Thermodynamic, economic and thermo-economic optimization of a new proposed organic Rankine cycle for energy production from geothermal resources

Neda Kazemi, Fereshteh Samadi

Department of Chemical Engineering, Shiraz Branch, Islamic Azad University, Shiraz, Iran

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Abstract

The main goal of this study is to propose and investigate a new organic Rankine cycle based on three considered configurations: basic organic Rankine cycle, regenerative organic Rankine cycle and two-stage evaporator organic Rankine cycle in order to increase electricity generation from geothermal sources. To analyze the considered cycles’ performance, thermodynamic (energy and exergy based on the first and second laws of thermodynamics) and economic (specific investment cost) models are investigated. Also, a comparison of cycles modeling results is carried out in optimum conditions according to different optimization which consist thermodynamic, economic and thermo-economic objective functions for maximizing exergy efficiency, minimizing specific investment cost and applying a multi-objective function in order to maximize exergy efficiency and minimize specific investment cost, respectively. Optimized operating parameters of cycles include evaporators and regenerative temperatures, pinch point temperature difference of evaporators and degree of superheat. Furthermore, Peng Robinson equation of state is used to obtain thermodynamic properties of isobutane and R123 which are selected as dry and isentropic working fluids, respectively. The results of optimization indicate that, thermal and exergy efficiencies increase and exergy destruction decrease especially in evaporators for both working fluids in new proposed organic Rankine cycle compared to the basic organic Rankine cycle. Moreover, the amount of specific investment cost in new organic Rankine cycle is obtained less than basic organic Rankine cycle during thermodynamic and economic optimization for R123. Finally, a profitability evaluation of new proposed and basic systems is performed based on total production cost and return on investment for three countries: Iran, France and America. Its results show that Iran has the maximum amount of return on investment.

1. Introduction

Geothermal energy, as a low-grade heat source, could be used to produce and convert energy into useful work (i.e., electricity) by using organic Rankine cycle (ORC) technology [1]. This technology is based on principles of renewable energy, in which it could reduce the emission of greenhouse gases (GHGs) caused by fossil fuels. In other words, an ORC is widely used in the geo-plant in order to generate power in an environment friendly manner [2].

In such systems, at first water is injected into the ground, which its injection wells were calculated geometrically and adsorbs the certain amount of heat from the ground layers (geothermal energy). Then, hot water is pumped from production wells to an evaporator. In the evaporator, heat is transferred from the hot water to an organic fluid which is also known as “working fluid”. As a result, the working fluid in the evaporator is vaporized or even superheated according to the amount of heat it takes. In the next stage, the saturated or superheated vapor expands through a turbine and produces electricity by an electrical generator which is transferred mechanical energy into electrical type. Afterwards, the expanded vapor is cooled by a condenser to liquefy and finally, to complete the cycle, working fluid is pumped to the evaporator [3].

Generally, the ORC system performance could be specified by thermodynamic efficiency which depends on several parameters, for instance the operating conditions, types of working fluids, the system components and also cycle configuration [4]. Accordingly, in recent decades, in order to improve the performance and
increase the efficiency of ORC system, various studies have been considered which include the investigation of different working fluids effect, the optimization of operating parameters with an appropriate objective function such as thermodynamic, economic or thermo-economic and also variations in the cycle configuration.

In this research, Yang and Yeh [5] analyzed an ORC system performance by using economic optimization. Their results showed that R600 gave the best performance under economic optimization among working fluids which have the lower global warming potential (GWP) like R290, R600a, R1233zd, R1234yf and R1234ze. Also, the pinch point temperature differences in the evaporator altered more compared to the condenser.

Shokati et al. [6] carried out a comparative study in order to investigate an exergoeconomic analysis by optimization of basic, dual-pressure and dual-fluid ORCs and Kalina for geothermal applications in power plants. They optimized parameters of mentioned cycles in order to maximize the energy production and minimize the cost of power generation. Their optimization results showed that dual-pressure ORC generated the maximum value of electricity and also Kalina cycle had the minimum value of unit cost of power produced.

Yang and Yeh [7] used thermo-economic optimization in order to evaluate the performance of R245fa, R1234yf, R1234ze, R152a, and R600a as working fluids in ORC systems. The results illustrated that R1234yf gave the best performance in thermo-economic investigation among all considered working fluids.

Dai et al. [8] investigated the performance of ORC for different working fluids using genetic algorithm. In their study, exergy efficiency was chosen as an objective function to optimize operating parameters including inlet temperature and pressure of the turbine. They found, the working fluid R236EA has the highest exergy efficiency (35.43%) and thermal efficiency (12.37%).

