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Thermodynamic Modeling and Exergy Analysis of a Heat Recovery System in Meshkinshahr Geothermal Power Plant, Iran

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ABSTRACT

A 55 MW geothermal power plant is projected to be installed in the Sabalan geothermal field. The drilling of wells to support the 55 MW power plant started in June 2008. While in the drilling phase of the project, SUNA has considered installing a demonstration pilot 4 MW power plant using steam condensing turbine in a single flash cycle. The steam fraction of the two-phase fluid produced from geothermal wells in Sabalan is less than 20 percent. In a single flash cycle the brine phase is discarded still contains a huge volume of energy.

The main objectives of this study are: to develop thermodynamic model and exergy analysis of a heat recovery system to produce electricity in an organic rankine cycle (ORC), parallel with single flash system using energy content of separated brine; pinpoint locations and quantities of exergy losses and wastes; and to determine the optimum performance parameters of the main components. The first and second law efficiencies and the productivity of geothermal fluid will be determined for single flash as well as for combined system. The Engineering Equation Solver (EES) software was used for developing and analyzing mathematical models of energy and exergy flows. Thermoeconomic analysis using obtained results can be considered for further work to help decision makers.

1. INTRODUCTION

Currently, the Sabalan Geothermal Field has four production wells and new wells are being dilled. A 4 MW demonstration pilot power plant is considered to be installed on well NWS-4 at this field. Data obtained from discharged well test shows that well NWS-4 produces two-phase geothermal fluid with mass flow of 56 kg/s and enthalpy of 954 kJ/kg at well head pressure of 5.5 bar (SKM, 2005).

Single flash cycle with steam condensing turbine will be used as energy conversion system at the power plant. Using a single flash system will result in discarding a significant volume of energy in form of brine from separator, due to low quality of produced two-phase fluid. However, the double flash cycle is not feasible due to low wellhead pressure. Using the energy content of saturated water discarded from separator in a binary ORC cycle paralleled with single flash is considered in this study. Figure 1 shows the process flow diagram of proposed system, the production fluid is separated to steam and brine through the separator, steam is directed to steam condensing turbine and brine is led to the vaporizer in binary ORC plant.

The energy performance of power generation systems is usually evaluated based on the first law of thermodynamics. However, compared to energy analysis, the exergy analysis can better and accurately show the location of inefficiencies. Integration of energy and exergy analysis can present a whole picture of the system performance.

2. SUMMARY OF THERMAL DESIGN OF THE PROPOSED SYSTEM

The thermodynamic design of the proposed system has been established in EES software.

2.1 Single Flash Cycle

The following plant operating parameters are used for the thermal design:

$p_{sep} = 5$	Separator pressure - [bar-a]
$p_{cond} = 0.1$	Condenser Pressure - [bar-a]
$\eta_{turb} = 80$	Turbine isentropic efficiency- [%]
$\eta_{pump} = 50$	Pump isentropic efficiency - [%]
$T_{db} = 10$	Wet-bulb temperature- [°C]

All pressure and heat transfer losses are neglected.

The subscript numbers refer to state locations on figure 1.

The fraction and flow rate of the steam and brine can be defined by mass and heat balance of the separator as follows:

$$\dot{m}_1 x_1 = \dot{m}_2 x_2 + \dot{m}_3 x_3 \quad (1)$$

$$\dot{m}_1 h_1 = \dot{m}_2 h_2 + \dot{m}_3 h_3 \quad (2)$$

Where the $\dot{m}$ and h are the mass flow and enthalpy of the stream at their specified state on the system. The subscript numbers denote the state position of stream at figure 1.

The turbine power production is:

$$\dot{W}_{turb} = \dot{m}_2 (h_2 - h_4) \quad (3)$$

$$\eta_{turb} = \frac{h_2 - h_4}{h_2 - h_{4s}} \quad (4)$$

Where h_4 and h_{4s} are the enthalpy values at the turbine exit states for actual and isentropic processes, respectively.

