimal parameters. Heat load area at the lowest expense of average cost per unit exergy is obtained by carrying out the single-objective optimization.

Keywords:

Geothermal water

Radiant floor heating

Organic Rankine cycle

CHP

**Nomenclature**

|  |  |  |  |
| --- | --- | --- | --- |
| A | area, |  | Boltzmann’s constant |
|  | boiling number |  | thickness, m |
| c | average cost per unit of exergy, $() |  | compressor pressure ratio |
|  | specific heat, kJ |  | |
|  | cost rate, $ | 1-16 | state points |
| CHP | combined heating and power | g1-g4 | state points |
| CRF | capital recovery factor | av | average temperature |
| CEPCI | chemical engineering plant cost index | BM | bare module |
| D | diameter, m | cond | condenser |
| e | exergy, kJ | comp | compressor |
|  | exergy flow rate, kJ | D | destruction |
|  | exergy flow rate per year, kJ | e | effective |
| f | friction factor | elec | electricity |
| F | configuration factor | es | equivalent diameter |
|  | shape factor | ev | evaporation |
| G | mass flow rate, kg | ex | exergy |
| h | convection heat transfer coefficient, W | F | fuel |
|  | enthalpy, kJ | g | geothermal |
| H | depth, m | he | heat exchanger |
|  | interest rate | hf | hot flow |
| l | length, m | in | inside |
|  | mass flow rate, kg | 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 |
|  | reduced pressure | ot | organic Rankine turbine |
|  | heat transfer rate, kW | out | outside |
|  | heat transfer rate per year, MWh | pump | pump |
| T | temperature, K | r | radiant floor heating |
| U | overall heat transfer coefficient, W | rq | radiant floor heating rate |
| v | velocity, m | s | single phase |
|  | power, kW | t | tube |
|  | power per year, MWh | turb | turbine |
| x | vapor quality | vg | vapor generator |
|  | annually levelized cost value, $ | w | wall |
| Greek symbol | | wt | turbine work |  |
|  | heat load ratio | well | well |
|  | efficiency, % |  |  |
|  | heat conductivity, W |  |  |
|  | density, kg |  |  |
|  | dynamic viscosity, |  |  |

1. **Introduction**

The increasing development of the globe economy and industry has attracted more and more attention to the energy problem. The environment impacts of using conventional energy such as coal, nature gas and oil have urged people to find alternative ways to fulfill the demand of the energy consumption. The geothermal energy is a stable and low-carbon energy which can be developed to relieve the serious air pollution. Geothermal energy is the internal heat of the earth and has huge reserve. The total amount of the geothermal energy can supply the global energy use for about 2.17 million based on the global consumption rate of approximately 1.7×108 GWh/yr [1]. The utilization of geothermal energy has been carrying on all around the world [2-5].

The research works related to the geothermal energy have been carried out. A power plant which used the geothermal energy as the heat source and CO2 as the turbine working medium was designed by Nagasree et al. [6] in Switzerland. They analyzed the performance of the thermodynamic energy convention as well as the capital cost of the system with the increasing of the drilling depth. Nasruddin et al. [7] investigated the potential of geothermal energy for electricity generation in Indonesia and they found that with the increasing of the electricity consumption, the utilization of renewable geothermal energy was worthwhile.

Geothermal energy is mainly utilized to produce electricity and many systems have been designed to make full use of the thermal energy. Zhao et al. [8] modeled a dual-loop geothermal energy source system for power which had a steam cycle as the main cycle and an organic Rankine cycle (ORC) as the subsystem. They concluded that thermodynamic optimization and the exergoeconomic optimization of the system were 44.22% and 42.89%, respectively. Zhao et al. [9] investigated combined cooling and power (CCP) system which was driven by the geothermal energy. Organic Rankine cycle and steam cycle were employed in the system. The output of the refrigeration was 345.8kW, the steam turbine output power was 923.0kW and the organic Rankine cycle turbine output power was 100.0kW. Sun et al. [10] proposed geothermal energy source system for electricity which had a dual-pressure Organic Rankine cycle. They compared three working fluids and concluded that the fluid R245fa had a better performance. Genetic algorithm was used to gain the optimization parameter. They concluded that the dual-pressure cycle system had a better performance than the single pressure. 818.6kW electricity power output can be produced and the efficiency was 5.85%. Li et al. [11] established geothermal energy system for power with the combination of the transcritical CO2 cycle. They analyzed the design performance as well as the off-design performance of the system. They got the conclusion that in the design stage the thermal efficiency and the exergetic efficiency of the system are 8.51% and 29.59%, respectively. Also, in the off-design stage, it can get a maximun value of the thermal efficiency by changing the geothermal resource mass flow rate. Wang et al. [ 12] employed Kalina cycle using ammonia-water working fluid to the geothermal energy system for power. Mathematical model was built for the system and the total efficiency of the system was 37.01% under the given condition. A cascade power generation system driven by geothermal which had a high pressure Kalina cycle and a low pressure Kalina cycle was designed by Hadi et al. [13]. The net power, thermal efficiency, exergy efficiency and the total cost of the system were obtained. Zare et al. [14] investigated a novel Kalina cycle system driven by the geothermal water. In the system, the exhaust heat of the Kalina cycle is utilized by the employing thermoelectric generators (TEGs). The performance of the novel system was compared with the conventional Kalina cycle. They reached the conclusion that the net output power increased 7.3% and both the exergy and the energy increased.

