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Energy and Exergy Analysis of Sabalan Pilot Single Flash Geothermal Power Plant

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ABSTRACT

Following the changes in the SUNA's policy from the implementation and ownership of renewable energies projects to hand over them to private sector, and due to the lack of experience and interest of domestic companies, it was necessary for SUNA to perform the construction of a geothermal pilot power plant to demonstrate the potential of the Sabalan geothermal reservoir and the capacity of drilled wells to gain experience and attract private investors. For this purpose, a domestic company, using the experience of Italian consultant in the form of an EPC contract, has started to design and execute a pilot plant of 5 MW at well NWS-6 on the site D of Meshgin Shahr geothermal field.

In this paper thermodynamic model and exergy analysis of the proposed pilot single flash cycle was developed, and pinpoint locations and quantities of exergy losses and wastes, and the optimum performance parameters of the main components were determined. The first and second law efficiencies and the productivity of geothermal fluid were estimated for single flash system.

The Engineering Equation Solver (EES) 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 the decision makers.

1. INTRODUCTION

Currently, the Sabalan Geothermal Field has seven production wells and 5 MW pilot demonstration plant is installed on well NWS-6 of this field. Data obtained from discharged well test shows that well NWS-6 produces two-phase geothermal fluid with mass flow rate of 56 kg/s and enthalpy of 1,150 KJ/Kg at well head pressure of 10.5 bar-a (EDC, 2013).

Single flash cycle with steam condensing turbine will be used as energy conversion system at the power plant. Using 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. The energy content of saturated water discarded from separator will be considered for use in the power plant energy demands points such as NCG vacuum system and plant heating system.

Therefore, a throttling valve and a secondary low-pressure separator is employed at stage 3 of Figure 1 to supply motive steam needed for NCG vacuum system.

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 secondary separator.

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.

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

$p_{sep} = 9$	Main Separator pressure - [bar-a]
$p_{sep_{Sec}} = 6$	Secondary Separator Pressure - [bar-a]
$pcond=0.07$	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:

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

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

Similarly, the fraction and flow rate of the outlet steam and brine in secondary separator can be as follows:

$$\dot{m}_3 x_6 = \dot{m}_{10} x_{10} + \dot{m}_{11} x_{11} \qquad (3)$$

$$\dot{m}_3 h_3 = \dot{m}_{10} h_{10} + \dot{m}_{11} h_{11} \qquad (4)$$

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:

$$\dot{W}_{turb} = \dot{m}_2 (h_2 - h_4) \qquad (5)$$

$$\eta_{turb} = \frac{h_2 - h_4}{h_2 - h_{4s}} \qquad (6)$$

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

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

$$\dot{Q}_{cond} = \dot{m}_2 (h_4 - h_5) \qquad (7)$$

The mass flow of cooling water is found by:

$$\dot{m}_{CW} = \frac{\dot{Q}_{cond}}{(h_8 - h_7)} \qquad (8)$$

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

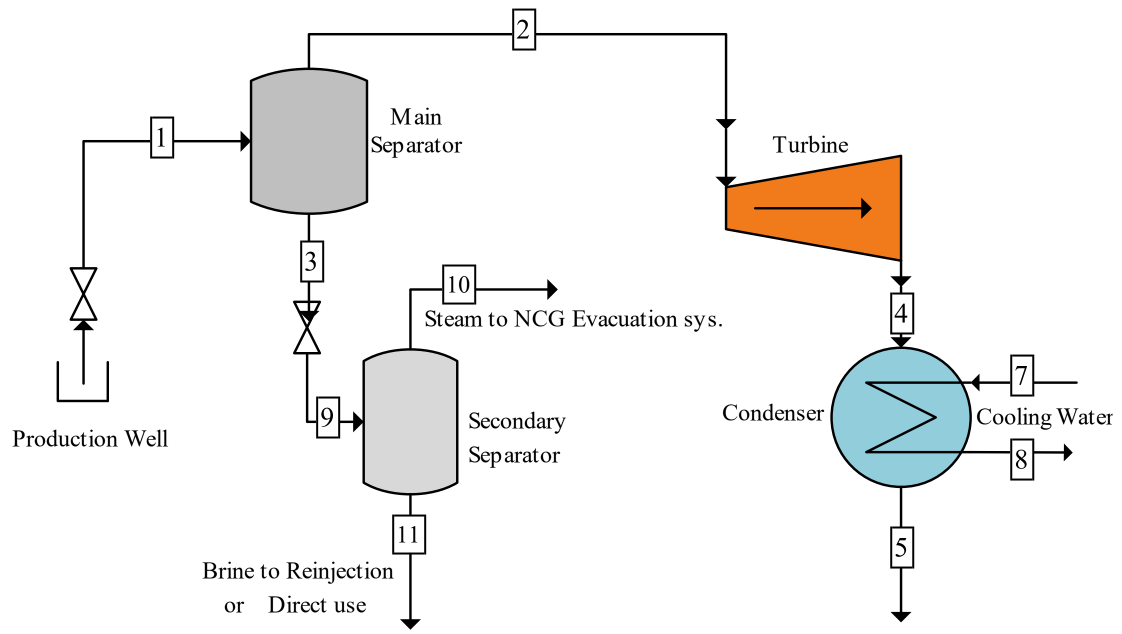

Figure 1: The process flow diagram of single flash cycle with secondary separator.

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

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

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.

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

$$E = \dot{m}(h - h_0 - T_0(s - s_0)) \quad (10)$$

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 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:

$$\eta_e = \frac{E_{desired}}{E_{input}} \quad (11)$$

$$E_{input} = E_{output} + E_{destroyed} \quad (12)$$

$$E_{output} = E_{desired} + E_{Waste} \quad (13)$$

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:

$$\eta_{\varepsilon\_overall} = \frac{\dot{W}_{net}}{E_1} \quad (14)$$

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

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

$$\eta_{\varepsilon,Turb} = \frac{\dot{W}_{Turb}}{E_2 - E_4} \quad (15)$$

The difference between the numerator and denominator in Eq. (15) is simply the exergy destruction in the turbine.

$$I_{Turb} = (E_2 - E_4) - \dot{W}_{Turb} \quad (16)$$

Condenser in the plant are essentially heat exchanger 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, we obtain

$$\eta_{\varepsilon\_cond} = \frac{E_8 - E_7}{E_4 - E_5} \quad (17)$$

Where the exergy rates are given in Table 1. The difference between the numerator and denominator in Eq. (17) is the exergy destruction in the condenser. That is,

$$I_{cond} = (E_4 - E_5) - (E_8 - E_7) \quad (18)$$

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 exergy balance for the single flash cycle can be written as:

$$E_1 + E_7 = \dot{W}_{net} + E_{11} + E_5 + E_8 + \Sigma I_{process} \quad (19)$$

4. RESULTS AND DISCUSSION

The thermodynamic design and exergy analyses of the 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.

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 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 16834 KW. Of this available exergy, 10097 KW is contained in the steam which is connected to the condensing turbine of the single flash cycle and 6737 KW exergy exists in the brine which is led to the secondary separator.

Secondary separator produces 1.5 kg/s steam with exergy of 1307 KW to be used as motive steam in NCG evacuation system and 5379 KW of exergy is disposed by 43.2 kg/s of brine which can be used in buildings heating system.

The gross work developed by the turbine is 5832 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 58%. The total exergy in the steam exhausted into the condenser is 3249 KW. The exergy destruction through the condenser is defined as the exergy drop of the steam across the condenser and was found to be 3184 KW.

The overall exergy efficiency of the single flash cycle was found to be 30% with reference to the total exergy from the connected well. For comparison, the overall energy efficiency (First Law 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.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 (40%) and it can be used as energy source of buildings heating system.

Table 2: 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 $(°C)$	P (bar)	h (kJ/kg)	s $(kJ/kg-K)$	***E*** ***(KW)***
0	geothermal	-	-	10	0.75	42	0.151	0
1	geothermal	two phases	56	175.5	9.01	1,150	2.1	16,834
2	geothermal	steam	11.23	175.5	9	2,772	6.62	10,097
3	geothermal	liquid	44.2	175.5	9	742.8	2.095	6,735
4	geothermal	steam	11.23	39	0.07	2162	6.62	3,249
5	geothermal	liquid	11.23	39	0.07	163.4	0.56	65
7	water	liquid	358	20	0.85	84	0.3	259
8	water	liquid	318	35	0.85	146.5	0.5	1,538
9	geothermal	two phases	44.2	158.8	6	743	2.1	6,733
10	geothermal	steam	1.55	158.8	6	125.7	6.76	1,307
11	geothermal	liquid	43.2	158.8	6	670	1.93	5,379

