Proceedings World Geothermal Congress 2015
Melbourne, Australia, 19-25 April 2015

Kamojang Geothermal Power Plant Unit 1-2-3 Evaluation and Optimization Based on Exergy Analysis

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Keywords: Exergy, efficiency, losses, modification, overall equipment effectiveness.

ABSTRACT

The development of Kamojang Unit 1-2-3 since the 1980’s marked the first utilization of Indonesia’s geothermal field. Kamojang Unit 1-2-3 constantly produces 140,000 kWe with high quality Equivalent Availability Factor (EAF) of 93% on average which comes from 1,000 t/h steam supply.

Within the production period, the Kamojang reservoir pressure and temperature has decreased by around 9.3 bar abs and 19°C from its initial condition which has affected the production rate by a 3% decrement per year.

In terms of the overall sustainability of the Kamojang geothermal field, especially from a power plant perspective, concerns have arisen related to the improvement of power plant efficiency. The exergy analysis method is used to describe and determine the overall system losses.

Based on an exergy analysis, Kamojang Unit 1-2-3 has system losses of about 104,431 kW of exergy and an exergy efficiency of 57.62%. The condenser has the largest loss from the system of more than 30% of exergy.

An improvement to the cooling tower system was estimated where the cooling water outlet temperature was slightly decreased from its existing condition of 33°C to 29.44°C. This can be done by modifying the water inlet flow rate to 90% of its original value (11,160 m3/h) and increasing air inlet flow rate to 110% of its original value (11,107.49 m3/h). The result shows a turbine steam flow rate decrease of about 3.22 kg/s or 11.59 t/h for a single unit.

1. INTRODUCTION

Since the 1980’s, the Kamojang geothermal power plant Unit 1-2-3 has marked a new era of renewable energy development in Indonesia. As the first geothermal power plant in Indonesia, Kamojang Unit 1-2-3 totally has produced a total of 29,617.78 GWh with 93% of Equivalent Availability Factor (EAF) on average. Through its thermal area of 21 km^2 which is known as one among five steam-dominated reservoirs in the world, Kamojang Unit 1-2-3 constantly produces 140,000 kWe to the Java-Bali interconnection grid. In 2008, the additional Unit 4 increased the field production to 200,000 kWe.

In order to maintain this production rate, 77 wells that have been developed since 1976 supply more than 1,500 t/h steam to the unit (Suryadarma et.al., 2005). Based on a recent reservoir simulation, the Kamojang reservoir has its capacity of 230,000 kWe for 25 years from 2012 (Enjinering Kamojang and ITB, 2009). This data was supplied from gravity (1999) and magneto telluric (2009) data delineation.

Over more than 30 years of production, the Kamojang reservoir began to evolve. Decline curve analysis showed that the evolution involved reservoir pressure and temperature reductions. The production decreases around 3% per year which is caused by the reservoir pressure declining by 9.3 bar abs and temperature by 19°C from the initial condition (Suryadarma et.al., 2010).

In terms of production and reservoir evolution, it has become a serious matter for Kamojang geothermal field to sustain and continue its production. From the point of view of power generation, this issue is concerned with several parameter including pressure, temperature, steam flow rate and quality.

Over the years of operation, power generation equipment degrades along with efficiency and effectiveness reductions. The main power generation equipment such as the turbine, generator, condenser and cooling tower will in time undergo degradation due to plant Equivalent Availability Factor (EAF) and Capacity Factor (CF). The demand of the Java-Bali grid that uses geothermal power as a base loader has further pushed the operation of Kamojang Unit 1-2-3.

Based on a thermodynamic perspective, degradation of equipment efficiency and effectiveness will cause losses. The more the losses, the more the steam is required to produce the same amount of electricity. So that, Kamojang Unit 1-2-3 needs a loss assessment in the overall power generation equipment and a way to improve on those losses. Based on the research, the assessment will be done by exergy analysis.

Exergy analysis has been used broadly in order to find out available exergy for the system or equipment. The exergy balance method, widely known as exergy analysis, could identify the equipment’s heat loss quantity and quality (Rosen, 2002).

Thus, concerning reservoir depletion and power generation sustainability, this research intends to focus on overall equipment losses based on exergy analysis and figure the best possible way to increase efficiency and effectiveness. Engineering Equation Solver (EES) will support system calculation and modification.

2. PROCESS FLOW DIAGRAM

The government of Indonesia developed the Kamojang geothermal field based upon a joint operation committee between PT. Pertamina Geothermal Energy (PGE) as the steam supplier and PT. Indonesia Power (IP) as the power producer (Figure 1). In order to generate a total of 140,000 kWe of electric power, PGE supplies steam from around 30 production wells. Those wells produce dry-steam with pressure around 6.7-6.8 bar abs. A pipeline interconnection system, namely Pipe Line 401-404 has the responsibility to deliver steam into the unit.

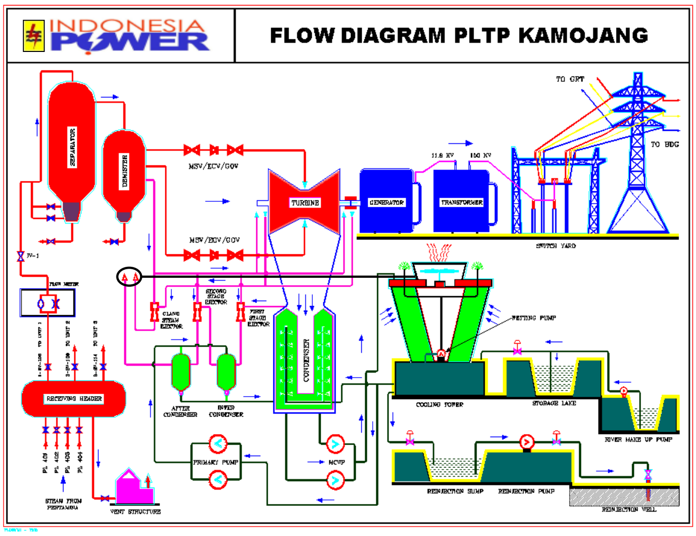

Figure 1. Kamojang Unit 1-2-3 P&I Diagram

The 140,000 kWe power generation requires more than 1,000 t/h of steam supply which is divided into 220 t/h for Unit 1 and around 420 t/h each for Unit 2 and 3. The Kamojang Unit 1-2-3 facility uses a steam receiving header to prevent steam flow fluctuation directly affecting the unit. The steam receiving header is a pressure vessel which is connected to the vent valve system to control steam pressure and flow. The vent valve system has 6-normally open valves which have their own priority to dispose of surplus steam pressure. Normally, PGE will deliver 10% above the minimum steam flow requirement so that the vent valve opening compensates for this amount of flow. Recently, due to the reservoir depletion issue, PGE supplies only slightly above the total steam needed.

