Proceedings World Geothermal Congress 2020+1
Reykjavik, Iceland, April - October 2021

Single and Double-flash Cycle Design of Menengai Prospect Geothermal Power plant, Nakuru- Kenya

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Keywords: Flash Cycle, Thermodynamics Laws, T_s Diagram, Optimization

ABSTRACT

This study evaluated the possible power output that can be obtained from a configuration of 15 production wells and one reinjection well in Menengai geothermal field by installing a single flash cycle power plant. Once the optimum conditions of the single flash model were determined, the same parameters were used as input for the double flash cycle design to investigate the possibility of obtaining a higher output compared to the single flash model. Engineering Equation Solver (EES) was used to solve the relevant mathematical equations.

Obtained Results confirmed that the single flash model gave more output of 113 MW compared to the double flash that produced 108 MW. Reinjection temperature for both models of 175^{0}C and 168^{0}C show minimal likelihood of there being heavy precipitate of polymer silicate in the brine line and reinjection well. Additionally, reinjection fluid mass for the double flash cycle is about 3 kg/s compared to 160 kg/s for the single flash cycle, a drastic decrease.

1. INTRODUCTION

Electric energy generation and distribution in Kenya falls exclusively under the domain of the Ministry of Energy. Generation in Kenya is on an upward trend increasing to 10,205 GWh in 2016/17 from 9,817 GWh the previous year. This growth is attributed to positive expansion in the commercial and industrial electricity consumption. Installed capacity in Kenya currently stands at 2,234.83MW (as at 2017) comprised of hydro-power at 36%, thermal diesel engines 31%, geothermal 29.1%, oil & gas 2.4%, while others like wind, solar and cogeneration comprise about 1.1%.

Geothermal energy utilization is commonly divided into two categories, i.e., electric production and direct application. 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 are determined by the geothermal system from which the fluid originates [1], [2], [3] and [4].

Kenya enacted the first Geothermal Resource Act of 1982 and which was later revised in the year 2012. The Act regulates access to and exploitation of geothermal resources for power generation. The license confers upon the licensee the right to explore, drill, extract and carry out all activities that are reasonably necessary for operations and maintenance of geothermal systems[5].

Kenya Electricity Generating Company Ltd (KENGEN) became the first company that actively began exploration of prospective sites in 1960 and Fig. 1 below shows the most promising sites it delineated including Menengai field highlighted in yellow. It then concentrated its activities of geothermal exploitation to Olkaria field in Fig.1, where installed capacity stands at 652MW. Kenya is currently ranked 9th globally in installed capacity of geothermal energy and thus geothermal has become the base-power load supply to the national grid [6] .

Menengai geothermal project is an undertaking spearheaded by the Geothermal Development Company (GDC), a government parastatal, whose sole mandate is to promote rapid development of geothermal resources in Kenya through surface exploration and drilling for steam and avail the steam to power plant developers for electricity generation [7].

GDC was able to drill production wells and one hot water reinjection well in Menengai with the expected total output of 105MW. The power plants are yet to be built thus work presented in this paper outlines the design of a single flash power plant system, then evaluate the probable performance of adding a double flash cycle plant and comparing the thermal efficiency of the two systems.

The Engineering Equation Solver (EES) was used to solve the mathematical equations

2. THEORETICAL ANALYSIS

First law of thermodynamics states that energy can neither be created nor destroyed. The second law states that conversion of energy is possible only if the total entropy increases. In order to analyze a power cycle based on specific working fluid, the thermodynamic properties of the fluid must be known [8]. Resource temperature imposes a primary constraint on a selection of appropriate conversion technology. Efficiency of conversion decreases significantly as fluid enthalpy decreases. If the entropy of a system is high, then low quality of a system is found. Steam turbines lose overall efficiency at lower temperatures due to low steam fraction.