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 & separators	1,6834	11,404	50	5,379	70
Turbine	10,097	5,832	1,027	3,249	58
Condenser	3,249	1,280	1,904	65	40
Overall single Flash cycle[(1)]	16,180	5,032	5,769	5,379	31
Overall single Flash Cycle[(2)]	10,097	5,032	5,000	65	50

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

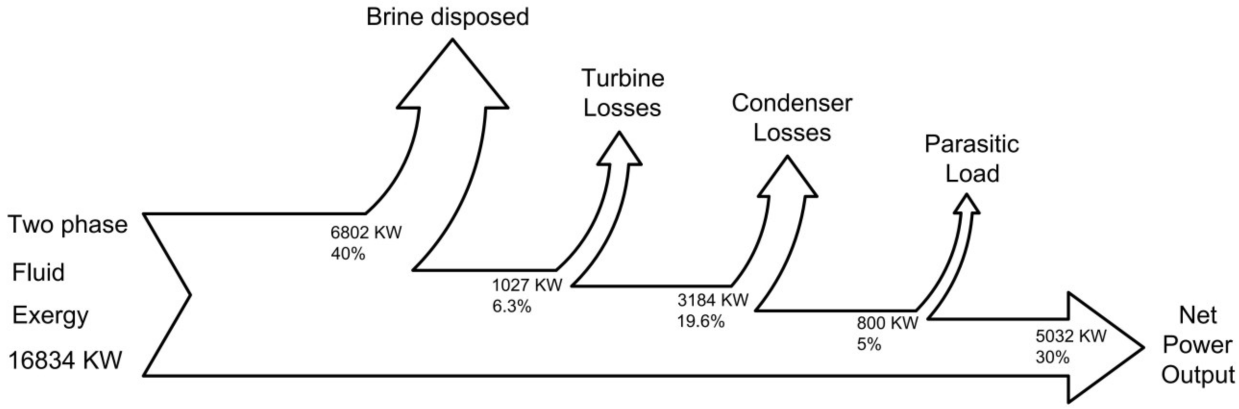

Figure 2: Exergy flow diagram of system, given as the percentages of two phase fluid exergy input.

5. CONCLUSION

Thermodynamic modeling and exergy analysis of a single flash cycle was carried out and the locations and quantities of exergy losses, wastes and destructions in the different states of the cycle were pinpointed. In addition, the exergy analysis enabled the degree of thermodynamic imperfections for the processes to be determined. EES 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 16,834 KW. The total exergy contained in the steam phase and connected to the single flash cycle was found to be 10,097 KW and the exergy contained in the waste brine is significantly high to be discarded and can be re-flashed in lower pressure to produce the motive steam of NCG evacuation system and the disposed brine can be utilized for cascading purposes.

2. The overall exergy efficiency for the single flash cycle plant is 30% and the overall energy efficiency is 7.8%, in both cases with respect to the fluid from the connected wells.

3. The overall exergy and energy efficiency for the cycle with respect to the steam content of the fluid is 50% and 16.5%, respectively.

4. The desired exergy output from the system in form of electricity is 5,032 KW which has a 4% increase in comparison with single flash cycle without secondary separation.

Exergy losses occurred in: disposed brine (5,379KW), turbines and pumps (1,200 KW) and the condensers (3,184 KW) with the percentages of 33.2%, 7.5% and 19.7%, 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 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 buildings heating system, hot spas and medicinal uses which can contribute to the local community from the economy and social acceptance point of view.

REFERENCES

DiPippo, R., Marcille, D.F: Exergy Analysis of Geothermal Power Plants. *Geothermal Resources Council Transactions* 8, (1984), 47–52.

Kanoglu, M.: Exergy Analysis of a Dual-level Binary Geothermal Power Plant, *Geothermics*, 31, (2002), 709-724.

Kotas, T.J.: The Exergy Method of Thermal Plant Analysis. *Anchor Brendon Ltd, Tiptree, Essex, Great Britain*, (1984), 293 pp.

Kwambai, C.B.: Exergy analysis of olkaria I power plant, Kenya. Report 5 in: *Geothermal Training in Iceland 2005*. UNU-GTP, Iceland, (2005).

EDC (Energy Development Corporation). : Discharge Testing of NWS6D Evaluation Report (2013). 25 pp.

Wark, K.J.: Advanced Thermodynamics for Engineers. *McGraw-Hill, New York*, (1995).