Energy and Exergy Analysis of Geothermal Steam Binary Power Generation

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ABSTRACT

Scaling and corrosion have presented problems in many geothermal systems. Dissolved materials in geothermal waters can exhibit aggressive corrosive properties or have the tendency to deposit large amounts of mineral scale. Evaporators of binary cycles are a very important part of the system and their indirect contact with geothermal fluid is one of the methods to overcome aggressive behavior of these fluids. The idea of this project was to use separated steam in an evaporator as a heat source for binary plant. In this research, the experiment was designed and results were collected, a model was developed and validated using experimental data. Energy and exergy efficiencies of the proposed plant was calculated to be 7.42% and 35.14%, respectively.

Introduction

The increase in energy demands, decline in hydrocarbon energy resources and the link between energy utilization and environmental impact have prompted calls for sustainable approaches to the development and management of the Earth's energy resources (Rosen and Dincer, 2001). Consequently, it has become increasingly important to understand the mechanisms that degrade the quality of energy and its resources, and to develop systematic approaches to improve energy conversion systems (Gong and Wall, 1997). Exergy is defined as energy, which can theoretically be converted into effective work or electricity. Exergy analysis is based on the assumption that only the exergy contained in any heat stream has value, then the non-convertible part of the heat stream has no value and is defined as anergy. Exergy analysis has been used as a powerful tool to identify and quantify energy degrading processes because it enables evaluation of the types, locations and quantities of the energy losses (Dincer and Rosen, 2007).

Geothermal energy utilization is commonly divided into two categories, i.e., electric production and direct application (Jalilinasrabady, 2015). The utilization method depends on parameters such as local demand for heat or electricity, distance from potential market, resource temperature, and chemistry of the geothermal fluid. These parameters are important to the feasibility of exploitation. Utilization of geothermal fluid depends heavily on its thermodynamic characteristics and chemistry. These factors depend on the geothermal system from which the fluid originates (Jalilinasrabady et., 2010).

Sustainable utilization of earth resources has been an attractive topic, which is under improvement and development. Geothermal energy is one of the renewable resources that need further studies for its sustainable utilization. Cascade use of geothermal fluid is the ideal way of its optimum usage, but uncertainty is always accompanying these projects (Jalilinasrabady et al., 2013). Plant operations of geothermal fields depend on demand for heat or electricity in the region and reservoir ability to support these utilization units.

In most of geothermal power plants there is considerable amount of exergy loss due to reinjection. Despite technical issues related to reservoir management, if this reinjection doesn't participate in sustainability of the reservoir, the injected fluid (usually with temperature around 100°C) can be considered as a total exergy loss (Jalilinasrabady and Itoi, 2012, Jalilinasrabady et al., 2011).

The Organic Rankine Cycle (ORC) has been used for binary systems at a number of geothermal fields. They are used for low temperature resources when flash plants are not economically viable. The basic technology is analogous to the steam Rankin cycle used in thermal power plants, except that the steam comes from a geothermal reservoir rather than a boiler. The most attractive geothermal fields for developers have been those that produce high temperature and high enthalpy fluids. These fields can deliver high quality steam at high pressure, which makes operation of the condensing steam turbines more efficient and reduces electricity production costs compared to fields that produce low-enthalpy fluids. Condensing steam plants are typically used for resources with temperatures higher than 200°C. For low-enthalpy resources, a low operating pressure is needed to obtain a reasonable amount of flashed steam. This increases the equipment size and makes the process more expensive compared to high-enthalpy fluids. Additionally, a significant proportion of the available energy in the produced fluids remains in the separated water. Turbine-generator unit capacities are typically in the 20–80 MW range, but are manufactured in sizes from less than 5 MW up to 110 MW (Dipippo, 2008).

Like all other sources of energy, geothermal has its own challenges in production, transmission, distribution and utilization. One of them is deposition of solids in the system, from the geothermal fluid (Bjornsson, 1989). Another problem is corrosion.