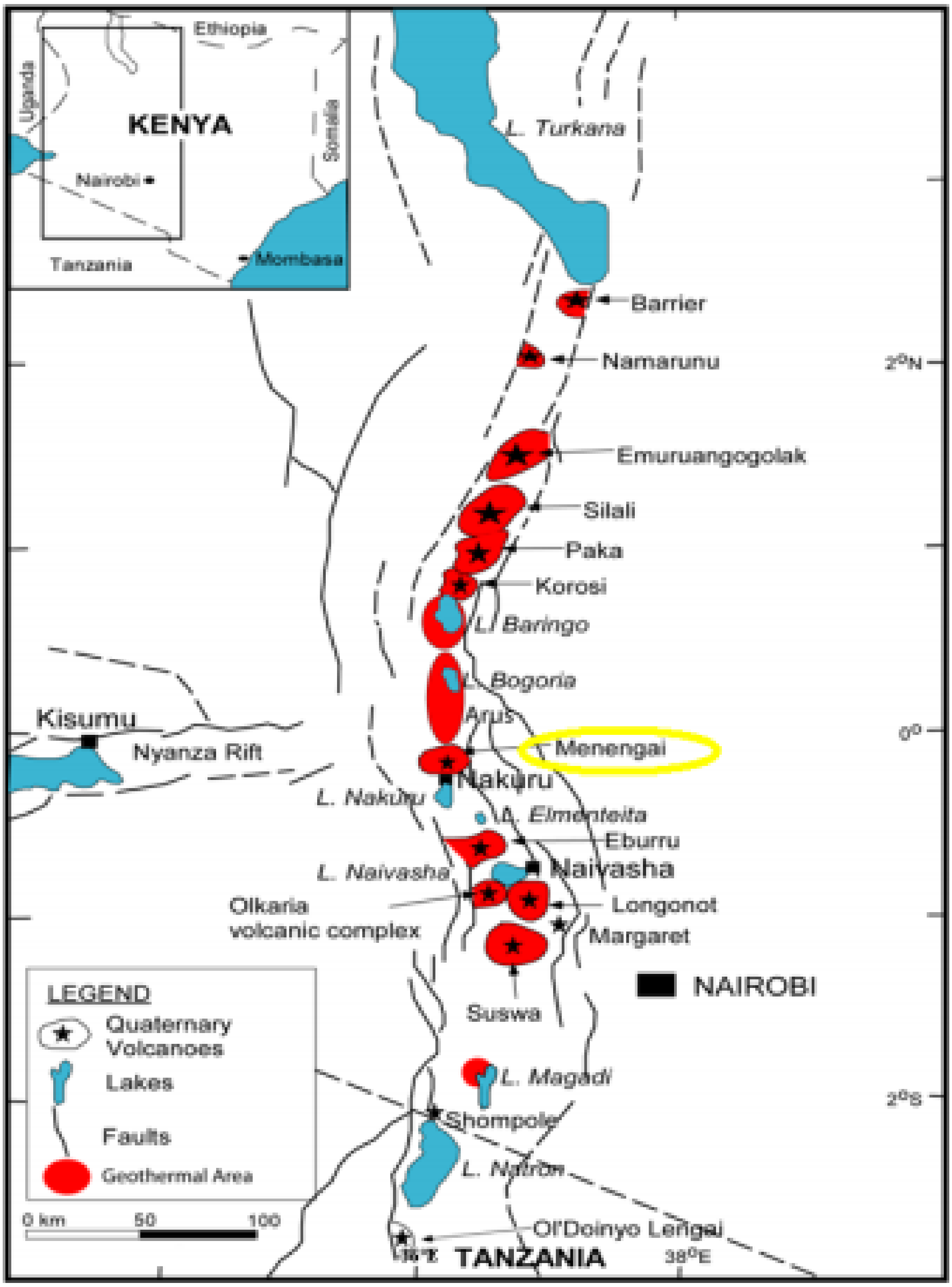
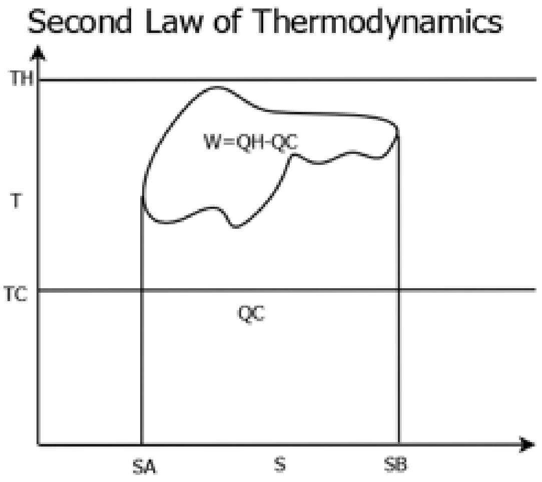


Figure 1: Location of Menengai in Kenya

Reducing the pressure of a mixture of hot water and steam causes it to boil (flash) increasing the mass of steam relative to the mixture. The higher the original enthalpy, and the lower the flash pressure, the greater the proportion of steam that results. This property of water increases the design of geothermal power plants. A range of pressure-temperature combinations at which both steam and liquid water exist can be found in the steam tables. At temperatures above saturation at isobaric conditions only superheated steam can exist and below saturation, only liquid water can exist.

Since plant optimization is desirable, the resource characteristics of temperature, size and phase conditions must be defined. Well output characteristics of flow and enthalpy versus WHP, fluid chemistry, environmental constraint, expected reservoir response from modeling, and financial analysis must be kept into consideration [9].



$$W = Q_H - Q_C \text{ (Work done)} \quad (1)$$

Q_H =Total heat transferred from hot reservoir

$=\int T dS$

Q_C =Heat transferred to cold reservoir

S = Entropy

T = Temperature

For maximum work, Q_C must be minimized using a condenser

Figure 2: Second Law of Thermodynamics

There are various geothermal power generation technologies employed such as direct dry steam, back pressure turbines, single, double, and multi flash, binary and hybrid binary & steam systems. Single flash and double flash systems are designed to have the steam from the turbine discharge to a condensing chamber that is maintained at a very low absolute pressure, typically about 0.06 bar for maximum power transfer. Exhaust steam must be condensed since an impractical amount of work would be required to pump the fluid from the low-pressure conditions to the liquid state[10]. For the double flash system, more generation is compared to single flash from the same resources. This process biggest limitation is fluid chemistry as a concentrated condensate can lead to scaling problems. Mitigation of scaling can be done by acid like sulfuric acid dosing prior to reinjection[11].

Four basic processes are considered when analyzing geothermal power plants;

- An isentropic process, where the entropy s is constant. This involves expansion in turbines as well as compressors and pumps. Work is delivered or consumed ideally with no losses to the environment or fluid.
- An isenthalpic process, where enthalpy h is constant. Here, no work or heat is delivered or consumed from the environment and therefore the energy of the fluid remains constant, such as found at expansion in throttling valve.
- An isobaric process, where the pressure p is constant. This is an ideal assumption where no pressure changes take place in the process.
- Heat exchange, where heat but not work is transferred to or from the fluid. This involves a change in both enthalpy and entropy, but in most cases the pressure is constant.

There are two types of balances that must be fulfilled for each component in the flash cycles Fig.2. Firstly, mass balance for a steady state system requires all sum of all mass flows $\dot{m}$, given in kg/s, into a component must be equal to the mass flow out of the component. Second is the energy balance where sum of energy flow into a component must be equal to the energy flow out of it. The flow of energy can be in the form of energy of working fluid denoted by $\dot{m}h$, work performed or consumed, W, and heat flowing into or from a component, Q.

3. METHODOLOGY

3.1 Work Segmentation

Well data was first obtained, sorted and organized from daily field monitoring reports. Steam gathering system design parameters were obtained from status reports from the contractor to the project engineer. The schematics and layouts of the proposed flash cycles components were drawn for the system. Equations for analysis of individual components were coded in EES. By manipulating the parameters in EES using available data, output was evaluated. Where data were missing, reasonable estimations of design values were used.

3.2 Steam field description

Menengai geothermal field employed a combination of satellite and individual configuration of pipeline design, Fig. 3. Most wells were clustered together based on topography and well output parameters for a single separator while other wells had individual separators in situ. The steam pipeline also increases with diameter as more lines merge into one another to form two steam pipelines flowing to a single header bar where each turbine will draw steam from. Brine flows by gravity from each separator station to the injection well.

