Energy and exergy analyses of single flash geothermal power plant at optimum separator temperature

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Abstract

In this work, a thermodynamic analysis consisting of energy and exergy analyses is carried out to determine the performance of a single flash geothermal power plant. A new derivation for determination of the optimum separator temperature, which results in maximum turbine power output, is achieved. The energy and exergy analyses are carried out at that optimum separator temperature. The thermodynamic derivation showed that the separator would result in maximum performance of the power plant when it operates at the average value of the production well and condenser temperatures. Moreover, the derivation was numerically validated by calculating three different values for the geothermal well temperatures. The results show that the highest exergy destruction rate is in the expansion valve followed by the steam turbine, the mixing process and the pump. The separator has exactly zero exergy destruction rate while the condenser has almost zero exergy destruction rate. The results also show that the exergy destruction rate for all components of the power plant decreases with reduction in the geofluid temperature of the production well except for the condenser. The maximum energy efficiency of the power plant is about 12.5, 11 and 9.5% when the geofluid of the geothermal well temperature is 300, 275 and 250°C, respectively.

Keywords: single flash; geothermal; exergy analysis; optimum separator temperature

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1. INTRODUCTION

Geothermal energy is a renewable energy obtained by using natural heat from the interior layers of the earth. The extracted energy from these layers is applicable for various purposes including desalination, heating, cooling and power generation depending on the temperature of geothermal fluid. At low temperatures (below 100°C), geothermal fluid can be used in drying technology, cooling load production using single effect lithium bromide absorption chiller, agriculture and domestic use. Owing to the merits of generating power from innovative renewable technologies, geothermal energy has received big interest in recent years.
the vapor cools down and condenses back to liquid form, after which the cycle then starts again.

Nowadays, some of scientists do efforts to combine geothermal source and Rankine cycle to gain cheap energy; as a consequence, organic Rankine cycle (ORC) power generation that uses low-temperature geothermal resources is introduced as an alternative for the presently used power production technologies. Development of ORC power generation and state-of-the-art drilling technologies and other factors make this renewable and unconventional energy source as one of the best future viable, alternate and available resources. Some new articles have been published concerning this issue. One of the helpful references is the work by Ahmadi et al. [1]. They did a review about ORC powered by geothermal resource. Firstly, they reviewed the applications of ORC for electricity production from geothermal energy resource. Secondly, hybrid optimization approaches for the purpose of maintaining long term performance of enhanced geothermal system reservoirs were summarized. Serafino et al. [2] presented a robust design optimization methodology for ORCs. The integration of an enhanced single flash geothermal cycle with a transcritical ORC was proposed by Hassani and Mosaffa [3]. To discover the feasibility and thermoeconomic improvement, the proposed system was studied and thermo economically compared with a subcritical ORC integrated with single flash geothermal. Their results showed that the heat source temperature plays a key role in the power production variation, while the separator and flash results showed that the heat source temperature plays a key role. Moreover, the proposed ORC was proposed by Hassani and Mosaffa [3] to discover the feasibility and thermoeconomic improvement, the proposed system was studied and thermo economically compared with a subcritical ORC integrated with single flash geothermal. Their results showed that the heat source temperature plays a key role in the power production variation, while the separator and flash results showed that the heat source temperature plays a key role.

Low efficiency of ORC with low-grade heat source has limited their usage in industry. In order to rise up the power output capacity and its efficiency, a new configuration composed of an ORC and ejector (EORC) was proposed by Li et al. [27] and Liu et al. [28] investigated a two-stage Rankine cycle for power generation including a water steam Rankine cycle and an organic Rankine bottoming cycle. Wang et al. [29], in another experimental study, investigated the performance of a low-temperature solar recuperative Rankine cycle system using working fluid of R245fa. The results showed that both thermal and collector efficiencies could be significantly improved by adjusting the working fluid flow rate to an appropriate level based on the solar heat flux. Abdelkaareem et al. [30] reviewed the latest developments in the renewable energy systems to power desalination plants. The review focused on desalination processes powered by solar, geothermal, wind and ocean energy. Toward the end, the work also outlined...
the existing challenges and made recommendations about future directions.