A research was carried out by Xi et al. [9] in order to maximize the exergy efficiency of ORC using single or double-stage regenerative. They presented that an ORC with two-stage regenerative had higher exergy efficiency comparing to a basic ORC. Roy and Misra [10] optimized different parameters and analyzed the performance of a regenerative organic Rankine cycle (RORC) for waste heat recovery. They recognized, the performance of R123 as a working fluid is better than R134a for this specified cycle within a superheating at a constant pressure of 2.5 MPa. Thermo-economic optimization of RORC in order to recover the amount of heat losses were investigated by Imran et al. [11]. The results of optimization showed that R245fa is the best possible working fluid under their cycle conditions and also, the thermal efficiency of ORC was improved by adding a regenerative to the cycle.

Li et al. [12] optimized the operating parameters of a two-stage evaporator organic Rankine cycle (TSORC). Their studies showed that the system performance would increase with geothermal water inlet temperature and different range of evaporator temperature. Moreover, they figured out the irreversible losses would decline to a minimum value in the evaporator by using two-stage evaporator within two different temperature range. Li et al. [13] investigated the performance of the ORCs in order to use two stage evaporation strategies (parallel and series). In their research, R245fa and geothermal water with a temperature range of 90–120°C were selected as working fluid and heat source, respectively. Although, they recognized both strategies could reduce the system irreversibility, significant reduction was found in series two-stage evaporator. Li et al. [14] proposed an ORC with double parallel evaporator which their research results stated a decrease in irreversible energy loss and an increase in production capacity from geothermal resources.

Zare [15] evaluated the performance of three different configurations (basic ORC, regenerative ORC and an ORC within an internal heat exchanger) through thermodynamic and exergoeconomic optimization. The results illustrated that although the ORC within an internal heat exchanger gave a premier performance in thermodynamic optimization, basic ORC showed the best behavior in economic optimization compared to the other considered cycles.

In this paper, three different points of view, including the change in basic ORC configuration (a new arrangement of the cycle equipment), optimization of cycles operating parameters by using three different objective functions (thermodynamic, economic and thermo-economic) and survey of the effect of various working fluids are considered to improve and increase the efficiency of a basic ORC. In other words, the restructuring of basic ORC equipment has been investigated by adding an evaporator and a regenerative, simultaneously. Also, three selected objective functions have been applied to optimize heat exchangers temperatures (excluding
condenser), pinch point temperature difference of evaporator and degree of superheat. Furthermore, two pure organic components, including isobutane (dry) and R123 (isentropic) were selected as working fluids.

2. Materials and methods

Although, a basic ORC could be used to produce useable work by using low-grade heat sources, its thermal efficiency is low due to system irreversibility [16]. Overall, irreversibility reduction is possible by changing the basic ORC configuration which has been investigated by previous works. One of the changes is adding a regenerative to ORC (RORC) in which, the amount of heat received by evaporator and the outlet work in the turbine would reduce, because a certain amount of superheated working fluid which is taken from the turbine is used to preheat the outlet subcooled working fluid from the pump [11]. In contrast, another method has been applied by adding an extra evaporator either in series or parallel to the evaporator of basic ORC which is known as TSORC [13]. In TSORC system, the mass flow rate of vaporized working fluid increases due to receive a large amount of heat from the geothermal water compare to the basic ORC. Therefore, in the mentioned cycle, the outlet work in the turbine raises. In other words, although use of this method would increase the electricity production, adding an extra evaporator would also increase the amount of heat received from the heat resource. The method which has been proposed in this article is use of both mentioned methods, simultaneously. Description of the basic ORC and new proposed ORC (NORC) are explained in details as follow:

2.1. Cycles description

2.1.1. Basic ORC

The main processes and temperature–entropy (T–S) diagram in the basic ORC are illustrated in Fig. 1. According to Fig. 1(a), the cycle includes an evaporator, a pump, a condenser and a turbine. Geothermal water and cooling water are used in the evaporator and condenser in order to heat and cool down the working fluid in the cycle, respectively. In addition, T–S diagram in Fig. 1(b) shows the trend of entropy variation for each process (in both isentropic and real states), geothermal and cooling water with their temperatures and also, the pinch point temperature differences in the condenser and evaporator.

2.1.2. New organic Rankine cycle

In new proposed organic Rankine cycle (NORC), series two-stage evaporator and also a regenerative are added to the basic ORC equipment to improve the performance. The configuration of NORC and its T–S diagram are depicted in Fig. 2. According to Fig. 2, compared to the basic ORC process, the outlet subcooled working fluid from pump 1 enters a regenerative which is used to heat the working fluid by an amount of superheated working fluid taken from the turbine. Afterwards, the saturated working fluid is pumped to the evaporator 2 (intermediate evaporator) to receive heat from the geothermal water. In this stage, a portion of working fluid gets into the turbine and the rest moves to the first evaporator. In the evaporator 1, at a constant pressure, the working fluid is heated by absorbing the energy from geothermal water and then is expanded through the turbine and produces electricity.

2.2. Assumptions

In this article, the following assumptions are applied for modeling of the mentioned cycles.

- Steady-state condition is considered for each process in the cycles.
- Heat and friction losses are neglected in the heat exchangers.
- Kinetic and potential energy are neglected for water as a heat source and sink media.
- Entirely developed flow is considered to calculate heat transfer parameters.
- Heat exchangers are selected from shell and tube type.
- Peng Robinson equation of state (PR EOS) is applied to calculate thermodynamic properties.
- Two pure organic fluids, including isobutane and R123 were selected as working fluids.