Geothermal energy can also be utilized to provide heat with the combination of heat pump. Li et al. [15] designed a system with the combination of organic Rankine cycle and heat pump cycle to realize the cascaded utilization of thermal energy. They simulated the combisystem with the T Transient System Simulation Program (TRNSYS) and got the results that the ORC turbine could provide 55.6% power for the heat load.

Combined heating and power (CHP) systems have also been wildly studied by researchers. Zhang et al. [16] modeled a new system which can produce heating and power. In the novel system, biomass energy and geothermal energy were employed to provide energy for the system. Power generation rate, carbon conversion ratio and warm water temperature were selected to optimized the system. Their results showed that the output of power was 276kW and the temperature of the water produced was 125℃. Zhang et al. [17] introduced a CHP system which was driven by solar and fuel. They reached the concluded that 2732.1MW electricity and 46.7MW heat could be provided.

It can be observed that in most cases of the geothermal energy was utilized to generate electricity. In a few cases the geothermal energy was used to provide heat load with the combination of heat pump. There is few CHP system driven totally by geothermal water. Also, in many heat load system, the fluctuation of the heat demand was not taken into consideration. In this paper, a CHP system driven by geothermal water is developed with the aim of generating electricity and providing heat load at the same time. The geothermal water exchanges heat with working fluid in an organic Rankine cycle. After that process, the geothermal water releases heat in a heat exchanger to provide heat load. To utilized the low-grade thermal energy, radiant floor heating is introduced. Considering the fluctuation of the heat demand in winter, a heat pump cycle is added to the system to fulfilled the increasing heat requirement. The exhaust geothermal water operates as low-temperature heat source in the heat pump cycle and water in a relatively high temperature can be produced to provide heat for the radiant floor heating cycle.

**2. System description**



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



**Fig. 2.** Schematic diagram of the improved CHP system 2

The basic schematic diagram of the CHP system for this study is shown in Fig. 1. The system combines an organic Rankine cycle (ORC) and a radiant floor heating cycle. As shown in Fig. 1, geothermal water is pumped from the head well which is high in temperature and pressure. A sand separater is employed to separate the sand in the water. Then the water flows into the vapor generator, where the organic working fluid exchange heat with the hot geothermal water. The organic working fluid which is pumped from the condenser is heated into high temperature and pressure state. The vapor generated in the vapor generator expands in the organic Rankine cycle (ORC) turbine in which the thermal energy of the working fluid is translated into electricity. The exhausted vapor of the turbine then enters the condenser where the vapor is condensed to liquid state. The geothermal water flows out of the vapor generator is still in a relative high temperature state and can be exploited to heat the radiant floor heating cycle. Because of the corrosivity of the geothermal water, a heat exchanger is employed to separate the geothermal water and the circulation water of the floor heating cycle. Finally, the geothermal water is reinjected back to the ground.

Fig. 2 shows the improved CHP system driven by geothermal source. To fulfill the increasing demand head load of the customers, more thermal energy should be produce by the CHP system. As a result, another radiant floor heating cycle is added to the CHP system combined with the utilization of a heat pump cycle. As shown in Fig. 2, after driven from the produce well, the geothermal water is divided into two separated parts with the help of valves. One part of the water flows into the vapor generator as the heat source of the organic Rankine cycle (ORC). Another part which is in high temperature and pressure enters the heat exchanger 1 to produce more heat load if necessary. The temperature of the water at the outlet of the heat exchanger 1 is still higher than the ambient temperature. To fully utilize the thermal energy, the low grade geothermal water operates as the heat source of the heat pump. In the heat pump cycle, water expands through the throttle valve and then absorbs heat from the geothermal water. Water in the pump cycle is converted to vapor state in evaporator. The compressor compresses the vapor to a state which is in high temperature. The vapor then enters the condenser where the vapor releases heat to drive another radiant floor heating cycle.

