Energy Optimization Modeling of Geothermal Power Plant
(Case Study: Darajat Geothermal Field Unit III)

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Abstract. Darajat unit III geothermal power plant is developed by PT. Chevron Geothermal Indonesia (CGI). The plant capacity is 121 MW and load 110%. The greatest utilization power is consumed by Hot Well Pump (HWP) and Cooling Tower Fan (CTF). Reducing the utility power can be attempted by utilizing the wet bulb temperature fluctuation. In this study, a modelling process is developed by using Engineering Equation Solver (EES) software version 9.430. The possibility of energy saving is indicated by Specific Steam Consumption (SSC) net in relation to wet bulb temperature fluctuation from 9°C up to 20.5°C. Result shows that the existing daily operation reaches its optimum condition. The installation of Variable Frequency Drive (VFD) could be applied to optimize both utility power of HWP and CTF. The highest gain is obtained by VFD HWP installation as much as 0.80% when wet bulb temperature 18.5°C.

1. Introduction
Darajat geothermal power plant is located in West Java, Garut residence, about 35 km southeast from Bandung on 1750 -2000 masl. Darajat unit III was operated in July 2007 for capacity 110 MWe. Since 2009, it has been uprated up to capacity 121 MW, load 110%. There are other geothermal power plants near Darajat Geothermal, those are Kamojang and Wayang Windu plants. Kamojang is located 9 km northeast from Darajat. Meanwhile Wayang Windu about 10 km west from Darajat[8].

Initial design of geothermal power plant was started by choosing suitable thermodynamics cycle, preparing power plant modeling, and then determination of technical specification of every equipment using a simulation. It aims to optimize the net output of a power plant by setting up the thermodynamics parameters[7]. During its operation, power plant systems should be analysed both for full and partial load. Partial load operation is conducted to determine power plant system condition during maintenance of some equipment without shutting down the power plant[9]. A Simulator is required to make decisions for such condition.

This study aims to optimize the overall plant efficiency of Darajat Unit III plant through modeling and simulating the wet bulb temperature fluctuation with particular attention to the refrigeration system. The modeling is validated by Process Flow Diagram (PFD) Darajat Unit III load 110 % (121 MW) for 1% NCG percentage, atmosphere pressure 0.8 bar, wet and dry bulb temperature 15.5°C and 21.77 °C, respectively, and relative humidity of 55 %.
2. Basic Theory

1.1. Heat and Mass Balance

Power plant facility is analysed based on heat and mass balance. There are some assumptions for analysing heat and mass balance[4]:

a. Fluid flows independent on time (steady state)

b. No heat transfer from/to every facility

c. No work produced/consumed, except turbine and pump

d. Kinetic and potential is ignored since enthalpy is far higher than those kinds of energy.

Heat and mass balance is analysed based on the following equations, [6]

\[
\frac{dm_{cv}}{dt} = \sum_{i} \dot{m}_{i} - \sum_{e} \dot{m}_{e}
\]

(1)

\[
\frac{dE}{dt} = \dot{Q}_{ev} - \dot{W}_{ev} + \sum_{i} \dot{m}_{i} \left( h_{i} + \frac{v_{i}^{2}}{2} + g_{z_{i}} \right) - \sum_{e} \dot{m}_{e} \left( h_{e} + \frac{v_{e}^{2}}{2} + g_{z_{e}} \right)
\]

(2)

Where:

\( \dot{Q} \) = Heat transfer to or from system (kW)

\( \dot{W} \) = Work produced or consumed by facility (kW)

\( \dot{m} \) =mass rate (kg/s)

\( h \) =enthalpy specific (kJ/kg)

\( v \) =fluid velocity (m/s)

\( g \) =gravitation constant (m/s²)

\( z \) = high level (m)

1.2. Energy Conversion System

Darajat plant energy conversion system is displayed by figure 1. Point 1 shows fluid outflow from well head which has steam quality 99.98%. Then, fluid flows in to steam scrubber and produces dry steam to be expanded into turbine. Meanwhile brine flows out from steam scrubber (point 3) to cooling tower before injected into the reservoir. Figure 1 illustrates simple scheme Darajat unit III process flow[1]. Figure 2 shows T-s diagram of Darajat unit III process flow. A complete Process Flow Diagram (PFD) of Darajat unit III is shown in figure 3.