2.3. Thermodynamic modeling

The first and second laws of thermodynamics could be considered to investigate how the basic ORC performance would be modified by changing its structure. In other words, the performance of ORC systems could be specified by thermal and exergy efficiencies analysis based on the first and second laws of thermodynamics, respectively. In this respect, thermal efficiency is determined according to Eq. (1):
Exergy efficiency could be used to evaluate the maximum thermal and exergy (useful and destroyed) efficiencies that have been modeled for each process in ORCs\cite{19}. In addition, constant design parameters for ORC systems are described as follow (according to Figs. 1 and 2):

\[m_{\text{dif}} = \dot{m}_h \left( h_{\text{hi}} \left( T_0 \right) - h_h \left( T_0 \right) \right) / \left( h_{\text{hi}} \text{ at Bubble Point} - h_{\text{hi}} \text{ at Regenerative} \right)\]  

(4)

\[X = (h_h \text{ at Regenerative} - h_h \text{ at Condenser}) / (h_h \text{ at Condenser} - h_h \text{ at Evaporator})\]  

(5)

\[\dot{m}_e = \dot{m}_{\text{dif}} \left( h_{\text{hi}} \text{ at Dew Point} - h_{\text{hi}} \text{ at Condenser} \right) / \left( h_{\text{hi}} \text{ at Condenser} - h_{\text{hi}} \text{ at Evaporator} \right)\]  

(6)

Moreover, in order to calculate exergy and energy efficiencies, certain parameters such as enthalpy, entropy and vapor pressure should be determined. Therefore, the values of enthalpy and entropy were obtained by using PR EOS and also, the vapor pressures of each working fluid were calculated using modified Wagner equation, according to Refs. \cite{20,21}, respectively. In addition, constant design parameters for ORC systems are indicated in Table 1.

### 2.4. Economic modeling

In this article, not only thermodynamic modeling of each cycle was considered but also their economic modeling investigated. Accordingly, specific investment cost (SIC) and return on investment (ROI) were used for economic analysis. In order to compute SIC and ROI, costs of equipment have to be evaluated. Therefore, firstly, basic parameters for calculating SIC and ROI are described as below:

2.4.1. Equipment cost

Bare module method was selected to calculate each equipment cost within the mentioned cycles \cite{22}:

- Heat Exchangers cost

\[C_{\text{HX}} = \frac{527.7}{397} \times C_{\text{D,HX}} \times \left[ B_{1,\text{HXC}} + B_{2,\text{HXC}} \times F_{\text{M,HXC}} \times F_{\text{P,HXC}} \right] \]  

(7)

\[\log C_{\text{D,HX}} = \left[ K_{1,\text{HXC}} + K_{2,\text{HXC}} \left( \log A_{\text{HXC}} \right) + K_{3,\text{HXC}} \left( \log A_{\text{HXC}} \right)^2 \right] \]  

(8)

\[\log F_{\text{M,HXC}} = \left[ C_{1,\text{HXC}} + C_{2,\text{HXC}} \left( \log P_{\text{HXC}} \right) + C_{3,\text{HXC}} \left( \log P_{\text{HXC}} \right)^2 \right] \]  

(9)

where respectively, \(C_{\text{D,HX}}, F_{\text{M,HXC}}, F_{\text{P,HXC}}, A_{\text{HXC}}\) and \(P_{\text{HXC}}\) are the cost, the material factor, pressure factor, surface area and pressure for heat exchangers which include evaporator, regenerative and condenser.

Eq. (8) could also be used for initial heat exchangers cost.

### Table 1

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Values</th>
</tr>
</thead>
<tbody>
<tr>
<td>Isentropic efficiency of pump, (\eta_{p,i}^{\text{ex}}) (%)</td>
<td>80</td>
</tr>
<tr>
<td>Isentropic efficiency of turbine, (\eta_{\text{t,i}}^{\text{ex}}) (%)</td>
<td>76</td>
</tr>
<tr>
<td>Electrical generator efficiency, (\eta_{\text{gen}}) (%)</td>
<td>95</td>
</tr>
<tr>
<td>Heat source and sink media</td>
<td>Water</td>
</tr>
<tr>
<td>Heat source and sink media, (T_{\text{hi}}) (k)</td>
<td>423.15</td>
</tr>
<tr>
<td>Heat sink and sink medium, (T_{\text{cs}}) (k)</td>
<td>298.15</td>
</tr>
<tr>
<td>Heat source inlet pressure, (P_{\text{hi}}) (bar)</td>
<td>293.15</td>
</tr>
<tr>
<td>Heat sink inlet pressure, (P_{\text{cs}}) (bar)</td>
<td>5</td>
</tr>
<tr>
<td>Temperature of condenser, (T_{\text{c}}) (k)</td>
<td>308</td>
</tr>
<tr>
<td>Temperature of the environment taken as standard-state value, (T_{0}) (k)</td>
<td>50</td>
</tr>
</tbody>
</table>

