Optimizing the Utility Power of a Geothermal Power Plant using Variable Frequency Drive (VFD) (Case Study: Sibayak Geothermal Power Plant)

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Abstract. Sibayak Geothermal Power Plant (SGPP) is one of the plants being developed by Pertamina Geothermal Energy (PGE) at the upstream phase. At the downstream phase, State-owned Electricity Company (PLN) through PT. Dizamatra Powerindo is the developer. The gross capacity of the power plant is 13.3 MW, consisting 1 unit of Monoblock (2 MW) developed by PGE and 2 units (2×5.65 MW) operated through Energy Sales Contract by PLN. During the development phase of a geothermal power plant, there is a chance to reduce the utility power in order to increase the overall plant efficiency. Reducing the utility power can be attempted by utilizing the wet bulb temperature fluctuation. In this study, a modeling process is developed by using Engineering Equation Solver (EES) software version 9.430. The possibility of energy saving is indicated by condenser pressure changes as a result of wet bulb temperature fluctuation. The result of this study indicates that the change of condenser pressure is about 50.8% on the constant liquid/gas (L/G) condition of the wet bulb temperature of 15°C to 25°C. Further result indicates that in this power plant, Cooling Tower Fan (CTF) is the facility that has the greatest utility load, followed by Hot Well Pump (HWP). The saving of the greatest utility load is applied through Variable Frequency Drive (VFD) instrumentation. The result of this modeling has been validated by actual operations data (log sheet). The developed model has also been reviewed through Specific Steam Consumption (SSC), resulting that constant L/G condition allows the optimum condition on of the wet bulb temperature of 15°C to 25°C.

Keywords: Geothermal, Sibayak, variable frequency drive, wet bulb

1. Introduction

Initial design of geothermal power plant is started on choosing suitable thermodynamics cycle, preparing power plant modeling, and then determining technical specification for every equipment component through simulation [9]. This procedure is conducted in order to optimize the net output of a power plant by setting up the thermodynamics parameters. During its operation, power plant systems should be analysed for both full and partial load. Partial load operation is conducted to know power plant system condition during some equipment maintenance without shutting down the power plant. A Simulator is needed to make some decisions for such condition.

[8] studied the performance of both full and partial load of Lumut Balai geothermal power plant with capacity 2× 55 MW using Engineering Equation Solver (EES) software. In the study, the simulation of partial load using both wet bulb temperature and Non-Condensable Gas (NCG) percentage were used as input variable parameter. The study shows that power plant performance is more significantly influenced by wet bulb temperature fluctuation rather than NCG percentage.

Wet bulb temperature fluctuation makes a chance for optimizing power plant system by reducing energy use from certain equipment. Variable Frequency Drive (VFD) is applied for both Main Cooling
Water Pump (MCWP) and Cooling Tower Fan (CTF) [3]. The installation of VFD to Hot Well Pump (HWP) in Darajat Geothermal Power Plant can be applied starts from wet bulb 15.5°C until 18.5°C. The highest gain of VFD HWP is obtained when the wet bulb temperature 18.5°C, about 0.80% compared to constant Liquid/Gas (L/G) [6].

This study aims to optimize the overall plant efficiency of Sibayak Geothermal Power Plant (SGPP) through modeling and simulating the wet bulb temperature fluctuation with particular attention to the refrigeration system.

2. Sibayak Geothermal Power Plant (SGPP)

SGPP is one of the plants being developed by Pertamina Geothermal Energy (PGE) at the upstream phase. At the downstream phase, State-owned Electricity Company (PLN) through PT. Dizamatra Powerindo is the developer. SGPP is located at 3°10’12.7” N and 98°29’17.9”E about 65 km southwest from Medan, capital city of North Sumatera. The gross capacity of the power plant is 13.3 MW, consisting 1 unit of Monoblock (2 MW) developed by PGE and 2 units of single flash system (2×5.65 MW) operated through Energy Sales Contract by PLN.