![Figure 1. Simple Process Flow Scheme of Darajat Unit III](image-url)
Figure 2. T-s Diagram of Darajat Unit III

Based on figure 2, steam mass flow rate in to turbine is expressed as $\dot{m}_{s,2} = x_1 \cdot \dot{m}_{s,1}$. The isentropic process of turbine results $s_2 = s_{4,s}$. Turbine outlet steam quality (point 4s) is calculated based on equation (3):

$$x_{4s} = \frac{s_{4s} - s_3}{s_6 - s_3}$$

Then, fluid enthalpy at point 4s is:

$$h_{4s} = x_{4s} \cdot h_6 + (1 - x_{4s}) \cdot h_3$$

Thus, turbine output is calculated by equation (5)

$$W_t = x_1 \cdot \dot{m}_{s,1} \cdot (h_2 - h_4)$$

Turbine efficiency is:

$$\eta_t = \frac{h_2 - h_4}{h_3 - h_{4s}}$$

Based on PFD, inlet turbine pressure (point 2) is 16.83 Bar where saturation temperature, both steam and NCG enthalpy are $T_2 = 203.9$ °C, $h_{s,2} = 2795 \frac{kJ}{kg}$, and $h_{NCG,2} = 159.7 \frac{kJ}{kg}$, respectively. Turbine isentropic process results $s_2 = s_6 = 6.402 \frac{kJ}{kgK}$. Therefore, enthalpy at point 6 as function of both entropy and pressure, $h_{s,6} = 2011 \frac{kJ}{kg}$. Turbine isentropic efficiency is 82.5%, therefore steam enthalpy at point 6 is

$$\eta_{turbine} = \frac{h_{s,2} - h_6}{h_{s,2} - h_{6s}}$$

Electricity power is calculated by

$$\dot{W} = \dot{m}_{s,3} \cdot (h_{mix,2} - h_{mix,6}) \cdot \eta_{generator}$$

Generator efficiency is 98%, thus gross electricity power is $\dot{W}_{gross} = 121$ MW.
Figure 3. Process Flow Diagram (PFD) of Darajat Unit III.
3. Power Plant Modeling

Some main facilities are modelled by heat and mass balances are shown below [4]:

3.1. Main Condenser System

Main condenser scheme is illustrated by figure 4. Mass balance of steam and water, NCG, and air are shown through equation 9, 10, and 11, respectively.

\[
2\dot{m}_{s,6} + \dot{m}_{w,33} + \dot{m}_{w,14} + \dot{m}_{w,15} + \dot{m}_{w,20} + \dot{m}_{w,27} + \dot{m}_{w,39} = \dot{m}_{w,7} + \dot{m}_{w,16}
\]

\[2\dot{m}_{\text{NCG,6}} = \dot{m}_{\text{NCG,7}} \tag{10}\]

\[\dot{m}_{w,46} + \dot{m}_{w,20} = \dot{m}_{w,7} \tag{11}\]

Figure 4 Main Condenser Scheme

Meanwhile, energy balance is shown below

\[
2(\dot{m}_{s,6} + \dot{m}_{\text{NCG,6}})h_{\text{hump,6}} + \dot{m}_{w,13} \cdot h_{w,13} + \dot{m}_{w,14} \cdot h_{w,14} + \dot{m}_{w,15} \cdot h_{w,15} + \dot{m}_{w,20} \cdot h_{w,20} + \dot{m}_{w,27} \cdot h_{w,27} + \dot{m}_{w,39} \cdot h_{w,39} + \dot{m}_{w,46} \cdot h_{w,46} = (\dot{m}_{s,7} + \dot{m}_{\text{NCG,7}})h_{\text{hump,7}} + \dot{m}_{w,7} \cdot h_{w,7} + \dot{m}_{w,16} \cdot h_{w,16} \tag{12}\]