The main objective of this work is to evaluate the performance of single flash geothermal power plant at optimum separator temperature at which the exergy destruction rates of all components of the power plant are determined. Hence, a novel derivation of the separator optimum temperature that gives maximum power output is derived.

2. THERMODYNAMIC ANALYSIS

The schematic diagram of single flash geothermal power plant is shown in Figure 1 where the components of the plant are expansion valve, separator, steam turbine, condenser and a pump. The geofluid (pure water) is flowing from the production well (state 1) as saturated liquid through an expansion valve where its pressure and temperature drop to cause two-phase flow. The two-phase flow (state 2) enters the adiabatic separator where the vapor is separated (state 5) and sent to the steam turbine. The remaining liquid in the separator (state 7) is sent back to a reinjection well or it can be used as waste heat source for low-temperature applications. The steam is expanded in the turbine to state 4, which is two-phase that is condensed to state 5 in the condenser. The flow leaving the condenser and the separator are mixed at state 8 prior to its use in other applications. The applied equations for modeling of this configuration are presented in the following subsections.

2.1. Mass balance

Mass rate balance equations for the system are written as follows:

\[ \dot{m}_1 = \dot{m}_2 = \dot{m}_8 \]  

\[ \dot{m}_3 = \dot{m}_4 = \dot{m}_5 = \dot{m}_6 = x_2 \dot{m}_2 \]  

\[ \dot{m}_7 = (1 - x_2) \dot{m}_2 \]  

where \( x_2 \) is the vapor quality at state 2.

2.2. Energy balance

The energy rate balance equations of the system components can be written in terms of the specific enthalpy, \( h \), as follows:

**Expansion valve:**

\[ h_1 = h_2 \]  

where

\[ h_1 = h_f(T_1) \]  

The steam quality at state 2 can be expressed as

\[ x_2 = \frac{h_2 - h_f}{h_{fg}} \]  

which is needed to calculate the mass flow rates of saturated vapor and saturated steam leaving the separator.

**Separator:**

\[ \dot{m}_2 h_2 = \dot{m}_3 h_3 + \dot{m}_7 h_7 \]  

where

\[ h_3 = h_g(T_{sep}) \]  

and

\[ h_7 = h_f(T_{sep}) \]  

**Steam turbine:**

\[ \dot{W}_T = \dot{m}_3 (h_3 - h_4) \]  

The specific enthalpy of state 4 can be obtained using the following equations:

\[ x_{4s} = \frac{s_{4s} - s_f}{s_{fg}} \]  

\[ s_{4s} = s_3 = s_g(T_{sep}) \]  

\[ h_{4s} = h_f + x_{4s} h_{fg} \]  

\[ h_4 = h_3 - \eta_T (h_3 - h_{4s}) \]  

where \( \eta_T \) is the isentropic efficiency of the steam turbine and the subscript \( s \) denotes the isentropic state.
Condenser:
\[ \dot{Q}_c = \dot{m}_4 (h_4 - h_3) \]  
(15)

where \( h_3 = h_f(P_{\text{cond}}) \).

Pump:

The pump work can be obtained from the following equations:

\[ s_{6a} = s_5 \]  
(16)

\[ h_{6a} - h_5 = v_5 (P_6 - P_5) \]  
(17)

\[ \eta_p = \frac{h_{6a} - h_5}{h_6 - h_5} \]  
(18)

where \( \eta_p \) is the isentropic efficiency of the pump and \( v_5 \) is the specific volume at state 5.

\[ \dot{W}_p = \dot{m}_5 (h_6 - h_5) \]  
(19)

2.3. Exergy analysis

The specific flow exergy for state \( i \) can be expressed as follows:

\[ e_{xi} = (h_i - h_o) - T_o (s_i - s_o) \]  
(20)

and the corresponding exergy rate as follows:

\[ \dot{E}_i = \dot{m}_i [(h_i - h_o) - T_o (s_i - s_o)] \]  
(21)