Towards the unit, steam flows through a separator which acts as particulate removal. The main function of this separator is to remove solid particles that are carried over by steam. In principle, the separator uses its vessel shape to develop centrifugal forces. To maintain the performance of separator, IP specifies a pressure differential parameter so that pressure loss during operation can be monitored. An orifice is placed after the separator to measure the amount of steam consumption alongside power generation. Yet, this measurement takes main and auxiliary steam flow rate into account.

Afterward, steam flows into a demister to make sure the steam is in dry condition. As the name implies, the demister catches moisture content in the steam by means of turbulence and impact forces between high velocity steam and the demister element. The nature of the collision between the steam and demister element slightly affects steam pressure towards separation of moist and dry steam. Then moisture content will be removed to a flash tank and the steam continues its power generation journey.

Steam is then divided into main steam flow and auxiliary steam. Main steam flows to the turbine through Left-Hand (LH) and Right-Hand (RH) piping. Each of these pipes is equipped with a Main Stop Valve (MSV) that acts as a safety valve in the situation of unit trouble or shutdown and a governor valve which controls steam flow rate to maintain turbine speed.

The turbine expands steam at an inlet pressure of 5.8 bar abs and outlet pressure of 0.12 bar abs on average for all three units. The turbine is coupled with a generator and its speed is set to 3,000 rpm for the purpose of synchronizing to the 50 Hz Java-Bali interconnection grid frequency. The generator produces 11.8 kV and 3,000 A; through a step-up transformer this will be raised to the 150 kV grid voltage. This is the boundary of the Kamojang Unit 1-2-3 facility since transmitting electric power becomes the responsibility of PT. PLN P3B.

Exhaust steam is condensed in the condenser by cooling water from a cooling tower. The condenser has its pressure set to a vacuum condition due to the Gas Removing System (GRS), cooling water temperature and flow rate. Condensate in the condenser hot well will be pumped by 2 x 100% Main Cooling Water Pump (MCWP) to the cooling tower. Condensate that has been cooled down in the cooling tower flows back by gravity and vacuum pressure to the condenser as cooling water.

In terms of unit shutdown, the cooling tower gets its supply water from a river make up pump that stores water temporarily in a storage lake. Some cooling water from the cooling tower flows to the reinjection sump that in time will be delivered to a reinjection well by the reinjection pump. Since 2008, IP no longer uses the mechanism of a reinjection pump because its pressure can be taken over by gravity.

The overall process is thermodynamically analyzed using a heat and mass balance due to energy and exergy on the system. The power generation equipment is described in an Engineering Equation Solver (EES) simulation in order to develop a method to assess the impact of potential modifications with a single click.

2.1 Steam Receiving Header

As has been described before, the steam receiving header is the entrance from the production wells to the power plant. It supports the unit against steam pressure fluctuation. In collaboration with a vent valve system which extracts surplus steam to the environment, the steam receiving header gets steam from four main pipes with more than 1,000 t/h and 6.7-6.8 bar abs.

It is called surplus steam because 10% extra steam is supplied from PGE, which might be used to compensate load fluctuation in the Java-Bali interconnection grid. This 10% surplus can be seen as major losses due to the reservoir depletion and sustainability issue of the Kamojang geothermal field.

The steam receiving header has its safety valve, which is called a rupture disc, in case of emergency due to unit trip, as the vent valve is not quite fast enough to release steam pressure within the required time frame. Some assumptions used in the exergy balance of the steam receiving header:

1. Steam pressure and flow rate from production wells is steady and constant.
2. Process is thermodynamically adiabatic.
3. Vent valve is an output and losses to the system.
4. There are no pressure drops in steam receiving header, just in separator.

Figure 2 gives an illustration of the steam receiving header in coordination with separator and vent valve system.

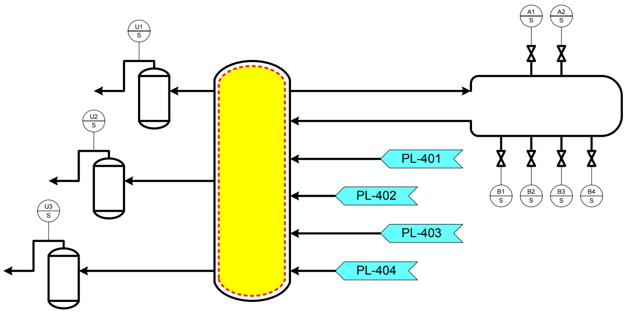

Figure 2. Steam Receiving Header Illustration

The exergy balance for this condition can be described by the equations:

$$\sum E_{in} = \sum E_{out} + \sum I_{SRH} \tag{1}$$

$$\dot{m}_{401}e_{401} + \dot{m}_{402}e_{402} + \dot{m}_{403}e_{403} + \dot{m}_{404}e_{404} = \dot{m}_{U1}e_{U1} + \dot{m}_{U2}e_{U2} + \dot{m}_{U3}e_{U3} + \dot{m}_{VV}e_{VV} + I_{SRH} \tag{2}$$

where *E* (kW) is exergy interaction along input-output of steam receiving header and *I* (kW) stands for irreversibility on steam receiving header. *m* (kg/s) is flow rate data, *e* (kJ/kg) is specific exergy, while *401*, *402*, *403*, *404* designate the main pipelines, and *U1*, *U2*, *U3* are outputs from the unit, *VV* and *SRH* are vent valve and steam receiving header.

The main function of the steam receiving header is to accommodate steam pressure and flow fluctuation in coordination with the vent valve system. From an exergy point of view, the performance of steam receiving header can be considered as the ratio of desired exergy output of steam leaving the unit to exergy from steam entering the steam receiving header.

$$\eta_{SRH} = \frac{\dot{m}_{U1}e_{U1} + \dot{m}_{U2}e_{U2} + \dot{m}_{U3}e_{U3}}{\dot{m}_{401}e_{401} + \dot{m}_{402}e_{402} + \dot{m}_{403}e_{403} + \dot{m}_{404}e_{404}} \tag{3}$$

2.2 Demister

The demister main function is to ensure steam dryness by catching moisture content in the steam formed by pressure drop in the steam pipeline. The demister (Figure 3) has a cone element that will extract saturated steam based on turbulence and impact forces between high velocity steam and the demister element.

The demister has two main outlets which are main steam and auxiliary steam. Main steam will go further to the turbine and auxiliary steam acts as motive steam for ejectors. The demister has its specification in Table 1.

