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BASIC DESIGN OF LUMUT BALAI 2×55 MW GEOTHERMAL POWER PLANT, INDONESIA

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ABSTRACT

This report covers research activities with the objective of producing basic design documents for the Lumut Balai geothermal power plant units 1 and 2, with a total electrical output of 2×55 MWe. The methodology to accomplish this objective is to initially create a geothermal power plant model, its application through simulations and a technical analysis. Once the model is set up, then the simulation can be carried out in order to understand the performance of the geothermal power plant when responding to several different operating conditions. The following operating conditions are used as variable input parameters in the presented power plant model, wet bulb temperature and non-condensable gas content of the steam supply. In addition the behaviour of the geothermal power plant, both for rated full load operation as well as partial load operation, is simulated for a comprehensive analysis of a heat and mass balance diagram and the technical specifications of the main equipment.

1. INTRODUCTION

Lumut Balai geothermal field on the island of Sumatera in Indonesia is part of the PT Geothermal Energy working area. The project comprises the development of upstream (steam field, steam gathering and reinjection system) and downstream facility construction (geothermal power plant) such that PT Pertamina Geothermal Energy can sell electricity to the national network grid company. This technical report focuses on the basic design of the geothermal power plant itself. The first step in the design procedure is to model the geothermal power plant using EES (Engineering Equation Solver), computer aided thermodynamic simulation software, followed by running simulations under several different conditions. After the heat balance diagrams are set up for different operating conditions, then the main equipment can be sized properly according to the required margins and matched with the requirements dictated by the selected thermodynamic cycle which, in this case, is a single-flash condensing Rankine cycle. The main equipment and systems include: a steam gathering and reinjection system using a steam turbine and a condenser; a gas removal system, including a hybrid system and a dual stage steam jet ejector; and a circulating water system including a circulating water pump and a cooling tower.

2. THE LUMUT BALAI GEOTHERMAL FIELD

2.1 Location and planned drilling activity

Drilling activity in Lumut Balai geothermal field (Figure 1) started in mid 2008, with well LMB 1-1 being the first well drilled. It is an exploration well and its profile is a straight-standard hole, but successive wells will be directional big holes. The first phase of geothermal development in Lumut Balai includes the drilling of 17 wells, both for production and reinjection. The steam produced will supply a two unit geothermal power plant with a total capacity of 2×55 MWe. PT Pertamina Geothermal Energy will sell electricity to the national network grid company through an energy sales contract. This kind of energy selling scheme is the second for a geothermal area owned and operated by PT Pertamina Geothermal Energy. The first one was for the Kamojang geothermal power plant unit 6, located in the West Java province, which sells a total of 60 MWe to the national grid company.

The Lumut Balai geothermal project is located on Sumatera Island, South Sumatra Province, Penindaian village, sub-district of Semende Darat Laut, Muara Enim regency, about 292 km southwest of Palembang. The project area can be reached by four-wheel drive vehicles via the asphalted road from Palembang to Simpang Meo and onwards to the project site in Penindaian village, approximately 32 km on gravel and some paved roads. The geothermal prospect is sited around Mt. Balai, Mt. Lumut and Mt. Pagut. The average altitude of the geothermal field location is around 1000 m above sea level (m a.s.l.). The Lumut Balai geothermal power plant project occupies an area which is planned for geothermal power plant units 1 & 2 and also for units 3 & 4 which will be developed in the next phase. The production wells are distributed on three wellpads. Reinjection wells are located at two different wellpads.

2.2 Basic design documents

Basic design documents are needed to ensure end-product quality and a budget and schedule that match project planning. Any defects in the product (in this case the geothermal power plant), such as excessive budget realisation and delays during project execution can lead to negative effects such as an increased financial burden borne by the company for a long term period.

Since the process of developing the geothermal power plant will involve several parties, e.g. an engineering consultant and an Engineering-Procurement-Construction contractor, the need for basic design documents becomes mandatory.

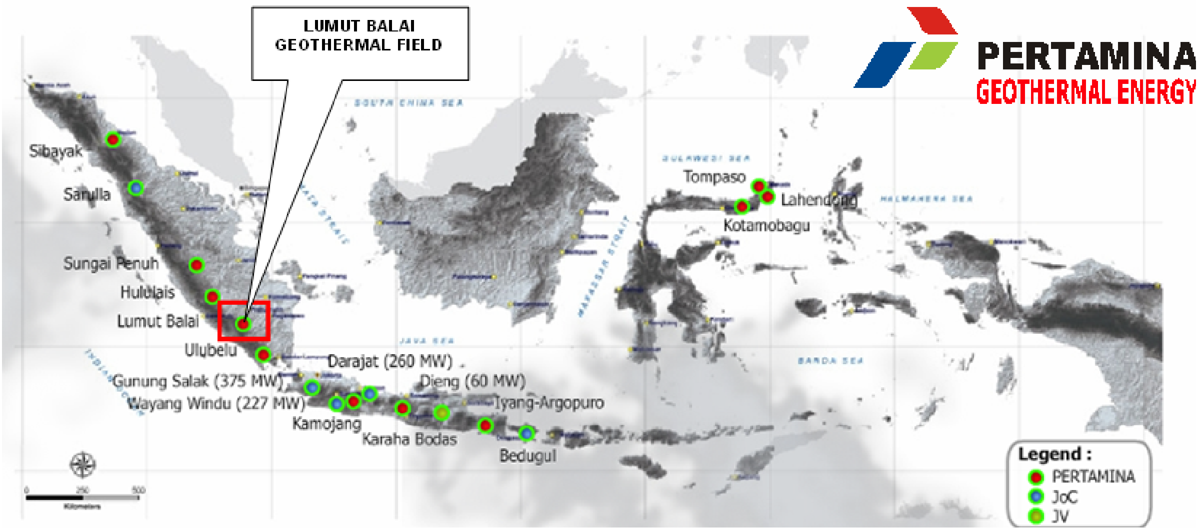

FIGURE 1: Location of Lumut Balai geothermal field,
PT Pertamina Geothermal Energy working area (PGE, 2010)

This research is intended to produce the following basic design documents for Lumut Balai geothermal power plant, units 1 & 2:

1. Heat and mass balance diagram; and
2. Technical specification for main equipment.

2.3 Power plant planning and design

The process of developing a new power plant from its inception to commercial operation is complex and dynamic. The power plant planning and design process described in this report is tailored to geothermal power plants. The basic steps are shown in Figure 2. The power plant design process changes depending on unique financial, engineering, environmental, and other requirements for a specific plant. One approach to the power plant design process is to design by function or system, purchase by component, construct by specialty contractor, and start up by system. Each of these steps is required in some form by all power plant designers. It is vital that the goals, objectives, and constraints for each project be carefully defined in the planning and analysis stage. Project planning and analysis encompass those strategic elements of a project that must be considered early in project development. Steam supply studies, system planning studies, site evaluation, transmission planning analyses, environmental feasibility analyses, and economic and financial feasibility analyses are integral to project planning and analysis for new geothermal power generation facilities.

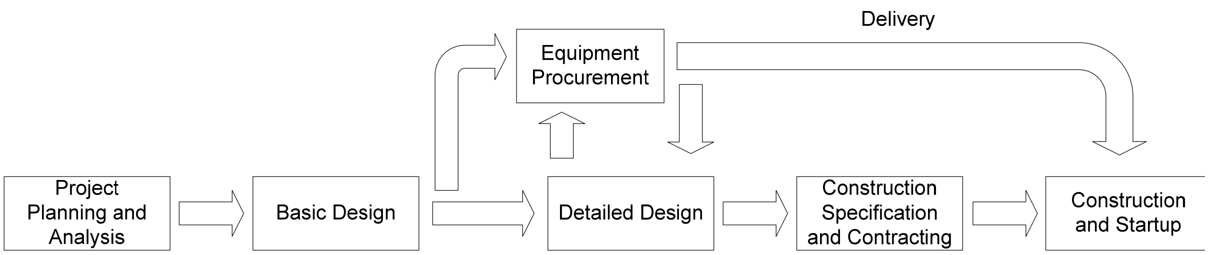

FIGURE 2: Power plant design process (Drbal et al., 1996)

The basic design stage encompasses a broad variety of activities. It consists of systematically defining and evaluating the basic conditions and constraints applicable to a specific system of an electricity generation plant. Basic design engineering starts as part of the project planning and analysis activities. The activities include thermodynamic cycle selection, creating a model of the power plant thermodynamic system, creating a heat and mass balance diagram, determining the performance of the power plant thermodynamic cycle and determining technical specifications for the main equipment. A detailed design phase includes determining the technical requirements for all plant components. It involves detailed consideration of equipment sizing, reliability constraints, and performance requirements for individual equipment, codes and standards, all directed toward successful specification, construction, and start-up. As the detailed engineering is completed for systems and equipment, procurement and construction specifications are developed to delineate specific technical and commercial requirements consistent with the overall design objectives. Finally, effective control of schedules, cost, design, and construction is essential to a successful power plant project. Project control activities include critical path scheduling of engineering and plant construction activities, cost control and cost risk assessment, design control, and construction control (Drbal et al., 1996).

3. BASIC THEORY

3.1 Heat and mass balance

The basic design for a geothermal power plant is required for supporting the development process for both a steam gathering system and the geothermal power plant. One of the most important documents in the basic design phase is a process flow diagram describing the heat and mass balance. This kind of diagram depicts the processes taking place in the geothermal power plant system. Two kinds of balances are represented in a heat and mass balance diagram: the balance of mass and the balance of energy. Some conditions shall be simulated when modelling the geothermal power plant in order to understand the effect of changing process parameters. Once the heat and mass balance diagram is fixed, then the requirements for steam supply and other stream mass flow can be confirmed; then subsequent design activities such as main equipment sizing can commence.

A heat and mass balance diagram is one of the key technical documents in the basic engineering design phase of geothermal power plant development and it is also used during subsequent phases, including Engineering-Procurement-Construction (EPC) contractor bidding, detailed design, field construction and the commissioning process. Furthermore, the heat and mass balance diagram is used during the operation and maintenance of the geothermal power plant in order to maintain the performance and allow continuous improvement during the commercial lifecycle of the geothermal power plant.

3.2 Steam supply pipeline system design

Pressure drop in a steam supply pipeline system:
Approximating the pressure drop value in a steam supply pipeline system from the separator to the power plant can be calculated using the well-known 'Babcock' or 'Guttermuth and Fischer' formula (Armstead, 1983):

$$\Delta p = 8.73 \cdot 10^{-8} (1 + \frac{0.0914}{d}) \frac{LVw^2}{d^5} \tag{1}$$

where Δp = Pressure drop (bar);
L = Pipe length (m);
V = Specific volume of steam (m^3/kg);
d = Internal diameter of pipe (m); and
w = Mass flow (kg/s).

This formula is applicable for dry saturated steam or steam with the presence of small quantities of liquid. For very wet water/steam mixtures, i.e. two-phase flow transmission, the formula gives too high a result.

The permissible pressure drop between the assumed economic well head pressure and the designed inlet pressure to the power plant must not be exceeded. Where a wellhead separator is installed, it would be wise to allow a pressure drop of about 10% of the absolute wellhead pressure to be absorbed in the separator and its associated pipework (Armstead, 1983).

Steam transmission velocity:
Steam velocity should not exceed the value determined by the following equation (Armstead, 1983):

$$v = \frac{93.03}{p^{0.54}} \tag{2}$$

where v = Steam velocity (m/s); and
p = Steam pressure (bar).

This formula is recommended by Russell James (control orifices replace steam traps on overland transmission pipelines) as the definition of 'moderate' velocity below which the rivulet of condensate flowing along the bottom of a steam pipe is considered to be more or less immune from being swept along in gulps by the faster moving steam. Apart from the risk of water-hammer, the re-entrainment of water caused by excessive steam velocity is apt to carry the water past the next downstream collection pot and so escape condensate removal. The reduced condensate removal efficiency would mean that the degree of purification would be far less than required. So long as the velocity is restricted to the value determined by the formula, a condensate removal efficiency of at least 70% for each collection pot (drain/steam trap) should be ensured (Armstead, 1983).

3.3 Thermodynamic cycle

Single flashing process:
Referring to Figure 3, the processing sequence begins with geofluid under pressure at state 1, close to the saturation curve. The flashing process is modelled at constant enthalpy, i.e. an isenthalpic process, because it occurs steadily, spontaneously, essentially adiabatically, and with no work involvement. We also neglect any change in the kinetic or potential energy of the fluid as it undergoes flashing. Thus, we may write $h_1 = h_2$, where h denotes specific enthalpy and the subscripts refer to the states shown in Figure 3.