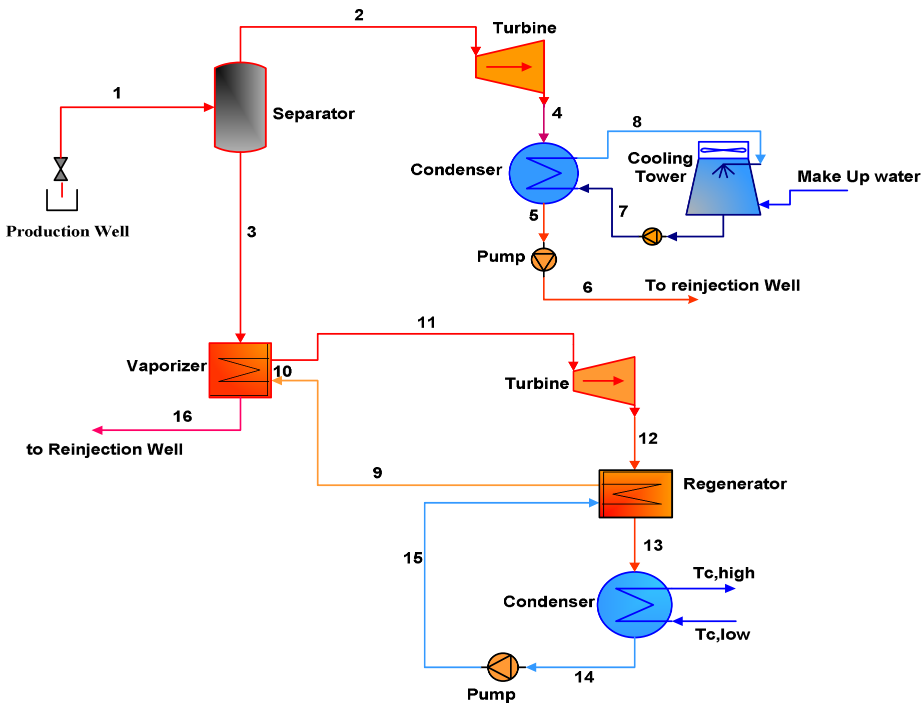

Figure1: The process flow diagram of proposed combined system

For the condenser, $\dot{Q}_{cond}$ the heat rejected by cooling water is:

$$\dot{Q}_{cond} = \dot{m}_2 (h_4 - h_5) \tag{5}$$

The mass flow of cooling water is defined by:

$$\dot{m}_{CW} = \frac{\dot{Q}_{cond}}{(h_8 - h_7)} \tag{6}$$

Where h_7 and h_8 are the enthalpies of cooling water at the inlet and outlet of condenser.

The power consumed by cooling water pump, $\dot{W}_{cwp}$ is calculated by:

$$\dot{W}_{cwp} = \dot{W}_{cwp,isentropic} . \eta_{pump} \tag{7}$$

$$\dot{W}_{cwp,isentropic} = v_{cw} \Delta p_{pump} \tag{8}$$

The energy conversion (or first law) efficiency for a heat engine operating cyclically between two thermal energy reservoirs is (Kotas, 1985):

$$\eta_{1law} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \tag{9}$$

Where $\dot{W}_{net}$ is the power delivered to the network and $\dot{Q}_{in}$ is the corresponding heat transfer to the engine per cycle.

2.2 Organic Rankine Cycle (ORC)

The working fluid operates in a contained, closed-loop cycle and is completely segregated from the heat source fluid. There are a number of possible variants of the cycle, in terms of heat exchange configuration, turbine configuration, etc, which may be selected as appropriate to the temperature and Physical state(s) of heat source fluid. In this case, the working fluid absorbs heat from a heat source (geothermal brine), via one shell and tube heat exchangers. This heat causes the working fluid to evaporate; producing the high-pressure vapor that is then expanded through a turbine-generator. The low pressure turbine exhaust vapor is then led to the regenerator. In this heat exchanger, residual sensible heat in the low-pressure turbine exhaust stream is used for initial preheating of the cold liquid from the motive fluid pump, thus increasing the cycle efficiency. The cooled regenerator exhaust vapor is then condensed, using water-cooled, shell-and-tube condenser. From the condenser, the liquid working fluid is pumped to high pressure and returned to the regenerator to close the cycle. The selection of the working fluid has great implications for the performance of a binary plant. While there are many choices available for working fluids, there are also many constraints on that selection that relate to the thermodynamic properties of the fluids as well as considerations of health, safety, and environmental impact (Dippipo, 2007). The plant uses isopentane as the working (binary) fluid but the results can be compared with those using other organic fluids such as n-pentane, isobutane, n-butane, etc.