Several assumptions have been made in order to analyze process thermodynamically:

1. Steam pressure and flow rate from production wells are steady and constant.
2. Process is thermodynamically adiabatic.
3. Vent valve is an output and losses to the system.

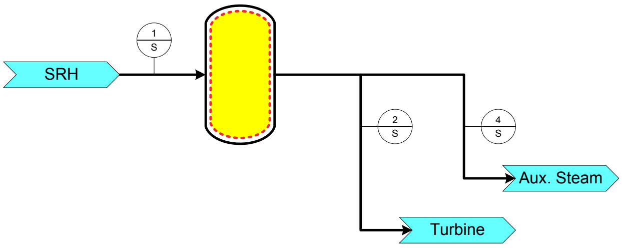

Figure 3. Demister Illustration

The exergy balance on demister can be expressed using this equation:

$$\sum E_{in} = \sum E_{out} + \sum I_{demister} \tag{4}$$

$$\dot{m}_{steam}e_{steam} = \dot{m}_{turb}e_{turb} + \dot{m}_{aux}e_{aux} + I_{dem} \tag{5}$$

E (kW) is exergy inlet and outlet through the demister with irreversibility on the demister symbolized with I_{dem} (kW). *m* (kg/s) stands for flow rate data, *e* (kJ/kg) is specific exergy, while *steam*, *turb*, *aux* designate steam that goes from steam receiving header to turbine and auxiliary steam.

Steam dryness on outlet is the performance measurement of the demister, so it can be stated that demister main function is to maximize the amount of dry steam leaving the outlet.

$$\eta_{demister} = \frac{\dot{m}_{turb}e_{turb} + \dot{m}_{aux}e_{aux}}{\dot{m}_{steam}e_{steam}} \tag{6}$$

2.3 Turbine

Kamojang Unit 1-2-3 uses a double flow geothermal condensing turbine with impulse blades on its first 2-stages. The turbine rotor was machined from a single solid forging material and customized for geothermal use.

A sufficient allowance is provided between the rated speed and critical speed. The most importance thing for a turbine blade is its strength against vibration in the corrosive geothermal steam. Low Pressure (LP) end blading in particular is designed to have natural frequencies free from resonance with any existing forces. Each stage applies a system of effective drain catchers to make sure steam quality is maintained through stages of expansion.

Steam enters the turbine through two main steam pipes, each containing a stop valve and a governing valve located in series in front of the turbine. Exhaust steam from the turbine is led into a condenser which is installed beneath the turbine.

The largest exergy loss happens due to expansion through turbine blade, facilitating conversion into mechanical energy in form of turbine rotation. The impact of this loss is energy reduction from chest pressure to condenser pressure.

Turbine and generator are coupled with fixed coupling that caused both to rotate at the same speed: 3,000 rpm. Mechanically the rotation of turbine and generator has breaking forces caused by excitation load from stator magnetic fields. This rotation is then converted into electrical energy, the amount of which depends on turbine and generator efficiency. Table 1 gives turbine manufacturing data for Kamojang Unit 1-2-3.

Table 1. Turbine Spec. for Kamojang Unit 1-2-3

Parameter	Unit 1	Unit 2-3
Manufacturer	Mitsubishi Heavy Industries	
Nominal capacity	30,000 kW	55,000 kW
Maximum capacity	37,500 kW	57,750 kW
Steam inlet pressure	6.5 bar abs	
Steam inlet temp.	161.9°C	
Steam outlet pressure	0.133 bar abs	0.10 bar abs
Steam flow rate	244,190 kg/h	388,300 kg/h

Some assumptions were taken in order to analyze turbine thermodynamic process:

1. Process is adiabatic (no heat loss).
2. There are no pressure losses due to friction.
3. There are no air leakages into turbine due to gland steam system.
4. Turbine inlet and outlet flow rate are the same.

The equation for exergy balance is:

$$\sum E_{in} = \sum E_{out} + W_{turb} + I_{turb} \tag{7}$$

where E (kW) is exergy inlet and outlet from the turbine with its irreversibility symbolized with I_{turb} (kW). W_{turb} (kW) is turbine work. The relationship between turbine work, generator output (W_{gen}) and generator efficiency (η_{gen}) can be stated as:

$$W_{gen} = W_{turb}\eta_{gen} \tag{8}$$

The principal function of the turbine is to convert exergy of the steam into electricity as efficiently as possible. Considering this, turbine performance will be evaluated based on the ratio between gross generator output to turbine exergy input. The term input exergy involves exergy through expansion from chest pressure to condenser pressure.

$$\eta_{turb} = \frac{W_{gen}}{\sum E_{in}} \tag{9}$$

2.4 Condenser

The Kamojang Unit 1-2-3 condenser is a spray jet direct contact condenser. Basically, condenser construction divides its internal part into a condensation chamber and a gas cooling zone. As the name implies, the condensation chamber is where contact between turbine exhaust steam and cooling water happens in form of condensation. The process occurs with direct contact, which results in condenser pressure dropping into vacuum pressure.

The gas cooling zone uses its trays of nozzles to spread out cooling water and cools down NCG carried by steam. This NCG content is then extracted by the Gas Removing System (GRS).

The condenser and turbine are connected together using an expansion joint in the exhaust steam channel. Cooling water from the cooling tower continually flows to the condenser because of the pressure difference between atmospheric and vacuum pressure. Cooling water flows are controlled by a condenser valve to manage the balance between vacuum pressure, level and condenser inlet-outlet temperature. Cooling water and condensate mixture is kept in the condenser hot well that will be delivered to the cooling tower using 2x100% Main Cooling Water Pump (MCWP).

Condensation involves interaction between exhaust steam with 0.1 bar abs and 48-50°C, cooling water from cooling tower that has temperature around 29-30°C, condensate mixture that will leave condenser, return condensate from inter-aftercondenser and also NCG. Cooling water flow rate should be managed to meet the requirements of the exhaust steam while keeping the balance of condenser pressure and level. Table 2 gives design data for Kamojang Unit 1-2-3 condenser.

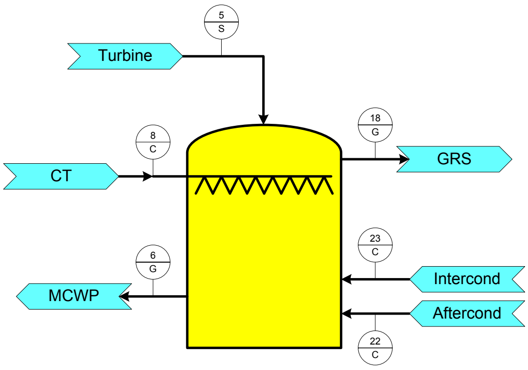

Figure 4. Condenser Illustration

Several assumptions are made to analyze thermodynamic process in condenser:

1. Condenser is not using thermal isolation, so that surrounding air leaks to condenser and it assumes 10% of total extracted steam and NCG by GRS.
2. GRS is able to extract all of NCG content in the steam.
3. GRS is not only carry NCG, but also some part of the steam.