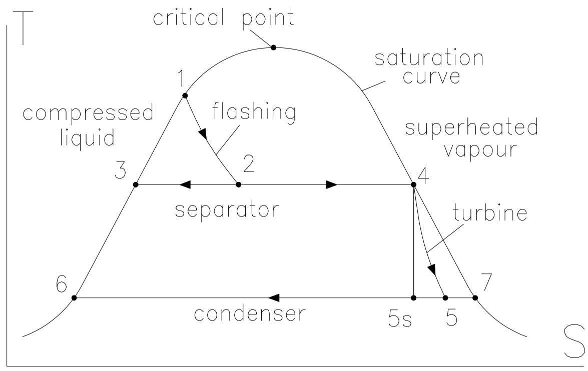

FIGURE 3: Temperature-entropy state diagram for a single-flash plant (DiPippo, 2005)

Separation process:
The separation process is modelled at constant pressure, i.e. an isobaric process, once the flash has taken place. The quality or dryness fraction, x, of the mixture that forms after the flash, state 2, can be found from:

$$x_2 = \frac{h_2 - h_3}{h_4 - h_3} \tag{3}$$

by using the so-called lever rule from thermodynamics. This gives the steam mass fraction of the mixture and is the amount of steam that goes to the turbine per unit total mass flow into the separator.

Optimum separator temperature: An approximation formula
Regarding the process diagram shown in Figure 3, the optimum value for the separator temperature is given approximately (DiPippo, 2005) as:

$$T_{3,opt} = \frac{T_1 + T_6}{2} \tag{4}$$

Since this rule indicates that the temperature range between the reservoir and the condenser is divided into two equal segments, this rule is sometimes called the "equal-temperature-split" rule. This approximate rule applies to all flash plants regardless of the number of flashes. For a double-flash plant, the rule says: the temperature difference between the reservoir and the first flash is equal to the temperature difference between the first flash and the second flash, and is also equal to the temperature difference between the second flash and the condenser.

Turbine expansion process:
The work produced by the turbine per unit mass of steam flowing through it, is given by:

$$w_t = h_4 - h_5 \tag{5}$$

assuming no heat loss from the turbine and neglecting the changes in kinetic and potential energy of the fluid entering and leaving the turbine.

The maximum possible work would be generated if the turbine operated adiabatically and reversibly, i.e. at constant entropy or isentropically. The process shown in Figure 3 from 4-5s is the ideal process. We define the isentropic turbine efficiency, η_t, as the ratio of the actual work to the isentropic work, namely:

$$\eta_t = \frac{h_4 - h_5}{h_4 - h_{5s}} \tag{6}$$

The power developed by the turbine is given by:

$$\dot{W}_t = \dot{m}_5 w_t = x_2 \dot{m}_{total} w_t \tag{7}$$

This represents the gross mechanical power developed by the turbine. The gross electrical power will be equal to the turbine power times the generator efficiency:

$$\dot{W}_e = \eta_g \dot{W}_t \tag{8}$$

All auxiliary power requirements for the plant must be subtracted from this to obtain the net, sellable power. These so-called parasitic loads include, but are not limited to, all pumping power and cooling tower fan power.

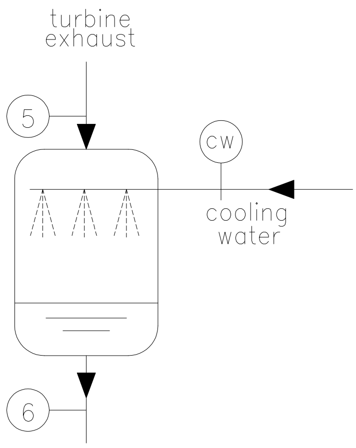

FIGURE 4: Direct contact condenser (DiPippo, 2005)

Condenser:
The primary purpose of the condenser is to condense the exhaust steam from the turbine. There are primarily two types of condensers: direct-contact and surface condensers. It is common to employ a direct-contact condenser for geothermal power plant applications, especially if there is a limited cooling water source, such as for geothermal power plants built in the highlands. A modern direct-contact condenser is of the spray type; early design was of barometric or jet type (El-Wakil, 1984).

A direct-contact condenser (Figure 4), as the name implies, condenses the steam by mixing it directly with the cooling water. This is done by spraying water into the steam in the condenser. Thus turbine exhaust steam mixes with cooling water to produce nearly saturated condensate. A mass balance on the system, where $\dot{m}$ denotes mass-flow rates, gives:

$$\dot{m}_6 = \dot{m}_5 + \dot{m}_{cw} \tag{9}$$

An energy balance gives:

$$\dot{m}_6 h_6 = \dot{m}_5 h_5 + \dot{m}_{cw} h_{cw} \tag{10}$$

From the above equations, the mass flow rate of the cooling water can be determined as follows:

$$\dot{m}_{cw} = \dot{m}_5 \left(\frac{h_5 - h_6}{h_6 - h_{cw}} \right) \tag{11}$$

Cooling tower:
The cooling tower must be designed to accommodate the heat load from the condensing steam. With reference to Figure 4, the steam condensate that has been pumped from the hot well of a condenser is sprayed into the tower (Figure 5) where it falls through an air stream drawn into the tower by a motor-driven fan at the top of the tower.

The ambient air enters with a certain amount of water vapour, determined by its relative humidity, and picks up more water vapour as the condensate partially evaporates. The evaporation process requires heat that comes from the water itself, thereby dropping its temperature.

The internal process involves the exchange of both heat and mass between the air and the water. Energy balance and mass balance for the cooling tower system shall be applied in order to determine how much mass flow of outside air is required. This value will be used later to determine cooling tower induced draft fan motor power requirements.

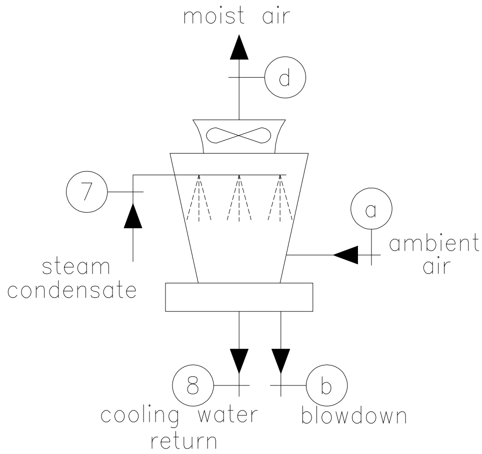

FIGURE 5: Mechanically induced draft wet cooling tower (DiPippo, 2005)

Energy balance for the cooling tower must take into account the water content of the incoming and leaving air streams:

$$(\dot{m}_a h_a + \dot{m}_{wa} h_a) + \dot{m}_7 h_7 = (\dot{m}_d h_d + \dot{m}_{wd} h_d) + \dot{m}_8 h_8 + \dot{m}_b h_b \tag{12}$$

There are two other equations needed to analyse the process: mass conservation of water and mass conservation of air. Recall that both the entering and leaving air streams contain water in the vapour phase (in different percentages). The water conservation equation is:

$$\dot{m}_{wa} + \dot{m}_7 = \dot{m}_{wd} + \dot{m}_8 + \dot{m}_b \tag{13}$$

The dry air goes through the cooling tower unchanged. The dry air conservation equation is:

$$\dot{m}_{ad} = \dot{m}_{aa} = \dot{m}_a \tag{14}$$

For a unit mass of dry air:

$$\left(h_a + \frac{\dot{m}_{wa}}{\dot{m}_a} h_a\right) + \frac{\dot{m}_7}{\dot{m}_a} h_7 = \left(h_d + \frac{\dot{m}_{wd}}{\dot{m}_a} h_d\right) + \frac{\dot{m}_8}{\dot{m}_a} h_8 + \frac{\dot{m}_b}{\dot{m}_a} h_b \tag{15}$$

where the terms $\dot{m}_{wa}$ and $\dot{m}_{wd}$ represent the water content of the incoming and leaving air streams, respectively.

These can be found from the specific humidity, ω, of the air streams:

$$\dot{m}_{wa} = \omega_a \dot{m}_a \tag{16}$$

and

$$\dot{m}_{wd} = \omega_d \dot{m}_d \tag{17}$$

which gives:

$$h_a + \omega_a h_a + \frac{\dot{m}_7}{\dot{m}_a} h_7 = h_d + \omega_d h_d + \frac{\dot{m}_8}{\dot{m}_a} h_8 + \frac{\dot{m}_b}{\dot{m}_a} h_b \tag{18}$$

Re-arranging Equation 18, gives the following form:

$$\omega_a h_a + \frac{\dot{m}_7}{\dot{m}_a} h_7 = (h_d - h_a) + \omega_d h_d + \frac{\dot{m}_8}{\dot{m}_a} h_8 + \frac{\dot{m}_b}{\dot{m}_a} h_b \tag{19}$$

The dry air going through the cooling tower is unchanged. The circulating water loses mass by evaporation. The water vapour in the air gains mass due to evaporated water:

$$\omega_d - \omega_a = \frac{\dot{m}_7}{\dot{m}_a} - \frac{\dot{m}_8}{\dot{m}_a} - \frac{\dot{m}_b}{\dot{m}_a} \tag{20}$$

For incoming air and leaving air, the following approximation can be used due to the low pressure and temperature commonly encountered in the tower:

$$(h_d - h_a) = c_p (T_d - T_a) \tag{21}$$

Equation 19 can now be written in the form:

$$\omega_a h_a + \frac{\dot{m}_7}{\dot{m}_a} h_7 = c_p (T_d - T_a) + \omega_d h_d + \left(\frac{\dot{m}_7}{\dot{m}_a} - \frac{\dot{m}_b}{\dot{m}_a} - (\omega_d - \omega_a) \right) h_8 + \frac{\dot{m}_b}{\dot{m}_a} h_b \tag{22}$$

where ω_a = Specific humidity of cold air entering cooling tower;
ω_d = Specific humidity of hot air leaving cooling tower;
h_a = Enthalpy of cold dry air entering cooling tower (kJ/kg);
h_d = Enthalpy of dry air leaving cooling tower (kJ/kg);
h_7 = Enthalpy steam condensate (hot water) entering cooling tower (kJ/kg);
h_8 = Enthalpy of cold water (cooling water return) leaving cooling tower (kJ/kg);
h_b = Enthalpy of blowdown portion leaving cooling tower (kJ/kg);
$\dot{m}_7$ = Mass flow of hot water (steam condensate) entering cooling tower (kg/s);
$\dot{m}_a$ = Mass flow of cold air entering cooling tower (kg/s);
$\dot{m}_b$ = Mass flow of blowdown portion leaving cooling tower (kg/s);
c_p = Specific heat of dry air (kJ/kg°C);
T_d = Temperature of air leaving cooling tower (°C); and
T_a = Temperature of air entering cooling tower (°C).

With the respect to Equation 11, the mass flow rate of $\dot{m}_7$ is equal to $\dot{m}_{cw}$ so the mass flow rate of dry air, $\dot{m}_a$, which enters and leaves the cooling tower can be estimated.

Range and approach of the cooling tower:
A cooling tower is also characterised by two other parameters: the *range* and *approach*. The *range* is the change in water temperature as it flows through the cooling tower, namely T_7-T_8. With reference to Figure 5, the *approach* is the difference between the water outlet temperature and the wet-bulb temperature of the incoming air, namely, T_8-$T_{wb,a}$.

Cooling tower height:
Contact time between water and air is governed largely by the time required for the water to discharge from the nozzles and fall through the tower to the basin. The contact time therefore becomes a function of the height of the tower. Should the contact time be insufficient, no amount of increase in the ratio of air to water will produce the desired cooling. It is, therefore, necessary to maintain a certain minimum height of a cooling tower. When a wide approach of 8-11°C to the wet-bulb temperature and a 13.9-19.4°C cooling range are required, a relatively low cooling tower will suffice. A tower in which the water travels 4.6-6.1 m from the distributing system to the basin is sufficient. When a moderate approach and a cooling range of 13.9-19.4°C are required, a tower in which the water travels 7.6-9.1 m is adequate. Where a close approach of 4.4°C with a 13.9-19.4°C cooling range is required, a tower in which the water travels 10.7-12.2 m is required. It is usually not economical to design a cooling tower with an approach of less than 2.8°C (Perry and Green, 2008).

Cooling tower makeup water requirement:
The makeup requirements for a cooling tower consist of the summation of evaporation loss, drift loss, and blowdown. Therefore:

$$\dot{m}_{mu} = \dot{m}_e + \dot{m}_d + \dot{m}_b \tag{23}$$

where $\dot{m}_{mu}$ = Mass flow of makeup water;
$\dot{m}_e$ = Evaporation loss;
$\dot{m}_d$ = Drift loss; and
$\dot{m}_b$ = Blowdown.