In addition, \(m_{\text{dif}}, m_{\text{e}}, X\) as a mass flow rate of working fluids, cooling water and also, a fraction of mass flow rate of working fluids into the regenerative are described as follow (accoridng to Figs. 1 and 2):
• Cost of turbine

\[ C_T = \frac{527.7}{397} \times C_{0T} \times F_{M1} \]  
(10)

\[ \log C_{0T} = \left[ K_{11} + K_{22}(\log W_1) + K_{33}(\log W_2)^2 \right] \]  
(11)

where \( C_{0T} \) and \( F_{M1} \) are the initial cost and the material factor of turbine, respectively.

• Cost of pump

\[ C_P = \frac{527.7}{397} \times C_{0P} \times \left[ B_{1p} + (B_{2p} \times F_{Mp} \times F_{Pp}) \right] \]  
(12)

\[ \log C_{0P} = \left[ K_{1p} + K_{22}(\log W_p) + K_{33}(\log W_p)^2 \right] \]  
(13)

\[ \log F_{Pp} = \left[ C_{1P} + C_{22}(\log P_p) + C_{33}(\log P_p)^2 \right] \]  
(14)

where \( C_{0P} \), \( F_{Pp} \) and \( P_p \) are initial cost, pressure factor and pressure of pump, respectively. In Eqs. (7)–(14), \( B_{1p}, B_{2}, K_{1}, K_{2}, K_{3}, C_{1}, C_{2} \) and \( C_{3} \) are constants related to the material which was used in each equipment within the cycle and their values are given in Table 2. Moreover, 527.7 and 397 are chemical engineering plant cost indexes for the year 2015 and 2001, respectively [23].

2.4.2. Heat exchanger surface area calculation

In order to calculate cost of heat exchangers, the type of heat exchanger should be known. Moreover, a heat exchanger should be chosen according to the operating conditions as well as its heat transfer surface area. Generally, the amount of generating capacity in a geo-plant is one of the main crucial parameters for choosing the best types of heat exchangers. In this basis, for productive capacity (\( W_{nt} \)) of lower than tens of kilowatts, the plate heat exchangers are most suitable, whereas shell and tube heat exchangers could be used for productive capacity of higher than hundreds of kilowatts [12]. Hence, in this study, shell and tube heat exchanger was chosen in which, flow in the tube and liquid in the shell [24] and its specifications are shown in Table 3.

Table 2

<table>
<thead>
<tr>
<th>Constants</th>
<th>Equipment</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Heat exchanger</td>
</tr>
<tr>
<td>( K_1 )</td>
<td>4.3247</td>
</tr>
<tr>
<td>( K_2 )</td>
<td>-0.3030</td>
</tr>
<tr>
<td>( K_3 )</td>
<td>0.1634</td>
</tr>
<tr>
<td>( C_1 )</td>
<td>0.03881</td>
</tr>
<tr>
<td>( C_2 )</td>
<td>-0.1127</td>
</tr>
<tr>
<td>( C_3 )</td>
<td>0.0818</td>
</tr>
<tr>
<td>( B_1 )</td>
<td>1.6300</td>
</tr>
<tr>
<td>( B_2 )</td>
<td>1.6600</td>
</tr>
<tr>
<td>( F_M )</td>
<td>1.0000</td>
</tr>
<tr>
<td>( F_S )</td>
<td>1.7000</td>
</tr>
</tbody>
</table>

Table 3

Shell and tube heat exchanger data.

| Inner tube diameter, \( d_i \) (mm) | 10.92 |
| Outer tube diameter, \( d_o \) (mm) | 12.7 |
| Tube pinch, \( P_{im} \) (mm) | 19.05 |
| Fouling factor \( \left( m^2 \cdot C/W \right) \) | 0.0001761 |
| Hot water | 0.0001761 |
| Cold water | 0.0001761 |
| Refrigerant (vapor) | 0.0001761 |
| Refrigrerant (liquid) | 0.0003522 |
| Total fouling resistance with two-phase | 0.00067 |

Accordingly, Kern’s method was used to determine the heat transfer surface area of evaporator, regenerative and condenser [26]. And also, inner and outer heat transfer coefficients were calculated by using Gnielinski equation [27] and Kern’s method [26], respectively.

2.4.3. Specific investment cost (SIC)

The value of SIC could be obtained from Eq. (15) described as follow [28]:

\[ SIC = F_s \times \frac{TCB}{W_{net}} \]  
(15)

where \( F_s \) is a correction factor for overhead cost and its values are illustrated in Table 2. Moreover, \( TCB \) is the total bare module cost described in equation (16) as follow:

\[ TCB = \sum_{i=1}^{n} C_i \]  
(16)

In this study, heat source cost (geothermal energy) and cost of working fluid in mentioned cycles were neglected.