Table 2. Condenser Spec. for Kamojang Unit 1-2-3

Parameter	Unit 1	Unit 2-3
Manufacturer	Mitsubishi Heavy Industries	
Design pressure	0.133 bar abs	0.100 bar abs
Design cooling water temp.	29°C	27°C
Design exhaust steam temp.	49.6°C	42.8°C
Exhaust steam flow rate	232,700 kg/h	376,910 kg/h
NCG flow rate	2,350 kg/h	1,885 kg/h
NCG outlet temp.	32°C	29°C

Exergy balance from the process in condenser can be found using the equation:

$$\sum E_{in} = \sum E_{out} + \sum I_{cond} \tag{10}$$

$$\dot{m}_{turb}e_{turb} + \dot{m}_{CT}e_{CT} + \dot{m}_{inter-after}e_{inter-after} = \dot{m}_{MCWP}e_{MCWP} + \dot{m}_{GRS}e_{GRS} + I_{cond} \quad (11)$$

Where *E* (kW) is exergy inlet and outlet through condenser with I_{cond} (kW) being the irreversibility on condenser. *m* (kg/s) stands for flow rate data, *e* (kJ/kg) is specific exergy, while *turb* stand for exhaust steam from the turbine, *CT* is cooling water from the cooling tower, *inter-after* explains return condensate from the inter-aftercondenser, *MCWP* refers to the condensate to the cooling tower, *GRS* shows extracted NCG by the gas removing system.

The main function of the condenser is to maximize exergy from the exhaust steam and condense steam into condensate using cooling water. The performance of condenser is the ratio of exergy gained by cooling water to the exergy lost by the exhaust steam.

$$\eta_{cond} = \frac{\dot{m}_{CT}(e_{MCWP} - e_{CT})}{\dot{m}_{turbine}(e_{turbine} - e_{MCWP})} \quad (12)$$

2.5 Inter-aftercondenser

The Inter-aftercondeser is an integrated part of the Gas Removing System (GRS) which will extract Non Condensable Gas (NCG) content in the condenser. Steam acts as motive steam for first and second ejector that then withdraw NCG by means of velocity. Motive steam enters the first stage ejector in the condition of sub sonic velocity. Steam pressure is suddenly dropped and velocity begins to increase until it reaches sonic velocity (Mach number = 1) in the nozzle throat when steam comes to the converging part of the nozzle. The cross section difference on the diverging part of the nozzle has the shock wave pressure decrease and steam velocity increase so it reaches supersonic condition. NCG as the entrained fluid is extracted through a suction chamber due to steam velocity changes. Its velocity also increases along with the steam velocity. Motive steam and NCG (gas-steam mixture) interacts within the suction chamber and converging part of the diffuser.

The gas-steam mixture then follows the process through thermodynamic interaction in the intercondenser. The main purpose of the intercondenser is to condense motive steam from the ejector. It uses direct-contact condensation between the gas-steam mixture and primary cooling water. The intercondenser consists of trays and nozzles that make water sprays, expanding the heat transfer area. The process results in NCG which will be extracted by the second stage ejector, and condensate which will flow back to the condenser.

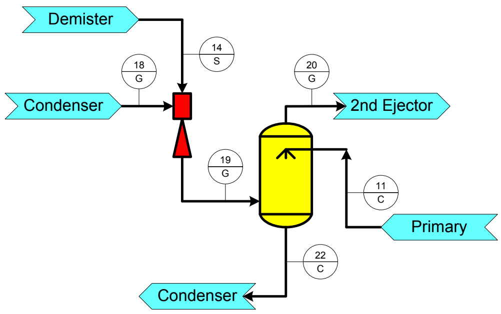

Figure 5. Intercondenser Illustration

Some assumptions have to be used in analyzing the process for the inter-aftercondenser:

1. Condenser air leakage becomes intercondenser cooling load.
2. Inter-aftercondenser are not thermally isolated, so that surrounding air leaks into its vessel and assumes the same amount as it was on heat balance.

The exergy balance on the inter-aftercondenser is shown in the equation:

$$\sum E_{in} = \sum E_{out} + \sum I_{inter-after} \quad (13)$$

$$\dot{m}_{condenser}e_{condenser} + \dot{m}_{primary}e_{primary} = \dot{m}_{inter-after}e_{inter-after} + \dot{m}_{GRS}e_{GRS} + I_{inter-after} \quad (14)$$

m (kg/s) stands for flow rate data, *e* (kJ/kg) is specific exergy, while *cond* stand for exhaust steam from the turbine, *CT* is cooling water from the cooling tower, *inter-after* explains return condensate from the inter-aftercondenser, *MCWP* refers to the condensate to cooling tower, *GRS* shows extracted NCG by the gas removing system.

Similar to the process in the first stage ejector, motive steam is used to remove NCG from the intercondenser and the gas-steam mixture is sent to the aftercondenser. The intercondenser uses primary cooling water to condense motive steam through a direct-contact condensation mechanism. Condensate as a process output flows back to condenser. The cooling tower stack has the responsibility to remove any remaining NCG to the environment. This removal process has to strictly consider environmental regulations with the aim of ecological sustainability.

The Inter-aftercondenser is a pressure vessel equipped with trays which has the main function of distributing cooling water. The mechanism of direct contact condensation within the inter-aftercondenser is highly affected by the quality of cooling water itself. Cooling water for the inter-aftercondenser comes from the primary cooling water system which is extracted from the cooling tower.

The main function of the inter-aftercondenser is to effectively condense NCG entrained steam by maximizing exergy from the NCG. Therefore, the performance of the inter-aftercondenser is related to the ratio of exergy gained by primary cooling water to exergy loss of NCG.

$$\eta_{inter-after} = \frac{\dot{m}_{primary}\left(e_{inter-after} - e_{primary}\right)}{\dot{m}_{condenser}\left(e_{condenser} - e_{inter-after}\right)} \quad (15)$$

2.6 Cooling Tower

Kamojang Unit 1-2-3 uses an induced draft cross flow cooling tower. This cooling tower has a configuration where the air stream flows horizontally through water drops from the top of the cooling tower. Condensate coming from the condenser is delivered by the main cooling water pump to a hot basin in the top of the cooling tower and then distributed to fill grid and fill bar using water nozzles.