Well data has been summarized in the table below:

Table 1: Well Data for Menengai power project

Well ID	Separator	WHP(Bar)	TOTAL MASS FLOW (T/HR)	TOTAL MASS FLOW (kg/s)	WATER FLOW (T/HR)	ENTHALPY (KJ/KG)	STEAM FLOW (T/HR)	STEAM FLOW (kg/s)
1		11.49	220.1333	61.14815	128.98	1352.5286	74.37667	20.66019
2	1	13.9	200.5767	55.71574	21.01	2438	170.9733	47.49259
3		6.46	166.91	46.36389	125.13	983	27.16	7.544444
4		10.31	66.62	18.50556	0	2674.99	64.25	17.84722
5		10.45	28.99	8.052778	2.14	2508	25.67	7.130556
6	2	9.14	130.95	36.375	52.8	1764	69.84	19.4
7		20.11	48	13.33333	4.52	2463	41.46	11.51667
8		10.89	49.93	13.86944	17.28	1894	29.68	8.244444
9	3	12.87	39.06	10.85	12.75	1939.5	24.1	6.694444
10		16.02	40.68	11.3	10.89	2071	27.6	7.666667
11	4	20.83	47.69	13.24722	0	2675	46	12.77778
12		5.25	94.28	26.18889	50.98	1457	35.92	9.977778
13	5	12.18	75.47	20.96389	3.82	2561	68.7	19.08333
14		9.42	88.53	24.59167	3.4	2588.19	81.74	22.70556
15	6	13.65	81.83	22.73056	54.11	1183	21.06	5.85

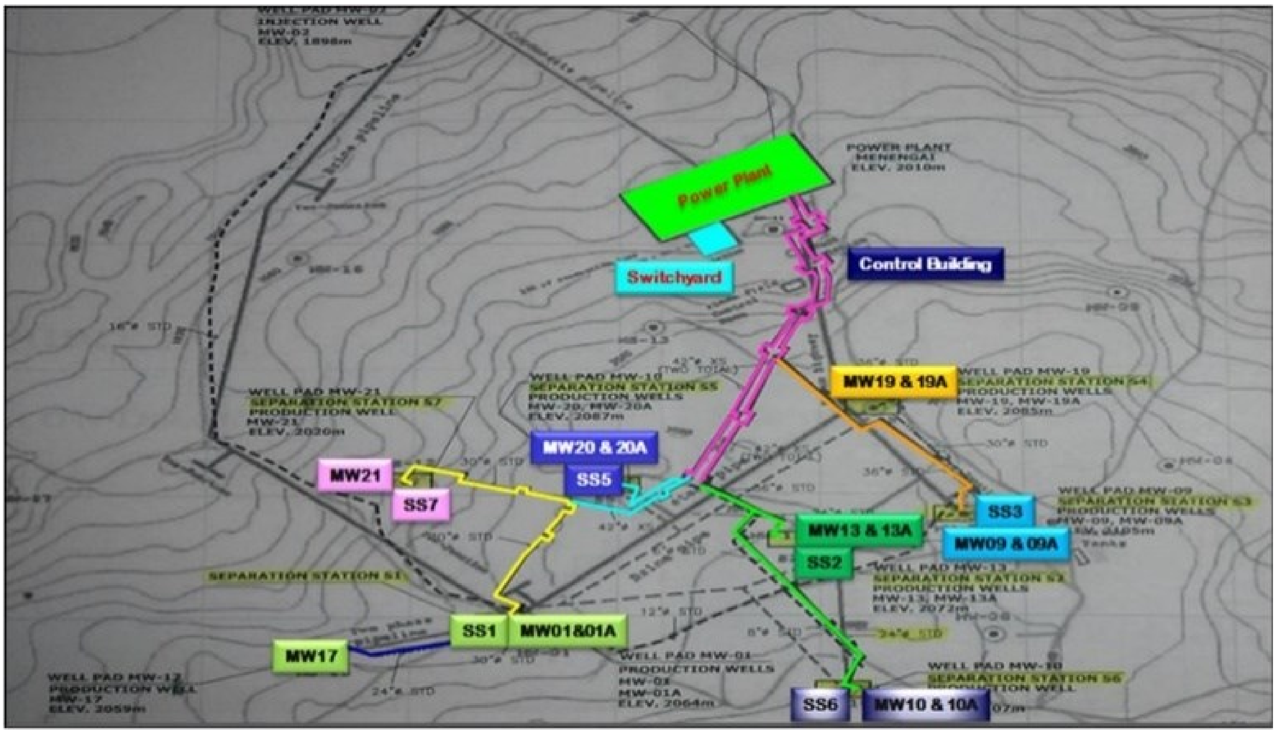


Figure 3: Steam field network in Menengai

4. ANALYSIS

4.1 Single flash power plant design

Single flash steam plants are usually the first systems installed at a developed liquid-dominated geothermal field like Menengai [12]. Turbines for single-flash units in this paper was considered to be rated at 35MW each whereby three turbines would generate 105MW.

For analysis of the single flash, the thermodynamic principals of energy conservation and the principle of mass conservation was applied. The process followed Temperature – Entropy (T – s) diagram below:

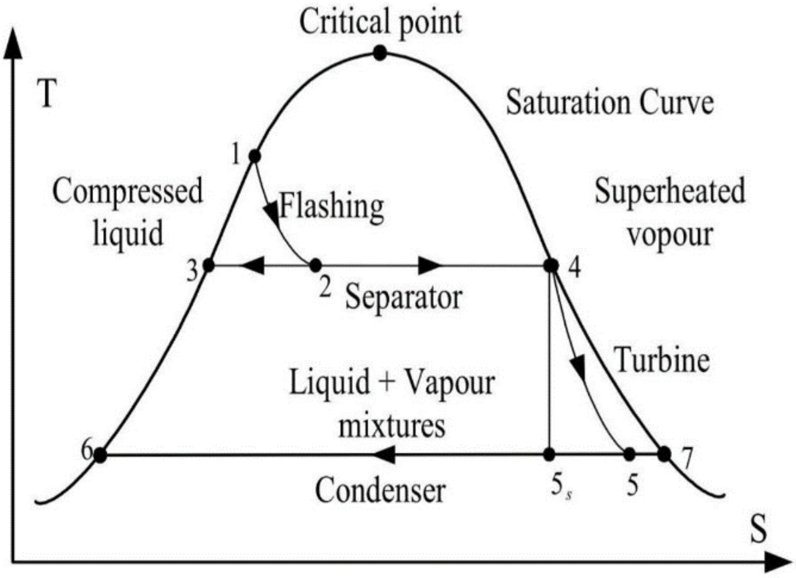


Figure 4: The T-s diagram for single flash cycle

4.1.1 The Wells

Well data properties have been listed in Table.1. Output of the wells was initially obtained using the James Lip pressure method, thus the flow rates and enthalpies were used as input in EES. Well temperatures were inferred from the steam tables. It is assumed isenthalpic flow as there are no heat losses from the well to the surroundings. All of the well output was consolidated by using weighted average method resulting to one inlet temperature, enthalpy, and mass flow rate.

4.1.2 Steam Separators

This equipment are cylindrical such that two-phase water is introduced tangentially at well head pressure having high velocity, and due to difference in density and centrifugal force, steam rises up to be collected in the steam pipe, while brine drops to be collected in the brine line.

Flashing process in Fig. 4 begins at the separator shown as 1 to 2, steam and liquid are separated at a specific pressure P[1] set at 9.0 bar. Dryness fraction was computed using the equation [13]:

$$x[1] = \frac{\dot{m}_2}{\dot{m}_1} = \frac{h_2 - h_3}{h_4 - h_3} \qquad (1)$$

In the separator, the enthalpy of the fluid from the well head is assumed to be the same by making the process isenthalpic from the well head. After the separator, the steam quality was set as one (1) since only steam powers the turbine. Also, the pressure into the steam gathering pipes P[2] is equated to the separator pressure P[1]. Brine mass flow rate from the separator flows at the separator pressure P[1] and has steam quality of zero (0) since it is in saturated liquid state, and is computed using equation:

$$\dot{m_6} = \dot{m}_1 - \dot{m}_2 \qquad (2)$$

4.1.3 Turbines

For the turbine calculations shown as 4 to 5 in Fig. 4; at the turbine entry, steam quality was set at one (1) and mass flow rate of steam was taken to be equal to mass flow rate after separator assuming negligible loss of mass flow through the dimistifier. Pressure loss was set at 3.0bar through the steam gathering pipes before being admitted to the turbine. Mass flow rate from the turbine is assumed to be equal to the steam flow rate from the separator as there is no mass loss in the steam expansion process. Similarly, in an ideal turbine, we consider the process to be isenthalpic where entropy is constant. However, ideal conditions are never quite achievable hence an efficiency factor η_t is empirically set as 0.85, that is 85% turbine efficiency [14].

Turbine efficiency is calculated as:

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{s4}} \qquad (3)$$

Work done by the turbine is given by;

$$\dot{W}_T = \dot{m}_3 \times (h[3] - h[4]) \times \eta_t \qquad (4)$$

4.1.4 At the Condenser

A condenser is an improvement of the atmospheric exhaust design and is by far the most common plant concept employed for geothermal power generation [10]. Instead of steam being discharged to the atmosphere, it is discharged to a condensing chamber whose pressure is set to be as low as possible close to vacuum pressure, at 0.06bar [15]. Direct condensers are generally used for condensing the turbine exhaust steam. These incorporate banks of spray nozzles through which cooling water is passed in order to condense the steam. Effect of non-condensable gases (NCGs) was considered negligible hence steam ejector equipment was not designed for.

Mass balance was required from first principal of thermal dynamics whereby one unknown variable was the mass flow rate of cooling water from cooling tower. Temperatures and enthalpy were derived from the steam table at isobaric conditions of 0.06 bar. Solving eqn. 6. Simultaneously by substituting eqn. 5 inside, cooling tower mass flow rate to the condenser was obtained.

$$\dot{m}_5 = \dot{m}_2 + \dot{m}_{20} \tag{5}$$

$$[\dot{m}_2 * h_2] + [\dot{m}_{20} * \dot{h}_{20}] = [\dot{m}_2 + \dot{m}_{20}] * h_5 \tag{6}$$

The condenser was assumed to have a 3^0C temperature drop between mass flow rate in and out. Density of the condensate, σ, was extrapolated from the steam tables such that the volume flux Q can be computed;

$$\dot{Q}_5 = \left(\frac{\dot{m}_5}{\sigma}\right) * 3600 \tag{7}$$

Work done by the condenser was calculated from;

$$W_{cond} = \frac{\dot{m}_5 * (h_4 - h_5)}{1000} \tag{8}$$

4.1.5 Condenser Pump Work

Condensate is pumped to the cooling tower and assumptions for pump efficiency, 0.8 and pump motor efficiency of 0.95 were assumed. The expected head potential assigned by the pump manufacturer was assumed to be 43m. Equation 8 below was used;

$$W_{cond.pump} = \left(\frac{\dot{m}_5 * 9.81 * h_p}{\eta_{cp} * \eta_{cm}}\right) \Big/ 1000 \tag{9}$$

4.1.6 Cooling tower computations

Two types of cooling towers can be considered depending on the power plant being set up, the natural draught (convection) or mechanically forced draft (cooling fans). Hot water collected at the condenser is cooled and recycled to the condenser. Efficiency depends on the atmospheres wet bulb temperature.

Advantages of the mechanical draught are that they are reasonably cheap to build, the fan speeds may be altered to adjust for ambient conditions and turbine loads but the disadvantages are the power consumption of fans and high maintenance costs.

The natural draft cooling like the one found in Matsukawa, Japan has the advantage of low maintenance, and low power consumption cost but disadvantageous in terms of high capital cost of construction, unsightly tall tower and inflexible compared to mechanical draught towers[9].

At the cooling tower, there were mass flow rates from the turbine that needs cooling, mass flow rates of cooling water, air inlet and exhaust steam. When conducting the mass balance from thermodynamic first principals, two variables of mass flow rates existed and were solved simultaneously.