The geothermal brine leaving the vaporizer is directed to the reinjection well where it is reinjected back into the ground.

The following plant operating parameters are used for the thermal design:

$\Delta T_{pp,vap} = 10$ Temperature difference at vaporizer pinch point-[°C]

$\Delta T_{pp,regen} = 10$ Temperature difference at regenerator pinch point-[°C]

$\Delta T_{pp,cond} = 10$ Temperature difference at condenser pinch point-[°C]

$T_{16} = 70$ Brine temperature at vaporizer outlet-[°C]

$T_{cw,low} = 20$ Cooling water temperature at inlet of condenser-[°C]

$T_{cw,high} = 30$ Cooling water temperature at outlet of condenser-[°C]

$\eta_{turb} = 80$ Turbine isentropic efficiency- [%]

$\eta_{pump} = 50$ Pump isentropic efficiency - [%]

The heat amount delivered from the geothermal water is determined by:

$$\dot{Q}_{in,ORC} = \dot{m}_3 . C_{p,Water} . (T_3 - T_{16}) \quad (10)$$

Where $\dot{m}_3$ is the mass flow of geothermal brine in the vaporizer, $C_{P,water}$ is specific heat of water at constant pressure and T_3 and T_{16} are the temperature values of geothermal brine at inlet and outlet of vaporizer, respectively.

The mass flow of isopentane in binary cycle $(\dot{m}_{isopentane})$ can be calculated as:

$$\dot{m}_{isopentane} = \frac{\dot{Q}_{in,ORC}}{(h_{11} - h_9)} \quad (11)$$

h_9 and h_{11} are the enthalpy values of isopentane at inlet and outlet of vaporizer, respectively.

The power production of turbine in ORC is:

$$\dot{W}_{turb,ORC} = \dot{m}_{isopentane}(h_{11} - h_{12}) \quad (12)$$

$$\eta_{turb,ORC} = \frac{h_{11} - h_{12}}{h_{11} - h_{12s}} \quad (13)$$

h_{11} is the enthalpy of the ORC turbine inlet and h_{12} and h_{12s} are the enthalpy values at the turbine exit state for actual and isentropic processes, respectively.

The heat exchange taking place in the regenerator can be determined as:

$$\dot{Q}_{regen} = \dot{m}_{isopentane}(h_{12} - h_{13}) \quad (14)$$

The heat rejected in condenser $\dot{Q}_{cond,ORC}$, is:

$$\dot{Q}_{cond,ORC} = \dot{m}_{isopentane}(h_{13} - h_4) \quad (15)$$

The motive pump power is defined as:

$$\dot{W}_{pump} = \dot{W}_{pump,isentropic} . \eta_{pump} \quad (16)$$

$$\dot{W}_{pump,isentropic} = v_{isopentane} . (P_{high} - P_{low}) \times 100 \quad (17)$$

The pressure on high pressure side (P_{high}) can be found as:

$$P_{High} = P(isopentane, x = 0, T = T_{10}) \quad (18)$$

Where x is the quality of isopentane and T_{10} is the boiling temperature of isopentane in vaporizer.

The pressure on low pressure side (P_{low}) is defined as:

$$P_{low} = P(isopentane, x = 0, T = T_{cw,high}) \quad (19)$$

$T_{cw,High}$ is the temperature of cooling water at outlet of condenser.

The first law efficiency of ORC is determined by:

$$\eta_{1Law,ORC} = \frac{\dot{W}_{net,ORC}}{\dot{Q}_{in,ORC}} \quad (20)$$

2.3 Single flash Combined with ORC

The first law efficiency of combined system is defined as:

$$\eta_{1law,combined} = \frac{\dot{W}_{combined}}{\dot{Q}_{in,combined}} \quad \textbf{(21)}$$

3. EXERGY ANALYSIS

The concept of exergy (sometimes called available work) relates to the maximum work (or power) output that could theoretically be obtained from any system relative to given surroundings. We often refer to the state of the surroundings as the dead state because when fluids are in thermodynamic equilibrium with the surroundings there is no potential for doing work, and the fluid may be considered "dead."(R.Dippo, 2004).