In the cooling tower, air and water are intimately mixed to provide heat transfer. Therefore, psychometry is the basis for analysis of heat transfer in cooling tower, especially wet type. Heat transfer in the cooling tower occurs by two major mechanisms which are transfer of sensible heat from water to air (convection) and transfer of latent heat by the evaporation of water (diffusion). Both of these mechanisms operate at the air-water boundary layer. The total heat transfer can also be expressed in terms of the change in enthalpy of each bulk phase. The heat transfer at the boundary layer is equal to the heat transfer in the bulk phase

All of the process components elaborate and define exergy balance on cooling tower as the equation:

$$\sum E_{in} + \sum W_{pump} + \sum W_{fan} = \sum E_{out} + \sum I_{CT} \quad (16)$$

E (kW) are inlet-outlet exergy through cooling tower, W_{pump} (kW) stands for main cooling water pump power, W_{fan} (kW) is fan power requirement and I_{CT} (kW) can be defined as cooling tower irreversibility.

The main responsibility of the cooling tower is to cool down condensate by transferring its exergy content to the surrounding air. If the cooling tower is considered as a heat exchanger, then the aim of the system is to maximize extracted exergy from the condensate by using the air stream. Therefore, the cooling tower efficiency can be written as:

$$\eta_{II,CT} = \frac{\dot{m}_{air\ inlet}\left(e_{air\ outlet} - e_{air\ inlet}\right)}{\dot{m}_{MCWP}\left(e_{MCWP} - e_{condenser}\right)} \quad (17)$$

Thus, cooling tower performance could be shown better with a Coefficient of Performance (COP), which is the ratio of condensate exergy loss to total power consumption of main cooling water pump and cooling tower fan.

$$COP_{CT} = \frac{\dot{m}_{MCWP}\left(e_{MCWP} - e_{condenser}\right)}{W_{pump} + W_{fan}} \quad (18)$$

3. EXERGY ANALYSIS

3.1 Environment as Dead State

The environment has its important role in exergy analysis. It is used as a thermodynamic reference through pressure, temperature and even chemical composition. The environment is assumed to be a very large simple compressible system modelled as a thermal reservoir with a uniform and constant temperature (T_0) and pressure (P_0) (Kwambai, 2005). The environment should not be affected by any means of a process that happens inside of it. The exergy content of a system will be higher if the system deviated to environmental conditions. This study uses the existing Kamojang field condition, which then deviates to the environmental condition. Table 3 shows existing environment condition of Kamojang geothermal field.

Table 3. Kamojang Environment Reference

Parameter	Value	Unit
Altitude	± 1,500	masl
Atmospheric pressure	850	mbar
Atmospheric temp.	17-20	°C
Relative humidity	50	%

3.2 Overall Equipment Effectiveness

Table 4 shows the result of exergy analysis on the overall power generation equipment. The 246,431 kW available exergy of the input steam is converted into electric power of 142,000 kW. This means the overall power generation system has a functional efficiency of 57.6%. The total irreversibility and waste exergy from the system is 47,331 kW, which can be seen as power generation inefficiency.

Table 4. Exergy Calculation Result on Kamojang Unit 1-2-3

Point	Parameter				
	Exergy Input	Desired Exergy Output	Irreversibility	Waste Exergy	Exergetic Efficiency
	kW	kW	kW	kW	%

SRH	246,431	235,391	4,312	6,728	95.52
Demister	236,454	231,038	5,417.2		97.71
Turbine	223,526	157,777	14,722	51,027	70.59
Condenser	2,321	1,030.3	1,282	8,619	44.39
Intercondenser	2,508.9	820,3	1,675.4	13,233	32.70
Afterconderser	74,649	58,297	15,942	9,521	78.09
Overall Exergetic Efficiency	246,431	142,000			57.62

Considering the recent condition of the Kamojang geothermal field, optimizing the power generation system is a must, especially steam flow requirements. Kamojang Unit 1-2-3 has been operated since 1980's which means the probability of system degradation in term of efficiency and optimization is higher.

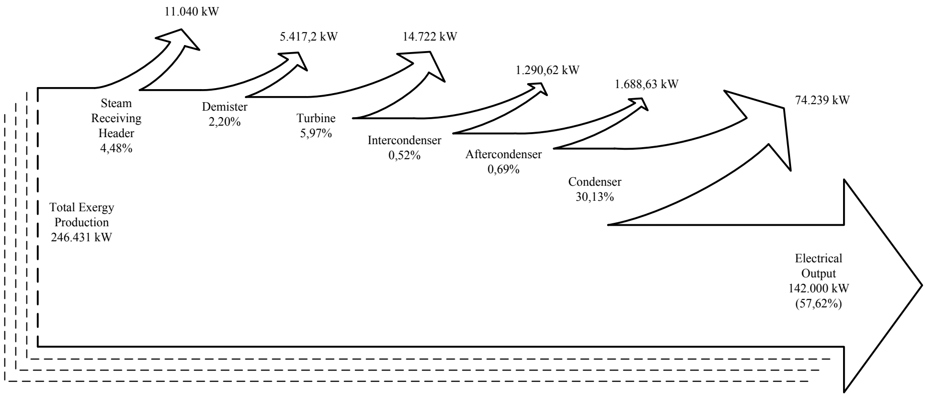

Figure 6. Sankey Diagram of Kamojang Unit 1-2-3

Figure 6 gives the overall losses of Kamojang Unit 1-2-3 using Sankey diagram. 100% (246,431 kW) of available exergy is reduced due to equipment losses. Each equipment in the system, consisting of the steam receiving header, demister, turbine, condenser, inter-aftercondenser and cooling tower, has its own losses due to component degredation or failure that in time contributes to overall power generation losses. The data shows that the condenser has the largest losses with more than 74,000 kW, equivalent to 30.13%. Each unit mentioned above will be analyzed sequentially.

3.3 Steam Receiving Header

The steam receiving header produces losses of 11,040 kW or around 4.48% which comes from the steam pressure venting process in the vent structure. This amount of losses in the vent valve process is an important consideration in the sustainability of the Kamojang geothermal field. The vent valve system should manage to remove surplus steam, which comprises 10% of the 1,100 t/h supply. This steam management system wastes quite a lot of steam. Recent technology applied to Kamojang Unit-4 enables zero venting criteria. This means there is no steam waste in the vent structure during normal unit operation (Agani, 2010). The steam flow rate is directly controlled by a pressure control valve on the steam receiving station that in time will deliver the amount of steam needed to produce 60 MWe.