Conditions precedent were the approach temperature 5^0C, which is the difference between cooling water, relative humidity of the area ω_{24} at 0.522 (kg H_2O/kg dry air), average air temperature of 15^0C, density of air at 0^0C at 1.3kg/m^3, atmospheric pressure of 101325 (atm) and the elevation altitude of the power plant at 2010m. Humidity ratio air into and out of the cooling tower was derived from air property tables by recalling from the EES software.

Mass balance was obtained by;

$$\dot{m}_{21} + (\dot{m}_{24} * \omega_{24}) = \dot{m}_{22} + (\dot{m}_{25} * \omega_{25}) \tag{10}$$

For the energy balance, enthalpy of fluid in the cooling tower, and enthalpy of air was extrapolated from property tables inbuilt in EES.

$$(\dot{m}_{21} * \dot{h}_{21}) + (\dot{m}_{24} * \dot{h}_{24}) = (\dot{m}_{22} * \dot{h}_{22}) + (\dot{m}_{25} * \dot{h}_{25}) \tag{11}$$

Mass evaporated in the cooling tower is obtained by;

$$\dot{m}_{loss.evap} = \dot{m}_{21} - \dot{m}_{22} \tag{12}$$

Mass evaporated from cooling tower as a ration of inlet steam mass flow (75-80% normal);

$$MER = \left(\frac{\dot{m}_{loss.evap}}{\dot{m}_3}\right) * 100 \tag{13}$$

4.1.7 Cooling tower fan motor power calculation

Conditions precedent that were set include cooling tower fan efficiency, η_{CTF} 0.7, cooling tower fan motor efficiency, η_{CTM} 0.80, and pressure change across the cooling fan, ΔP_{CTF}, 328bar. The air volume at the set altitude of 2010m was derived from property tables inbuilt in EES.

Work done by the cooling tower fan was calculated by;

$$W_{CTF} = \frac{(\dot{m}_{24} * \Delta P_{CTF} * V_{CT})}{(\eta_{CTF} * \eta_{CTM})} \Big/ 1000 \quad (14)$$

4.1.8 Power losses due to other equipment

These values were pre-assumed;

W_{RWP}=800W, power for steam gathering system reinjection pumps

$W_{loss.transformers}$=155W, dependent on substation configuration

$W_{loss.others}$=500W, used for power station lighting, workshop, compressors, and related things

$W_{loss.NCG}$=75W, Non-Condensable Gas extraction system (usually in the range of 1.2-2MW for a 50MW plant)

4.1.9 Net power produced

$$W_{net} = W_T - W_{CP} - W_{CTF} - W_{loss.NCG} \quad (15)$$

$$W_{paras.load} = W_{CP} + W_{CTF} + W_{loss.NCG} \quad (16)$$

4.2 Double Flash Power Plant Design

In order to increase the cycle efficiency further, a secondary pressure step is introduced which utilizes the saturated water from the first separator. The pressure of the water is the same as the wellhead pressure and is then lowered in a throttling valve, generating a mixture of water and steam at a lower pressure level. The steam is then separated from the mixture and fed into a low-pressure turbine along with the steam from the high-pressure turbine outlet.

The plant is more complex, costlier, requires more maintenance, but the extra power output often justifies the installation of such plants. Since many aspects of a double flash plant are similar to a single flash plant, the format of analysis generally follows the same as that of single flash[12].

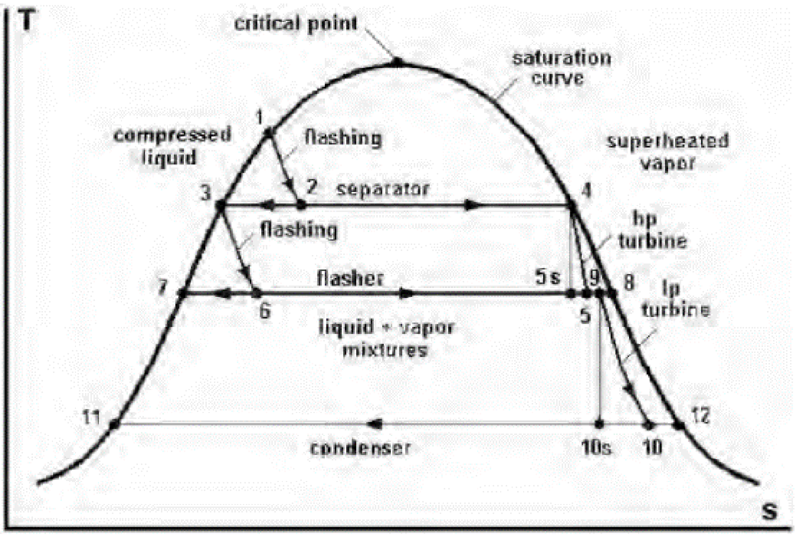


Figure 5: The T – s diagram for double flash cycle

With reference to Fig. 5. the two flash process 1-2 and 3-6, were analyzed in the same way as the flash process for the single flash plant.

Conditions precedent were the fluid enthalpy of 2036kJ/kg and mass flow rate of 225kg/s.

4.2.1 At the Well Head

The pressure was set at 12.0bar. Temperature, enthalpy of water and steam, and entropy of steam were derived from the steam tables integrated with EES.

The steam quality index was computed as;

$$x_3 = \frac{(h_t - hl_1)}{(hg_1 - hl_1)} \quad (17)$$

4.2.2 For the Turbine Computations

The pressure at the primary separator was set at 0.9 bar of the well head pressure. Process 4-9 in Fig.5 above occurs in the high-pressure turbine. The governing pressure in the high-pressure turbine was set at 6.0bar. At the low-pressure turbine, the pressure from the flasher was set at 0.1 bar. The same concept of computations was inputted in EES for these turbine processes as indicated in eqns. 1-4. Mass flow balance was computed obeying the first law of thermodynamics whereby all variables were known such as steam quality index shown in eqn. 1.