SGPP is situated on Singkut Caldera about 1400 up to 2200 masl. It is surrounded by three active volcanoes: Pintau Mt, Sibayak Mt, and Pratektekan Mt. SGPP exploration field was begun by Pertamina from 1989 to 1991. Initial exploration well was located between Sibayak Mt. and Pratektekan Mt. Well drilling was held by Pertamina since 1991 until 1997 consisted of 7 production wells (SBY-1, SBY-3, SBY-4, SBY-5, SBY-6, SBY-7, SBY-8) and 3 reinjection wells (SBY-2, SBY-9, and SBY-10).

Reservoir’s temperatures are ranging between 240 and 300°C, and the pressure is about 45 bar-g. Reservoir’s enthalpy is up to 1100 kJ/kg and the mass flow rate from well about 51.4 kg/s and pressure 9 bar [7]. Through simulation using EES, [7] obtained that the increase of turbine output power is in tandem with the decrease of condenser pressure. The increase of turbine output power is also related to the increase of utility load. As the result, lower condenser pressure will reduce turbine output power for constant steam supply. Through modeling, optimum output turbine (about 20.858 MW) for 2 units power plant is obtained when separator, condenser, and inlet turbine pressure are on 8 bara, 0.08 bara and 7.7 bar, respectively. However, based on actual condition, turbine and condenser pressure is operated on 7.5-8.0 and 0.1 - 0.3 bara, respectively.

3. Theoretical Basics

3.1. Power Plant Efficiency

The higher NCG percentage, the lower the efficiency of power plant system because NCG will affect the condenser pressure. Condenser pressure is maintained by the increase of NCG extraction. Gas extraction system needs motive steam (for steam jet ejector) or utility power (for vacuum pump) which is increased as well as NCG percentage increase [1]. Through developed simulator, performance of SGPP is analyzed both by net power and steam supply. The performance is called Specific Steam Consumption (SSC) [8] which defines the amount of steam that is needed to generate 1 MWh electricity.

SGPP is developed by single flash power plant, whose flow process through T-s diagram is shown in figure 1[2]. Based on figure 1, the process starts from well head (point 1). Fluid undergoes pressure decrease and isenthalpic ($h_i = h_f$) inside the separator (point 2). Steam from separator flows to demister and enters to the turbine (point 3). While brine from separator (point 4) and demister (point 7) flows to cooling tower before injected through reinjection well. Turbine expansion process is shown from point 3 to 5. Point 3 to 5s shows isentropic process inside turbine ($s_3 = s_{5s}$). Therefore, steam quality at point 5s is
SGPP is developed by single flash power plant, whose flow process through T-s diagram is shown in

\[ x_{3s} = \frac{s_{3s} - s_{s1}}{s_{s1} - s_{sJ}} \]  

(1)

Enthalpy at point \( h_{3s} \):

\[ h_{3s} = x_{3s} \cdot h_{s1} + (1-x_{3s}) \cdot h_{sJ} \]  

(2)

Turbine isentropic efficiency is denoted as \( \eta_t \). Turbine exit enthalpy \( (h_{4}) \) is calculated using:

\[ \eta_t = \frac{h_{1} - h_{3}}{h_{1} - h_{s}} \]  

(3)

Thus, turbine output power is:

\[ W_t = \dot{m}_3 \cdot (h_1 - h_{3}) \]  

(4)

Where \( \dot{m}_3 \) denotes steam flow rate from well head, \( \dot{m}_3 = x_s \cdot \dot{m}_s \)

Exit steam turbine is condensed by cooling water from cooling tower (point 8). Condensate temperature is obtained by:

\[ \dot{m}_4(h_6 - h_{6}) = \dot{m}_s(h_{5} - h_{6}) \]  

(5)

3.2. Variable Frequency Drive (VFD)

The lower wet bulb temperature for constant cooling tower operation, the lower the temperature of circulating cooling water. Thus, VFD should be applied to reduce utility power of Hot Well Pump (HWP). Meanwhile, VFD Cooling Tower Fan (CTF) is applied to maintain both cooling water temperature and mass rate while wet bulb fluctuating. Maintaining of both temperature and mass rate are controlled by the adjustment of cooling air mass rate [3]. This study analyzes plant optimizing by VFD CTF only.