Irreversibility in the steam receiving header, which wastes 4,312 kW exergy to the environment, could be caused by steam leakage. As it has four main pipe inlets from PL 401-404, four outlets and two safety valves, steam usually leaks from the valve packing, due to its rubber characteristic that in time will become stiff and crack. In order to minimize the losses, IP replaces this packing annually, especially in terms of steam receiving header maintenance. The steam receiving header is common equipment for all three units, so its maintenance requires total shutdown which was recently done in 2010.

From the perspective of efficiency, the steam receiving header system with more than 95% can be considered as good performance. Based on steam pressure along the system, the pressure drop across the steam receiving header and separator can be kept as low as possible. This is due to steam dryness and quality that is maintained strictly by PGE.

3.4 Turbine

A turbine is always an attractive choice for power generation. Available exergy of 223,526 kW is converted into 157,777 kW of electricity. This turbine available exergy has been degraded by around 3.25% compared to the available exergy from the demister output of 231,038 kW. The Main Stop Valve (MSV) and Governor Valve (GV) affect steam pressure drop through valve throttling. The main thermodynamic process occurring in a turbine is expansion. The better the expansion process through the turbine blades, the larger the power output becomes. This explains 14,722 kW of irreversibility that is possibly caused by the expansion across the blades. Steam flows into the turbine stationary and moving blades which will have a direct impact on pressure drop. The blade shape is an important factor on the expansion process. Kamojang Unit 1-2-3 has been operated for more than 25 years; therefore there is a possibility of plastic deformation of the turbine blades.

This deformation is caused by solid particles entrained in the steam, silica deposition and steam pressure drop. Solid particles that cannot be perfectly caught by the separator scratch the turbine blades and in time will slightly change the steam dynamic. Silica in the steam is condensed especially on the 1st and 2nd impulse stage blading due to the large pressure drops. The development of silica deposition will cause a decrease in turbine performance and will affect the unit's ability to produce power. This is expressed in the form of production derating.

The increase in the irreversibility of the expansion process means a decrease in generator power output. This results in a larger steam flow rate needed to maintain nominal production (140,000 kWe). Figure 7 shows the data of unit steam flow rate for the period of 2010 to 2013.

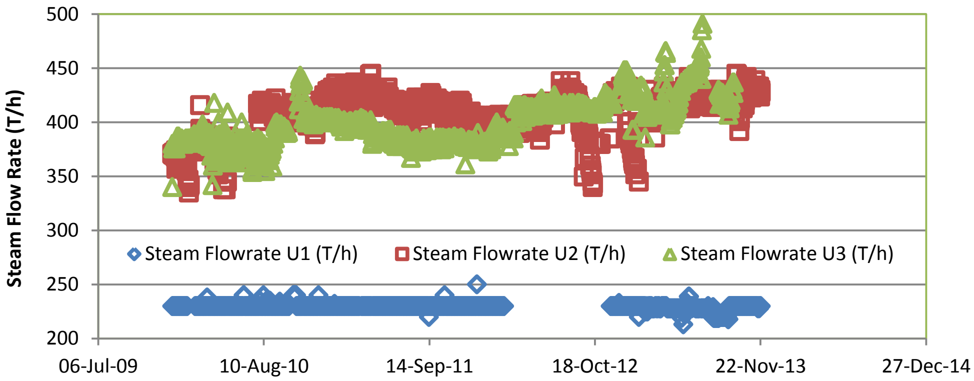

Figure 7. Turbine Steam Flow Rate 2010-2013

Based on the data, the recent operation of Kamojang Unit 1-2-3 has required relatively larger steam flow rates. If the data is compared with design and commissioning data, the steam flow rate has increased. According to the design heat balance, the turbine needs 390.24 t/h of steam in order to deliver 55,000 kWe of generator power, while commissioning data shows that the unit uses 338.5 t/h steam to produce 55,200 kWe. In comparison, the data on November 23rd 2013 showed 428 t/h was needed for Unit 2 to produce the nominal output of 55,000 kWe.

This means steam consumption has increased by more than 30 t/h over the years to produce the same nominal power. This data has highlighted an important aspect of the Kamojang geothermal field sustainability issue. Sustainability from a power generation point of view is a matter of the equipment effectiveness in producing the nominal amount of power with the same or even lower steam flow rate.

3.5 Inter-Aftercondenser

Irreversibillities on the inter-aftercondenser in total remove almost 3,000 kW of exergy. The irreversibility is probably caused by plugging trays in that equipment. The heat transfer mechanism in the inter-aftercondenser is direct contact condensation using the tray's nozzle to distribute cooling water in form of drops. The condensation process involves cooling water from the primary cooling water system, which usually has relatively high concentration of sulphuric compound. NCG and partially carried steam from the ejector interacts with cooling water in the condensation process and produces condensate. Eventually, cooling water also carries sludge from the cooling tower and possibly plugs the inter-aftercondenser tray's nozzle.

Sludge is formed by the combination of the sulphur content in condensate with a relatively convenient water temperature around 30°C. This temperature is suitable for bacteria to grow, with sulphur as a key factor accelerating their growth. The particular bacteria that grow are responsible for sludge formation. IP plans to implement a non-oxidizing biocide in order to control bacterial growth.

Besides chemical means, the inter-aftercondenser tray's nozzle plugging problems could be solved by mechanical cleaning. But to do so, the unit must be in a shutdown condition. Meanwhile, downtime due to a decrease in the inter-aftercondenser performance affects the Kamojang Unit 1-2-3 production rate.

3.6 Condenser

Cooling water from the cooling tower contains 74,649 kW available exergy and produces 58,297 kW desired exergy in form of condensate. Irreversibility in the system of 15,942 kW comes from exhaust steam heat rejection.

According to the condenser's functional efficiency, it is able to condense more than 78% of the exhaust steam, which comprises its main heating load. This result shows a good condenser performance which is potentially a result of the direct contact condensing mechanism of Kamojang Unit 1-2-3. The heat transfer area of the direct contact condenser is substantially affected by spray water size from the condenser nozzle, so it will also impact the overall condeser performance. Nozzles on the condenser are located in two different places: the condensation zone and gas cooling zone's tray.

A condensation problem normally arises from nozzles and trays plugging or blocking. Similar to the situation in the inter-aftercondenser, sludge from the cooling tower enters these nozzles. In the case of nozzles plugging, spray condensation changes to film condensation due to the nozzles' inability to distribute cooling water to spray water. The impact can be seen in the condensate temperature and condenser pressure.

3.7 Cooling Tower

The cooling tower receives condensate in a hot basin which is delivered by 2x100% Main Cooling Water Pump (MCWP) from the condenser. Fills will distribute condensate into water drops in order to expand the heat transfer area. Atmospheric air in the form of wind enters the cooling tower, cools down the condensate and exits as moist air. Condensate that has been cooled down is sent back to the condenser and is used as primary cooling water. Besides its duty to lower the condensate temperature, the cooling tower also acts as the NCG and leakage air final process where they are released into the environment through the cooling tower stack.