4.2.3 At the Condenser

Volume of water necessary to cool the exhaust steam into the condenser was obtained in eqn. 18

$$Q_{cooling} = \dot{m}_5 * (h_5 - h_6) \quad (18)$$

Temperature of cooling water in the condenser was set at 10^0C and cooled by 5^0C out of the condenser. From these temperatures, their respective water saturated enthalpies were derived from steam tables, and the mass flow rate of cooling water, m_c could now be solved simultaneously by plugging eqn. 18 into eqn. 19.

$$m_C * (h_{co} - h_{cin}) = Q_{cooling} \quad (19)$$

Surface area, A, was computed by setting the U-value, rate of transfer of heat (in watts) through one square meter of the cooling tower, divided by difference in temperature, as 2000 W/m^2.^{o}C.

4.2.4 Pumping of cooling water

Density of cooling water was derived from the steam tables, gravity constant kept at 9.82 m/s^2, and pump efficiency η set at 0.85.

$$W_{pump.cooling.water} = \frac{(\sigma_m * g * \Delta H_{cw})}{\eta_{pump}} \Big/ 1000 \quad (20)$$

4.2.4 Energy Balance

This conforms to 2nd law of thermodynamics to obtain the mass of air draught that passes through the cooling tower thus maintaining the optimal conditions. The equations used are replicate of eqns. 10 – 11.

4.2.4 NCG extraction

The most commonly used gas removal system is steam jet ejector, which removes the NCGs from the condenser and compress them to the atmospheric pressure with the expense of steam. An ejector is a type of vacuum pump or compressor. Since an ejector has no valves, rotors, pistons or other moving parts, it is a relatively low-cost component, is easy to operate and requires relatively little maintenance but consumes a considerable amount of steam.

A steam jet ejector operates on the venturi principle. The motive steam is expanded through the nozzle to the design suction pressure. The pressure energy of the steam is converted to velocity energy and on leaving the nozzle at high supersonic velocities the steam passes through the suction chamber and enters the converging diffuser or entrainment, as gas and associated water vapor. Because of the capacity of a single ejector is fixed by its dimensions, a single unit has practical limits on the total compression and throughout it can deliver.

Auxiliary power increases and net power output decreases with increasing NCG fraction. The plant which is employed with compressors generates the highest net power output at each NCG fraction. Wet bulb temperature is an important parameter to determine the motive steam flowrate for the NCG removal system. The performance of power plants changes throughout the year depending on the wet bulb temperature as a function of outdoor temperature and relative humidity. Wet bulb temperature is the most important controlling parameter on cooling towers. Since cooling towers are parts of the gas removal systems, which maintain the cooling water for condenser where the NCGs are extracted from, the wet bulb temperature is vital requiring close monitoring.[16]

5. RESULTS AND DISCUSSION

5.1 Optimization

Graphical plots below show the relationships between various parameters with the aim of determining the optimum conditions of operations. Figure 5. shows the optimal working conditions for condenser pressure for the case of single flash cycle would be at 0.07bar as depicted by the rising graph. The plot also shows that the higher the steam flow rate, the more the net output as shown by the falling graph in Fig. 5 below.