Since dry air mass passing through a cooling tower is obtained using Equation 22, the evaporation loss, $\dot{m}_e$, can be also calculated. According to El-Wakil (1984) then:

$$\dot{m}_e = \dot{m}_a(\omega_d - \omega_a) \tag{24}$$

According to Perry and Green (2008), drift loss can be estimated by:

$$\dot{m}_d = 0.0002 \times \textit{amount of water supplied to the tower} \tag{25}$$

Blowdown discards a portion of the concentrated circulating water due to the evaporation process in order to lower the system's concentration of solids. The amount of blowdown can be calculated according to the number of cycles of concentration required to limit scale formation. "Cycles of concentration" is the ratio of dissolved solids in the recirculating water to the dissolved solids in the makeup water. Since chlorides remain soluble on concentration, cycles of concentration are best expressed as the ratio of the chloride content of the circulating and makeup waters. Thus, the blowdown quantities required are determined from:

$$\dot{m}_b = \frac{\dot{m}_e - (cycles - 1)\dot{m}_d}{cycles - 1} \tag{26}$$

Cycles of concentration involved with cooling tower operation normally range from three to five cycles (Perry and Green, 2008).

Non-condensable gas removal system - Selection criteria:
The non-condensable gases that are present in geothermal steam, and which accumulate in the condenser, must be pumped out from the condenser separately using gas removal equipment in order to maintain condenser vacuum and heat exchange process effectiveness. The expansion process in a steam turbine is degraded if there is an increase in condenser pressure due to non-condensable gas accumulation. The non-condensable gases are generally disposed of by mixing them with the cooling tower discharge air plume. The appropriate equipment to be used for gas removal is dependent on the proportion of non-condensable gases present in the steam.

At a low gas proportion (less than 1.5% by weight), steam jet ejectors (Figure 6) are generally the most economical choice. These are reliable but are relatively inefficient.

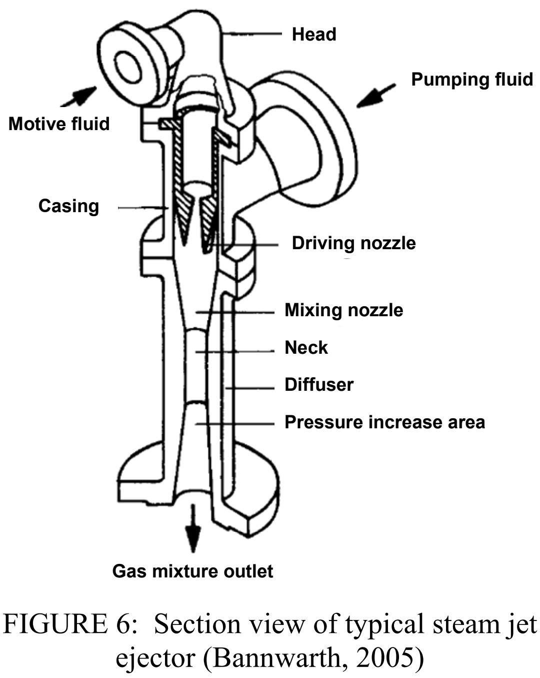

FIGURE 6: Section view of typical steam jet ejector (Bannwarth, 2005)

At a higher gas proportion, the high steam consumption of the comparatively inefficient steam ejectors leads to the selection of higher capital cost, and lower auxiliary consumption alternatives. Generally, for non-condensable gas contents between approximately 1-3% by weight, the most economic option will be a hybrid system involving a first-stage steam jet ejector and second-stage compression with a liquid ring vacuum pump. The liquid ring vacuum pump is essentially a constant-volume flow device, so physically large and expensive units would be required if these were also to be used for first-stage compression.

At non-condensable gas contents above approximately 3.5% by weight, it is generally more economical to use multi-stage centrifugal compressors. These are usually coupled directly to the turbine through a gearbox in order to obtain typical shaft speeds of 10,700 rpm for the high pressure (HP) stages and 5,300 rpm for the low-pressure (LP) stages.

For non-condensable gas contents exceeding about 12% mass of the steam, it is generally most economical to use a back pressure turbine rather than a condensing steam turbine because of the large amount of power required to extract the gases from the condenser (Dickson and Fanelli, 2003).

Steam jet ejector:
For the non-condensable gas removal system, it is necessary to know the consumption of steam as a motive fluid for sucking non-condensable gases from the condenser using the venturi principle. The following steps are used to approximate the quantity of steam required for driving a 1st stage steam jet ejector (Branan, 1999):

1. Determine compression ratio for 1st stage compression.
2. Determine power equivalent to compress non-condensable gas from the steam jet ejector suction inlet to discharge outlet for 1st stage compression. In order to achieve this, it is necessary to first calculate adiabatic head according to the following equation:

$$H_{AD} = \frac{ZRT_1}{(K-1)/K}\left[\left(\frac{P_2}{P_1}\right)^{(K-1)/K} - 1\right] \tag{27}$$

The polytrophic and adiabatic efficiency are related as follows:

$$E_A = \left[\frac{\left[\left(\frac{P_2}{P_1}\right)^{(K-1)/K} - 1\right]}{\left[\left(\frac{P_2}{P_1}\right)^{(K-1)/KE_p} - 1\right]}\right] \tag{28}$$

where H_{AD} = Adiabatic head (kN·m/kg);
Z = Average compressibility factor;
R = 8.314 $kJ\cdot kg^{-1}\cdot K^{-1}$/(molecular weight);
T_1 = Suction temperature (K);
P_1 = Suction pressure (bar);
P_2 = Discharge pressure (bar);
K = Adiabatic exponent, C_p/C_v;
E_p = Polytrophic efficiency; use 75% for preliminary calculation; and
E_A = Adiabatic efficiency.

The power needed to compress non-condensable gases can then be calculated as follows:

$$P_{AD} = \frac{\dot{m}_{NCG} H_{AD}}{E_A} \tag{29}$$

where P_{AD} = Power equivalent to compress non-condensable gases (kW); and
$\dot{m}_{NCG}$ = Non-condensable gas mass flow (kg/s).

3. The quantity of steam required for driving a 1st stage steam jet ejector is the theoretical amount that can deliver the previously calculated power equivalent, using operating steam conditions, from the operating steam inlet to the discharge outlet.

$$\dot{m}_{os} = \frac{P_{AD}}{h_{osi} - h_{do}} \tag{30}$$

where $\dot{m}_{os}$ = Mass flow of operating steam (kg/s);
h_{osi} = Enthalpy of operating steam at inlet (kJ/kg); and
h_{do} = Enthalpy of steam jet ejector discharge outlet (kJ/kg).

For a two-stage steam ejector system the above calculation steps is repeated to obtain the quantity of steam required for driving the 2nd stage steam jet ejector.

Intercondenser and aftercondenser:
An intercondenser is a vessel installed after the 1st stage steam jet ejector, while an aftercondenser is a vessel installed after the 2nd stage steam jet ejector.

The purpose of both an intercondenser and an aftercondenser is to condense exhausted operating steam and steam that carried over while the non-condensable gas was sucked by the venturi effect of the steam jet ejector. The process involves mixing discharged fluids from the steam jet ejector with sprayed cooling water. The vapour part is condensed and then flows to the condenser. The non-condensable gas is separated and flows to a gas side outlet.

Liquid ring vacuum pump:
During normal operation, a liquid ring vacuum pump will be used for 2nd stage compression according to the hybrid system scenario. In case of an emergency condition, i.e. liquid ring vacuum pump break down, 2nd stage compression will be replaced by a 2nd stage steam jet ejector. According to Bannwarth (2005), the power for driving the liquid ring vacuum pump is obtained by:

$$P_{LRVP} = \frac{0.028 p_i \dot{V}}{\eta_{is} \eta_{motor}} \ln \frac{p_o}{p_i} \tag{31}$$

where P_{LRVP} = Motor power for driving liquid ring vacuum pump (kW);
p_i = Inlet suction pressure (bar);
$\dot{V}$ = Suction capacity at suction pressure (m^3/h);
p_o = Outlet compression pressure (bar);
η_{is} = Isothermal coupling efficiency; and
η_{motor} = Motor efficiency.

The volume portion of the impeller cells available to the gas to be sucked is calculated according to the proportional partial pressure of the water vapour with the aid of the following equation (Bannwarth, 2005):

$$\varphi = \frac{p_i - p_s}{p_i} \tag{32}$$

where φ = Portion of pumped gas;
p_i = Inlet suction pressure; and
p_s = Saturated vapour pressure of the operating liquid.

Apart from energy transfer, sealing the impeller cells, clearances between the impeller, the port plate and the casing, the operating liquid is also necessary for the absorption and removal of the heat accrued in the pump. Besides the compression heat, further heat flows may accrue in the liquid ring pump as a result of vapour condensation, absorption of gases, or chemical reactions between the process gas and the ring liquid, as well as the cooling of sucked gases with a higher temperature. The total heat quantity to be removed can be arithmetically calculated according to the following equations (Bannwarth, 2005):

$$\dot{Q}_{tot} = \dot{Q}_{comp} + \dot{Q}_{cond} + \dot{Q}_{cool} \tag{33}$$

$$\dot{Q}_{comp} = 0.9 P_{LRVP} 3600 \tag{34}$$

$$\dot{Q}_{cond} = \dot{m}_v h_{fg} \tag{35}$$

$$\dot{Q}_{cool} = \dot{m}_G c_p \left(T_{i,G} - T_{o,cw}\right) \tag{36}$$

where $\dot{Q}_{tot}$ = Heat flow to be removed from the pump (kJ/h);
$\dot{Q}_{comp}$ = Isothermal compression flow and heat loss flow (kJ/h);
$\dot{Q}_{cond}$ = Condensation heat flow (kJ/h);
$\dot{Q}_{cool}$ = Heat exchange gas/operating liquid (kJ/h);
P_{LRVP} = Power consumed by the liquid ring vacuum pump (kW);
$\dot{m}_v$ = Mass flow of the condensing vapour (kg/h) ;
$\dot{m}_G$ = Mass flow of the sucked gas (kg/h);
c_p = Specific heat of the sucked gas (kJ/kg·K);
$T_{i,G}$ = Inlet temperature of the sucked gas (K); and
$T_{o,liq}$ = Outlet temperature of the operating liquid (K).

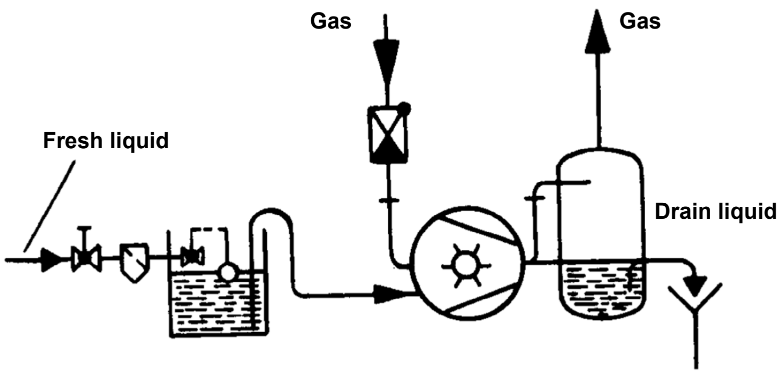

FIGURE 7: Fresh liquid operation; Liquid ring vacuum pump without recirculation (Bannwarth, 2005)

A liquid ring vacuum pump in the geothermal power plant will be put into operation without liquid recirculation (Figure 7), a so-called fresh liquid operation mode. With this mode of operation, only fresh liquid from the supply network or an existing supply system (e.g. re-cooling water) is supplied to the vacuum pump. This brings about a particularly intense cooling of the pump and the process gas, as it is required for the generation of low suction pressures. A liquid separator for the discharge of gas and liquid should be installed. The fresh liquid fed to the pump will not be reused in this mode of operation which is usually applied when economical water is available as the operating liquid and can be used for the gases and vapours to be discharged. The supplied liquid flow absorbs the total heat accruing in the pump.

Owing to this continuous supply of fresh operating liquid, there is a permanent heat transport from the pump which keeps the temperature of the ring liquid constant. About 90% of the arising heat quantities are removed through the ring liquid. Due to the higher heat capacity of the operating liquid compared to the gas to be pumped, most of the energy passes to the liquid ring during the heat exchange between these two matters, causing the temperature of the compressed gas to be only slightly higher than the temperature of the new operating liquid entering the pump. Consequently, the compression typical for this design of liquid ring vacuum pumps is almost isothermal. During the compression of dry gas with water as the ring liquid, depending on the operating pressure, an increase in temperature of about 3-10°C with liquid ring vacuum pumps is to be expected when compared to the inlet temperature of the operating liquid (Bannwarth, 2005).

Pumps:
Several pumps are used in a geothermal power plant such as a cooling water pump, an auxiliary cooling water pump and a closed circuit cooling water pump. The formula to calculate the power requirements for driving a water pump on a volume flow rate basis (Perry and Green, 2008) is:

$$P_{pump} = \frac{HQ\rho}{\eta_{pump}\eta_{motor}3.670\cdot 10^5} \tag{37}$$

or, on a mass flow rate basis:

$$P_{pump} = \frac{H\dot{m}}{\eta_{pump}\eta_{motor}3.670\cdot 10^5}$$

where P_{pump} = Motor power to drive the pump (kW);
ρ = Density of fluid (kg/m^3);
Q = Volume flow rate (m^3/h);
$\dot{m}$ = Mass flow rate (kg/h);
H = Total developed head (m);
η_{pump} = Pump efficiency; and
η_{motor} = Motor efficiency.