Disregarding kinetic and potential energy changes, the specific flow exergy of geothermal fluid at any state (plant location) can be calculated from

$$E = \dot{m}(h - h_0 - T_0(s - s_0)) \quad (22)$$

Where T_0 is the environment (dead state) temperature, *h* and *s* are the enthalpy and the entropy of the geothermal fluid at the specified state, and h_0 and s_0 are the corresponding properties at the restricted dead state (Kanoglu, 2002).

3.1 Exergetic Efficiencies

The first law efficiency alone is not a realistic measure of the performance of plant, thus the second law of efficiency needs to be defined. There are various definitions for the exergetic efficiency in the literature. The definition is used in this study is called rational efficiency and defined by

Kotas (1995) as the ratio of the exergy recovered (or the desired product) to the exergy supplied to the system or process:

$$\eta_e = \frac{E_{desired}}{E_{input}} \quad (23)$$

$$E_{input} = E_{output} + E_{destroyed} \quad (24)$$

$$E_{output} = E_{desired} + E_{Waste} \quad (25)$$

Where $E_{desired}$ = Sum of desired exergy outputs (net positive work by the system);

$E_{destroyed}$ = Exergy rate lost in the system as a result of irreversibilities;

E_{waste} = Exergy exiting the system which still has capacity to do work.

When this concept is applied to a power plant as a whole, the overall exergetic efficiency reduces to a very simple formula, namely, the ratio of the net power output to the exergy of the motive fluid serving as the energy source for the plant (DiPippo and Marcille, 1984). Thus; the exergetic efficiency of the single flash cycle based on the two phase fluid exergy input to the plant can be calculated as:

$$\eta_{\varepsilon\_overall\_SF} = \frac{\dot{W}_{net,SF}}{E_1} \quad (26)$$

Where the $\dot{W}_{net,SF}$ is the net power output of single flash cycle and E_1 is the exergy rate of two phase geothermal fluid in state 1.

The overall exergetic efficiency for the combined system will be calculated as:

$$\eta_{\varepsilon,Combined} = \frac{\dot{W}_{Combined}}{E_1} \quad (27)$$

The exergetic efficiency of a turbine is defined as a measure of how well the stream exergy of the fluid is converted into actual turbine work output. Applying this to the single flash condensing turbine, we obtain

$$\eta_{\varepsilon,SF-Turb} = \frac{\dot{W}_{SF,Turb}}{E_2 - E_4} \quad (28)$$

The difference between the numerator and denominator in Eq. (28) is simply the exergy destruction in the turbine.

$$I_{SF-Turb} = (E_2 - E_4) - \dot{W}_{SF,Turb} \quad (29)$$

Vaporizer, regenerator and condensers in the plant are essentially heat exchangers designed to perform different tasks. The exergetic efficiency of a heat exchanger may be measured by the increase in the exergy of the cold stream divided by the decrease in the exergy of the hot stream (Wark, 1995). Applying this definition to the condenser in single flash cycle, we obtain

$$\eta_{\varepsilon\_cond,SF} = \frac{E_8 - E_7}{E_4 - E_5} \quad (30)$$

Where the exergy rates are given in Table 1. The difference between the numerator and denominator in Eq. (30) is the exergy destruction in the condenser. That is,

$$I_{cond,sf} = (E_4 - E_5) - (E_8 - E_7) \quad (31)$$

However the exergy drop of the working fluid across the condenser can be expressed as the exergy destruction in the condenser. That is, the exergy gained by the cooling water is not considered (Kanoglu,2002).