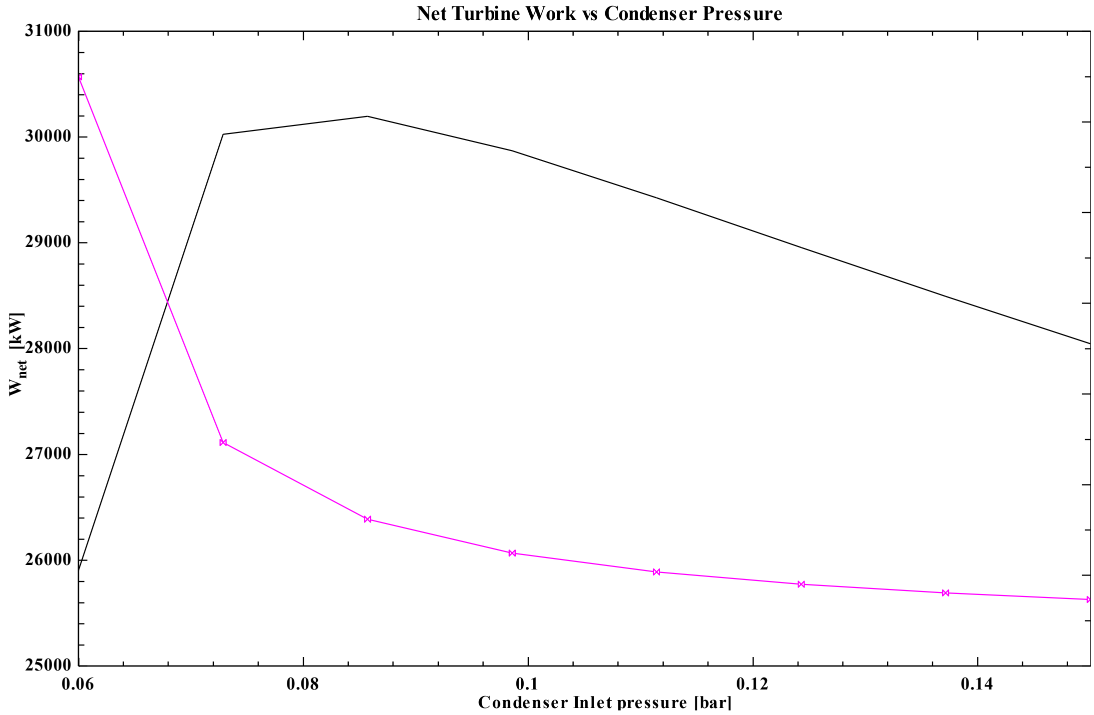


Figure 5: Actual Turbine Work Vs Separator Pressure for Single Flash Cycle

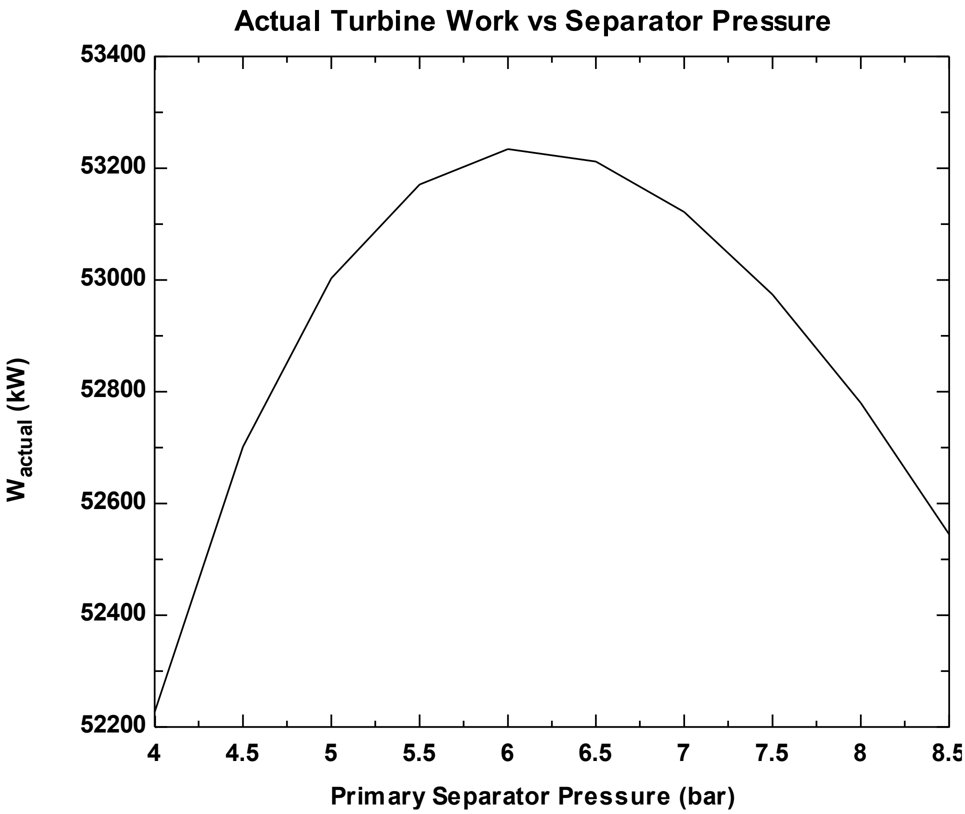


Figure 6: Actual Turbine Work Vs Separator Pressure for Double Flash Cycle

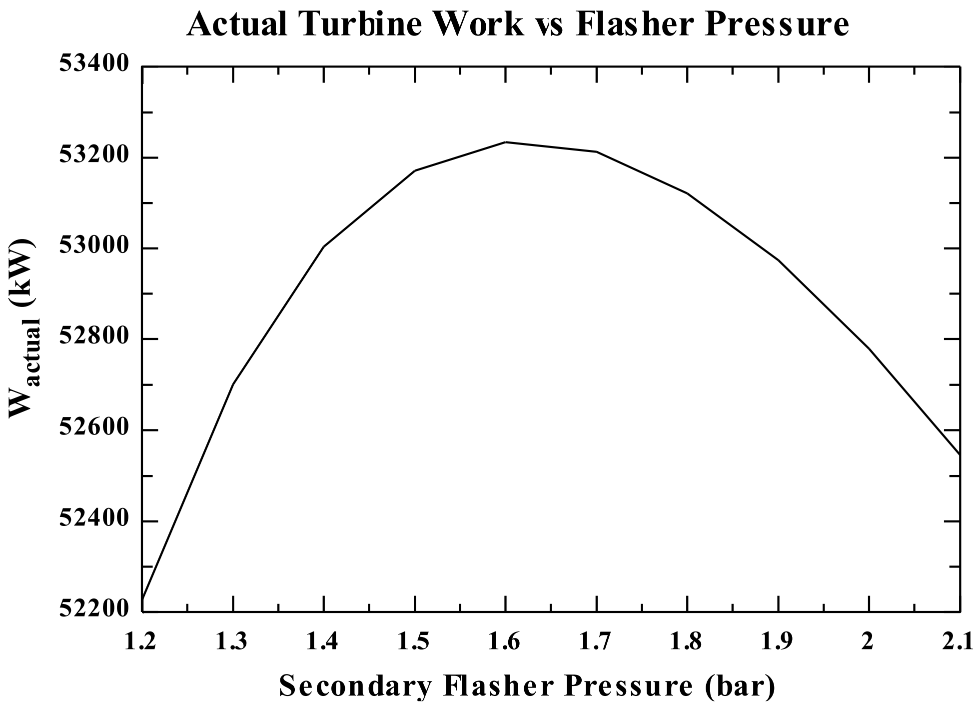


Figure 7: Actual Turbine Work Vs Flasher Pressure for Double Flash Cycle

Figures 5 & 6. shows the optimal operating conditions of the single and double flash cycles for maximum turbine work output against separator pressure should be at between 5 and 6 bar respectively. For the case of the double flash cycle, the secondary steam flasher optimum operating conditions is at between 1.6 and 1.7 bar as shown in Fig7. above.

In the analysis, a total of 15 wells as shown in Table. 1 were used to design the plant. The well outputs were based on monitoring results taken at each day and was assumed to be valid. Analysis was made using EES (in the appendix). Graphical representation is shown in Fig.5 and the results presented in Table.2.

Table 2: Governing results of single and double flash design in Menengai field

Mass Flow Rate (kg/s)	Steam Flow Rate (kg/s)	Aggregated Well Head Pressure (Bar)	Separator Pressure (Bar)	Steam Pipeline Pressure Loss (Bar)	Turbine Admission Pressure (Bar)	Condensor Pressure (Bar)	HP Turbine Output (kW)	LP Turbine Output (kW)
383.23	224.6	10	9	3	6	0.006	108732	0
383.2	224.6	12	10.8	3	6 & 0.1	0.006	108732	4875

Data collected for geothermal power plants around the world show the average separator pressure is 6.2 bar abs. for single flash plants. While the average separators pressures are 6.7 and 2 bar abs. respectively for double flash plants [17].