Scaling and corrosion have presented problems in many geothermal systems. Dissolved materials in geothermal waters can exhibit aggressive corrosive properties or have the tendency to deposit large amounts of mineral scale. Either property can seriously shorten the life of pipes in the production well or the reinjection well. Scaling and corrosion constitute technical barriers to the utilization of geothermal resources and it can be said that these are two of the most important geothermal utilization problems that require the close attention (Papic, 1991).

Evaporators of binary cycles are very important part of the system and their indirect contact with geothermal fluid is one of the methods that can be used to overcome aggressive behavior of these fluids. It is common practice to use geothermal brine as a heat source to heat up the water in a secondary loop. Since the scaling is a problem with geothermal fluid, if the steam and brine are separated, remaining steam can be suitable heat source to be sent to evaporator. The idea of this project was to use this steam in an evaporator as a heat source for a binary plant. In this research, the experiment was designed and results were collected, a model was developed and validated using experimental data.

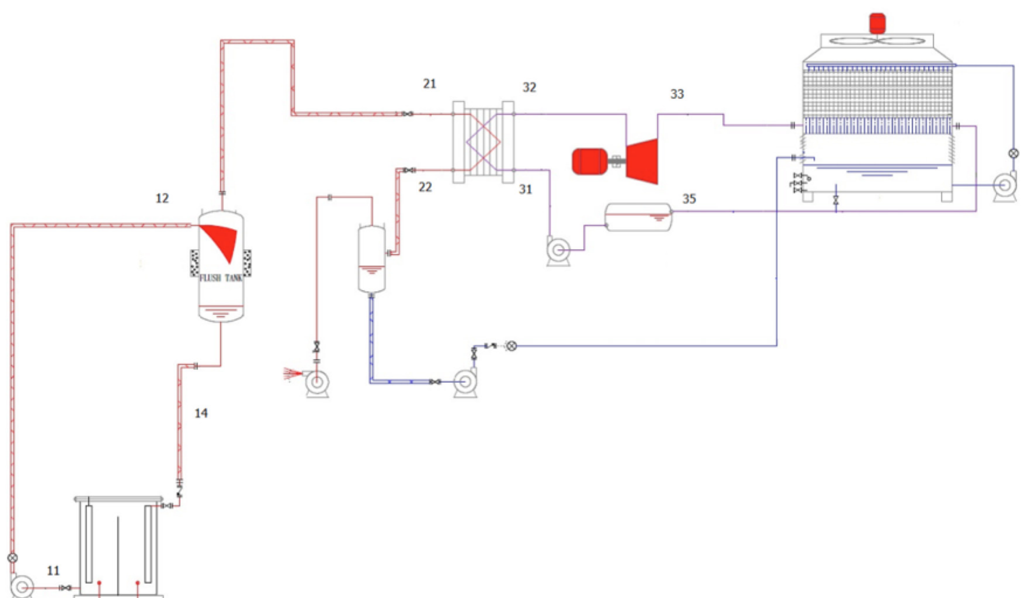

Figure 1. schematic diagram of proposed binary geothermal power plant.

Energy and Exergy

The ORC cycle is designed to use steam as a heat source in evaporation. Figure 1, shows simplified diagram of proposed plan. Thermodynamic model was developed and optimized using actual data from an experiment that was conducted on site.

Obtained results from the model shows the actual turbine power output is 8.39 kW while working fluid circulation pump is consuming 1.818 kW. The net power output from ORC cycle is 6.572 kW, Approximately 2.382 kW is consumed by parasitic uses in the units including fan power and other pumps except working fluid circulation pump. Subtracting the parasitic load from the net power of ORC cycle gives the net power output from the power plant equal to 4.19 kW.

$$w_{output,orc} = w_{actual,turbine} - w_{pump} \tag{1}$$

$$w_{output,orc} = 8.39 - 1.818 = 6.572\,\text{kW}$$

$$w_{output,plant} = w_{output,orc} - w_{parasitic} \tag{2}$$

$$w_{output,plant} = 6.572 - 2.382 = 4.19\,\text{kW}$$

The other parameters discussed in this study are shown in table 1. The state numbers refer to state locations in figure 1.