Heat exchanger:
A plate heat exchanger will be installed as a closed circuit cooling water heat exchanger. The heat exchanger is part of the closed circuit cooling system which transfers heat from the generator coolers, lube oil coolers and compressed air coolers and rejects the heat to the circulating water system through an S plate-type heat exchanger.

Plate heat exchangers have some advantages over shell-tube heat exchangers. Among these are superior thermal performance, ease of maintenance, expandability and multiplex capability, and their compact design. The preliminary size of a plate type heat exchanger can be calculated as shown by Rafferty and Culver (1991).

3.4 Performance of a geothermal power plant

Specific steam consumption:
The performance of a geothermal power plant is usually measured in terms of steam rate, i.e. the total mass of steam required to produce one kilowatthour of output (Kestin, 1980):

$$SSC = \frac{\dot{m}_{steam}}{P_{gen}}3{,}600 \tag{38}$$

The above definition is the so-called *SSC gross* and is applicable without taking the parasitic load or auxiliary power into account. If the auxiliary power for all running equipment to support geothermal power plant operation is to be considered, then the term turns into *SSC net*. The formula then becomes:

$$SSC_{net} = \frac{\dot{m}_{steam}}{P_{gen} - P_{aux}} 3{,}600 \tag{39}$$

where SSC = Specific steam consumption (kg/kWh);
SSC_{net} = Specific steam consumption net (kg/kWh);
$\dot{m}_{steam}$ = Steam supply at geothermal power plant boundary (kg/s);
P_{gen} = Electrical power output at generator terminal (kW); and
P_{aux} = Total auxiliary power (kW).

Geothermal utilisation efficiency:
The performance of the entire plant may be assessed using the second law of thermodynamics by comparing the actual power output to the maximum theoretical power that could be produced from the given geothermal fluid. This involves determining the rate of exergy carried into the plant with the incoming geofluid. The specific exergy, e, of a fluid that has a pressure, p, and a temperature, T, in the presence of an ambient pressure, p_0, and an ambient temperature, T_0, is given by (DiPippo, 2005):

$$e = h(T,p) - h(T_0,p_0) - T_0[s(T,p) - s(T_0,p_0)] \tag{40}$$

When this is multiplied by the total incoming geofluid mass flow rate, we obtain the maximum theoretical thermodynamic power or the exergetic power:

$$\dot{E} = \dot{m}_{total} e \tag{41}$$

The ratio of the actual net power to the exergetic power is defined as the utilisation efficiency or the second law (exergetic) efficiency for the plant:

$$\eta_u = \frac{\dot{W}_{net}}{\dot{E}} \tag{42}$$

All types of power plants can be compared on the basis of the utilisation efficiency, no matter the source of the primary energy - be it coal, oil, nuclear, biomass, hydro, solar, wind, or geothermal. Plants can also be designed to maximise η_u when the value of the primary energy (or exergy) is a significant factor in the economics of the operation (DiPippo, 2005).

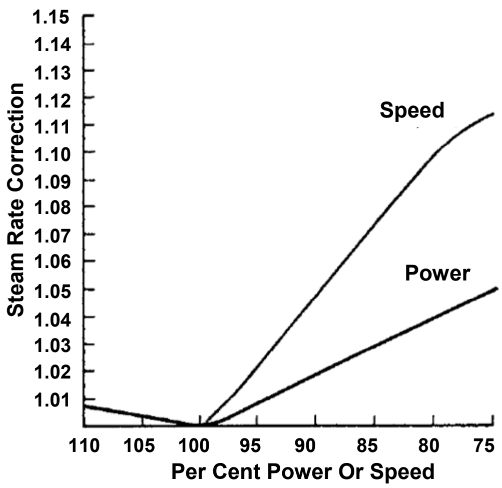

FIGURE 8: Steam turbine partial load correction (Bloch and Singh, 2005)

3.5 Consideration of partial load operation mode

The equipment is designed to give the best efficiency at a specific design point. The efficiency will decrease for operation at a partial load rate. For this research, the equipment was modelled with the parameter that decreased at partial load operation, the steam turbine. The steam rate consumption was altered from its least-steam rate consumption point according to Figure 8. The condition affecting steam turbine performance is only load (output power) while the effect of changing the steam turbine speed is neglected since, for the power generation unit, the speed will be maintained at a specific speed, e.g. 3,000 rpm.

4. DEVELOPING THE MODEL

4.1 Methodology

1. Develop a thermodynamic cycle model of a geothermal power plant within a computer-aided thermodynamic simulation software environment, in this case Engineering Equation Solver (EES). The geothermal power plant is modelled under selected design parameters and conditions based on mature and proven technology for similar unit size.
2. Run simulations with several different conditions for the above mentioned model of the geothermal power plant.
3. Analyse the characteristics of the thermodynamic cycle for a geothermal power plant at rated full load operation and partial load operation.

4.2 Scope of work and limitations

The basic design objectives for these analyses are:

1. To produce a heat and mass balance for a geothermal power plant for rated full load operation and partial load operation; and
2. To produce technical specifications of main equipment for a geothermal power plant with 2×55 MW_e units.

Some limitations have been made:

1. The focus of the research is mainly on the geothermal power plant and its characteristics during operations at rated full load and partial load at different conditions;
2. The site and climatic data such as wet bulb temperature, humidity and site altitude are assumed;
3. The assumptions on reservoir and well characteristics are based on moderate levels;
4. "Rated full load" is defined as gross output power at generator terminal; and
5. An economic analysis is beyond the scope of this research.

4.3 Design parameters and conditions

Design parameters and conditions for the basic design of the geothermal power plant are as follows:

1. The thermal steam cycle is a single-flash condensing cycle.
2. The reservoir is water-dominated and the temperature is 270°C.
3. Non-condensable gases (NCGs) are dominated by CO_2.
4. NCG content is 1.325% at design conditions for rated full load. However, the impact of different NCG amounts (0.65, 1.325 and 2%) will be taken into consideration during the simulation process.
5. Wet bulb temperature is 25°C at design conditions for rated full load. Nonetheless, the impact of different wet bulb temperatures (18-32°C) will also be simulated.
6. Relative humidity is 85%.
7. The altitude of the geothermal power plant site is 1100 m a.s.l.
8. Gross output power at generator terminals is 55 MW of electricity for each unit of the geothermal power plant. Total gross output power is 110 MWe.
9. A separator is installed in each wellpad. The steam will be supplied to the power plant site as single-phase saturated steam while the brine will be delivered through a hot reinjection line and reinjected into hot reinjection wells. Condensate produced within the geothermal power plant will be delivered through a cold reinjection line and reinjected into cold reinjection wells.
10. Optimum separator pressure will be determined using the "equal-temperature-split" rule based on a condenser pressure of 0.1 bar.

11. The condenser type is a direct contact spray condenser.
12. The cooling system type is mechanically induced draft wet cooling tower.
13. The gas removal system type is a hybrid system that employs a steam jet ejector for first stage compression and a liquid ring vacuum pump as a second stage compression system. As a redundant unit, the steam jet ejector system will be put online for the second stage compression unit just in case the liquid ring vacuum pump fails to operate. When the second stage compression unit is a steam jet ejector, the gas removal system changes over to a dual stage steam jet ejector.

4.4 Geothermal power plant process outline

The single-flash steam cycle with a condensing turbine is recommended for the project so that the brine in the reinjection system can be kept at high pressure and high temperature. The single-flash steam cycle is suitable for water-dominated resources of relatively low NCG content as presented in Lumut Balai geothermal field. The single-flash steam cycle can deliver a large capacity of output power. It has an advantage over the double flash steam cycle in preventing silica scale formation in the reinjection line.

The technology for a single-flash steam plant is fully developed, well proven and the mainstay of the geothermal power industry. It is often the first power plant installed at a newly-developed liquid-dominated geothermal field. As of July 2004, there were 135 units of this kind in operation in 18 countries around the world. Single-flash plants account for about 29% of all geothermal plants. They constitute nearly 40% of the total installed geothermal power capacity in the world (DiPippo, 2005).

Within the geothermal power plant system, the steam flows to a steam turbine through a mist eliminator, a strainer, main stop valves, and control valves. Exhaust steam from the turbine flows into a direct contact type main condenser through the exhaust duct. An expansion joint is installed between the turbine and the condenser to accommodate erection allowance and thermal expansion. In the condenser, exhaust steam is condensed by direct contact with cold water from the cooling tower basin, and a mixture of cold water and condensate is sent to the top of the cooling tower by circulating water pumps. In the cooling tower, the mixture is cooled down and sent back to the condenser by gravity and condenser vacuum. NCGs are cooled down in the main condenser to reduce the accompanying steam and are extracted by the gas removal system and sent to the cooling tower fan stacks for dispersion into the atmosphere. The gas removal system consists of a 1st stage steam jet ejector, an intercondenser, a 2nd stage LRVP and a LRVP separator. The backup for the LRVP system is a steam jet ejector and an aftercondenser. After first stage compression, an intercondenser is installed to condense motive steam. The separator will be installed after the liquid ring vacuum pump to separate NCGs from the LRVP working fluid. For the case of a second stage compression using a steam jet ejector, the aftercondenser will be used to condense motive steam from the NCG stream. The motive steam for the ejectors is drawn from the main steam line. Drainage from the intercondenser, the LRVP separator and the after-condenser is led into the main condenser. Cold water from the cooling tower basin is used not only for the condenser cooler but also for the turbine oil cooler, generator air cooler, air compressor cooler, the intercondenser, the LRVP separator and the aftercondenser. The cooling tower is of the multi cell mechanically induced-draft wet, counter-flow type. The cooling tower is equipped with a maintenance stair and lifting facilities. At the outlet of the cooling tower basin, a mesh screen is installed to prevent foreign particles from entering into the system. Excess water from the cooling tower is sent to a settling basin and then to cold reinjection wells.

4.5 Simulation and calculation of specific conditions

The geothermal power plant was modelled to work as a base load electric power generation unit within the national electric grid network. Based on this kind of electric power generation, some conditions were simulated accordingly to understand the change in parameters that might result due to fluctuations in ambient conditions dominated by wet bulb temperature and well characteristics such as variations in NCG content in the steam supply.

Even though the geothermal power plant is to be a base load electric power generation unit, under some circumstances the power plant should be able to operate in partial load mode. The conditions in which the geothermal power plant might be operated under partial load mode were simulated including a scenario where the equipment is out of service. The geothermal power plant was modelled with the steam turbine at partial load with constant circulating cooling water flow rate and a resulting fall in the circulating water temperature. This has an impact on the condenser pressure and the resulting effect on the power plant performance was analysed.

5. RESULTS

5.1 Thermal cycle model of the geothermal power plant

The screenshot of a thermal cycle model of the geothermal power plant developed within the EES environment is shown in Figure 9. A graphical user interface modelling-approach through a feature in EES named Diagram Window was used to better visualise the overall system and to speed up the engineering work in the basic design phase in constructing a heat and mass balance diagram during simulation under different specific conditions.

5.2 Analysis of the geothermal power plant operation under specific conditions

The characteristics of the geothermal power plant operation, including a change in its parameters, were obtained by running a simulation of the thermal cycle model under specific conditions.