2.4.4. Return on investment (ROI)

Determination of this type of profitability is usually defined as a ratio of net profit to total cost of investment [29]:

\[ ROI = \frac{(1 - t_{corp})(S_{annual} - C_{TPC})}{TCI} \]  
(17)

where \( S_{annual} \), \( C_{TPC} \), \( t_{corp} \) and \( TCI \) are the annual sales revenue, the total production cost (through electricity generation), the corporate tax rate and the cost of total capital investment, respectively. The corporate tax rate was considered to be 25.00% for Iran, 33.33 for France and 35.00% for United States in year 2015 [30]. The equations which related to the total capital investment and total production costs are listed in Tables 4 and 5, respectively.

• Annual sales revenue

\[ S_{annual} = M_a C_{el} \]  
(18)

\[ M_a = H_{annual} W_{net}^{elec} \]  
(19)

\[ W_{net}^{elec} = W_{net}^{elec} - W_{P}^{elec} \]  
(20)

\[ H_{annual} = 0.9 \times 365 \times 24 \]  
(21)

In these equations, \( M_a \) shows the annual electricity generation according to the net electrical power output. Also, \( C_{el} \) which illustrates the electricity price for industry in year 2015, was assumed to be 0.16, 0.11 and 0.07 $/kWh for Iran [31], France and United States [32], respectively.

Table 4

Components of total capital investment.

| Total bare module cost, \( TCB \) | \( TCB = \sum C_{fix} + C_1 + C_P \) |
| Cost of site preparation, \( C_{site} \) | \( C_{site} = 0.05TCB \) |
| Cost of service facilities, \( C_{serv} \) | \( C_{serv} = 0.05TCB \) |
| Allocated costs for utility plants and related facilities, \( C_{util} \) |  
| Total direct permanent investment, \( C_{TPC} \) | \( C_{TPC} = TCB + C_{site} + C_{serv} + C_{util} \) |
| Cost of contingencies and contractor’s fee, \( C_{cont} \) | \( C_{cont} = 0.18C_{TPC} \) |
| Total depreciable capital, \( C_{TDC} \) | \( C_{TDC} = C_{TPC} + C_{cont} \) |
| Cost of land, \( C_{land} \) | \( C_{land} = 0 \) |
| Cost of royalties, \( C_{royal} \) | \( C_{royal} = 0 \) |
| Cost of plant startup, \( C_{startup} \) | \( C_{startup} = 0.1C_{TDC} \) |
| Total permanent investment, \( C_{TPI} \) | \( C_{TPI} = C_{TDC} + C_{land} \) |
| Working capital, \( C_{wpc} \) | \( C_{wpc} = 0 \) |
| Total capital investment, \( C_{TIC} \) | \( C_{TIC} = C_{TPI} + C_{wpc} \) |
2.5. Optimization

The basic ORC operating parameters optimized in this paper include evaporation temperature, degree of superheat (D.S) and pinch point temperature difference in evaporator. In addition to basic ORC parameters, regenerative temperature and temperature of the second evaporator were also optimized for NORC. Furthermore, selection of a suitable objective function plays a crucial role for optimization of these operating parameters. Therefore, three different objective functions proposed in this paper contain: thermodynamic, economic and thermo-economic. Accordingly, the objective functions for thermodynamic and economic optimization are described as below:

\[ F_1(x) = \text{Maximize } (\eta_{ex}) = \frac{W_i - W_p}{m_i(h_{hso} - h_{lso} - T_0(s_{hso} - s_{lso}))} \]  

\[ F_2(x) = \text{Minimize } (SIC) = F_3 \times \frac{T_{C_b}}{W_{net}} \]

Moreover, linear weighted evaluation function was selected for thermo-economic optimization (a multi objective function) [33]:

\[ F(x) = \alpha F_1(x) + \beta F_2(x) \]

where \( \alpha \) and \( \beta \) are weight coefficients of the mentioned objective function:

\[ \alpha = \frac{(F_2 - F_1^*)}{(F_1^* - F_2^*)} \]

\[ \beta = 1 - \alpha \]

where according to Eqs. (25) and (26), \( F_1^* \) is the maximum value of \( F_1 \), \( F_2^* \) is the value of function \( F_2 \) when \( F_2 \) obtained a minimum value, \( F_2^* \) is the minimum value of \( F_2 \) and \( F_1^* \) is the value of function \( F_2 \) when \( F_1 \) obtained a maximum value [33].

The mentioned objective functions were optimized by genetic algorithm [9] in which constraints and bounds of operating parameters are illustrated in Table 6. And also, thermodynamic properties of working fluids are shown in Table 7 [34].