**3. Mathematical models**

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 is saturated liquid and the state at the outlet of the evaporator is saturated vapor.

3.1. Energy analysis

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 CHP system

|  |  |
| --- | --- |
| Component | Energy equation |
| Vapor generator |  |
| ORC turbine |  |
| Condenser 1 |  |
| Pump 1 |  |
| Heat exchanger 1 |  |
| Pump 2 |  |
| Radiant floor heating 1 |  |
| Evaporator (system 2) |  |
| Compressor (system 2) |  |
| Condenser 2 (system 2) |  |
| Valve 4 (system 2) |  |
| Pump 3 (system 2) |  |
| Radiant floor heating 2 (system 2) |  |

According to the first law of thermodynamic, the total thermal efficiency of the CHP system 1 is given by

(1)

The thermal efficiency of the improved CHP system 2 is expressed as

(2)

3.2. 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

(3)

where 、 and are the parameter of the ambient state.

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

(4)

Thermal systems are always connected with exergy inputs associated directly or indirectly with fuels or other energy resources, such as geothermal 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 [18]

(5)

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 CHP system

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
| Component |  |  |  |  |
| Vapor generator |  |  |  | / |
| ORC turbine |  |  |  | / |
| Condenser 1 | / | / |  |  |
| Pump 1 |  |  |  | / |
| Heat exchanger 1 |  |  |  | / |
| Pump 2 |  |  |  | / |
| Radiant floor heating1 |  |  |  | / |
| Reinjection | / | / | / |  |
| Evaporator (system 2) |  |  |  | / |
| Compressor (system 2) |  |  |  | / |
| Condenser 2 (system 2) |  |  |  | / |
| Valve 4 (system 2) | / | / |  | / |
| Pump 3 (system 2) |  |  |  | / |
| Radiant floor heating 2 (system 2) |  |  |  | / |
| Reinjection (system 2) | / | / | / |  |

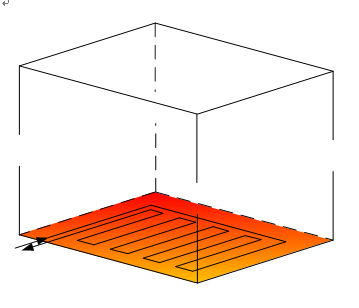
The exergy efficiency of the whole system is given by the expression

(6)

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

(7)

3.3. Radiant floor heating analysis



**Fig .3** Radiant floor heating diagram

According to the blood circulation inside the human body, the average temperature of the foot is lower than the temperature of the head. Different from conventional heat system, the radiant floor heating system provides heat from the low floor up to the air which is more comfortable in winter. Also, the temperature of the hot water flows in the radiant floor heating system is relative lower than that of the traditional system. Based on the two main reasons mentioned above, the radiant floor heating system is chosen to provide the heat load in this paper. The radiant floor heating system is showed in Fig. 3.

In order to simply the system model, a set of assumptions are employed in the following.

1. The system is in a steady state.
2. The environment outside the building is -5℃ and the air temperature inside the building is 18℃。

3.3.1. Heat loss of the building

To maintain the stable temperature in the house in winter, the heat loss in the building should be equal to the heat load from the heat source. Because of the low temperature of the outside environment, heat will transfer through the wall and the roof of the building. In addition, with the regular circulation of the air inside and outside, heat will loss by means of convection.

1. Heat loss through the wall

The heat transfer equation through the wall is given as

(8)

where is the area of the wall towards to the cold air.

The overall heat transfer coefficient of the building is expressed as

(9)

where is the convection heat transfer coefficient of the inside air and is the convection heat transfer coefficient of the outside cold air. and denote the thermal conductivity of the material of the wall and the thickness of the material, respectively. The parameters of the wall are listed in Table 3.

**Table 3**

Material parameters of the wall

|  |  |  |
| --- | --- | --- |
| Material of layer | Thickness/mm | Thermal conductivity/W |
| 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 |

1. Heat loss through the inflow of cold air

The heat transfer rate by the inflow of the cold air is given as

(10)

where is the specific heat of the air at a mean temperature and is the average mass of the air which flows into the building within a day.