5.2.1 Constant gross output power (rated full load operation) using hybrid gas removal system

As shown in Figure 10, the net output power and the auxiliary power for different NCG content, 0.65, 1.325 and 2%, were simulated for a range of wet bulb temperatures from 18 to 32°C. A hybrid system was selected as a default gas removal system. The net output power decreased as the wet bulb

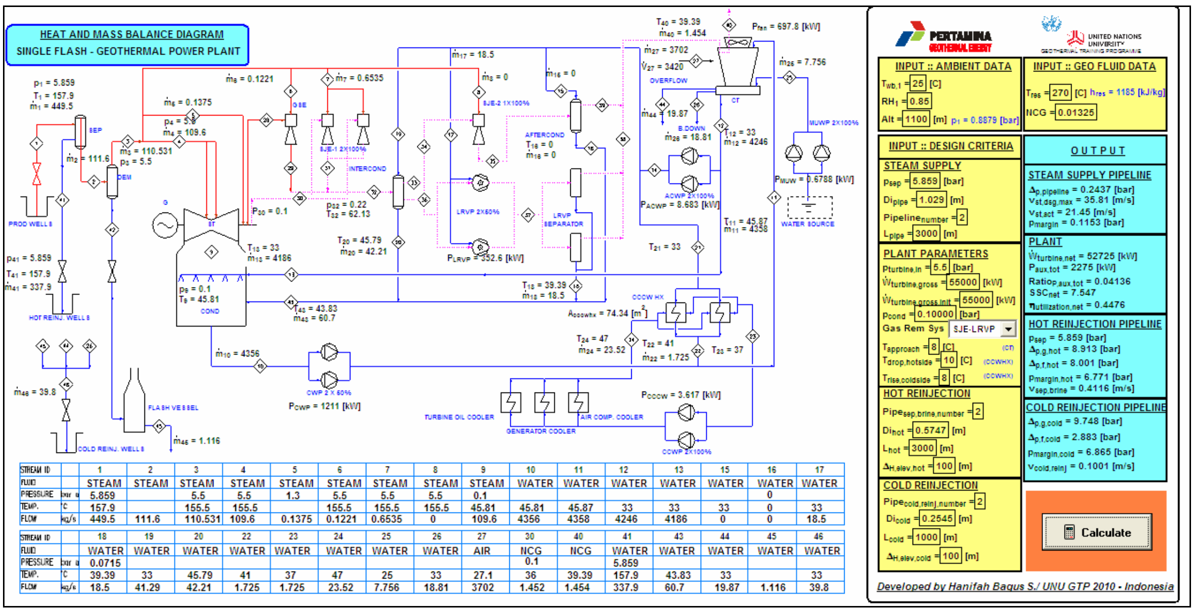

FIGURE 9: Graphical user interface of a thermal cycle model of the geothermal power plant

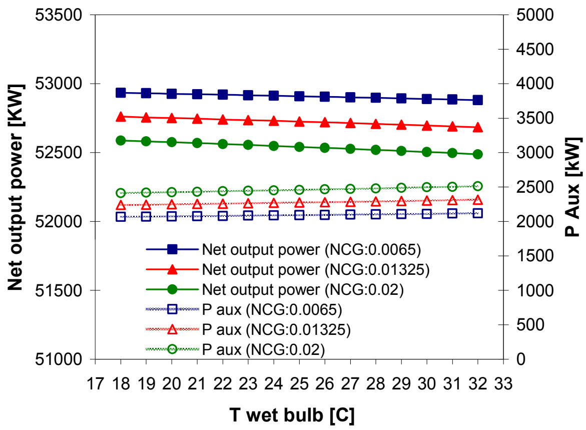

FIGURE 10: Net output power and auxiliary power at constant gross output power for different NCG content using a hybrid gas removal system

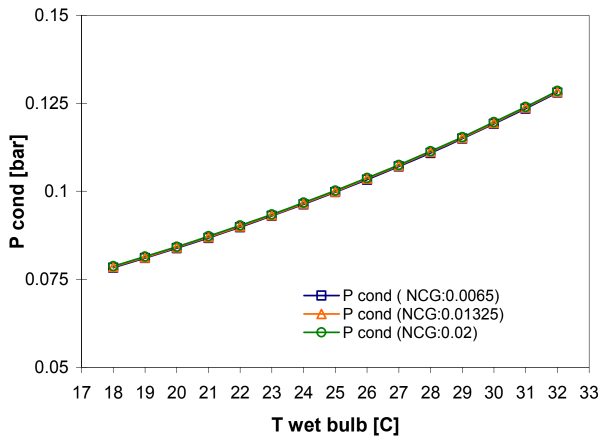

FIGURE 11: Condenser pressure at constant gross output power for different NCG using a hybrid gas removal system

temperature increased. The increase of wet bulb temperature affects the temperature of the circulating cooling water since the cooling tower is designed for specific ambient air temperature, cooling range and liquid-to-gas ratio. An increase in the wet bulb temperature would lead to a temperature increase of circulating water entering and leaving the condenser. The higher the circulating water temperature leaving the condenser, the higher the condenser pressure; therefore, the gross output power would decrease for the same amount of steam flow supplied to the steam turbine. To maintain the gross output power at 55 MW, the amount of steam supply to the geothermal power plant, especially steam entered into the turbine, should be increased and this would increase the amount of NCGs that need to be removed from the condenser. The higher the amount of NCGs, the higher the LRVP power needed to evacuate NCGs from the condenser. Therefore, the auxiliary power deducted from the gross power output increases, resulting in lower net output power.

As seen in Figure 11, the change in the condenser pressure is primarily affected by the change in the wet bulb temperature while the change in the condenser pressure is less susceptible to variation in NCG content of the steam supply. Figure 12 shows the effects of NCG content variation on auxiliary power component distribution. The LRVP power consumption increases as the NCG content increases. The auxiliary power component distribution calculated for each NCG proportion is based on an average value within a wet bulb temperature range of 18-32°C.

The amount of steam that must be supplied to the geothermal power plant in order to maintain constant gross output power is presented in Figure 13. The steam supply in terms of the SSC net is also provided in this figure. An increase in NCG content and wet bulb temperature would lead to higher

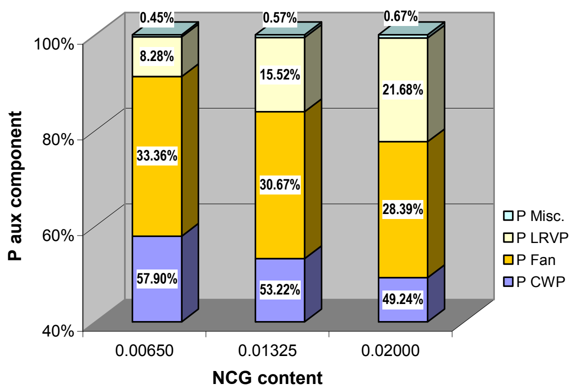

FIGURE 12: Distribution of auxiliary power components at constant gross output power for different NCG content using hybrid gas removal system

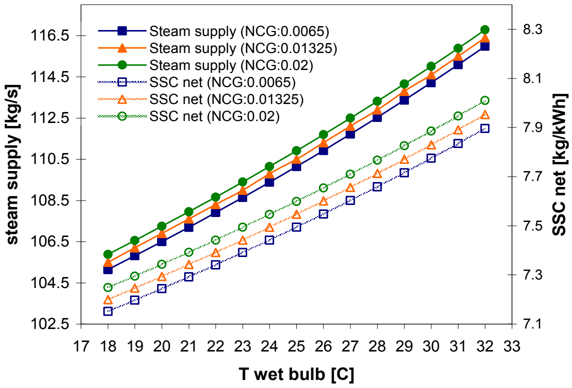

FIGURE 13: Steam supply and specific steam consumption at constant gross output power for different NCG content using a hybrid gas removal system

SSC net, so the geothermal power plant operation becomes less efficient, although it maintains its constant gross output power.

5.2.2 Constant gross output power (rated full load operation) with a dual stage steam jet ejector gas removal system

The net output power is only slightly decreased within the range of different wet bulb temperatures and different NCG content when using a dual stage steam jet ejector gas removal system, as shown in Figure 14. The decreasing rate of net output power is much smaller than in the hybrid systems because the DSJE system has lower auxiliary power since it utilises a steam jet ejector instead of LRVP for 2nd stage gas removal equipment. The steam supply and SSC net for the geothermal power plant using DSJE system are shown in Figure 15.

In comparison with the hybrid system, the steam requirement for a geothermal power plant with DSJE system is larger (Table 1) at 32°C wet bulb temperature and 2% NCG content. This condition was the toughest circumstance for which the geothermal power plant was simulated. However, this behaviour prevailed within a wet bulb temperature range of 18-32°C.

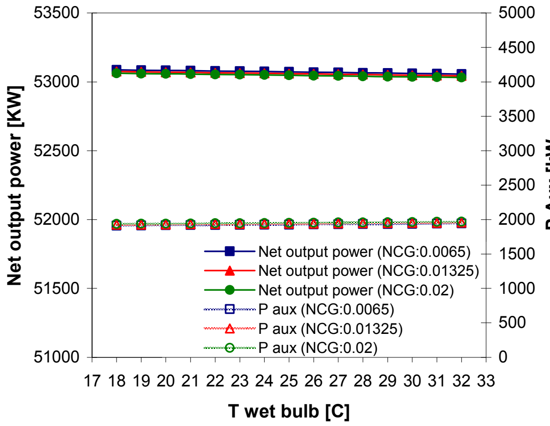

FIGURE 14: Net output power and auxiliary power at constant gross output power for different NCG using a DSJE gas removal system

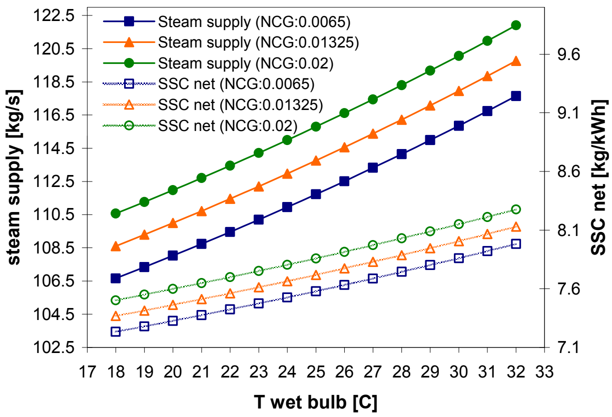

FIGURE 15: Steam supply and specific steam consumption at constant gross output power for different NCG content using a DSJE gas removal system

TABLE 1: Steam supply and SSC net for a geothermal power plant with hybrid and DSJE gas removal systems at 32°C wet bulb temperature and 2% NCG content

Parameters	Hybrid system	DSJE system
Steam supply (kg/s)	116.79	121.91
SSC net (kg/kWh)	8.01	8.28

Referring to Figure 16, the condenser pressure of the geothermal power plant with DSJE gas removal system was mainly affected by variations in the wet bulb temperature while the effect of NCG content variations on condenser pressure was smaller. However, the range of variation in the condenser pressure for different NCG content at certain wet bulb temperatures was wider than in the hybrid system (Figure 11). The wider range of the condenser pressure variation results from the change in NCG content for the geothermal power plant using the DSJE system, because a higher steam flow is required to operate not only the 1st stage steam jet ejector but also the 2nd stage steam jet ejector. Higher steam supply to the cycle increases the thermal load in the circulating water system. For constant circulating water flow rate, this condition increases the temperature of the circulating water, therefore the condenser pressure for a geothermal power plant maintaining constant gross output power with a DSJE system is higher for certain wet bulb temperatures and NCG content. The higher condenser pressure leads to a higher steam flow requirement to maintain a certain gross output power.

It means that at the same wet bulb temperature and same NCG content, the geothermal power plant with a DSJE gas removal system requires more steam flow at the steam turbine inlet than the geothermal plant with a hybrid gas removal system.

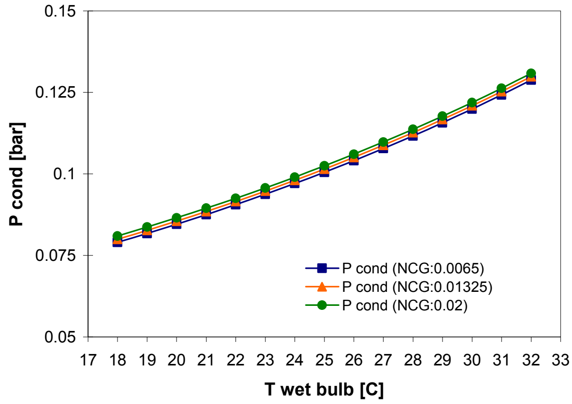

FIGURE 16: Condenser pressure at constant gross output power for different NCG content using a DSJE gas removal system

5.2.3 Partial load operation

The conditions for partial load operation are as follows:

1. One of the 50% capacity liquid ring vacuum pumps is out of service.
2. One of the 50% circulating water pumps is out of service.
3. One of five cooling tower cells is out of service.

All of the conditions for partial load operation are at the same level of NCG content which is 1.325% and use a hybrid gas removal system since these are the basic conditions under which the geothermal power plant would be run under normal operations.

Generally the gross and net output power decreases as the wet bulb temperature increases. Out of the three scenarios, the largest decline in output power is when one of the 50% CWPs is out of service, followed by a significant decline when one of the 50% LRVPs is out of service. The smallest decline in output power is when one of the five cooling tower cells is out of service. The calculated impacts on power output under specific conditions are shown in Figure 17.

The steam supply requirement for partial load is shown in Figure 18. Steam supply curves are proportional with the power output curves (Figure 17) and from these figures it can be deduced how steam supply decreases as output power decreases. A quite interesting phenomenon appeared in the SSC net curves for the scenarios when one 50% capacity liquid ring vacuum pump is out of service and when one of the five cooling tower cells is out of service (Figure 18). Those SSC net curves almost overlapped one another. Yet the output power difference between those two conditions is around 8% in average (Figure 17).

This state can be attained with lower condenser pressure, see Figure 19, where one of the 50% capacity liquid ring vacuum pumps is out of service while keeping the circulating water at the same flow rate as that for a rated full load operation.