<p>| Table 6 |
| Constraints and bounds for optimization. |</p>
<table>
<thead>
<tr>
<th>Parameters (constraints)</th>
<th>Lower bound</th>
<th>Upper bound</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature of evaporator 1</td>
<td>335 (K)</td>
<td>485 (K)</td>
</tr>
<tr>
<td>Temperature of evaporator 2</td>
<td>335 (K)</td>
<td>485 (K)</td>
</tr>
<tr>
<td>Temperature of regenerative</td>
<td>335 (K)</td>
<td>485 (K)</td>
</tr>
<tr>
<td>Degree of superheat</td>
<td>0 (K)</td>
<td>20 (K)</td>
</tr>
<tr>
<td>PPTD in evaporator 1</td>
<td>5 (bar)</td>
<td>20 (bar)</td>
</tr>
<tr>
<td>Pressure of evaporator 1</td>
<td>5 (bar)</td>
<td>30 (bar)</td>
</tr>
</tbody>
</table>

| Table 7 |
| Thermodynamic properties of working fluids [34]. |
| Substances | \( T_0 \) (K) | \( P_0 \) (bar) | \( \omega \) |
| Isobutane | 407.85 | 36.40 | 0.19 |
| R123 | 456.83 | 36.62 | 0.28 |
| Water | 647.09 | 220.64 | 0.34 |

3. Results and discussion

3.1. Verification

Table 8 shows a comparison of thermodynamic parameters calculated by PR EOS with NIST Reference Database (ASHRAE Standard) [34]. Accordingly, to this table, the results of present study indicate a good agreement with NIST data bank.

Also, Table 9 illustrates comparison results between basic ORC of this study and previous researches. Pressure and thermal efficiency in this table show that the modeling results have a good agreement between the present work and pervious ones. Furthermore, exergy efficiency values versus evaporator temperature, obtained from present model are compared with study carried out by Shengjun et al. [38] for R123 fluid in Fig. 3. According to this figure, it is obvious that exergy efficiency obtained in this article has a low error as well as a similar pattern in Ref. [38].

3.2. Optimization results and cycles performance investigation

In order to make a valuable comparison between the performance of both basic ORC and NORC, it could better to use optimization results of cycles with the same objective function. Accordingly, Table 10 shows the optimization results of basic ORC and NORC with both thermodynamic and economic objective functions and two different working fluids. According to the table, the results of thermal and exergy efficiencies show that the NORC gives a better performance than the basic ORC. In addition, by comparing the results of optimized parameters, it can be concluded that in both thermodynamic and economic optimization, the evaporating temperatures are between the boundaries and also, the pinch point gets the minimum value of its optimization bands. However, the degree of superheat for two objective functions behaves oppositely, so that degree of superheat tends to be the minimum in thermodynamic objective function and the maximum in economic objective function.

According to Section 2.1, the NORC was designed based on RORC and TSORC reviewed by previous researchers. In this basis, in order to investigate the deviation performance of NORC in the present work in respect to RORC and TSORC performances, Eq. (27) was employed:

\[ \% \text{ Deviation} = \frac{100(P_{ORC} - P_{NORC})}{P_{ORC}} \]

where \( P_{ORC} \) and \( P_{NORC} \) denote the parameters of ORC and other cycles. Table 11 shows the results of NORC, RORC and TSORC in respect of the basic ORC. According to this table, the new cycle has more exergy efficiency than the other cycles for both working fluids.

According to the results of thermodynamic and economic objective functions (see Table 10), it was found that these two objective functions have the contrary behaviors during the optimization of degree of superheat. For this purpose, in order to investigate these two functions simultaneously, thermo-economic objective function was used. Table 12 indicates the results which obtained through thermo-economic optimization of basic ORC and NORC for isobutene and R123. The value of \( \omega \) shown in the table was used as the weight coefficient of thermo-economic objective function.
Thermodynamic and economic optimization results of Isobutane and R123 in basic ORC and NORC.

Comparing the error of entropy and enthalpy in this study and NIST databank.

According to Eq. (25), in order to calculate

\[
\text{with Ref. [38].}
\]

Comparison of exergy efficiency obtained through modeling in this study

range of 0–20°C heat which was obtained for this objective function was in the

posed cycle is higher than the basic ORC. Also, the degree of super-

Eventually, thermo-economic weighting function and the corresponding

cycles in this paper (new and basic ORC) were compared with basic

economic objective functions.

Table 10

Comparing the results of thermodynamic optimization in RORC, TSORC and NORC

based on basic ORC for Isobutane and R123.

Table 11

Comparing the results of thermodynamic optimization in RORC, TSORC and NORC

for isobutane. These results have analyzed in thermo-economic optimization

with the inlet temperature of geothermal water 423.15 K. According to this figure, it is observed that not only exergy efficiency of new cycle is more than two other basic cycles but also, the performance of basic ORC investigated in this work is better than the work carried out by Heberle and Brüggemann, due to different assumptions and methods of optimization.

According to Table 13, thermal efficiency and specific investment cost for new and basic ORCs in this paper are compared with considered cycles (basic ORC, regenerative ORC and double stage regenerative ORC) in Ref. [11] for R123. Due to Table 13, it can be concluded that new and basic cycles in this paper have more thermal efficiency and less SIC than each cycle studied in Ref. [11]. These differences can be seen in assumptions and optimization conditions.