Exergy rate balances for each component can be written in terms of the flow exergy rate as well as the exergy destruction rate as follows:

Expansion valve:

\[ \dot{E}_1 = \dot{E}_2 + \dot{E}_{d,v} \]  
(22)

Separator:

\[ \dot{E}_2 = \dot{E}_3 + \dot{E}_7 + \dot{E}_{d,s} \]  
(23)

Steam turbine:

\[ \dot{E}_3 = \dot{E}_4 + \dot{W}_T + \dot{E}_{d,T} \]  
(24)

By applying Equations (10) and (21), the exergy rate destruction of the turbine, \( \dot{E}_{d,T} \), is obtained as follows:

\[ \dot{E}_{d,T} = \dot{m}_3 T_o (s_4 - s_3) \]  
(25)

where \( T_o \) is the surrounding temperature (298.15 K).

Condenser:

\[ \dot{E}_4 = \dot{E}_6 + \dot{Q}_{\text{cond}} \left( 1 - \frac{T_o}{T_{\text{cond}}} \right) + \dot{E}_{d,c} \]  
(26)

Applying Equations (15) and (21) gives the condenser exergy destruction rate as follows:

\[ \dot{E}_{d,c} = \dot{m}_4 T_o (s_5 - s_4) + \frac{T_o}{T_{\text{cond}}} \dot{Q}_{\text{cond}} \]  
(27)

Pump:

\[ \dot{E}_5 + \dot{W}_p = \dot{E}_6 + \dot{E}_{d,p} \]  
(28)

The net power output of these power plants can be expressed as follows:

\[ \dot{W}_{\text{net}} = \dot{W}_T - \dot{W}_{P_1} \]  
(29)

The energy and exergy efficiencies, respectively, are as follows:

\[ \eta_{en} = \frac{\dot{W}_{\text{net}}}{\dot{m}_1 (h_1 - h_{\text{ref}})} \]  
(30)

\[ \eta_{ex} = \frac{\dot{W}_{\text{net}}}{\dot{E}_{\text{net}}} \]  
(31)

where \( h_{\text{ref}} \) is the geofluid specific enthalpy at reference temperature (298.15 K) and pressure (1 bar).

2.4. Optimum separator temperature

The steam quality at state 2 can be written in the following form as follows:

\[ x_2 = \frac{h_1 - h_5}{h_3 - h_7} \]  
(32)

\[ h_1 = h_2 \]  
(33)

Since states 1 and 7 are saturated liquid, then the steam quality is rewritten as follows:

\[ x_2 = \frac{C_{pl} (T_1 - T_7)}{h_3 - h_7} \]  
(34)

where \( C_{pl} \) is the specific heat of the geothermal fluid in liquid phase.

The specific enthalpy at state 4 is the following:

\[ h_4 = h_f + x_4 h_{fg} \]  
(35)

Substituting Equation (2) into Equation (10), the turbine power output is written as follows:

\[ \dot{W}_T = \dot{m}_1 x_2 (h_3 - h_4) \]  
(36)

Then substituting Equation (34) into Equation (36) gives the following:

\[ \dot{W}_T = \dot{m}_1 C_{pl} (T_1 - T_7) \frac{h_3 - h_4}{h_3 - h_7} \]  
(37)
The specific enthalpies at states 3 and 7 can be approximated by the following equations, respectively.

\[ h_3 = \Delta h_{\text{vap}} + C_{pv}T_3 \]  
(38)

where \( C_{pv} \) is the specific heat of the geothermal fluid in vapor phase and \( \Delta h_{\text{vap}} \) is the heat of vaporization of the geofluid. The temperatures in Equations (38) and (39) are in °C.

In order to prevent blades corrosion at the last stages of the expansion in the turbine, turbine exit steam quality should be kept above 0.8; therefore, the enthalpy at state 4 can be approximated as the saturated vapor enthalpy at the condenser temperature.

\[ h_4 = \Delta h_{\text{vap}} + C_{pv}T_4 \]  
(40)

where the temperature \( T_4 \) is in °C.