The exergetic efficiency of the heat exchangers in the binary cycle is calculated as below:

$$\eta_{\varepsilon,vap} = \frac{E_{11} - E_9}{E_3 - E_{16}} \quad (32)$$

$$\eta_{\varepsilon,regen} = \frac{E_{12} - E_{13}}{E_9 - E_{15}} \quad (33)$$

$$\eta_{\varepsilon,Condens} = \frac{E_{C,High} - E_{C,Low}}{E_{13} - E_{14}} \quad (34)$$

The exergetic efficiency and exergy destruction for the motive pump in binary cycle are calculated from the following relations:

$$\eta_{\varepsilon,Pump} = \frac{E_{15} - E_{14}}{\dot{W}_{pump}} \quad (35)$$

$$I_{pump} = \dot{W}_{pump} - (E_{15} - E_{14}) \quad (36)$$

The exergy balance for the single flash cycle, ORC and combined system can be written as:

$$E_1 + E_7 = \dot{W}_{net,SF} + E_3 + E_5 + E_8 + \Sigma I_{SF,processes} \quad (37)$$

$$E_3 + E_{C,Low} = E_{16} + \dot{W}_{ORC} + E_{C,High} + \Sigma I_{ORC} \quad (38)$$

$$E_1 + E_7 + E_{C,law} = \dot{W}_{combined} + E_5 + E_{16} + E_8 + E_{C,high} + \Sigma I_{Combined} \quad (39)$$

RESULTS AND DISCUSSION

The thermodynamic design and exergy analyses of the proposed system have been established in EES software. In Table 1, temperature, pressure, mass flow, enthalpy, entropy and exergy rate data for geothermal fluid, working fluid, and cooling water are given according to their state numbers specified in Fig. 1.

Table 1. The Exergy Rates and Other Properties at Various Plant Locations. State Numbers Refer to Fig 1.

STATE No.	FLUID	PHASE	$\dot{m}$ (kg/s)	T $(^\circ C)$	P (bar)	h (kJ/kg)	s $(kJ/kg-K)$	***E*** ***(kW)***
0	geothermal	-	-	10	0.75	42	0.151	0
0'	isopentane	-	-	10	0.75	-383	-1.8	0
1	geothermal	two phase	56	155.5	5.5	954	2.62	12351
2	geothermal	steam	8.33	151.8	5	2749	6.79	6815
3	geothermal	liquid	47.67	151.8	5	640	1.86	5441
4	geothermal	steam	8.33	45.8	0.1	2249	6.82	1996
5	geothermal	liquid	8.33	45.8	0.1	192	0.64	73
6	Geotherma;	liquid	8.33	46.2	1.1	193.5	0.65	75
7	water	liquid	410	32.8	2	137	0.47	1493
8	water	liquid	410	42.8	2	179	0.61	3023
9	isopentane	liquid	36.26	49.85	7.36	-291.7	-1.5	255
10	isopentane	liquid	36.26	100.8	7.36	-160.6	-1.1	1279
11	isopentane	vapor	36.26	100.8	7.36	111.7	-0.4	3547
12	isopentane	vapor	36.26	55.85	1.09	55.85	-0.36	1087
13	isopentane	vapor	36.26	40.7	1.09	21	-0.47	930
14	isopentane	liquid	36.26	30	1.09	-338.7	-1.652	58.5
15	isopentane	liquid	36.26	30.7	7.36	-336.1	-1.648	98
16	geothermal	liquid	47.67	70	5	293	0.955	1111
C,high	water	liquid	255	30	-	125.7	0.44	857.5
C,low	water	liquid	255	20	-	83.8	0.29	202.3

State 0 and 0' are the restricted dead states for the geothermal and working fluids, respectively. They correspond to an environment temperature of 10°C and an atmospheric pressure of 0.75 bar-a, which are the annual mean values measured at the plant area. For geothermal fluid, the thermodynamic properties of water are used. The thermodynamic properties of working fluid, isopentane, are obtained from EES software. The temperature of brine at vaporizer outlet is restricted by the silica amorphous saturation point and should not be lower than 70°C to prevent the silica deposition inside the vaporizer. The data listed in Table 1 was created by the thermodynamic design model when running it in EES software.

The summarized results of the exergy analysis for the single flash cycle are presented in Table 2. The results show that the total available exergy in two phase geothermal fluid produced by the well is 12354 kW. Of this available exergy, 5441 kW exergy exists in the brine which is disposed of at the separator; 6815 kW is contained in the steam and connected to the condensing turbine of the single flash cycle. The gross work developed by the turbine is 4166 kW. The exergetic efficiency of the turbine is defined as the ratio of the desired exergy output to the input exergy of the turbine and calculated to be 61%. The total exergy in the steam exhausted into the condenser is 2648 kW. The exergy destruction through the condenser is defined as the exergy drop of the steam across the condenser and was found to be 2576 kW. This value is 38% of the steam exergy input to the single flash cycle and is discarded to the cooling water.