Fig. 5 illustrates the irreversibility of each equipment in four mentioned cycles including, basic ORC, RORC, TSORC and NORC. According to this figure, it can be concluded that the destroyed exergy in the basic ORC was reduced by changing cycle configuration, especially in the new proposed cycle. It should be noted that

Table 8

Comparing the error of entropy and enthalpy in this study and NIST databank.

Table 9

Comparison of thermal efficiency obtained through modeling in this study with other papers.

For each cycle and fluid individ-

a

\[
\% R_{\text{Error}} = \frac{R_{\text{NIST}} - R_{\text{This Work}}}{R_{\text{This Work}}}
\]

b

\[
\% H_{\text{Error}} = \frac{H_{\text{NIST}} - H_{\text{This Work}}}{H_{\text{This Work}}}
\]

Fig. 3. Comparison of exergy efficiency obtained through modeling in this study with Ref. [38].

According to Eq. (25), in order to calculate \( \xi \), thermodynamic and economic objective function values reported in Table 10, could be used. Therefore, by calculating \( \xi \) for each cycle and fluid individually, thermo-economic weighting function and the corresponding parameters were optimized. By comparing the results of Table 12, it could be concluded that the thermodynamic efficiency of proposed cycle is higher than the basic ORC. Also, the degree of superheat which was obtained for this objective function was in the range of 0–20°C, due to an adverse effect of thermodynamic and economic objective functions.

Table 10

Thermodynamic and economic optimization results of Isobutane and R123 in basic ORC and NORC.

Table 11
Table 12
Investigation of parameters and results of thermo-economic optimization obtained through linear weighting function.

<table>
<thead>
<tr>
<th>Substance System</th>
<th>$T_{evap1}$ (K)</th>
<th>$T_{evap2}$ (K)</th>
<th>$T_{reg}$ (K)</th>
<th>D.S (K)</th>
<th>$\Delta T_{pp}$ (K)</th>
<th>$\eta_{th}$</th>
<th>$\eta_{ex}$</th>
<th>SIC ($/W$)</th>
<th>$x$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Isobutane ORC</td>
<td>396.159</td>
<td>–</td>
<td>–</td>
<td>8.319</td>
<td>5.064</td>
<td>0.1183</td>
<td>0.4709</td>
<td>3.405</td>
<td>0.983</td>
</tr>
<tr>
<td>NORC</td>
<td>396.315</td>
<td>373.988</td>
<td>329.926</td>
<td>9.516</td>
<td>5.085</td>
<td>0.1235</td>
<td>0.5047</td>
<td>3.543</td>
<td>0.980</td>
</tr>
<tr>
<td>R123 ORC</td>
<td>394.416</td>
<td>–</td>
<td>–</td>
<td>17.774</td>
<td>5.002</td>
<td>0.1312</td>
<td>0.4822</td>
<td>2.378</td>
<td>0.985</td>
</tr>
<tr>
<td>NORC</td>
<td>399.347</td>
<td>383.103</td>
<td>335.908</td>
<td>18.556</td>
<td>5.108</td>
<td>0.1397</td>
<td>0.5253</td>
<td>2.459</td>
<td>0.966</td>
</tr>
</tbody>
</table>

Fig. 4. Comparison of thermodynamic efficiencies (thermal and exergy) for two considered cycles obtained through thermo-economic optimization in this study with Ref. [39] for isobutane.

Table 13
A Comparison of thermal efficiency for two considered cycles obtained through thermo-economic optimization in this study with Ref. [11] for R123.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>$\eta_{th}$</td>
<td>0.1312</td>
<td>0.1397</td>
<td>0.7264</td>
<td>0.8724</td>
<td>0.9204</td>
<td></td>
</tr>
<tr>
<td>SIC ($/W$)</td>
<td>2.3780</td>
<td>2.4590</td>
<td>3.5560</td>
<td>3.7490</td>
<td>4.0570</td>
<td></td>
</tr>
</tbody>
</table>

Fig. 5. Comparison of exergy destruction in each cycles equipment.

Table 14
The investigation of thermodynamic parameters obtained through thermo-economic optimization for Isobutane and R123 in studied cycles.