The calculation of cooling water mass rate \((\dot{m}_{w,20})\) is determined by comparing water mass flow rate entering cooling tower with cooling air, defined as L/G [3]. It is 1.049 for wet bulb temperature 15.5°C. Therefore, both condensate mass flow rate \((\dot{m}_{w,16})\) and temperature \((T_{16})\) can be obtained.

3.2. Cooling Tower

The flowing of water, air, and NCG inside cooling tower is illustrated in figure 5.
Water/steam, NCG, and air mass balance is shown in equation (13), (14), and (15), respectively:

\[
\dot{m}_{w,12} + \dot{m}_{w,36} + \dot{m}_{w,38} + \dot{m}_{w,22} = \dot{m}_{w,39} + \dot{m}_{w,30}
\]  
(13)

\[
\dot{m}_{\text{NCG},12} + \dot{m}_{\text{NCG},34} + \dot{m}_{\text{NCG},36} = \dot{m}_{\text{NCG},30}
\]  
(14)

\[
\dot{m}_{a,12} + \dot{m}_{a,34} + \dot{m}_{a,36} + \dot{m}_{a,30} = \dot{m}_{a,30}
\]  
(15)

Energy balance:

\[
\begin{align*}
h_{w,36} + \omega_{36} \cdot C_{p_r,36} T_{wb,36} + \frac{1}{m_w} (\dot{m}_{w,18} \cdot C_{p_r,18} \cdot T_{18} + \dot{m}_{w,22} \cdot C_{p_r,22} \cdot T_{22} + \dot{m}_{w,36} \cdot h_{w,36}) & = \\
\end{align*}
\]

\[
\begin{align*}
h_{w,36} + \omega_{30} \cdot C_{p_r,30} T_{wb,30} + \left( \omega_{36} - \omega_{30} + \frac{1}{m_w} (\dot{m}_{w,18} + \dot{m}_{w,22}) \right) \cdot C_{p_r,19} \cdot T_{19}
\end{align*}
\]  
(16)

As explained before that L/G is 1.049. Therefore water mass rate entering cooling tower \((m_{w,19})\), inlet \((m_{a,36})\) and outlet air mass rate \((m_{a,30})\) can be obtained.

3.3. Utilities Load

Some kinds of utilities load are shown below:

3.3.1. Hot Well Pump. Hot well pump is used to remove condensate water to cooling tower. The scheme of hot well pump is shown in figure 6.
Hot well pump utility power is calculated by [5]:

\[ \hat{W}_{\text{SCW-P-001A}} = \frac{1}{2} v_{16} \cdot g \cdot \text{Head} \]

For given data, hot well pump’s power is \( \hat{W}_{\text{SCW-P-001A}} = 1077 \text{ kW} \). Based on manual data, the efficiency of pump and pump motor is 84.72% and 87.85%, respectively. The calculated head pump is 24.44 m. Meanwhile the head of pump for \( m_{w,16} \) calculated is 24.44 m.

3.3.2. Fan Cooling Tower. There are nine existing cells inside the cooling tower. The efficiency of each fan and its motor is 76.1% and 74.25%, respectively. Fan utility power is calculated by:

\[ \hat{W}_{\text{fan}} = \frac{m_{\text{air,16}} \cdot v_{\text{air,36}} \cdot \Delta p}{\eta_{\text{fan}} \cdot \eta_{\text{motor}}} \]

Variable \( \Delta p \) defines pressure drop inside cooling tower. Total fan utility power is \( \hat{W}_{\text{fan}} = 1666 \text{ kW} \). By using the similar approach, total utility power is 4310 kW.