Flowing capacity difference through a pump is controlled by an open valve or by a pump rotation arrangement. The capacity difference is shown in figure 2 where figure 2(a) shows the decreasing of capacity \( Q_1 \) to \( Q_2 \) by lowering of open valve. For this way, pump utility power remain constant. While figure 2(b) shows that decreasing capacity is controlled by the speed of pump rotation. This way can make the utility power difference. Optimization of HWP utility load is applied trough pump rotation difference.
4. **Power Plant Modeling**

As a part of the modeling stage, SGPP is described as a Process Flow Diagram (PFD) in figure 3. Red line shows steam flowing, blue line shows condensate and cooling water flowing. While purple line shows NCG flowing. This model is then validated with the log sheet data from unit 1 of the single flash of Sibayak Plant (5 MW) using heat and mass balance [5].

4.1. **Steam Turbine**

Steam flowing during turbine expansion is shown through stream number 4 to 6 in figure 3. Assumed that there is no heat transfer through turbine. Thus, work rate that turbine produced is:

\[ W_{\text{turbine}} = \dot{m}_4 \cdot (h_h - h_i) \cdot \eta_{\text{generator}} \]  

(6)

4.2. **Condenser**

Condensation process is maintained by cooling water spraying directly to steam exit turbine as shown in figure 3 through stream line 16. Using heat and mass balance in the condenser, equation (7), (8), and (9) are obtained.

- NCG mass balance: \( \dot{m}_{\text{NCG},6} = \dot{m}_{\text{NCG},10} \)

(7)

- Water and steam mass balance: \( \dot{m}_{x,6} + \dot{m}_{w,15} + \dot{m}_{w,16} = \dot{m}_{w,7} \)

(8)

- Energy balance:

\[
\left( \dot{m}_{x,6} + \dot{m}_{\text{NCG},6} \right) h_{\text{mix},6} + \dot{m}_{w,15} \cdot h_{w,15} + \dot{m}_{w,16} \cdot h_{w,16} = \dot{m}_{w,7} \cdot h_{w,7} + \dot{m}_{\text{NCG},10} \cdot h_{\text{NCG},10}
\]

(9)

Equations (7) to (9) are used to obtain both condensate temperature and mass flow rate.

4.3. **Cooling Tower**

Cooling tower is designed to reduce condensate heat. It is processed by the decrease of condensate temperature through cooling air absorbing around cooling tower. Then cooling water will be injected and some of it circulated again for condensing process. Mechanical draft cooling tower is used in SGPP. Utility power of cooling tower fan is formulated in equation (10):

\[ W_{\text{motorfan}} = \frac{\dot{\nu}_{\text{air}} \cdot \Delta p}{\eta_{\text{fan}} \cdot \eta_{\text{motorfan}}} \]

(10)

Where \( \dot{\nu}_{\text{air}} \) denotes air volumetric rate (m³/s) and \( \Delta p \) as pressure drop inside cooling tower. Cooling process in cooling tower is influenced by wet bulb temperature and air relative humidity. Cooling water temperature that can be reached is higher than wet bulb temperature. The difference of those temperatures is called approach. While the difference between inlet and exit temperature of cooling water is called as range. The value of both range and approach depends on cooling tower design.
Figure 3. Process Flow Diagram (PFD) SGPP Unit I

Notes:
- Water Flow
- Steam Flow
- NCG Flow
- Oil Flow
- Heat Exchanger
- Air Cooler
- Oil Cooler

From Steam Field
Separator
To Settling Basin
Cooling Tower
To Reinjection Well
To 2nd Turbine
1# LRVP
2# LRVP
1# Hot Water Pump
2# Hot Water Pump
3# Hot Water Pump

To Separation
Steam Demister
1# Cooling Water Pump
2# Cooling Water Pump

From Separator
From Demister
Water Basin
Settling Basin
Cooling Tower
To Reinjection Well
The concept of cooling tower heat transfer is developed based on the difference on potential enthalpy between air and water [4]. It is assumed that each water particle is wrapped up by air surface. The difference enthalpy between air and water causes cooling process. While cooling process, heat is released by water is same as heat received by cooling air. Therefore,

\[ m_{w} \cdot c_{p} \cdot (T_{w,\text{in}} - T_{w,\text{out}}) = m_{a} \cdot (h_{a,\text{out}} - h_{a,\text{in}}) \]  

(11)

Where \( m_{w} \) and \( m_{a} \) define mass flow rate of water and air, respectively. Whereas, \( c_{p} \), \( T \), and \( h \) denote heat capacity, temperature, and enthalpy, respectively.