<table>
<thead>
<tr>
<th>Substance</th>
<th>System</th>
<th>$P_{evap1}$ (bar)</th>
<th>$P_{evap2}$ (bar)</th>
<th>$P_{reg}$ (bar)</th>
<th>$P_{cond}$ (bar)</th>
<th>$m_w1$ (kg/s)</th>
<th>$m_w2$ (kg/s)</th>
<th>$X_1$</th>
<th>$m_t$ (kg/s)</th>
<th>$X_1$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Isobutane</td>
<td>ORC</td>
<td>29.906</td>
<td>–</td>
<td>–</td>
<td>4.628</td>
<td>33.805</td>
<td>–</td>
<td>–</td>
<td>240.766</td>
<td></td>
</tr>
<tr>
<td>NORC</td>
<td></td>
<td>29.985</td>
<td>20.191</td>
<td>8.062</td>
<td>4.628</td>
<td>32.932</td>
<td>9.782</td>
<td>0.143</td>
<td>260.819</td>
<td></td>
</tr>
<tr>
<td>R123</td>
<td>ORC</td>
<td>12.290</td>
<td>–</td>
<td>–</td>
<td>1.303</td>
<td>41.230</td>
<td>–</td>
<td>–</td>
<td>153.090</td>
<td></td>
</tr>
<tr>
<td>NORC</td>
<td></td>
<td>13.541</td>
<td>9.746</td>
<td>3.095</td>
<td>1.303</td>
<td>33.327</td>
<td>20.983</td>
<td>0.134</td>
<td>174.732</td>
<td></td>
</tr>
</tbody>
</table>
the highest amount of exergy destruction in the basic ORC is related to evaporator due to this figure and Ref. [40] which could be reduced to the lowest possible value in NORC.

According to optimized parameters obtained from thermo-economic objective function and also, the modeling equations described in the Section 2, important operating parameters such as heat exchangers pressure, mass flow rate of working fluid, mass flow rate of cooling water and thermal efficiency were calculated and tabulated in Table 14. These results could be used in design and simulation of various processes in the ORC systems.

3.3. Parameters investigation of NORC

According to the sections 2–5 and 3–2, temperatures of heat exchangers (the primary and secondary evaporators and regenerative), degree of superheat and pinch point temperature difference in the primary evaporator were optimized for the NORC using three different objective functions. Changes in any of these parameters could affect the cycle performance and its costs. Consequently, in this section, the effects of crucial parameters variation on thermodynamic efficiency and also SIC of NORC are studied.

Fig. 6 demonstrates the variation of NORC exergy efficiency versus degree of superheat and primary evaporator temperature for isobutene fluid. The value of parameters include pinch point temperature, temperature condenser, the second evaporator temperature and the temperature of regenerative were taken from Table 12. According to the figure, at a constant degree of superheat, by increasing the evaporator temperature, the exergy efficiency increases. In contrast, at a constant evaporating temperature, by increasing the degree of superheat, the exergy efficiency reduces. Furthermore, the variation of regenerative and secondary evaporator temperatures are affected the NORC exergy efficiency which depicted in Figs. 7 and 8.

Fig. 9 shows SIC variation in respect to the changes in the primary evaporator temperature and the degree of superheat. According to the figure, increasing the degree of superheat at a constant value of evaporating temperature and also, increasing the evaporation temperature at a constant degree of superheat give rise to a reduction in the SIC of proposed cycle.

Figs. 10 and 11 respectively represent the variations of exergy efficiency and SIC of NORC in respect with degree of superheat.
and pinch point. According to this figure, it can be concluded that for a given value of pinch point, by increasing the degree of superheat, the exergy efficiency decreases while the SIC reduces, and also, at a constant degree of superheat, with increasing the pinch point, the exergy efficiency decreases and SIC increases.

The effect of changes in the condenser temperature and the degree of superheat on the variation of exergy efficiency is shown in Fig. 12. According to the figure, reducing the temperature of condenser and the degree of superheat in the optimum thermo-economic condition leads to an increase in the exergy efficiency.

3.4. ROI results

In this section, ROI is investigated using the thermo-economic parameters of optimization results for three countries, Iran, France and America. Table 15 shows the amount of ROI for basic ORC and NORC in these three countries. The results of comparative study show that, the amount of ROI in Iran for both working fluids is more than two other countries. The high value of electricity price and low corporate tax rate in Iran can give rise to this result.

4. Conclusions

In this study, geothermal energy was investigated with the aim of producing more electricity and also reducing the emission of greenhouse gases which is one of the main causes of global warming. Therefore, a new ORC was designed and its performance was evaluated by comparing with basic ORC results in optimum conditions for two different types of working fluids (isobutane as dry and R123 as isentropic). The optimization of cycles’ operating parameters was carried out using three methods: thermodynamic, economic and thermo-economic objective functions. Optimized operating parameters contain the temperature of evaporators and regenerative, degree of superheat and pinch point temperature difference. Furthermore, an economic investigation based on ROI was performed in order to evaluate practical applications of considered ORCs for Iran, France and USA. The following results were concluded:

- Thermodynamic properties obtained from PR EOS had a good agreement with NIST standard reference database and literatures.
- In mentioned optimization procedure, evaporator temperature and pinch point temperature difference tended to be maximum and minimum in their optimization bands, respectively. However, degree of superheat illustrated a different pattern in economic (SIC) and thermodynamic (exergy) optimization.
- According to NORC results compared to the basic ORC, RORC and TSDORC performances, it was observed that NORC had higher efficiency (exergy and energy) than other cycles for both working fluids and three objective functions.
- Destroyed exergy in the basic ORC was reduced by designing a novel ORC configuration studied in this paper (NORC).
- The amount of ROI calculated for Iran, France and USA had decreasing trends, respectively.

**Acknowledgment**

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