6. CONCLUSION

The well configuration of Menengai field was combined to investigate the deterministic power output capable. Firstly, single flash design was conducted as shown in Fig. 8. Thereafter, the design was modified to scrutinize the possibility of operating a double flash system as shown in Fig. 9.

The optimum operating conditions were set for both single and double flash cycles to give the ideal power output from the turbine. Single flash cycle produced 109 MW while double flash cycle gave 113 MW for the same steam flow rate from the wells.

Reinjection fluid temperature from the single flash is 175°C while that of double flash is 158°C. Both of these combinations show that possibility of silica scaling is higher as the hotter the resource, the higher the silica concentration in the incoming geofluid, the greater the supersaturation in the waste brine, from either type of flash plant, and the greater the likelihood of precipitation in the flash vessel, in the piping leading to the injection wells or in the formation [12].

Results show the single flash system operate as close as possible to the optimal conditions such that adding the double flash cycle did very little improvement to it. If parasitic loads were minimized accordingly, both the single flash and double flash cycle would give more or less the same output.

Since the single flash cycle is the most cost-effective system to set up unlike the costlier double flash, Menengai geothermal prospect would be best suited for the single flash cycle until more wells are added to the system, then it would be possible to realize significantly more output from the double flash for the same input.

Single flash cycle

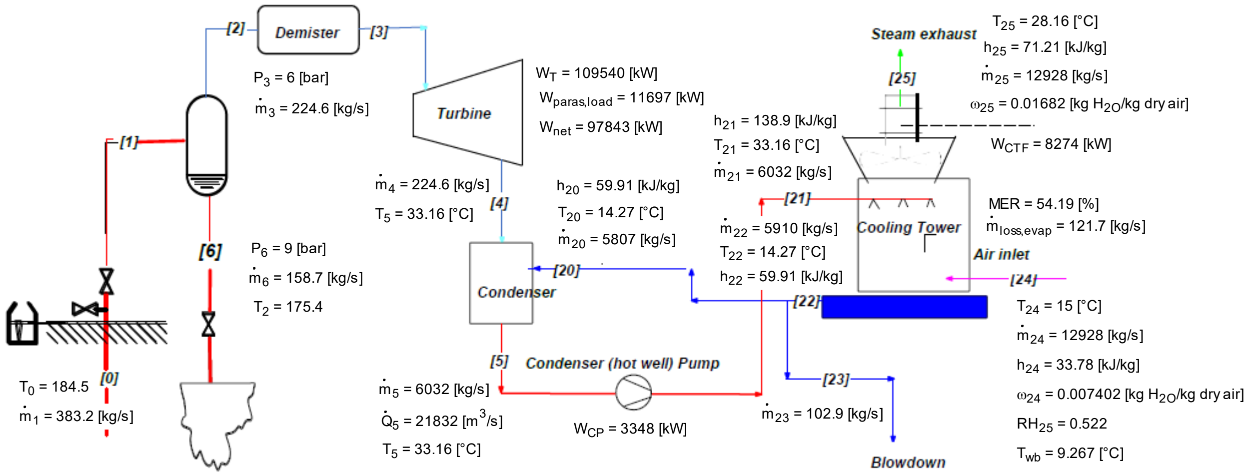


Figure 8: The single flash cycle

Double flash cycle

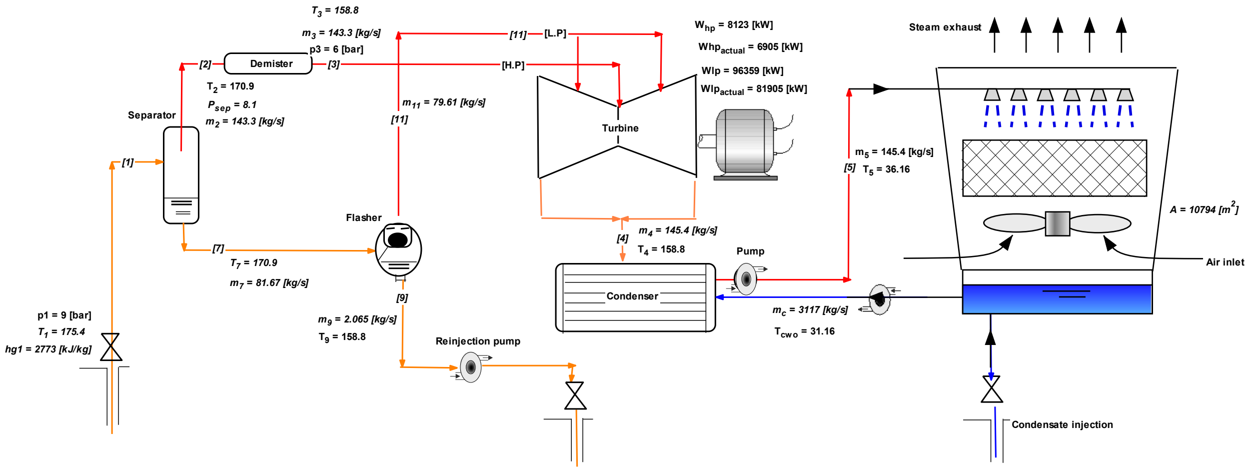


Figure 9: The double flash cycle

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Nomenclature

C_p	*specific heat capacity of water (kJ/kg K)*
h	*enthalpy (kJ/kg)*
$\dot{m}_1$	*total mass flow rate (kg/s)*
$\dot{m}_2$	*steam mass flow rate (kg/s)*
P	*pressure (MPa)*
P_c	*critical pressure (MPa)*
P_s	*saturation pressure (MPa)*
Q_{con}	*heat transfer in the condenser (kW)*
Q_{in}	*heat transfer in evaporator (kW)*
rho	*Density*
s	*entropy (kJ/kg K)*
T	*temperature (^0C)*
T_c	*critical temperature (^0C)*
W_{net}	*net power output (kW)*
W_p	*work of the feed pump (kW)*
W_t	*power of the turbine (kW)*
η_t	*isentropic efficiency of a turbine*
η_{Plant}	*overall plant efficiency*
η_p	*efficiency of the pump*
Δp	*pressure head (Pa)*
ρ	*water density of water (kg/m^3)*