Substituting the enthalpy expressions of Equations (38–40) into Equation (37) gives the turbine power output as follows:

\[ \dot{W}_T = \dot{m}_1C_{pl}\frac{(T_7 - T_4)(T_1 - T_7)}{\Delta h_{\text{vap}} + (C_{pv} - C_{pl})T_7} \]  
(41)

To find out the optimum separator temperature, the derivative of \( \dot{W}_T \) with respect to \( T_7 \) should be equal to zero as follows:

\[ \frac{d\dot{W}_T}{dT_7} = 0 \]  
(42)

The above equation will yield the optimum separator temperature \( T_{7,\text{opt}} \) as follows:

\[ T_{7,\text{opt}} = \frac{\Delta h_{\text{vap}}(T_1 + T_4) + (C_{pv} - C_{pl})T_1T_4}{2\Delta h_{\text{vap}} + (C_{pv} - C_{pl})} \]  
(43)

It can be proved numerically for water as geofluid that \( 2\Delta h_{\text{vap}} \gg (C_{pv} - C_{pl}) \) and \( \Delta h_{\text{vap}}(T_1 + T_4) \gg (C_{pv} - C_{pl})T_1T_4 \), then \( T_{7,\text{opt}} \) is obtained as follows:

\[ T_{7,\text{opt}} = \frac{T_1 + T_4}{2} \]  
(44)

Equation (44) shows that the single flash geothermal power plant gives maximum turbine power output when the separator is operating at the average temperature of production well and condenser temperatures.

3. RESULTS AND DISCUSSION

In order to proceed with the analysis, the input parameters required to solve the mass, energy and exergy equations should be given. These parameters are the mass flow rate and temperature at state 1, the separator temperature, the outlet pump pressure, condenser temperature or pressure and the isentropic efficiency of the steam turbine and the pump. The geofluid temperature at state 1 is considered as 250, 275 and 300°C for the three cases considered in this study, the mass flow rate from the well is 50 kg/s, the isentropic efficiency of steam turbine and pump is 0.85, the outlet pump pressure is 1.5 MPa and the condenser temperature is 50°C. The separator is operating at the optimum temperature, which is derived in Equation (44).

Tables 1, 2 and 3 show the thermodynamic properties for each state at \( T_1 = 250°C \) and \( T_{\text{sep}} = 150°C \), \( T_1 = 275°C \), \( T_{\text{sep}} = 162.5°C \), \( T_1 = 300°C \), \( T_{\text{sep}} = 175°C \), respectively. The tables show that the increase in the temperature of the geothermal well results in higher saturated vapor mass flow rate entering the turbine. This increase in the mass flow rate through the turbine is due to the increase in the optimum separator temperature, which results in higher vapor quality at the separator inlet.

The effect of the separator temperature on the power output of the steam turbine for three different values of geothermal well temperature is shown in Figure 2. This figure shows that the turbine power output increases as the separator temperature increases till it reaches a maximum point, after which it declines with further increase in the separator pressure. The power output reaches its maximum value when the separator temperature is equal to the average temperature of the geothermal well and condenser temperatures as shown in Equation (44). The figure also shows that increasing the geothermal well temperature results in higher turbine power output. The variation of power plant thermal efficiency with the separator temperature is shown in Figure 3, which shows that there exists a maximum value of thermal efficiency at the optimum separator pressure as the case in Figure 2. Moreover, the figure shows that increasing the geothermal well temperature results in higher thermal efficiency of the power plant.

Figure 4 shows the exergy efficiency of the single flash geothermal power plant with separator temperature. The figure shows that increasing the separator temperature results monotonical increase in the exergy efficiency. Figure 4 also shows that increasing the geothermal well temperature at the same separator temperature results in lower exergy efficiency, which is logical from the definition of the exergy efficiency given in Equation (31) from which it can be proved that the difference in exergy of the dominator is increasing as the geothermal well temperature increases.