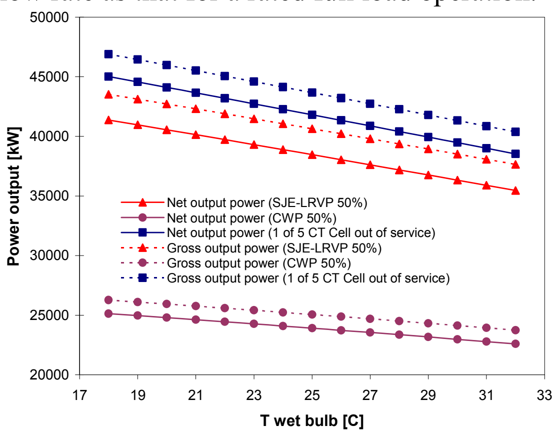

FIGURE 17: Power output at partial load, NCG content 1.325% using hybrid gas removal system

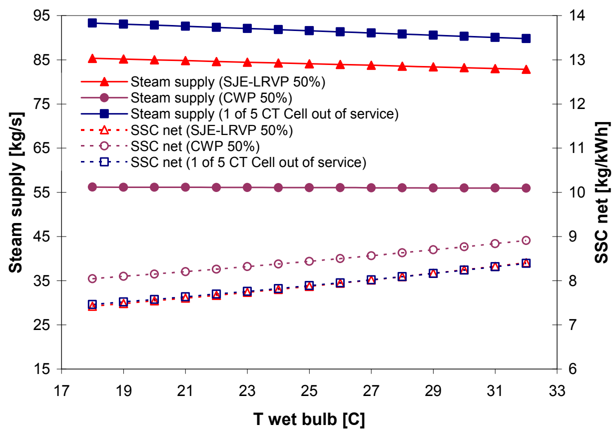

FIGURE 18: Steam supply and SSC net at partial load operation, NCG content 1.325% using a hybrid gas removal system

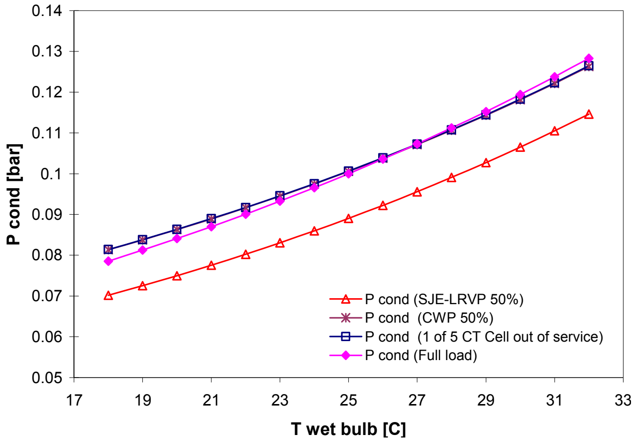

FIGURE 19: Condenser pressure at partial load, NCG content 1.325% using a hybrid gas removal system

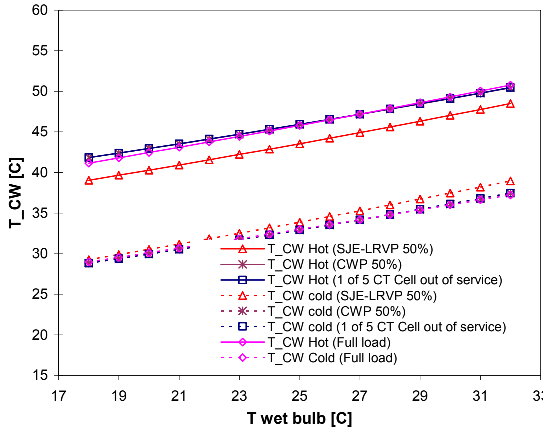

FIGURE 20: Circulating water temperature at partial load, NCG content 1.325% using a hybrid gas removal system

At partial load operation, the output power will be lower than for a rated full load operation resulting in lower mass flow of exhausted steam to be condensed. For this reason, the cooling load borne by the condenser becomes lower while the circulating water mass flow is constant, so the circulating water temperature leaving the condenser will go down accordingly, shown in Figure 20. The condenser pressure will be lower as the temperature of the circulating water leaving the condenser decreases.

5.3 Heat and mass balance diagram of the geothermal power plant

The heat and mass balance diagram for specific conditions are presented in Appendix I. The summary of these heat and mass balance diagrams for each 55 MW unit of the geothermal power plant are shown in Table 2.

TABLE 2: Summary of heat and mass balance diagram for each 55 MW unit

No.	Parameter	Rated full load			Partial load		
		NCG: 1.325% T_{wb}: 25°C HS	NCG: 1.325% T_{wb}: 25°C DSJE	NCG: 2% T_{wb}: 32°C DSJE	NCG: 1.325% T_{wb}: 25°C HS 1×50% LRVP out of service	NCG: 1.325% T_{wb}: 25°C HS 1×50% CWP out of service	NCG: 1.325% T_{wb}: 25°C HS 1 of 5 CT cell out of service
1	Gross pow. output (kW)	55,000	55,000	55,000	40,638	25,059	43,671
2	Net power output (kW)	52,725	53,060	53,031	38,463	23,916	41,808
3	Auxiliary power (kW)	2,275	1,940	1,969	2,175	1,143	1,864
4	Auxili. power ratio (%)	4.14	3.53	3.58	5.35	4.56	4.27
5	SSC net (kg/kWh)	7.547	7.718	8.276	7.872	8.438	7.886
6	Utilization efficie. (%)	44.76	43.77	44.06	42.91	40.03	42.83
7	Total steam flow (kg/s)	110.5	113.8	121.9	84.1	56.06	91.58
8	Turbine inlet flow (kg/s)	109.6	109.9	115.9	83.35	55.47	90.78
9	Turbine inlet press. (bar)	5.5	5.5	5.5	5.5	5.5	5.5
10	Condenser press. (bar)	0.1	0.1015	0.1308	0.0890	0.1005	0.1006

5.4 Technical specification of main equipment

Generally the equipment of the geothermal power plant should be designed to bear operational conditions at rated full load. Reasonable margin adjustments while sizing the equipment are indispensable since the geothermal power plant operational characteristics as a base load power generation unit are dictated by the network grid requirement; on the other hand, the performance of the geothermal power plant highly depends on climatic parameters such as wet bulb temperature and conditions of the steam supply, like the NCG content.

The technical specifications of the main equipment are determined in Section 5.4 and its sub-sections based on the analyses that were made for geothermal power plant behaviour, simulated under specific operating conditions (as presented in Section 5.2), taking into consideration the heat and mass balance diagram (attached in Appendix I) and the summary in Table 2. The technical specifications of the main equipment for the 2×55 MW geothermal power plant are presented in a table in Appendix II.

5.4.1 Steam turbine

The steam turbine will be a single cylinder, double-flow, horizontal shaft, condensing unit. For a geothermal power plant, it is essential to apply protection for erosion/corrosion so it is assumed that the rotor shall have erosion/corrosion protection overlays in the gland seal areas and nozzle stationary blade labyrinth seal areas.

The pressure at the steam turbine inlet is 5.5 bar and the steam is saturated. Steam mass flow that can be supplied to a steam turbine is determined according to Figure 21. Steam supply for a guaranteed rated full load at 25°C and a 1.325% NCG proportion is 109.6 kg/s. The upper margin for the steam supply is 5.75%; therefore, the steam turbine should be able to operate at a rated load with a steam supply up to 115.9 kg/s. The lower margin for the steam supply is 4.38%; the steam turbine should be able to operate at a rated load with a steam supply down to 104.8 kg/s.

5.4.2 Condenser

The condenser is of a direct contact, spray type connected to the turbine exhaust with an expansion compensator. It is designed to condense all the steam from the turbine for each of the load conditions. It is divided into two zones: condensation and gas cooling. The non-condensable gases from the condenser, after being cooled down to a temperature about 3 degrees higher than the cold water from the cooling tower, is extracted by the gas extraction system and discharged by piping through a stack to the atmosphere in the cooling tower.

Referring to Figures 11 and 16, the condenser pressure range can be determined. The circulating water flow is 4,356 kg/s.

5.4.3 Circulating water system

The circulating water pump is canned-type, vertical, wet suction, with mixed flow. The circulating water mass flow for each pump is 2,178 kg/s with a total developed head of 20 m. The configuration is 2×50% capacity. The power required for each motor is 606 kW.

The cooling tower is a multi cell mechanically induced draft wet, counter flow type and is comprised of five cells. The total power drawn by the cooling power fan is determined in Figure 22 and divided into five cooling tower fans; therefore, the cooling tower fan motor power becomes 144 kW. The liquid-to-gas ratio of the cooling tower is 1.202. At design conditions (wet bulb temperature 25^{o}C, NCG 1.325%, hybrid gas removal system), the approach temperature of the cooling tower is 8^{o}C and the cooling temperature is 13^{o}C. The liquid-to-gas ratio will be maintained constant at any conditions while the approach temperature and cooling temperature may vary.

5.4.4 Gas removal system

Two types of a gas removal system will be installed. The hybrid system will be used during normal operation. The hybrid system is comprised of a 1st stage compression using a steam jet ejector; for 2nd stage compression, a liquid ring vacuum pump is employed.

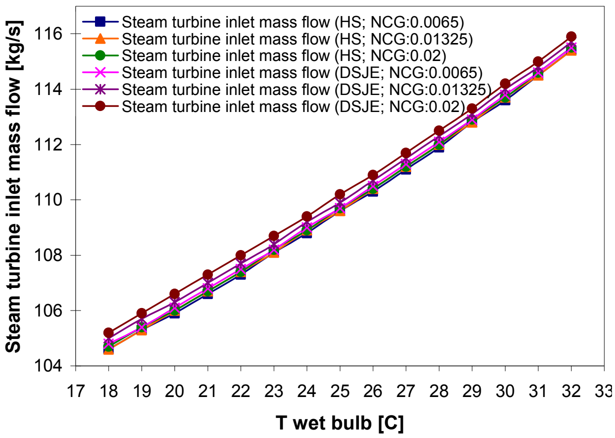

FIGURE 21: Steam turbine inlet mass flow at constant gross output power for different NCG content and different gas removal systems

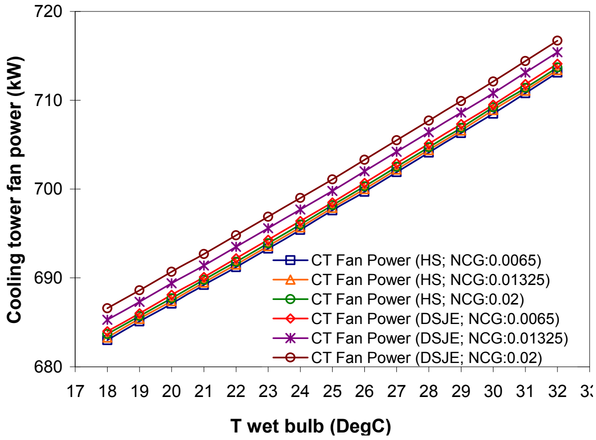

FIGURE 22: Cooling tower fan power at constant gross output power for different NCG content and different gas removal systems

The capacity of the 1st stage steam ejector should be able to handle NCG content within a supplied steam range of 0.65% to 2% by weight. The steam supply for the steam jet ejector for both 1st stage and 2nd stage is determined in Figure 23. The steam supply for the 1st stage ejector is 1.06 kg/s and for 2nd stage ejector it is 4.68 kg/s. The configuration of the 1st stage ejector is 2 x 100% while for 2nd stage ejector it is 1×100%.

The LRVP will be sized to cover operational conditions according to Figure 24. The motor power for each LRVP is 285 kW. The LRVP configuration is 2×50%.