4. Result
Power plant performance is determined by Specific Steam Consumption (SSC) steam supply to produce 1 kWh electricity. SSC is calculated by [9]:

\[ \text{SSC}_{\text{wet}} = \frac{\dot{m}}{\hat{W}_{\text{net}}} \cdot 3600 \]

4.1. Variable Frequency Drive (VFD) Application
Wet bulb temperature fluctuation provides a chance for power plant system optimization by using Variable Frequency Drive (VFD)[2]. It is installed to hot well pump and/or cooling tower fan.

4.1.1. VFD Application for Hot Well Pump (VFD HWP). The lower wet bulb temperature, the lower exit cooling water temperature will be. Therefore cooling water mass rate in to condenser should be reduced to keep constant pressure. VFD HWP should be installed to reduce hot well pump utility power. Fluid flowing capacity in to a pump can be controlled either by an open valve or by a pump rotation[2]. The difference of those methods is displayed in figure 7.

Figure 7 Flowing Capacity Arrangement in A Pump by (a) Open Valve, (b) Pump Rotation

Note: Percentage denotes valve opening
Figure 7(a) shows the increased capacity $Q_1$ to $Q_2$ is controlled by open valve only. It yields constant utility power. Meanwhile figure 7(b) shows that increased capacity is controlled by the speed of pump rotation. This method could reduce utility power for lower capacity.

The ratio between pump rotations is determined by

$$Q_1 = n_2 \left( \frac{D_1}{D_2} \right)^3$$

(20)

$Q_1$ : First stage of pump capacity
$Q_2$ : Second stage of pump capacity
$n_1$ : Pump speed rotation on first stage
$n_2$ : Pump speed rotation on second stage
$D_1$ : Pump diameter on first stage
$D_2$ : Pump diameter on second stage

Pump constant diameter ($D_1 = D_2$) results eq. (21):

$$\frac{Q_1}{Q_2} = \frac{n_1}{n_2}$$

(21)

Therefore frequency changes as result of stator field rotation of pump motor

$$n_s = \frac{120f}{P}$$

(22)

Where $f$ denotes frequency of stator current. Meanwhile, $P$ denotes the number of pole inside electric motor. Another parameter is slip. It is occurred as result of the difference of rotation motor with stator field which formulated as:

$$Slip = \frac{n_s - n_p}{n_s}$$

(23)

Where $n_p$ denotes pump rotation (rpm)

4.1.2. VFD Application for Cooling Tower Fan (VFD CTF). It is applied to maintain constant condenser pressure by the adjustment of both cooling water temperature and mass rate in relation to wet bulb temperature. The adjustment is controlled by cooling tower fan. Most of equations that are used both for VFD HWP and VFD CTF are relative same. The only difference is the reduction ratio of fan[2] which is formulated as equation (24)

$$R = \frac{n_{motor}}{n_{fan}}$$

(24)

Where

R : Gearbox reduction ratio
$n_{motor}$ : Motor fan rotation speed (rpm)
$n_{fan}$ : Fan rotation speed (rpm)

4.2. Power Plant Analysis

Estimation of cooling water temperature is obtained by cooling tower heat transfer theory. Both cooling water and condensate temperature, for wet bulb temperature, range from 9°C to 20.5°C, displayed in figure 8. It shows that the temperature difference of the two unit processes are relatively constant over the increasing wet bulb temperature since the mass rate of the cooling water is also relatively constant.
The higher cooling water temperature, the higher condenser pressure is. The increase of condenser pressure is shown in figure 9. It displays that condenser pressure increases 23.6% in relation to the increase of wet bulb temperature from 9°C up to 20.5°C.

The increase of condenser pressure will affect electricity generation which is displayed in figure 10. The increase of condenser pressure yields the decrease of 2.9% gross output power.
Using similar approach, SSC net increases approximately 3.19% until 3.24%, as shown in figure 11.