The overall exergy efficiency of the single flash cycle was found to be 33% with reference to the total exergy from the connected well. For comparison, the overall energy efficiency for the single flash cycle was found to be 7.5%. The large difference in the efficiencies shows that most of the energy received at the wells, exits the plant while still containing substantial exergy. The turbines showed high exergy efficiencies because most of the exergy is exhausted into the condensers and not consumed or destroyed. This means that an improvement of the turbines will enable them to extract more work from the fluids or alternatively, devise other ways of using the exergy from the fluids exiting the system (Kwambai, 2005).The greatest exergy losses occur in the condensers where most of the exergy is rejected and destroyed.

The exergy in the brine is significantly high (44%) and is the main objective of this study to be used to do more work with a binary system.

The exergy content of the brine which is led to the binary plant vaporizer is 5438 kW. Of this available exergy, 1484 kW exits the vaporizer to re-inject to the re-injection well. The outlet temperature of the brine in the vaporizer is restricted due to amorphous silica deposition temperature and is set at 70°C. This exergy with the temperature of 70°C is significant and can be used by directly in open systems.

The exergy analysis results for the binary ORC plant are summarized in table 3. The results show that the second Law efficiency of the ORC plant is calculated to be 33% based on the exergy input to the plant and 45.3% based on the exergy input to the isopentane Rankine cycles. This means that more than 27% of the exergy of the brine is discarded as waste heat. The exergy input to the isopentane Rankine cycle is 3954 kW. Of this exergy, 1792 kW is converted to power and 2162 kW is destroyed in the plant. The causes of exergy destruction in the ORC plant include vaporizer-regenerator losses, turbine-pump losses and the exergy of isopentane lost in the condenser. The main exergy destruction takes place in vaporizer and condenser.

The summary of the exergy analysis for combined system of single flash cycle with ORC binary plant is presented in table 4. The results show that the desired exergy output from combined system in form of electricity is 5840 kW which has a 44% increase in comparison with single flash cycle. 1557 kW of the exergy in the form of brine is discarded as waste heat which is 72% less than that from single flash cycle. The exergy destruction in the combined plant was found to be 4885 kW. The exergy flow diagram of combined system includes the exergy destruction percentages in the components, is shown in figure 2. The substantial exergy loss takes place in condensers with the percentage of 25.7%. The first law and exergy efficiencies of the single flash cycle, ORC and combined system and the productivity of the geothermal fluids in the various cases are presented in table 5. The first law efficiency of combined system is 11% and has an increase of 3.5% in comparison with single flash cycle, where the second law efficiency of combined system shows the 14.7% increase compared to single flash. The productivity of the geothermal fluid produced by the well is increased from 20 to 29 kWh/Ton in combined system related to single flash.

Table 2. Summary Results for the Exergy Analysis of the Single Flash Cycle.

Process or System	Exergy Input (kW)	Desired Exergy output (kW)	Exergy Destroyed (kW)	Waste Exergy Output (kW)	Exergetic Efficiency (%)
Wellhead & separator	12351	6815	99	5441	56
Turbine	6814	4166	653	1996	61
Condenser	1996	1530	1923	72	79
pump	73	2	0.3	75	72.2
Overall single Flash cycle(1)	12351	4059	2779	5516	33
Overall single Flash Cycle(2)	6815	4059	2681	75	59.6

(1) based on the exergy of two phase fluid
(2) based on the exergy of the steam

Table 3. Summary Results for the Exergy Analysis of the ORC.

component	Exergy Destruction (kW)	Efficiency (%)
Vaporizer	1036	76
Turbine	435	59
Regenerator	35	86
Condenser	871	75
Pump	35	53.5
Overall ORC (1)	3526	33
Overall ORC (2)	2412	42.7

(1) based on the exergy available in Brine
(2) based on the exergy input to ORC

Table 4. Summary Results for the Exergy Analysis of the Combined System.