States 0 and 0' refer to the restricted dead states for the geothermal and working fluid respectively. They correspond to an ambient temperature of 12°C and an atmospheric pressure of 84 kPa, which are Tokyo's reference environmental conditions.

Figure 2 shows diagram window of developed model which corresponds to recorded data from actual site performance.

Table1. Parameters at major stages of the plant.

State	Fluid	Phase	Temp. T (°C)	Pressure P (kPa)	Specific Enthalpy h (kJ/kg)	Specific Entropy s (kJ/kg K)	Mass Flow $\dot{m}$ (kg/s)	Specific Exergy ex (kW)	Exergy Rate $\dot{E}x$ (kW)
0	Brine	Dead state	12	86	50.44	0.1804	-	0	-
0′	R134a	Dead state	12	86	265.8	1.082	-	0	-
11	Brine	Liquid	99.5	60.11	417	1.301	1.598	47.02	75.14
12	Brine	2 phase	88.7	60.8	417	1.303	1.598	46.45	74.23
21	Brine	Steam	87.7	60.8	2657	7.502	0.03864	518.81	20.05
32	R134a	Steam	87.8	2856	288.1	0.9062	0.4346	72.43	31.48
33	R134a	2 phase	32.2	603	264	0.9267	0.4346	42.48	18.46
35	R134a	Liquid	20.5	603	80.03	0.3088	0.4346	34.71	15.08
31	R134a	Liquid	32.5	2856	84.21	0.3109	0.4346	38.29	16.64
22	Brine	Liquid	86.8	60.8	363.6	1.156	0.03864	34.97	1.35
14	Brine	Liquid	86.29	60.8	361.4	1.15	1.559	34.48	53.75