![Figure 11 Wet Bulb Temperature Impact to SSC Nett on Constant L/G](image1)

Figure 11 Wet Bulb Temperature Impact to SSC Nett on Constant L/G

Figure 12 shows the impact of VFD HWP to maintain condenser pressure 0.087 bar. Both cooling water and condensate temperature increase directly proportional with wet bulb temperature. The increase temperature is quite different with constant L/G. The increase of bulb temperature yields the less difference of those temperatures where caused by the amount of cooling water mass rate to maintain constant condenser pressure 0.87 bar.

![Figure 12 Cooling Water and Condensate Temperature on VFD Hot Well Pump](image2)

Figure 12 Cooling Water and Condensate Temperature on VFD Hot Well Pump

Both cooling water and condensate mass rate is revealed in figure 13. Their increase are 52.92% and 45.12%, respectively.
Based on HWP design, normal capacity of each pump is $11.420 \text{ m}^3/\text{hour}$ which provide energy saving of HWP utility power for a certain wet bulb temperature. Figure 14 shows that pump utility power range from 1930 kW up to 2146 kW for wet bulb temperature below 15.5°C. VFD HWP reduces 11.20% utility power. VFD can be applied up to wet bulb temperature 18.5 °C in case of the ability of HWP to operate until frequency 60 Hz. It is displayed in figure 14 by shaded region.

VFD can also be installed to cooling tower fan (CTF) electrical motor. Cooling air mass rate is adjusted to keep constant condenser pressure. It is displayed in figure 15.
Cooling air mass rate increases 273.93% to maintain exit cooling water temperature 23.9°C. Based on cooling tower design, each cell can absorb cooling air as much as 771.6 kg/s.

Normal cooling air is absorbed by 8 of 9 existing cells. Figure 16 reveals fan quantities to adjust the amount of cooling air as described before. It shows that VFD CTF is applied for wet bulb below 16.5°C.

Total fan utility power is represented in figure 17.
The possibility of energy saving of CTF is applied below wet bulb temperature 15.5°C as well as VFD HWP. It reduces energy up to 37.97% when wet bulb temperature 9°C. Meanwhile, VFD CTF can be applied up to wet bulb temperature 17°C in case of motor fan ability to operate until 60 Hz. Both gross and net power of each scenario is represented in figure 18. It shows that both gross and net power constant L/G is higher than other VFD applications.

![Figure 18 Gross and Nett Power of Each Scenario](image)

SSC net of each scenario is considered to decide power plant performance. SSC net defines the amount of steam to generate 1 MWh net electricity. The less steam supply required to generate 1 MWh electricity, the more efficient the power plant is. SSC net consideration is displayed in figure 19. When the wet bulb temperature is lower than 15.5°C, constant L/G SSC net is lower than others. Otherwise, when the wet bulb more than 15.5°C, SSC net VFD HWP is lower than constant L/G.

In case of motor pump ability to operate until frequency 60 Hz, VFD HWP can be applied starts from wet bulb temperature 15.5°C until 18.5°C. It is shown in Figure 19. The highest gain of VFD HWP is obtained when wet bulb temperature 18.5°C, about 0.80% compared to constant L/G.

![Figure 19 SSC Net](image)
5. Conclusions and Suggestions

5.1. Conclusions

Having developed and simulated some variables, some conclusions are described below:

- Based on consideration of wet bulb fluctuation, the existing daily operation reaches its optimum performance.
- In case of motor pump ability to operate until frequency 60 Hz, VFD HWP can be applied starts from wet bulb 15.5 °C until 18.5 °C. The highest gain of VFD HWP is obtained when wet bulb temperature 18.5 °C, about 0.80% compared to constant L/G.

5.2. Suggestions

- Besides technical consideration, economical approach should be considered for power plant optimization.
- This study mainly focuses on simulation based on full load. Further study should elaborate partial load in order to keep better performance during maintenance.

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