Cooling tower is modeled based on figure 3 which is flown by stream numbers 23, 24, 25, 27, and 29. It has 3 cells. The first and second fans have the same output power, each fan 75 kW while the third fan output is 37 kW.

Following equations are applied based on heat and mass balance in cooling tower:

- NCG mass balance: \( m_{\text{NCG,23}} = m_{\text{NCG,27}} \)  

(12)

- Dry air mass balance: \( m_{u,\text{24}} = m_{u,\text{27}} = m_{a} \)  

(13)

- Wet air mass balance: \[ \omega_{2a} m_{w,\text{24}} + m_{w,\text{23}} = m_{w,\text{25}} + m_{w,\text{29}} + \omega_{2b} m_{u,\text{27}} \]  

(14)

- Energy balance:

\[ \omega_{2a} h_{w,\text{24}} + m_{w,\text{23}} h_{w,\text{23}} = h_{w,\text{27}} - h_{u,\text{24}} + \omega_{2b} h_{w,\text{27}} + m_{w,\text{29}} h_{w,\text{29}} + m_{u,\text{25}} h_{w,\text{25}} - m_{a} \]  

(15)

Equation (13) to equation (15) provide cooling air mass rate \( (m_{a,\text{24}}) \) and cooling water mass rate \( (m_{w,\text{25}}) \), also L/G. Where L/G denotes to the ratio of condensate water mass rate \( (m_{w,\text{25}}) \) to cooling airmass rate \( (m_{a,\text{24}}) \), about 1.099. Variable \( \omega \) and \( h \) defines specific humidity and enthalpy, respectively.

4.4. Gas Removal System

Liquid Ring Vacuum Pump (LRVP) is one of gas removal systems. It is used in Sibayak. The higher NCG concentration, the higher work rate of LRVP to remove the gas.

4.5. Pump

There are several main pumps used in Sibayak, those are HWP, Cooling Water Pump (CWP), and reinjection pump. Pump work rate is calculated by:

\[ W_{\text{motorpump}} = \frac{\dot{m}_{\text{water}} \cdot v_{\text{water}} \cdot \Delta p}{\eta_{\text{pump}} \cdot \eta_{\text{motorpump}}} \]  

(16)

Where \( \dot{m}_{\text{water}} \) denotes water mass rate (kg/s), \( v \) as specific volume (m³/kg), \( \Delta p \) as pressure difference (kPa), and \( \eta \) denotes the efficiency.

5. Result

5.1. Constant Liquid/Gas (L/G)

Estimation of cooling water temperature is obtained by cooling tower heat transfer theory. Both cooling water \( (T_{j,\text{a}}) \) and condensate temperature \( (T_{j}) \), for wet bulb temperature range from 15°C to 25°C, are displayed in figure 4(a). The higher wet bulb temperature, the higher both cooling water temperature. Cooling water temperature range from 18.64°C up to 26.94°C.

Meanwhile, condensate temperature is 30.66°C up to 39°C. Figure 4 (a) shows that the temperature difference of the two unit processes are relatively constant over the increasing wet bulb temperature. This phenomenon occurs because of the mass rate of the cooling water is relatively constant. The higher the temperature of cooling water, the higher the pressure of condenser. The increasing of
condenser pressure is shown in figure 4(b). For wet bulb range 15°C to 25°C, the increase of condenser pressure is 0.1357 – 0.2 bar, which is about 47.38%. The increase of condenser pressure will affect the gross power where the higher condenser pressure, the lower gross power will be, for constant inlet turbine pressure.

As wet bulb temperature increases, both gross and net power slightly decrease for constant steam mass flow rate. Figure 5 shows the change of gross and net power. Gross power decreases from 5.64 MW to 5.201 MW, about 7.78%. In addition to gross power decrease, utility power also decreases from 5.289 MW down to 4.856. Most of utility power is consumed both by cooling water pump and cooling tower fan. Figure 5 also shows the increase of SSC net as the wet bulb temperature increases. SSC net rises from 11.74 kg/MJ to 12.79 kg/MJ, about 9%.