Subsequently, the total heat loss of each room can be calculated by adding the heat loss through the wall and heat loss through the inflow of cold air.

3.3.2. Heat supply of the building

The heat can be transfer to the room by means of radiation and convection.

1. Radiation heat transfer

The heat transfer through radiation is expressed as

(11)

where is the configuration factor and is the Boltzmann’s constant. is the average temperature of the floor surface.

1. Convection heat transfer

The heat transfer equation [19] of convection has the following form

(12)

where the value of and can be obtained in Table 4

**Table 4**

The value of and in equation (12)

|  |  |  |
| --- | --- | --- |
| condition |  |  |
| Heat transfer to the upper layer | 2.13 | 1.31 |
| Heat transfer to the lower layer | 0.14 | 1.25 |

The average temperature of the floor surface is determined by

(13)

where is the effective heat load to the building, being given by

(14)

where is the total heat load to the building and is the ratio number.

3.3.3. The hot water pipe of the radiant floor heating system

Fig. 4 denotes the structural profile of floor heating. Under the surface layer is the fine aggressive concrete. The hot water pipes are in the fine aggressive concrete to transfer the heat to both the upper layer and the lower layer. To increase the effective heat load and prevent much heat transfer through the lower layer, thermal insulation material is put under the fine aggressive.



**Fig. 4** floor heating pipes diagram

The basic modal of the hot water pipe is shown below

(15)

where is the overall heat conductivity of the floor coverings, being given as

(16)

is the shape factor and can be determined as [20]

(17)

where is the spacing between the pipe; is the diameter of the pipe and is the depth of the pipe.

is the mean temperature of the pipe, being given as

(18)

Subsequently, the heat loss of a certain building can be calculated with the known values of and . The effective heat load should be equal to the heat loss of the building to maintain the steady temperature of the building. As a result, the total heat load required by the building can be determined. The length of the pipe required to heat the building can be obtained.

3.4. 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 single-phase heat transfer process takes place in the heat load heat exchanger and the vapor generators. The two-phase heat transfer process occurs in the vapor generator 1 of the Organic Rankine cycle and the vapor generator 2 in the heat pump.

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.

3.4.1. Single-phase heat transfer process

For each subsection, the heat transfer rate is given by

(19)

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

(20)

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

(21)

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

(22)

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

(23)

(24)

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

(25)

The Reynolds number can be calculated as

(26)

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

(27)

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

The Prandtl number can be determined as

(28)

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

(29)

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

(30)

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

(31)

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

(32)

3.4.2. 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]

(33)

where is the boiling number, being determined as

(34)

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

(35)

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.

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 |  |  |

3.5. 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][36] for each component of CHP system

|  |  |  |
| --- | --- | --- |
| Component | Cost balance | Auxiliary relation |
| Geothermal resource |  | / |
| Vapor generator |  |  |
| Turbine |  |  |
| Condenser 1 |  |  |
| Pump 1 |  | / |
| Heat exchanger 1 |  |  |
| Pump 2 |  | / |
| Floor heating pipe 1 |  |  |
| Evaporator (system 2) |  |  |
| Compressor (system 2) |  | / |
| Condenser 2 (system 2) |  |  |
| Valve 4 (system 2) | / |  |
| Pump 3 (system 2) |  | / |
| Floor heating pipe 2 (system 2) |  |  |

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 which is the condenser in the organic Rankine cycle, being expressed as

(55)

1. **Results and discussion** 
   1. Validation

In order to make sure the accuracy of the mathematical model of the system, data from Ref. [38] and Ref. [39] are used to validate the calculation. The validation results of the organic Rankine cycle are listed in Table 7 and the results of the heat pump cycle are listed in Table 8. It can be observed the data in this study are highly consistent with data obtained by Wang et al. and Liang et al.

**Table 7**

Comparison of the organic Rankine cycle

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Term | Ref. [38] | Present | Term | Ref. [38] | Present |
| Fluid | R245fa | R245fa | (kW) | 10.615 | 10.615 |
|  | 1.4293 | 1.4293 | (kW) | 0.615 | 0.63 |
| (℃) | 107.75 | 107.75 | (%) | 8.38 | 8.39 |
|  | 0.1874 | 0.1874 |  |  |  |
| (℃) | 31.29 | 31.29 |  |  |  |
|  | 0.4988 | 0.5 |  |  |  |

**Table 8**

Comparison of the heat pump cycle

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
| Term | Ref. [39] | Present | Term | Ref. [39] | Present |
| Fluid | R134a | R134a | (℃) | 37.8 | 0.5 |
|  | 298 | 298 |  | 958 | 298 |
| (℃) | 0.5 | 0.5 | (%) | 26 | 26.48 |
|  | 958 | 958 |  | 1 | 1 |
| (℃) | 51.7 | 51.7 |  |  |  |
| (℃) | 37.8 |  |  |  |  |
|  | 958 |  |  |  |  |

4.2. Conditions of the system for simulation.

The thermodynamic parameters of the working fluid are calculated under the environment of MATLAB with the help of REFPROP 9.1 [35]. The condition of simulation for the CHP system 1 and the improved CHP system 2 is listed in Table 9.