Description	Value
Exergy Input To the Sysytem (kW)	12351
Desired Exergy Output (kW)	5863
Waste Exergy Or Undesired Output (kW)	1189
Exergy Destruction (kW)	5299
(%) Exergetic Efficiency	47.6

Table 5. First Law and Exergetic Efficiencies of the Single Flash Cycle, ORC and Combined System Plants.

System	First law Efficiency (%)	Exergetic efficiency (%)
Single Flash	19.5	59.6
ORC	12.5	33
Combined	15.5	47.7

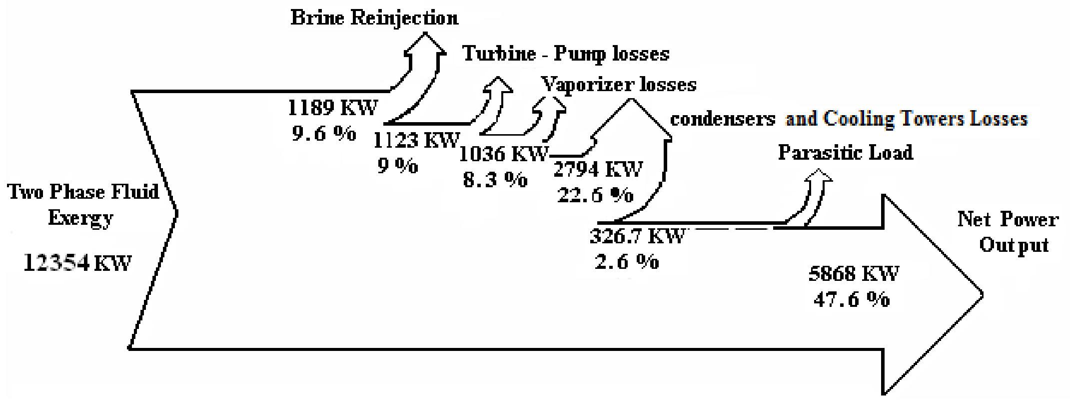

Figure 2: Exergy flow diagram of combined system, given as the percentages of two phase fluid exergy input

CONCLUSION

A significant volume of energy will be discarded as heat waste in form of re-injected brine by using a single flash cycle in Meshkin shahr geothermal power plant in Iran. Thermodynamic modeling of a combined system of single flash cycle with binary ORC plant and exergy analysis of the combined system was carried out and the locations and quantities of exergy losses, wastes and destructions in the different processes of the plant were pinpointed. In addition, the exergy analysis enabled the degree of thermodynamic imperfections for the processes to be determined. EES software was used for developing and analyzing of mathematical models of energy and exergy flows.

From the results, the following conclusions have been drawn:

1. The total exergy available from production well was calculated to be 12351 kW. The total exergy contained in the steam phase and connected to the single flash cycle was found to be 6814 kW and the exergy contained in the waste brine is significantly high to be discarded and can be utilized to obtain more work in a binary plant.

2. The overall exergy efficiency for the single flash cycle plant is 33% and the overall energy efficiency is 7.5%, in both cases with respect to the fluid from the connected wells. The productivity of the two phase fluid in single flash cycle was found to be 20 kWh/ton.

3. The overall exergy efficiency for the ORC binary plant is 33% and the overall energy efficiency is 14.8%, in both cases based on the brine led to the Binary plant. The productivity of the brine in binary plant was found to be 10.5 kWh/ton.

4. The overall exergy efficiency for the combined system is 47.7% and the overall energy efficiency is 11%, in both cases with respect to the fluid from the connected well. The productivity of the two phase fluid in combined system was found to be 20 kWh/ton.

5. The desired exergy output from combined system in form of electricity is 5840 kW which has a 44% increase in comparison with single flash cycle.

6. Exergy losses occurred in: brine re-injection (1556 kW), turbines and pumps (558 kW) , vaporizer and regenerator (961 kW) and the condensers (3183 kW) with the percentages of 12.6%, 4.5%, 7.7% and 25.7%, respectively based on the exergy available in two phase fluid produced by the well. A substantial amount of exergy is exhausted to the condensers because of the big difference between the cooling water and operating fluid temperatures. The wasted brine needs to be investigated regarding the silica deposition issue but can further be utilized for direct uses such as hot spas and medicinal uses which can generate a lot of revenue.

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