![Figure 4](image-url) (a) Cooling Water and Condensate Temperature on Constant L/G, (b) Condenser Pressure on Constant L/G

![Figure 5](image-url) SSC Gross/Net Power and SSC Net on Constant L/G
5.2.  

**VFD Application in Cooling Tower Fan (CTF)**

The installation of VFD to cooling tower fan aims to keep constant condenser pressure, 0.2 bara. Constant condenser pressure is obtained by maintaining both cooling water temperature and mass rate. Figure 6(a) shows the effect of VFD installation to maintain cooling water temperature. Both cooling water and condensate temperature are maintained at 26.94°C and 39°C, respectively, as wet bulb temperature increases. VFD also aims to maintain cooling water mass rate, which is shown in figure 6(b), as much as 568.2 kg/s. Meanwhile, cooling air is increased to maintain constant cooling water temperature and mass rate, in order to maintain constant condenser pressure. The increase is 238.9 kg/s up to 544.8 kg/s, about 56.15%.

![Figure 6](image)

**Figure 6.** (a) Cooling Water & Condensate Temperature and Condenser Pressure on VFD CTF, (b) Cooling Water, Condensate, and Cooling Air Mass Rate on VFD CTF

Cooling air mass rate is adjusted by the number of cooling tower fan. Figure 7(a) illustrates the number of utilized fans. SGPP has three units of fan, consisting of two 75 kW fans (fan b and c) and a unit of 37.3 kW fan (fan a). In wet bulb temperature range 15°C up to 20°C, both fan a and b should be utilized. Meanwhile, for wet bulb temperature 20°C - 25°C, all fans should be utilized simultaneously.

The impact of VFD installation is compared to constant L/G to find out optimization possibility. Figure 7(b) shows the difference of fan work rate of those two scenarios. The highest energy saving is obtained by VFD when wet bulb 15°C, that is 56.17%. Meanwhile when wet bulb temperature is 25°C, VFD does not provide any impact. Based on figure 7(b), VFD should be installed for wet bulb less than 25°C.

![Figure 7](image)

**Figure 7.** (a) Utility Fan Power on VFD CTF, (b) Utility Fan Power on Constant L/G and VFD CTF
The decision of the best scenario considers not only utility fan power but also gross and net power. Figure 8(a) displays that both net and gross power in constant L/G are higher than VFD CTF. Therefore, VFD does not affect the whole wet bulb temperature. Another consideration for choosing the best scenario is by evaluating the SSC net. SSC is defined as 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 8(b). The highest SSC net difference is more efficient than VFD CTF operation for wet bulb temperature less than 25°C. However, by figure 8(b), it is possible to assume that constant L/G SSC net is higher than VFD CTF operation for wet bulb temperature less than 25°C. The model is developed in wet bulb temperature range 15°C up to 20°C. Through the model developed, the highest fan utilization energy saving is obtained by CTF operation for wet bulb greater than 15°C. For complete verification of this guess, need more complete data are required.

6. Conclusion
Having developed the model and simulated some variables, conclusions are obtained that the developed model of SGPP approaches the real condition based on validation of daily operational data (log sheet). The impact of VFD installation is compared to constant L/G to find out optimization possibility. Through the model developed, the highest fan utilization energy saving is obtained by VFD when wet bulb 15°C, that is 56.17%. However, the decision of the best scenario considers not only utility fan power, but also steam supply and net power which is formulized in term of SSC. The model shows that the SSC net of VFD is higher than constant L/G which concludes that constant L/G operation is more efficient. The model is developed in wet bulb temperature range 15°C up to 20°C where constant L/G is more efficient rather than VFD CTF. Constant L/G gives the highest saving 6.64 % when wet bulb 15°C.

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Acknowledgment
This study was made possible by funding support from Del Institute of Technology under the framework of Internal Research Grant. The authors are also supported by PT. Dizamtra Powerindo, especially Mr. Sofjan Hadi, by providing access to operational supporting data for this research.