**Table 9**

Condition of simulation for the CHP system

|  |  |
| --- | --- |
| term | Value/description |
| Geothermal water temperature (℃) | 85 |
| Geothermal water pressure (kPa) | 200 |
| Mass flow of geothermal water (kg/s) | 36.11 |
| Pinch point temperature difference in vapor generator (℃) | 5 |
| ORC turbine inlet pressure (kPa) | 350 |
| Ambient temperature (℃) | 10 |
| Ambient pressure (kPa) | 101.3 |
| Condensation temperature of condenser (℃) | 10 |
| Isentropic efficiency of ORC turbine (%) | 80 |
| Isentropic efficiency of pump 1 (%) | 75 |
| Inlet temperature of cooling water (℃) | 10 |
| Outlet temperature of cooling water (℃) | 15 |
| Inlet temperature of radiant floor heating water (℃) | 45 |
| Outlet temperature of radiant floor heating water (℃) | 35 |
| Outlet temperature of in heat exchanger of geothermal water (℃) | 30 |
| Isentropic efficiency of pump 2 (%) | 75 |
| Heat load per area (kW/m²) | 0.07 |
| Geothermal water ratio for ORC cycle (%) (system 2) | 90 |
| Outlet temperature of in heat exchanger of geothermal water (℃) (system 2) | 30 |
| Outlet temperature of in evaporator of geothermal water (℃) (system 2) | 18 |
| Compressor pressure ratio (system 2) | 6 |

**5. Results and discussion**

5.1. Thermodynamic and exergeconomic analysis



**Fig. 5** Effects of ORC turbine inlet pressure on thermodynamic performance of the system

Fig. 5 shows the effects of the ORC turbine inlet pressure on the net work of the CHP system, the net work of the improved CHP system and the radiant floor heating of the two systems. In addition, the exergy efficiencies of the two systems are also analyzed.

Both the net work of the two systems decreases with the increase of the inlet pressure. It can be explained that with the increase of the inlet pressure of the ORC turbine, the saturated temperature of the working fluid increases, which leads to the increase of the specific enthalpy. At the same time, the specific entropy of the working fluid will increase as well which lead to the increase of the specific enthalpy of the exhaust working fluid of the ORC turbine. As organic working fluid, which is from steam with a great increase of the specific enthalpy between two isobars, the net increase of the specific enthalpy is relatively smaller. The increase of the saturated temperature will also lead to the decrease of the mass flow rate of the working fluid which has a much larger effect of the output of the ORC turbine. Combining the two aspects mentioned above, the decrease of the net work of the two systems can be explained. In addition, the net work in CHP system 1 is much larger than that of the CHP system 2. It is because of the consumption of the compressor work of the CHP system 2.

The decrease of the net work of the system means the less heating energy absorbed from the geothermal water, which will lead to the increase of the temperature of the outlet water of the vapor generator. This increase will bring much thermal exergy to the heat exchanger which increase the radiant floor heating a lot.

The net work of the system and radiant floor heating determine the exergy of the exergy efficiency of the system. But the order of magnitude of the radiant floor heating is much larger than the net work. Although the decrease of the net work, the great increase of the radiant floor heating contributes to the increase of the exergy efficiency of the system.



**Fig. 6** Effects of ORC turbine inlet pressure on exergeonomic performance of the system

Fig.6 shows the effects of ORC turbine pressure on the average cost per unit exergy for main steam and products of the system. It can be observed that the average cost per unit of exergy of steam outlet the compressor (c13) is not affected by the changing. That is because the ORC cycle is independent from the heat pump cycle. For the same reason, the average cost per unit of exergy of the water in the second radiant floor heating cycle (crq2) remains the same.