The exergy calculation shows Unit 1 has the highest COP of all with 9.7 while Units 2-3 have a COP around 8. This means that Unit 1 with a 3 fan cooling tower is able to cool down condensate temperature more effectively than the other two units. The COP considers the condensate temperature and flow rate, the pump to deliver condensate and the fan power to produce air movement. Unit 1 manages to lower condensate temperature using the lowest pump and fan power.

The cooling load for the cooling tower comes from the condensation requirement which directly affects condensate temperature and flow rate. Unit 1 produces less power than Unit 2-3, so steam and cooling water flow rate are necessarily also lower. Cooling tower performance due to condensate temperature depends on condensate flow rate, temperature and cooling air conditions (in terms of temperature, enthalpy, humidity ratio and specific volume). These parameters are then used to describe the cooling system on the cooling tower.

Furthermore, the cooling tower fan power is affected by the cooling air flow rate. This flow rate can be adjusted by setting the pitch angle of the cooling tower fan. Unit 1 has the minimum angle so that the power can be reduced.

4. POWER GENERATION OPTIMIZATION

According to overall equipment effectiveness based on exergy analysis, power generation performance is directly affected by condenser cooling water temperature. The lower this temperature becomes, the higher the condenser vacuum pressure will be. This temperature will also affect the condensation process in the inter-aftercondenser so that the overall performance of GRS should increase. As has already been mentioned, the cooling water temperature depends on the cooling tower performance. Therefore, cooling tower performance must be increased in order to optimize the power generation system.

This research only considers minor equipment modification to get higher cooling tower performance. One thing to keep in mind is that the recent operation conditions are quite different compared with design conditions in terms of the surrounding environment. The design 27°C of cooling water temperature can be easily achieved with cooling air temperature of around 12-15°C. But due to an increase in the surrounding temperature, the lowest cooling air temperature could only reach 19-22°C. An increase in the cooling tower performance may require equipment replacement to counter these changes.

In terms of steam supply sustainability, optimization means the same amount of power generation could be achieved with less steam supply. If the cooling tower can lower the cooling water temperature, it will lower the enthalpy of turbine steam exhaust by lowering condenser pressure. A calculation will be done upon Unit 2 as an example for the other two units.

Daeil Aqua Corporation (2003) gives two alternative methods of optimizing cooling tower performance without any equipment replacement. These are condensate flow rate management and cooling air adjustment. On Kamojang Unit 1-2-3, condensate flow rate is the responsibility of MCWP discharge valves, while fan pitch blade can be adjusted to get the desired cooling air flow rate.

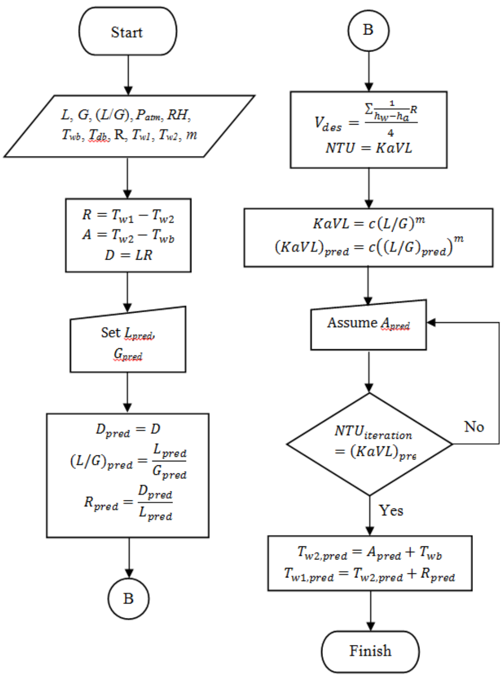

Figure 8. Modification Flowchart

Adiprana (2013) tests the possibilities of those two options. Condensate and cooling air flow rate are changed from 90% to 110% of the operation conditions. The results deliver a deep understanding about this option. Increasing condensate flow rate for more than 100% will decrease cooling water temperature from cooling tower. The tower characteristic will change due to the additional condensate flow rate. This tower characteristic could be considered as a heat transfer Number of Transfer Unit (NTU) which will decrease along with the tower characteristic. Conversely, an increase in the tower characteristic occurs as condensate flow rate is decreased to 90%.

The effect of air flow rate adjustment is different than condensate management. The cooling water temperature will be lowered as the fan blade pitch is increased due to more air entering the cooling tower but it will decrease as fan blade pitch is lowered. The increase comes from a 100% to 110% adjustment which corresponds to a fan blade pitch change from 8.3° to 9.13°, while the manual book gives 10-11° as a fan blade pitch reference. The option results in an increased power generation parasitic load because along with fan blade pitch increment, fan power will increase.

This research points out the coordination of condensate decrement together with fan blade pitch increment. According to Adiprana (2013), the target of cooling water temperature can be achieved with the equivalent condensate flow rate decrease to less than 100% (12,400 m3/h) or an increase in cooling air flow rate by adjusting fan blade pitch to more than 100% (around 8°) to produce a cooling air inlet flow rate of 10,097.72 m3/h. This research demonstrates the combination of managing condensate flow rate from 95-90% and simultaneously adjusting fan blade pitch from 105-110% with stepwise calculation, shown in Figure 8.

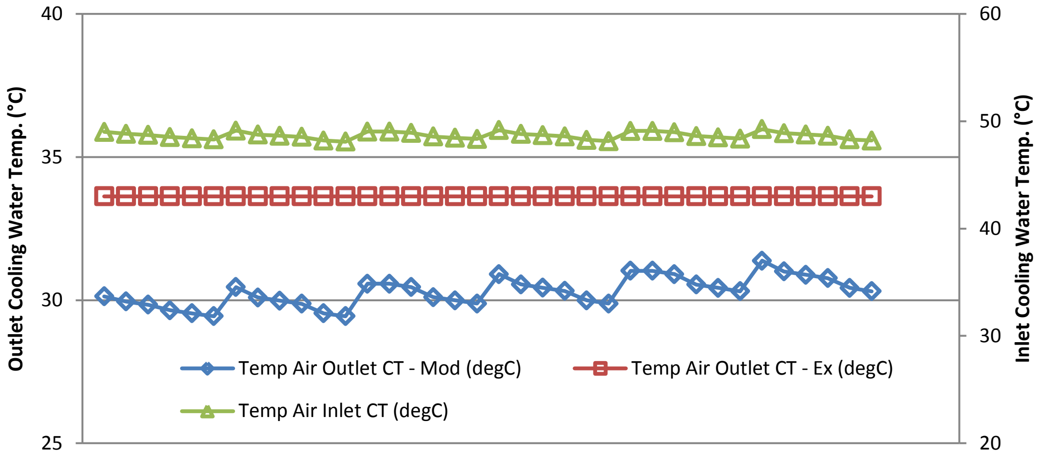

Figure 9. Modification Impact on Cooling Water Temp.