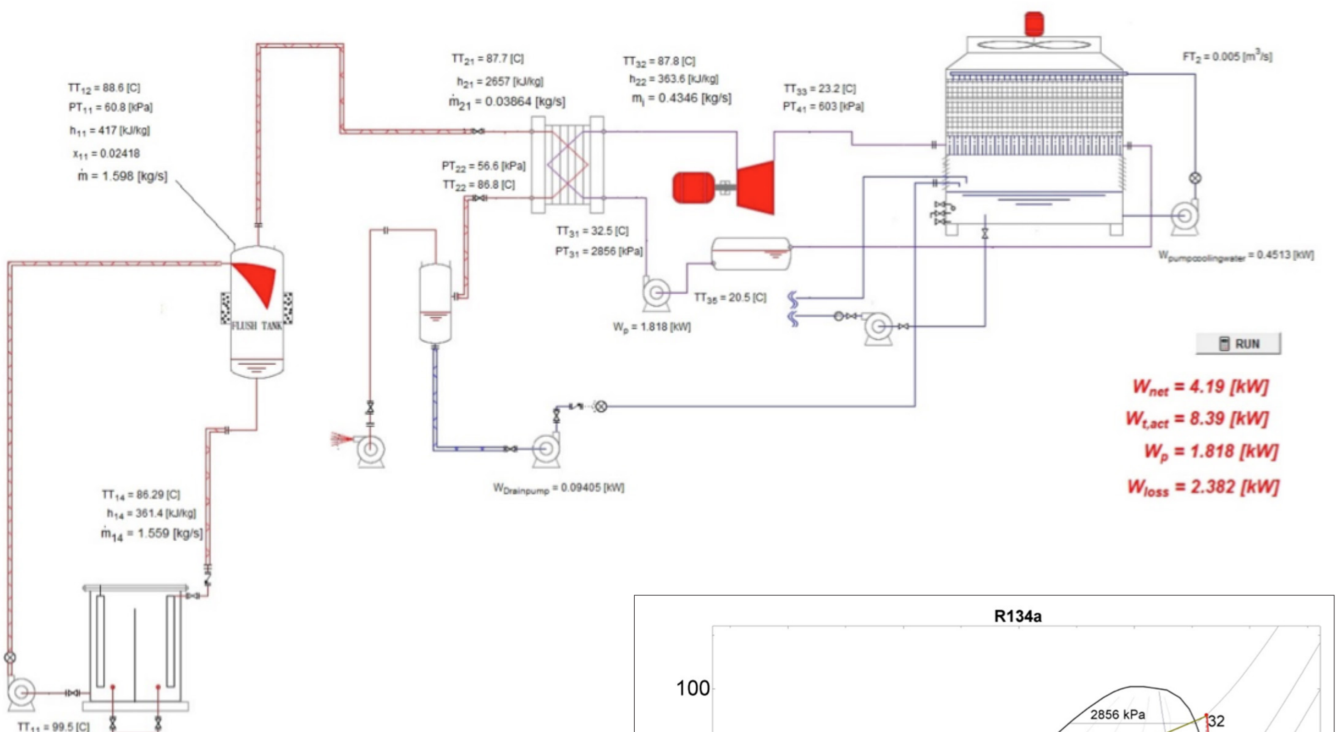

Figure 2. Calculation diagram window of developed model with key parameters.

Figure 3 illustrates T-S diagram of ORC cycle, and it can be seen that the working fluid is in the superheated region where actual power production happens (state 32-33). 80 l/min of 88°C hot water is being sent to pre-heat the ORC engine unit in actual plant performance to ensure that ORC working fluid cross to superheated region, since nature of this heat transfer is

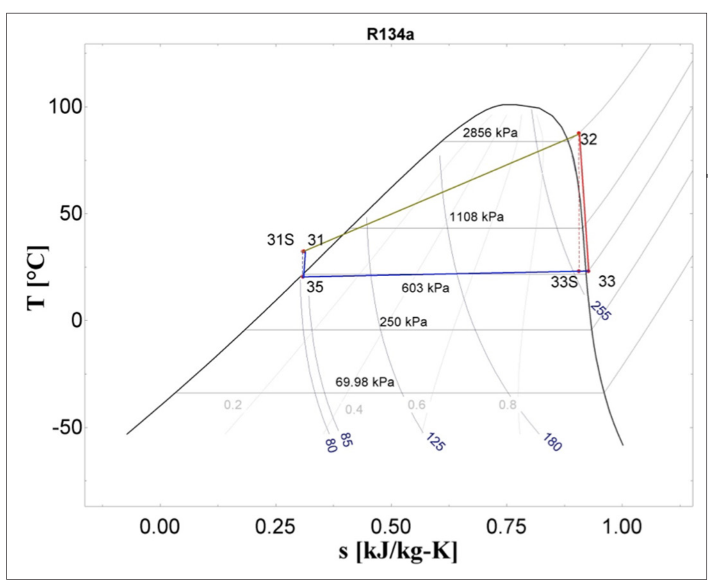

Figure 3. T-S diagram of ORC cycle.

not very clear, this amount of additional heat was not included in energy and exergy analysis, but it is interesting to know that even without this additional heat, superheating condition is being achieved.

1. Exergy Analysis

Exergy analysis has been applied for each component of the system such as the separator, turbines, condenser, etc. The exergy will be expressed as equal to the maximum work when the stream of substance is brought from its initial state to the environmental state defined by P_0 and T_0 by physical processes involving only thermal interaction with the environment.

The specific flow exergy of the geothermal fluid at any state can be calculated from:

$$ex = \left(h - h_0\right) - T_0\left(s - s_0\right) \tag{3}$$

Where h and s are the specific enthalpy and entropy of the geothermal fluid at the specific state, and h_0 and s_0 are the properties at the dead state. For the mass flow rate, the exergy flow rate can be written as (Kotas, 1995):

$$\dot{E}_x = \dot{m}\left(ex\right) \tag{4}$$

For a control volume, an exergy balance equation can be expressed as (Rosen, 1999):

$$E_{input} = E_{desired} + E_{waste} + E_{destroyed} \tag{5}$$

where:

- E_{input} : Total exergy inflow into the control volume (kW)
- $E_{desired}$: Total desired exergy output (net work output) (kW)
- E_{waste} : Sum of exergy from the system other than the desired (kW)
- $E_{destroyed}$: Sum of exergy lost in the system as a result of irreversibility (kW), (directly related to entropy generation: $E_{destroyed} = T_0 S$))

1.1 Heat Exchanger

The exergy efficiency of a heat exchanger shows the exergy increase of the cold stream divided by the exergy decrease of the hot stream. Applying this definition to the heat exchanger, the exergetic efficiency of HEX can be obtained as:

$$\varphi_{HEX} = \frac{\dot{E}_{x32} - \dot{E}_{x31}}{\dot{E}_{x21} - \dot{E}_{x22}} = \frac{31.48 - 16.64}{20.05 - 1.35} \times 100 = 79.36\% \tag{5}$$

The difference between the exergy of the inlet and outlet streams, is the exergy destruction rate in the equipment. (The difference between numerator and denominator).

$$\dot{I}_{HEX} = \left(\dot{E}_{x21} - \dot{E}_{x22}\right) - \left(\dot{E}_{x32} - \dot{E}_{x31}\right) \tag{6}$$

$$\dot{I}_{HEX} = \left(20.05 - 1.35\right) - \left(31.48 - 16.64\right) = 3.86\text{kW}$$

1.2 Condenser

The exergy efficiency similarly can be calculated for condenser. However the exergy destruction in the condenser is approximately equal to the exergy decrement of the working fluid across the condenser. This means the exergy increment of the air, which is small, is neglected.

$$\dot{I}_{cond} = \left(\dot{E}_{x33} - \dot{E}_{x35}\right) \tag{7}$$

$$\dot{I}_{cond} = \left(18.46 - 15.08\right) = 3.38\text{kW}$$

1.3 Turbine

The exergy efficiency of turbine shows how efficient the exergy of fluid passing through turbine is being converted to work.

$$\varphi_{tur} = \frac{\dot{w}_{tur,act}}{E_{x32} - E_{x33}} \tag{8}$$

$$\varphi_{tur} = \frac{8.39}{31.48 - 18.46} \times 100 = 64.44\%$$

The difference between the numerator and denominator in equation (8) is the exergy destruction rate in the turbine:

$$\dot{I}_{tur} = \left(E_{x32} - E_{x33}\right) - \dot{w}_{tur,act} \tag{9}$$

$$\dot{I}_{tur} = \left(31.48 - 18.46\right) - 8.39 = 4.63\text{kW}$$

1.4 Pump

The exergy efficiency and destruction rate for pump can be written as:

$$\varphi_{pump} = \frac{E_{x31} - E_{x35}}{\dot{w}_{pump}} = \frac{16.64 - 15.08}{1.818} \times 100 = 85.81\% \tag{10}$$

$$\dot{I}_{pump} = \dot{w}_{tur} - \left(E_{x31} - E_{x35}\right) \tag{11}$$

$$\dot{I}_{pump} = 1.818 - \left(16.64 - 15.08\right) = 0.258\text{kW}$$

1.5 Exergy Efficiency

Values for exergy efficiency can vary according to definitions, in this study three different scenarios were assumed for exergetic performance investigation. These scenarios differ in terms of exergy output/ input from/ to the plant. In scenario one, the ORC cycle, itself, has been considered. The work output from the cycle and the exergy value of steam entering the HEX are main parameters in this scenario. In scenarios 2 and 3, the work output is net power output (the practice load is deducted) and exergy input, are the exergy change in the HEX (same as scenario 1) and total brine exergy entering the plant, respectively. The exergy efficiency of the entire plant in scenarios 2 and 3, is based on the total brine exergy decrement across the HEX and brine exergy input into the plant, respectively.