With the increase of the inlet pressure of the ORC turbine, the less heat will be absorbed from the vapor generator as mentioned before, which leads to the decrease of the heat exchanger area of the vapor generator and the cost of the vapor generator. Lower cost of the vapor generator contributes to the decrease of the average cost per unit of exergy of the working fluid outlet the vapor generator (c3). The average cost per unit of exergy for the ORC turbine (cwt) is made of two aspects, which are the exergy-fuel-related part and the equipment-cost-related part. The decrease of the average cost per unit of exergy of the working fluid (c3) which flows into the ORC turbine to produce power cuts down the exergy-fuel-related part of cwt. For the equipment-cost-related part, the decrease of the output of the ORC turbine leads to the decrease of the cost of the turbine. However, the effect of the decrease of the output of the power of the ORC turbine is much greater than that of the cost. As a result, the equipment-cost-related part of the ORC turbine decreases as well. Hence, the average cost per unit exergy for the ORC turbine (cwt) decreases.

The increase of the pressure of the ORC inlet which increases the outlet geothermal water of the vapor generator as explained before will increase the exergy input of the heat exchanger. The increase in exergy input of the heat exchanger will cut down the equipment-cost-related part of the radiant floor heating water (crq1). The increase in temperature of the outlet geothermal water will increase heat exchanger area which has a tendency to increase the equipment-cost-related part of crq1. The effect of increase in exergy input is greater than the effect of increase in heat exchanger area. Hence, the average cost per unit of exergy of the radiant floor heating (crq1) decreases slightly.

The decrease in the average cost per unit exergy of the ORC turbine (cwt) and the decrease in the average cost per unit of exergy of the radiant floor heating (crq1) accounts for the decrease for the average cost per unit of exergy of the produce (cproduct) of the two systems.



**Fig. 7** Effects of pinch point temperature difference on thermodynamic performance of the system

Fig. 7 shows the effect of pinch point temperature difference on the performance of net work of the two system and the floor heating of the two systems. Also, the exergy efficiencies of the two systems are analyzed.

The increase of the pinch point temperature difference enables the mass flow rate of the working fluid to decrease resulting in the decrease of the ORC net work of the two systems. The decrease of the mass flow rate will decrease the heat absorbed from the geothermal water through the vapor generator leading to the increase of the outlet temperature of the geothermal water from the vapor generator. The increase of the input of thermal energy causes the radiant floor heating increasing rapidly as well as the exergy efficiencies of the two systems.

In addition, it can be obtained that the net work of the improved CHP system is less than zero which means that the output of the ORC turbine is less than the consumption of the compressor (ignore the work of the pumps).



**Fig. 8** Effects of pinch point temperature difference on exergoeconomic performance of the system

Fig. 8 shows the effects of pinch point temperature on the performance of the average cost per unit exergy for main components and products.

With the increase of the pinch point temperature difference, the mass flow rate of the working fluid decreases greatly, which leads to the decrease of exergy output of the working fluid flows out the vapor generator. The decrease of the mass flow rate also cuts down the heat transfer area needed by the vapor generator which decreases the cost of the vapor generator. However, the influence of the decrease of the exergy output is much greater. Hence, the average cost per unit exergy of the working fluid out let the vapor generator (c3) increases. The increase of the c3 rises the exergy-fuel-related part of the average cost per unit exergy of the ORC turbine (cwt), leading to its increase.

The increase of the exergy input of the radiant floor heating cycle 1 counteracts the increase of the cost of the heat exchanger which is caused by the increase of the heat transfer area. As a result, the average cost per unit of exergy of the radiant floor heating (crq1) increases slightly.

As the mass flow rate decrease sharply, the fictitious cost rates (c9) which takes into consideration the dissipative components decrease as well. The decrease of fictitious cost rates and the decrease of the radiant floor heating lead to the decrease of the products (cproduct) of the two systems.



**Fig. 9** Effects of temperature of the exhausted geothermal water on the thermodynamic performance of the system

Fig. 9 shows the effects of the geothermal water temperature outlet the heat exchanger on the performance of net power and floor heating of the two systems. In order to better analyze geothermal water temperature on the performance of the heat pump cycle, the work of the compressor is considered. The exergy efficiencies of the two systems are also analyzed.