5.4.5 Steam gathering and reinjection system

Separator and separator stations:
The separator pressure is determined using the "equal-temperature-split" rule. The pressure was found to be 5.859 bar. The enthalpy of the two-phase geothermal fluid flow is 1185 kJ/kg with a steam quality of 24.84%. The separator inlet mass flow will vary according to the operational type of geothermal power plant. There are three production wellpads that are served by three separator

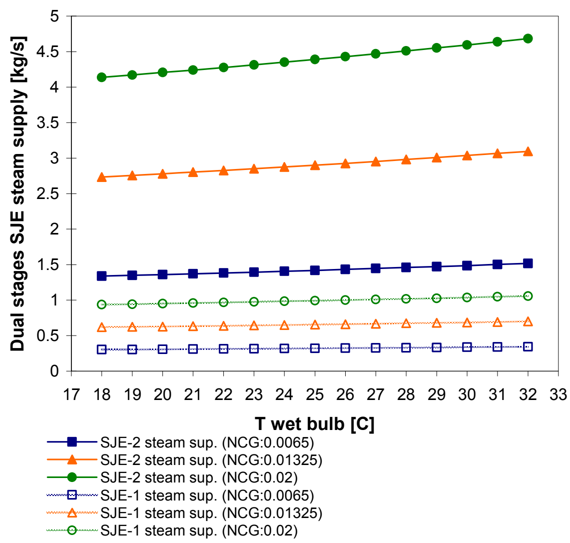

FIGURE 23: Steam supply for jet ejector at constant gross output power for different NCG content using a DSJE gas removal system

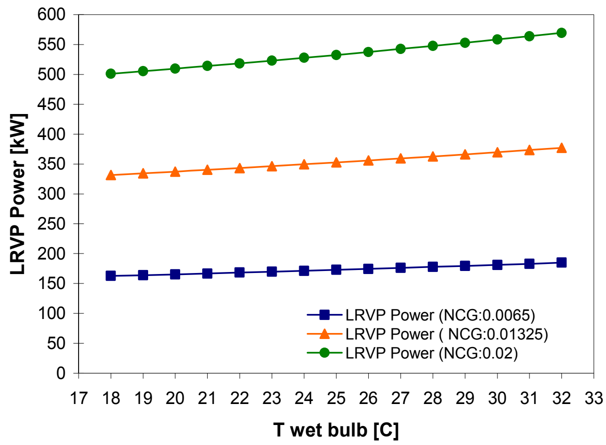

FIGURE 24: LRVP power at constant gross output power for different NCG content using a hybrid gas removal system

stations. Each production wellpad will supply two-phase flow fluid to one separator station. The calculation of the number of individual separators that should be installed in each separator station is beyond the scope of this research. The total two-phase flow rate requirement at design conditions to supply a 2×55 MW geothermal power plant is 2×449.5 kg/s or 899 kg/s. The design condition is at wet bulb temperature of 25°C, NCG content of 1.325% using a hybrid gas removal system. If the production wellpads are assumed to be scattered in three locations and each wellpad supplies the same quantity of two-phase flow fluid, then there are three separator stations that should be provided with the capacity of 299.3 kg/s each. When a margin is added, then the total capacity of each separator station is 330.53 kg/s, so the total capacity of the separator stations becomes 991.6 kg/s, enough to fulfil maximum steam requirements for maintaining a rated full load at a wet bulb temperature of 32°C and with a NCG content of 2% using a DSJE gas removal system. The total two-phase geothermal fluid flow rate requirement at the inlet of the separators needed to supply each 55 MW unit of the geothermal power plant for different conditions is shown in Figure 25.

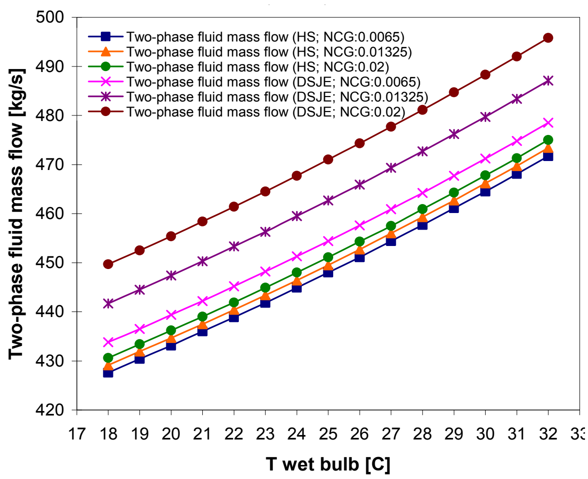

FIGURE 25: Two-phase fluid mass flow requirement at constant gross output power for different NCG content and different gas removal systems

Steam supply pipelines:
The steam supply for each 55 MW unit of the geothermal power plant will be served by two pipelines. In accordance with available commercial pipes in the market, the selected nominal diameter of the pipes is 1,000 mm schedule 40 (Nayyar, 2000). The length of pipeline between the separator station outlet header line to the power plant is assumed to be around 3,000 m. The pressure drop along the pipeline at maximum capacity is 0.30 bar with an actual steam velocity of 23.66 m/s and a maximum steam velocity of 35.81 m/s. The pressure drop is around 5.12 %, still within an acceptable range and able to fulfil the steam turbine inlet pressure requirement which is 5.5 bar.

Hot reinjection system:
The two-phase geothermal fluid will be separated into steam and brine streams after undergoing a flashing process in the separator. The steam will be transported to the power plant site through a steam transmission pipeline while the brine will be reinjected to the subsurface formation via hot reinjection wells. The brine will be transported from the separator to the reinjection wells through a hot reinjection pipeline by means of gravitational flow. As shown in Figure 26, the quantity of produced brine will vary depending on the operational mode of the geothermal power plant.

At design conditions, to produce the rated output power (wetbulb temperature 25°C, NCG content 1.325%, hybrid gas removal system), the quantity of brine is 2×337.9 kg/s produced after a flashing separation process for 2×55 MW geothermal power plant or a total of 675.8 kg/s. At a wet bulb temperature of 32°C, NCG content of 2%, using a DSJE gas removal system, the produced brine equals 2×372.7 kg/s for a total of 745.4 kg/s. In accordance with available commercial pipes in the market (Nayyar, 2000), the nominal diameter of selected pipes for the hot reinjection pipelines is 600 mm schedule 40. Four pipelines should be provided as hot reinjection pipelines for serving the 2×55 MW geothermal power plant. The length of each pipeline is about 3,000 m. Insulation is required to keep the skin temperature at the surface of pipeline at an acceptable level for complying with HSE requirements and to maintain high brine temperature so that silica scaling along the hot reinjection pipeline can be minimised or even avoided.

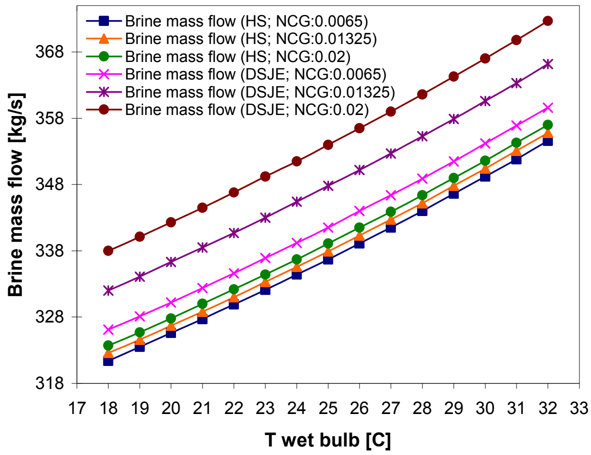

FIGURE 26: Brine mass flow at constant gross output power for different NCG content and different gas removal systems

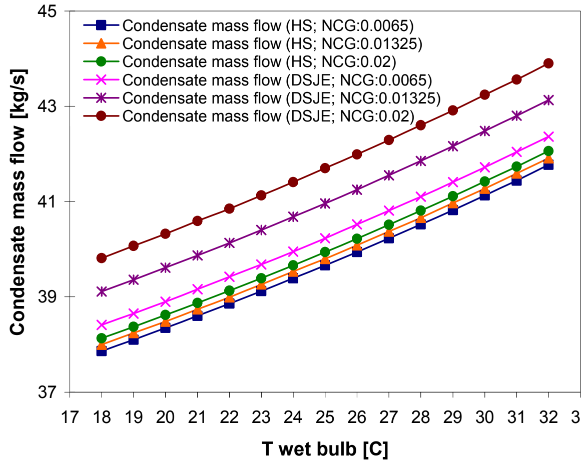

FIGURE 27: Condensate mass flow at constant gross output power for different NCG content and different gas removal systems

Cold reinjection system:
It is important to maintain a mass balance within the geothermal power plant boundaries, especially for condensate coming from overflow and blowdown in the cooling tower system. This situation is a result of using a direct-contact condensing system in which some portion of the condensed steam is re-circulated to the condenser as circulating water after undergoing a cooling process in the cooling tower. However, due to the suspended solid concentrations in the cooling fluid, it is necessary to dispose of a certain proportion of this circulating water. The condensate will be disposed of by reinjection into the cold reinjection wells. The condensate from the geothermal power plant, by design, will flow by gravity through cold reinjection pipelines. In accordance with available commercial pipes in the market (Nayyar, 2000), the selected nominal diameter for the cold reinjection pipelines is 250 mm schedule 40. Four pipelines shall be provided as cold reinjection pipelines for serving 2×55 MW geothermal power plants. The length of each pipeline is about 1,000 m. The condensate production rate can be seen in Figure 27.

6. CONCLUSIONS AND RECOMMENDATIONS

6.1 Conclusions

- The geothermal power plant is modelled using EES and the parameters are simulated under several conditions.
- As a base load power generation unit, the continuity of the power output of the geothermal power plant should be maintained under fluctuating wet bulb temperature and NCG content.
- Wet bulb temperature is the main factor that affects the performance of the geothermal power plant since it may vary even on a daily basis. The fluctuation of the wet bulb temperature will affect the circulating water temperature in the cooling tower which dissipates heat from the thermal system. The higher the wet bulb temperature, the higher the temperature of the circulating water; therefore, the condenser pressure will increase accordingly. Since the geothermal power plant should maintain constant gross power output, the steam supply should be increased since the pressure inside the condenser becomes higher. The rise in the steam supply to maintain power output means lower geothermal power plant efficiency in terms of the SSC net. Conversely, a higher efficiency will be achieved if the wet bulb temperature becomes lower.
- The NCG content of the steam supply will influence the performance of the geothermal power plant. NCG content is one important factor to be considered when designing a geothermal

power plant, even though fluctuations in the NCG content of the steam supply are less significant than wet bulb temperature fluctuations. The higher the NCG content, the higher the power needed to evacuate it from the condenser. This power can be derived from the enthalpy contained in the steam supply of the working fluid used to drive the steam jet ejectors or from electricity used to drive the LRVP. An increase in the steam supply for driving the steam jet ejector will lead to a higher thermal load in the circulating water system causing the condenser pressure to rise as the circulating water temperature increases. The higher the condenser pressure, the more steam supply is required to drive the steam turbine in order to maintain constant gross power output. As the steam supply and/or electrical auxiliary power increases, so decreases the geothermal power plant performance, in terms of the SSC net. Conversely, the performance increases if the NCG content of the steam supply becomes lower.

- In times of partial load operation, due to such things as sudden equipment failure or planned shut down on specific equipment, the simulation results can be used as guidelines for necessary action in order to run and maintain the unit efficiently and safely.
- A heat and mass balance diagram was constructed in a widely accepted manner for presenting technical parameters of the modelled geothermal power plant.
- Technical specifications for the main equipment of the geothermal power plant were fixed based on the heat and mass balance diagram and made available as input data for the detailed engineering design phase.

6.2 Recommendations

- It is preferable that exact on-site meteorological data such as wet bulb temperature, humidity and site altitude should be obtained in order to model the geothermal power plant and acquire more precise results.
- Definitive data on reservoir and well characteristics such as well productivity curves were not incorporated during this research. However, the geothermal power plant model developed within EES is expandable with regard to the thermodynamic simulation environment and may be integrated with reservoir and well characteristic data later on.
- It is possible that, at a later date, the requirement of the “rated full load” as stated in Section 4.2, scope of works and limitations, can be replaced with “net power output at high voltage side of step up transformer” instead of “gross output power at generator terminal”. Thereafter, the model should be modified as necessary.

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NOMENCLATURE

SEP	= Separator
DEM	= Demister
ST	= Steam turbine
G	= Generator
COND	= Condenser
GSE	= Gland steam ejector
SJE-1	= 1st stage steam jet ejector
SJE-2	= 2nd stage steam jet ejector
INTERCOND	= Intercondenser
AFTERCOND	= Aftercondenser
B.DOWN	= Cooling tower blowdown
ACWP	= Auxiliary cooling water pump
CCWP	= Closed circuit water pump
CCCW HX	= Closed circuit cooling water heat exchanger
CT	= Cooling tower
CWP	= Circulating water pump
DSJE	= Dual stage steam jet ejector
GPP	= Geothermal power plant
HS	= Hybrid system
LRVP	= Liquid ring vacuum pump
MUWP	= Makeup water pump
NCG	= Non-condensable gas

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APPENDIX I: Heat and mass balance diagrams