To have a clear idea about the reason of the variations presented in Figures 2 and 3, the effects of separator temperature on the steam quality at the separator inlet and mass flow rate in the turbine are presented in Figures 5 and 6, respectively. These figures show that increasing the separator pressure results in a continuous decrease in the steam quality at state 2; hence, the
mass flow rate in the turbine decreases, which can be seen from Equation (2). Increasing the separator temperature results in an increase in the saturated vapor enthalpy at the turbine inlet. For separator temperature below the optimum separator temperature, the increase in saturated vapor enthalpy at the turbine inlet is dominant, which results in increase in the turbine power output and thermal efficiency; however, for temperature higher than optimum separator temperature, the decrease in mass flow rate in the turbine is dominant; hence, the turbine power output and thermal efficiency decrease as shown in Figures 2 and 3.

The exergy destruction rates of all components of the single flash power plant are shown in Table 4 for three different geothermal well temperatures with their corresponding optimum separator temperatures. The table shows that the highest exergy destruction rate occurs in the expansion valve followed by the steam turbine, the mixing process and the pump for all the three cases.

The table also shows that the destruction rates of all components are decreasing with increasing the geothermal well temperature. The separator exergy destruction is exactly zero, which is due to the fact that the separator is adiabatic and it has zero entropy generation, which can be easily proved. The table shows that the condenser has almost zero exergy destruction due to the fact that the condenser is operating at low temperature.

The separator exergy destruction rate is zero as can be seen in Table 4. This can be proved thermodynamically by using second law of thermodynamics. The separator entropy generation rate is written as

\[ \dot{s}_{gen,sep} = \dot{m}_3 s_3 + \dot{m}_7 s_7 - \dot{m}_2 h_2 \]  (45)

The mass flow rates of states 3 and 7 are given by Equations (2) and (3)

\[ \dot{m}_3 = x_2 \dot{m}_2 \]  (2)

\[ \dot{m}_7 = (1 - x_2) \dot{m}_2 \]  (3)

Table 1. Thermodynamic properties of operating fluid at \( T_1 = 250^\circ C \) and \( T_{sep} = 150^\circ C \) (optimum separator temperature).

| State | \( T \) (K) | \( P \) (MPa) | \( h \) (kJ/kg) | \( s \) (kJ/kg.K) | \( \dot{m} \) (kg/s) | \( x \) |
|-------|-------------|-------------|-------------|-------------|----------------|-------|
| 1     | 523.2       | 3.974       | 1085        | 2.793       | 50             | 0     |
| 2     | 423.2       | 0.4757      | 1085        | 2.913       | 50             | 0.2143 |
| 3     | 423.2       | 0.4757      | 2746        | 6.838       | 10.71          | 1     |
| 4     | 323.1       | 0.01234     | 2303        | 7.181       | 10.71          | 0.8788 |
| 5     | 323.1       | 0.01234     | 209.3       | 0.7037      | 10.71          | 0     |
| 6     | 323.3       | 1.5         | 211.2       | 0.7049      | 10.71          | —     |
| 7     | 423.2       | 0.4757      | 632.3       | 1.842       | 39.29          | 0     |
| 8     | 428         | 0.275       | 542.0       | 1.634       | 50             | —     |

Table 2. Thermodynamic properties of operating fluid at \( T_1 = 275^\circ C \), \( T_{sep} = 162.5^\circ C \) (optimum separator temperature).

| State | \( T \) (K) | \( P \) (MPa) | \( h \) (kJ/kg) | \( s \) (kJ/kg.K) | \( \dot{m} \) (kg/s) | \( x \) |
|-------|-------------|-------------|-------------|-------------|----------------|-------|
| 1     | 548.2       | 5.943       | 1210        | 3.021       | 50             | 0     |
| 2     | 435.7       | 0.6579      | 1210        | 3.17        | 50             | 0.2524 |
| 3     | 435.7       | 0.6579      | 2761        | 6.729       | 12.62          | 1     |
| 4     | 323.1       | 0.01234     | 2277        | 7.103       | 12.62          | 0.8682 |
| 5     | 323.1       | 0.01234     | 209.3       | 0.7037      | 12.62          | 0     |
| 6     | 323.3       | 1.5         | 211.2       | 0.7049      | 12.62          | —     |
| 7     | 435.7       | 0.6579      | 686.5       | 1.968       | 37.38          | 0     |
| 8     | 134.8       | 0.313       | 566.5       | 1.677       | 50             | —     |