Figure 9 plots the calculation result for fan blade pitch modification (from 105-110%) and condensate flow rate decrement (from 95-90%) as blue dots while the brown mark is the existing cooling water temperature (33°C) and the green triangle stands for condensate inlet temperature from condenser. The graph shows that the lowest cooling water could possibly be reached using the lowest condensate flow rate (90%). Thus, the cooling water temperature is also increased due to the increase in air flow rate caused by higher fan blade pitch variation (110%). In comparison with the existing condition, cooling water temperature could best reach 28.81°C, or an increase of 4.19°C, with the combination of 90% condensate flow rate and 110% of air flow rate. The increment of the tower characteristic from the parameter adjustment will increase the cooling tower Number of Transfer Units (NTU).

Mitsubishi's commissioning data states the cooling tower fan blade pitch could be set around 10-11°. Maintenance data of existing fan blade pitch shows that recent IP regulation on parasitic load makes fan blade pitch reduce to around 8-9°. This modification is possible but may be strictly applied due to regulations. Based on the data on Table 7, fan blade pitch should be 9.7° in order to get air flow rate increment to 110% (11,107.49 m^3/h).

Table 5. Standard Fan blade pitch Setting

Angle	Average/Total					
	Current	Flow	Power	Vibration		
	Amp	m^3/h	kW	V	A	H
8,3	191,82	10.000,1	549,196	0,66	1,22	1,36
10	220	13.024,8	629,878	14,2	10,4	12
11	236,6	13.439,9	677,405	38,2	12,2	22

On the other hand, a decrease in condensate flow rate might be the result of controlling the Main Cooling Water Pump (MCWP) discharge valves. These valves are 2x100% parallel operating valves which maintain the amount of condensate delivered to cooling tower based on condenser vacuum and level. These valves should be set at 38.6% in order to achieve 90% of condensate flow rate decrement, which is equal to 11,160 m3/h.

5. IMPROVEMENT

The aim of modification in terms of sustainability is to achieve nominal power generation capacity with less steam supply. Cooling tower modification shall be effective in reducing the steam requirement. The cooling tower improvement impact could be seen based on the effect of this temperature to the condenser system performance. The condenser system which consists of exhaust steam, cooling water inlet, condensate outlet, inter-aftercondenser return and NCG extraction feels the difference due to an increase in cooling water temperature. The condensation process on the inter-aftercondenser shall improve with the changes. This is because the cooling water for the inter-aftercondenser comes from the condenser cooling water extraction.

Based on an EES simulation on condenser system, the most affected parameter of an increase in cooling tower effectiveness is the condenser vacuum pressure. As the condenser vacuum increases, turbine exhaust steam enthalpy will drop lower than its existing value. Thus, the enthalpy difference between the turbine inlet and outlet increases. Basically, the steam requirement shall be reduced as a result of this increase in enthalpy difference. It means that the required steam flow rate to deliver the same amount of power generation is lower.

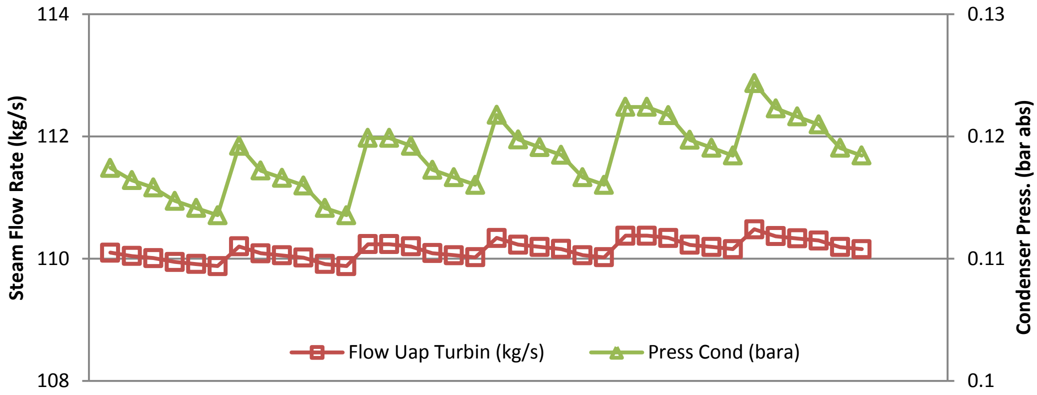

Figure 10. Modification Impact on Steam Flow Rate

Figure 13 shows the calculation result where condenser vacuum pressure changes due to the decrease in cooling water temperature. According to the graph, the condenser vacuum pressure will fluctuate to produce the nominal power generation output of 55,000 kWe and the impact of this is the variation of steam turbine supply. The best condition achieved is the one with the lowest cooling water temperature of 29.44°C which produces a condenser vacuum pressure of 0.114 bar abs, so that the turbine steam supply could be reduced to 109.88 kg/s. If it is compared with the existing turbine flow rate, which needs 113.1 kg/s, the turbine steam supply can be reduced by around 3.22 kg/s or 11.59 t/h for a single unit.

Operational data shows more than 1,000 t/h of steam supply is required to produce 140,000 kWe. If it is assumed that the entire plant has the same reduction in steam supply of around 11.59 t/h, the total steam required will decrease to 1,025.23 t/h. This results in the power generation exergetic efficiency increasing from 57.62% into 59.49%. This is caused by an increase in overall equipment effectiveness due to a slight change in cooling water temperature.

6. CONCLUSION

(1) Losses mapping on the Kamojang Unit 1-2-3 power generation equipment indicates condenser irreversibility as the largest loss, with a value of 15,942 kW. Based on equipment functional efficiency (exergetic efficiency), almost all of the equipment has a high efficiency value, except for the inter-aftercondenser which depends on cooling water cleanliness.

(2) Minor optimization was done with a change of 12,400 m^3/h condensate flow rate to 11,160 m^3/h by using the MCWP discharge setting and in combination with modification of fan blade pitch on the cooling tower which delivers 11,107.49 m^3/h of cooling air.

(3) The improvement is solid. Condenser vacuum pressure is decreased from 0.1362 bar abs into 0.1136 bar abs, which reduces the steam supply requirement by more than 11 t/h per unit to maintain nominal power generation capacity.

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