1.5.1 Scenario 1 (ORC Cycle)

In this scenario, only the ORC cycle output power and its components are considered to conduct exergy analysis. The exergy efficiency of the ORC cycle can be determined as:

$$\varphi_{ORC} = \frac{\dot{w}_{net,ORC}}{E_{x21} - E_{x22}} = \frac{6.572}{20.05 - 1.35} = 35.14\% \tag{12}$$

where the denominator represents decrement in steam flow across the HEX and numerator is the net power (ORC pump power is deducted). The total exergy loss rate applying this method can be expressed as:

$$\dot{I}_{ORC} = \dot{I}_{pump} + \dot{I}_{tur} + \dot{I}_{condenser} + \dot{I}_{HEX} \tag{13}$$

$$\dot{I}_{ORC} = 3.38 + 3.86 + 0.258 + 4.63 = 12.128\text{kW}$$

1.5.2 Scenario 2 (Overall Power Plant 1)

The exergy efficiency of the overall plant in this scenario is calculated based on exergy decrement in the HEX and can be expressed as:

$$\varphi_{overall\ \ plant} = \frac{\dot{w}_{output,plant}}{E_{x21} - E_{x22}} = \frac{4.19}{20.05 - 1.35} \times 100 = 22.41\% \tag{14}$$

where the net power output is obtained by subtracting the total parasitic power from the total net power output from the ORC cycle.

1.5.3 Scenario 3 (Overall Power Plant 2)

In this scenario the exergy efficiency of the overall plant is calculated based on brine exergy input into the plant and net power output from the plant. Then:

$$\varphi_{overall\ \ plant} = \frac{\dot{w}_{net}}{E_{x11}} = \frac{4.19}{75.14} \times 100 = 5.58\% \tag{15}$$

$$\dot{I}_{overall\ \ plant} = \dot{E}_{x11} - \dot{W}_{net} = 75.14 - 4.19 = 70.95\text{kW} \tag{16}$$

2. Energy Analysis

The energy efficiency of the ORC cycle is calculated based on the ratio of the net power output from the ORC cycle to the heat input rate to the HEX.

$$\eta_{ORC} = \frac{W_{net,orc}}{\dot{m}_{21}\left(h_{21}-h_{22}\right)} = \frac{6.572}{0.03864\times\left(2657-363.6\right)}\times 100 = 7.42\% \tag{17}$$

Energy efficiencies of overall plant according to scenarios 2 and 3 can be written as:

$$\eta_{Overall} = \frac{W_{output,plant}}{\dot{m}_{21}\left(h_{21}-h_{22}\right)} = \frac{4.19}{0.03864\times\left(2657-363.6\right)}\times 100 = 4.73\% \tag{18}$$

$$\eta_{overall} = \frac{W_{output,plant}}{\dot{m}_{11}\left(h_{11}-h_{0}\right)} = \frac{4.19}{1.598\times\left(417-50.44\right)}\times 100 = 0.71\% \tag{19}$$

Figure 4 shows the energy flow diagram for plant.

Table 2 shows the exergy flow from the plant components and exergy and energy efficiencies, using values from this table Grassman diagram was drawn as shown in Figure 5.

It can be seen from figure 5 that 91.25% of the exergy entering the plant is lost. The remaining 8.75% is converted to power, 36.24 % of which is used for parasitic loads in the plant. The exergy efficiency of the plant is 35.14% based on the exergy input to the ORC cycle and 5.58% based on the exergy input to the plant.

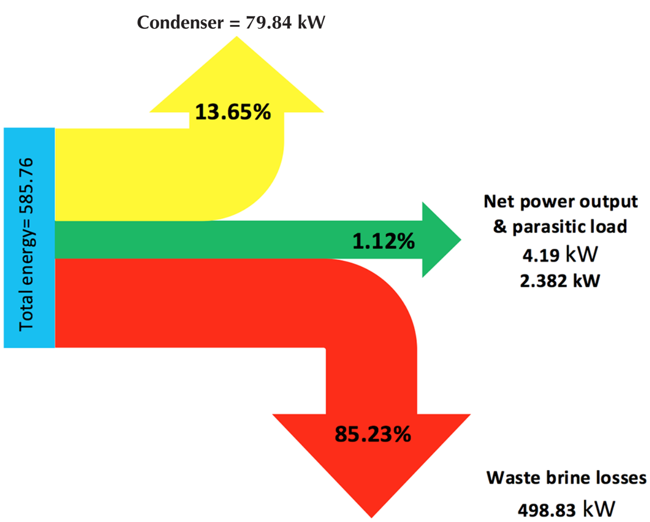

Figure 4. Energy flow diagram.