In order to keep a relatively steady performance of the radiant heating system and to remain a comfortable indoor environment in the building, the inlet water temperature and the outlet water temperature of the radiant floor heating cycle is 45℃ and 35℃, respectively. What’s more, the outlet geothermal water will heat the heat pump cycle to provide heat for the radiant floor heating cycle 2. It is clear that the increase in temperature of the geothermal water outlet the heat exchanger has a great effect of the first radiant floor cycle. Both the radiant floor heating and the exergy efficiency of the first system decrease sharply. However, with the increase of the outlet geothermal water temperature, the total radiant floor heating increases at the expense of the increase of the work consumption of the compressor. The net power of the improved CHP system decreases to less than zero with the increase of the outlet geothermal water temperature. The increase work consumption also leads to the decrease of the exergy efficiency of system 2.



**Fig. 10** Effects of return temperature of the geothermal water on the exergoeconomic performance of the system

Fig. 10 shows the increase of the outlet temperature of the geothermal water temperature. Because of the rapidly decrease of exergy input of the radiant floor heating cycle 1 which makes a large part of the product, the average cost per unit of exergy of the product for the system 1 (cproduct1) increases sharply. The decrease of the heat transfer combines with the decrease of the heat transfer area of the heat exchanger. As a result, the increase tendency of the average cost per unit exergy for the first radiant floor heating (crq1) is not very sharp. The effect about the increase of cost for the compressor caused by the rise of the outlet geothermal water temperature counteracts the effect of the increase input exergy. Hence, the average cost per unit of exergy of the second radiant floor heating (crq2) decreases slightly and nearly remains the same. Because of the increase of the average cost per unit of exergy in the first radiant floor heating cycle, the average cost per unit of exergy of the total product for the improved CHP system 2 increases.



**Fig. 11** Effects of the mass flow ratio on the thermodynamic performance of the system

Fig. 11 shows the effects of mass flow ratio of the geothermal water for the ORC cycle on the performance of the net power and radiant floor heating about improved CHP system 2. In order to fulfill the demand for heat load in winter, the mass flow ration is introduced to the improved CHP system 2. With the decrease of the mass flow ratio (from 1) the net power as well as the ORC turbine power decreases. The total radiant floor heating and the exergy efficiency of the system increase.



**Fig. 12** Effects of the mass flow ratio on the exergoeconomic performance of the system

Fig. 12 shows the effects of mass flow ratio on the performance of the average cost per unit exergy of the main components of the improved CHP system 2. With the increase of the mass flow ratio, the heat absorbed from the vapor generator increases, which leads to the increase of the exergy output of the vapor generator. The area of the vapor generator increases with the increases, too. But the effect of the exergy output is more determined. As a result, the average cost per unit of exergy of the working fluid outlet the vapor generator decreases, which cut down the exergy-fuel-related part of the average cost per unit of exergy of the ORC turbine. The rise of the ratio increases the output work of the ORC turbine as well as the cost of the component. Combining the two aspects, the equipment-cost-related part of the average cost per unit of exergy decreases, too. So, the total average cost per unit of exergy for the ORC turbine (cwt) decreases. The rise of the mass flow ratio has a great impact on the performance of the radiant floor heating cycles, which cuts down the floor heating rapidly. The decrease in radiant floor heating increases the average cost per unit of exergy of the floor heating (crq1). Meanwhile, the fictitious cost rates (c9) about the condenser increases. The two reasons both enable the average cost per unit of exergy of the system product to increase (cproduct2). Because the heat exchanger outlet temperature remains unchangeable, the average cost per unit exergy of the second radiant floor heating (crq2) shows no changes.



**Fig. 13** Effects of the compressor pressure ration on the thermodynamic performance of the system

Fig. 13 shows the effects of the compressor pressure ration on the performance of the heat pump cycle and the whole system. The increase of the compressor pressure ratio rises the pressure of the pressure outlet the compressor, which increases the saturated temperature of the water vapor. As a result, temperature difference in the condenser 2 increases, leading the increase of the mass flow rate in the second radiant floor heating. The radiant floor heating increases, consequently. The rise of the compressor work consumption results the drop of the net power of the system. The increase of the compressor requires much exergy consumption, resulting the exergy efficiency of the system decreases.



**Fig. 14** Effects of the compressor pressure ration on the exergoeconomic performance of the system

Fig. 14 indicates the effects of compressor pressure ratio on the average cost per unit exergy of the second radiant floor heating cycle and the product of the system.

With the increase of the pressure ratio, more radiant floor heating is produced in the floor heating cycle. The effect of the output heating is greater than the increase in the cost of components, leading to the drop of the average cost per unit of exergy of the heating (crq2). Although the increase in the pressure ratio will increase the cost of components, the output heating accounts for a large part of the products. The average cost per unit of exergy of the system product (cproduct2) decreases in a slow tendency.