Table 3. Thermodynamic properties of operating fluid at \( T_1 = 300^\circ C \), \( T_{sep} = 175^\circ C \) (optimum separator temperature).

| State | \( T \) (K) | \( P \) (MPa) | \( h \) (kJ/kg) | \( s \) (kJ/kg.K) | \( \dot{m} \) (kg/s) | \( x \) |
|-------|-------------|-------------|-------------|-------------|----------------|-------|
| 1     | 573.2       | 8.584       | 1344        | 3.253       | 50             | 0     |
| 2     | 448.2       | 0.8918      | 1344        | 3.436       | 50             | 0.2967 |
| 3     | 448.2       | 0.8918      | 2773        | 6.625       | 14.83          | 1     |
| 4     | 323.1       | 0.01234     | 2253        | 7.028       | 14.83          | 0.858 |
| 5     | 323.1       | 0.01234     | 209.3       | 0.7037      | 14.83          | 0     |
| 6     | 323.3       | 1.5         | 211.2       | 0.7049      | 14.83          | —     |
| 7     | 448.2       | 0.8918      | 741.2       | 2.091       | 35.17          | 0     |
| 8     | 139.0       | 0.352       | 584         | 1.729       | 50             | —     |
Table 4. *Exergy destruction rates (kW) of power plant components for different production well temperature.*

| Well Temperature ($^{\circ}C$) | Valve | Turbine | Mix pump | Condenser | Separator |
|--------------------------------|-------|---------|----------|-----------|-----------|
| $T_7 = 175$ | 2726  | 1780    | 680.2    | 5.157     | 4.40      |
| $T_7 = 162.5$ | 2219  | 1408    | 506.4    | 4.388     | 1.00      |
| $T_7 = 150$  | 1789  | 1060    | 362.6    | 3.725     | 8.740     |

Figure 2. Variation of turbine power output vs separator temperature for different production well temperatures.

Figure 3. Variation of thermal efficiency vs separator temperature for different production well temperatures.

Figure 4. Variation of exergy efficiency vs separator temperature for different production well temperatures.

Figure 5. Variation of steam quality at state 2 with separator temperature.
State 3 is saturated vapor and state 7 is saturated liquid, which are expressed as follows: $s_3 = s_f(T_{sep})$ and $s_7 = s_f(T_{sep})$. The specific enthalpy at state 2 is expressed as

$$h_2 = h_f + x_2(h_g - h_f)$$  \hspace{1cm} \text{(46)}$$

The substitution of Equations (2), (3) and (46) into Equation (45) results in zero entropy generation rate for the separator, i.e. $i_{gen,sep} = 0$. Hence, the separator exergy destruction rate is zero.

4. CONCLUSIONS

In this paper, single flash geothermal power plant was analyzed based on energy and exergy analyses to determine the best thermal performance of the power plant. Separator is among the most important components of the power plant on which the power plant performance is strongly dependent. A new thermodynamic methodology was used to determine the optimum separator temperature, which resulted in maximum power output of the steam turbine. It was found that the maximum power was achieved when the separator operates at the average temperature of production well and condenser temperatures. Hence, the maximum thermal efficiency of the power plant occurred at that optimum separator temperature was in the range of 9 and 12% for the considered geothermal well temperature range considered in this study. The exergy destruction rates of the expansion, steam turbine, mixing process and pump were in the range of 2726 and 1789, 1780 and 1060, 690 and 362, 5 and 7 kW, respectively, for the considered ranges of geothermal well temperature of 175 and 150°C. It is recommended from the results that the separator should operate at the optimum temperature for all geothermal well temperatures.

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