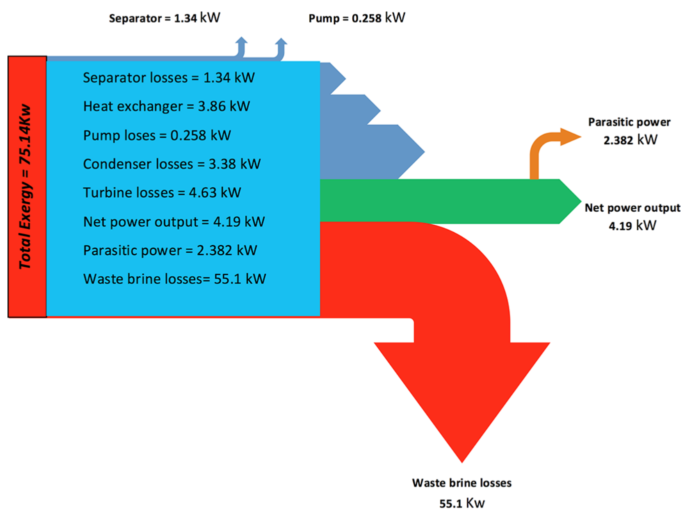

Figure 5. Grassman diagram of overall exergy flow.

Table 2. Exergy flow in various components of the plant.

Component	Exergy Destruction Rate (kW)	Exergy Efficiency (%)	Heat Transfer or Work Rate (kW)	Isentropic or Energy Efficiency (%)
Separator	1.34	98.79	0.27	-
Heat exchanger	3.86	79.36	88.62	-
Condenser	3.38	---	79.95	-
Pump	0.258	85.81	1.818	85.42
Turbine	4.63	64.44	8.39	64.28
ORC cycle (Scenario 1)	12.128	35.14	88.62	7.42
Overall plant (Scenario 2)	13.468	22.41	88.62	4.73
Overall plant (Scenario 3)	13.468	5.58	585.76	0.71

3. Conclusions

A mathematical model was developed for a binary cycle and optimized using actual data from a site experiment. Actual power produced by plant is 8.39 kW and net power produced by plant is 4.19 kW. Energy efficiency of ORC cycle is 7.42% and it is calculated as 4.73% considering parasitic loads.

Total available exergy was calculated to be 75.14 kW and exergy efficiency of ORC cycle is estimated to be 35.14% and it is 22.41% when considering parasitic loads.

Recommendations

1. Exhausted water from the evaporator is 86°C, this hot water has considerable potential and its usage should be taken into account.

2. For the purpose of pre-heating of ORC engine system, 80 l/min of hot water is being sent to this engine, as it can be seen from Figure 3, the working fluid reaches super-heated region without this heat, so its necessity should be evaluated and if it is necessary, exhausted hot water from the evaporator could be sent to this unit. This method will save the amount of hot water considerably.

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Nomenclature

		Unit
W	= power	[kW]
$\dot{m}$	= mass flow rate	[kg/s]
h	= specific enthalpy	[kJ/kg]
T	= temperature	[°C]
P	= pressure	[kPa]
s	= specific entropy	[kJ/kg K]
ex	= Specific Exergy	[kW]
$\dot{Ex}$	= Exergy rate	[kW]
$\dot{I}$	= Exergy destruction rate	[kW]

Greek Symbols

φ	= Exergy efficiency	(%)
η	= Energy efficiency	(%)

Subscripts

0	= ambient condition
tur	= Turbine
HEX	= Heat Exchanger
cond	= Condenser
ORC	= Organic Rankine Cycle
Plant	= power plant

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