5.2 Optimization and comparison

The analysis finding from Fig. 8-14 shows that the performance of the CHP system can be adjusted by changing the thermodynamic parameters. The ranges of the parameters are listed in Table 14. To harvest the optimal performance of the system, genetic algorithm (GA) [37] is employed to achieve single-objective optimization. The average cost per unit of exergy is considered as the objective function.

**Table 10**

Ranges for thermodynamic parameters

|  |  |  |
| --- | --- | --- |
| Thermodynamic parameters | Range for system 1 | Range for system 2 |
| Inlet ORC turbine pressure (kPa) | 350-450 | 350-450 |
| Pinch point temperature difference (℃) | 7-12 | 7-12 |
| Degree of superheat for in ORC cycle (℃) | 10-20 | 10-20 |
| Geothermal water temp after heat exchanger (℃) | 37-40 | 37-40 |
| Mass flow ratio (%) | / | 90-100 |
| Compressor pressure ratio | / | 5-7 |
| Reinjection water temp (℃) | / | 18-30 |

**Table 11**

Single-objective optimization results

|  |  |  |
| --- | --- | --- |
| Term | Sytem1 | System 2 |
| Inlet ORC turbine pressure (kPa) | 450.0 | 450.0 |
| Pinch point temperature difference (℃) | 7.0 | 7.0 |
| Degree of superheat for in ORC cycle (℃) | 11.399 | 11.40 |
| Geothermal water temp after heat exchanger (℃) | 37.0 | 37.0 |
| Mass flow ratio (%) | / | 0.9 |
| Compressor pressure ratio | / | 6.956 |
| Reinjection water temp (℃) | / | 24.976 |
| Net power output (kW) | 294.7421 | 0 |
| Radiant floor heating (kW) | 3547.20 | 5935.7 |
| Heat load area (m²) | 50674.29 | 84795.71 |
| Exergy efficiency (%) | 58.61 | 46.0 |
| Average cost per unit exergy for system product ($/(MWh)) | 4.6102 | 4.740 |

The results of the optimization are listed in Table 15. It can be obtained that the average cost per unit exergy of the system product are 4.6102$/(MWh) and 4.740$/(MWh) for CHP system 1 and improved system 2, respectively. The net power output of CHP system 1 is 294.7421kW. The decrease of the average cost per unit exergy for improved system 2 product happens with the decrease of the net power output of the system. But the electricity energy should come from the ORC turbine which means that the net power output should be zero or greater. Hence, the net power of the system output is zero. The two radiant floor heating of the two systems are 3547.20kW and 5935.7kW, respectively. The heating provided by the heat pump cycle is 2388.50kW. The two exergy efficiencies of the systems are 58.61% and 46.0%. It can be observed that the heat load area of the improved CHP system is 34121 m² more than the basic CHP system which means that the improved CHP system can provide extra 67.33% heat load in winter.



**Fig. 5** Schematic diagram of conventional heat load system

Fig. 5 shows the conventional radiant floor heating system driven by the geothermal water. As shown in the diagram, all the geothermal water energy is used to provide the heat load and there is no electricity generation, which is similar to the improved CHP system when operating at the lowest average cost per unit of exergy. The floor heating load rate and the areas of the heating load about the two systems are listed in Table 12.

**6. Conclusion**

In this study, a CHP system 1 and an improved CHP system 2 driven by the geothermal water are developed. The CHP system 1 generates electricity and provides heat load. A heat pump cycle is employed to utilize the exhaust geothermal water energy to fulfill the increase demand of heat load. The effects of significant parameters are examined. And the main conclusions are summarized as follows:

1. The inlet pressure of the ORC turbine, the pinch point temperature difference on the vapor generator and the outlet geothermal water at the heat exchanger have great effects on the thermodynamic performance of CHP system1 and improved CHP system 2. The mass flow ratio and the compressor pressure ratio have significant effect of the improved CHP system 2 and the heat pump cycle.
2. The average cost per unit exergy reflects both the thermodynamic and the economic aspects of the systems. The average cost per unit exergy of the radiant floor heating cycle 1 is much lower than that of the radiant floor heating cycle 2 because of the consumption of the compressor.
3. Two kinds of organic working fluid are compared in this study. And R245fa is chosen as the working fluid for the ORC cycle for the high electricity output and steady required heat transfer area. Through the genetic algorithm, single-objective optimization results are obtained. The lowest exergy cost per unit exergy of the system product is got. Meanwhile, the exergy efficiencies of the systems are also high.

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