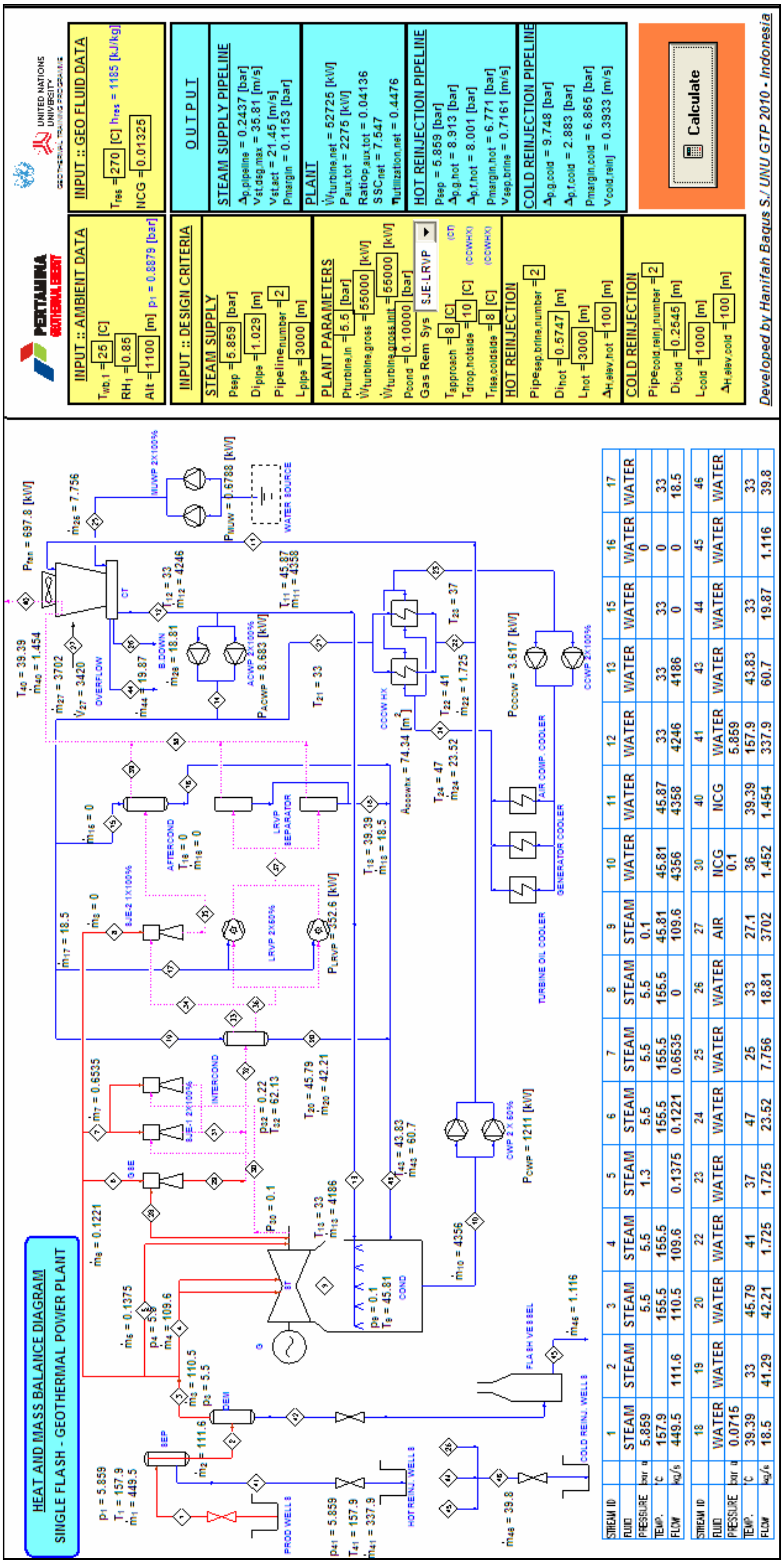
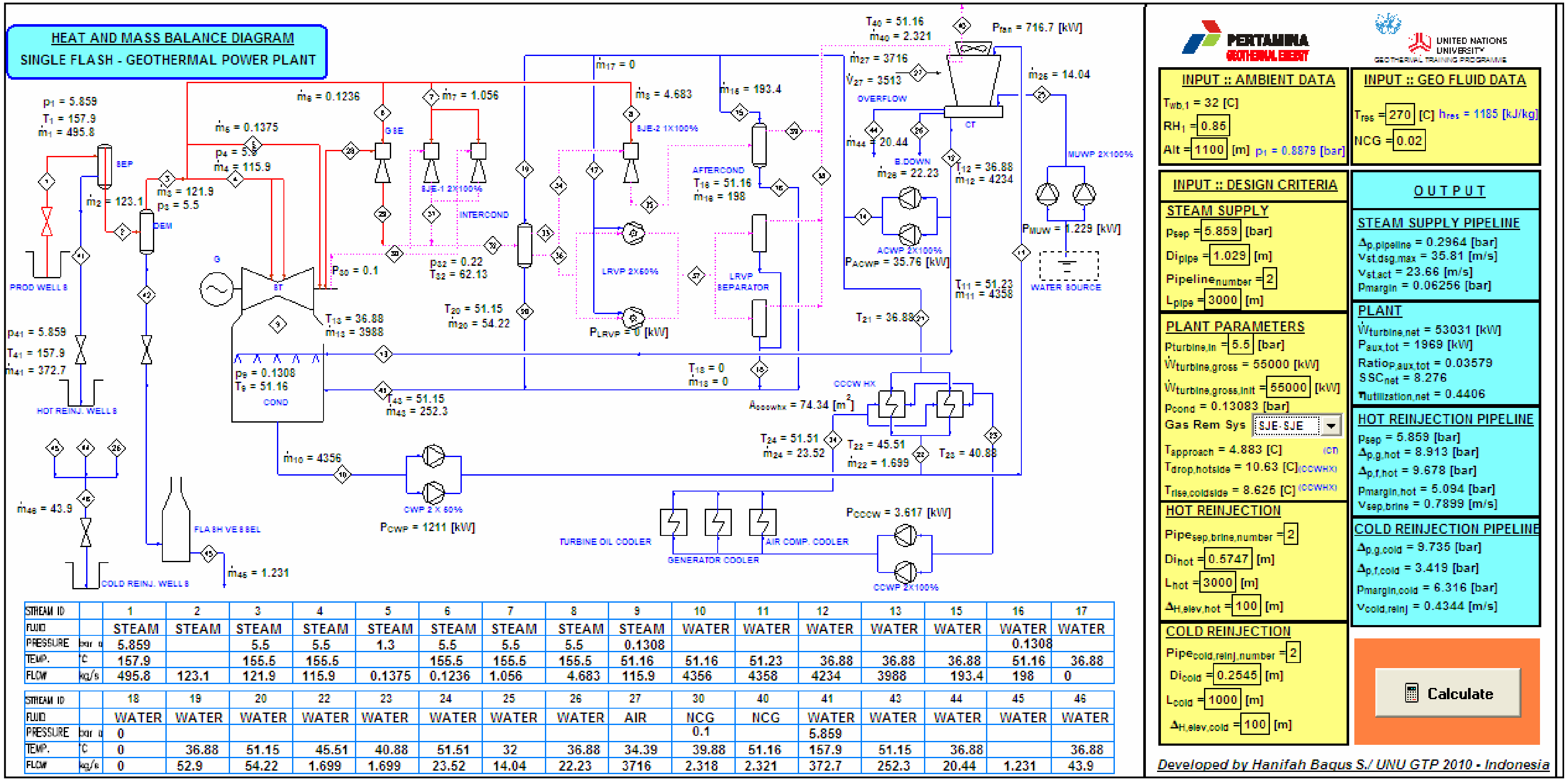


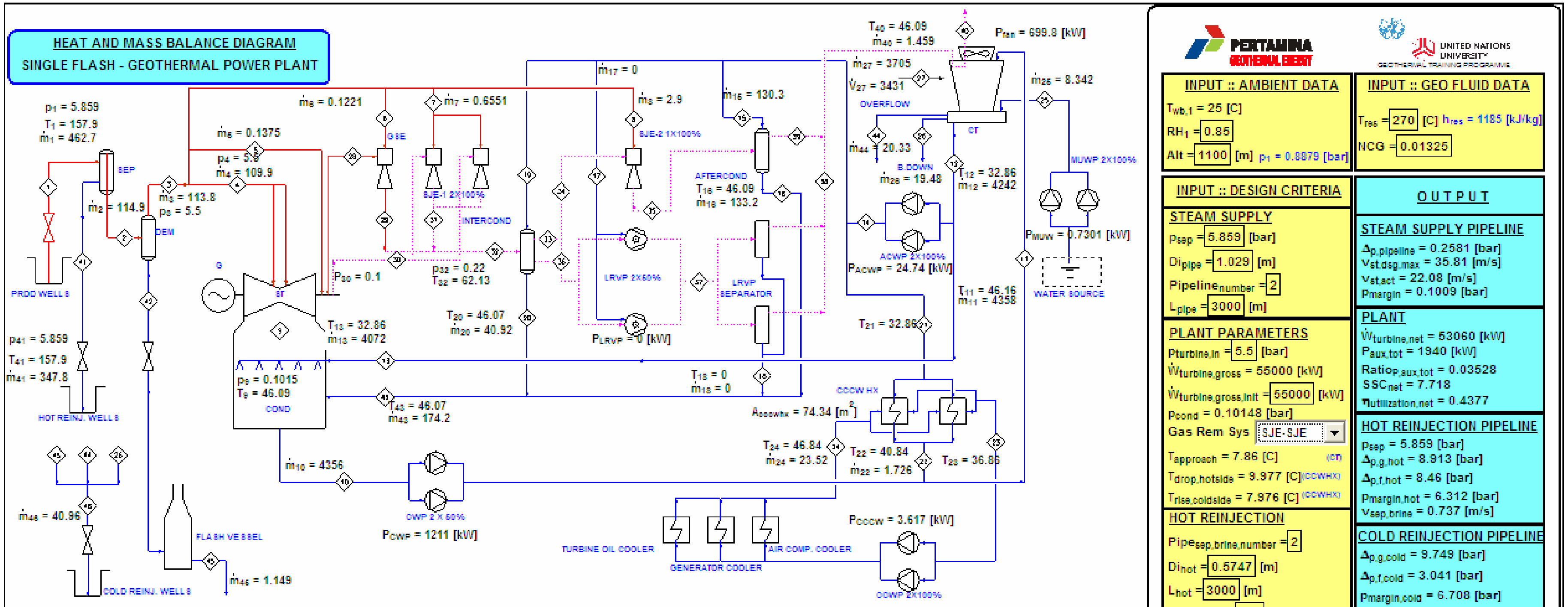
FIGURE 1: Heat and mass balance for rated full load operation at wet bulb temperature 25°C, NCG content 1.325%, with hybrid gas removal system



STREAM ID		1	2	3	4	5	6	7	8	9	10	11	12	13	15	16	17
FLUID		STEAM	STEAM	STEAM	STEAM	STEAM	STEAM	STEAM	STEAM	STEAM	WATER	WATER	WATER	WATER	WATER	WATER	WATER
PRESSURE	bar a	5.859		5.5	5.5	1.3	5.5	5.5	5.5	0.1308						0.1308	
TEMP.	°C	157.9		155.5	155.5		155.5	155.5	155.5	51.16	51.16	51.23	36.88	36.88	36.88	51.16	36.88
FLOW	kg/s	495.8	123.1	121.9	115.9	0.1375	0.1236	1.056	4.683	115.9	4356	4358	4234	3988	193.4	198	0

STREAM ID		18	19	20	22	23	24	25	26	27	30	40	41	43	44	45	46
FLUID		WATER	WATER	WATER	WATER	WATER	WATER	WATER	WATER	AIR	NCG	NCG	WATER	WATER	WATER	WATER	WATER
PRESSURE	bar a	0									0.1		5.859				
TEMP.	°C	0	36.88	51.15	45.51	40.88	51.51	32	36.88	34.39	39.88	51.16	157.9	51.15	36.88		36.88
FLOW	kg/s	0	52.9	54.22	1.699	1.699	23.52	14.04	22.23	3716	2.318	2.321	372.7	252.3	20.44	1.231	43.9

FIGURE 2: Heat and mass balance for rated full load operation at wet bulb temperature 32°C, NCG content 2.00%, with dual stage steam ejector gas removal system



STREAM ID		1	2	3	4	5	6	7	8	9	10	11	12	13	15	16	17
FLUID		STEAM	STEAM	STEAM	STEAM	STEAM	STEAM	STEAM	STEAM	STEAM	WATER	WATER	WATER	WATER	WATER	WATER	WATER
PRESSURE	bar a	5.859		5.5	5.5	1.3	5.5	5.5	5.5	0.1015						0.1015	
TEMP.	C	157.9		155.5	155.5		155.5	155.5	155.5	46.09	46.09	46.16	32.86	32.86	32.86	46.09	32.86
FLOW	kg/s	462.7	114.9	113.8	109.9	0.1375	0.1221	0.6551	2.9	109.9	4356	4358	4242	4072	130.3	133.2	0

STREAM ID		18	19	20	22	23	24	25	26	27	30	40	41	43	44	45	46
FLUID		WATER	WATER	WATER	WATER	WATER	WATER	WATER	WATER	AIR	NCG	NCG	WATER	WATER	WATER	WATER	WATER
PRESSURE	bar a	0									0.1		5.859				
TEMP.	C	0	32.86	46.07	40.84	36.86	46.84	25	32.86	27.1	35.86	46.09	157.9	46.07	32.86		32.86
FLOW	kg/s	0	40.01	40.92	1.726	1.726	23.52	8.342	19.48	3705	1.457	1.459	347.8	174.2	20.33	1.149	40.96

FIGURE 3: Heat and mass balance for rated full load operation at wet bulb temperature 25°C, NCG content 1.325%, with dual stage steam ejector gas removal system

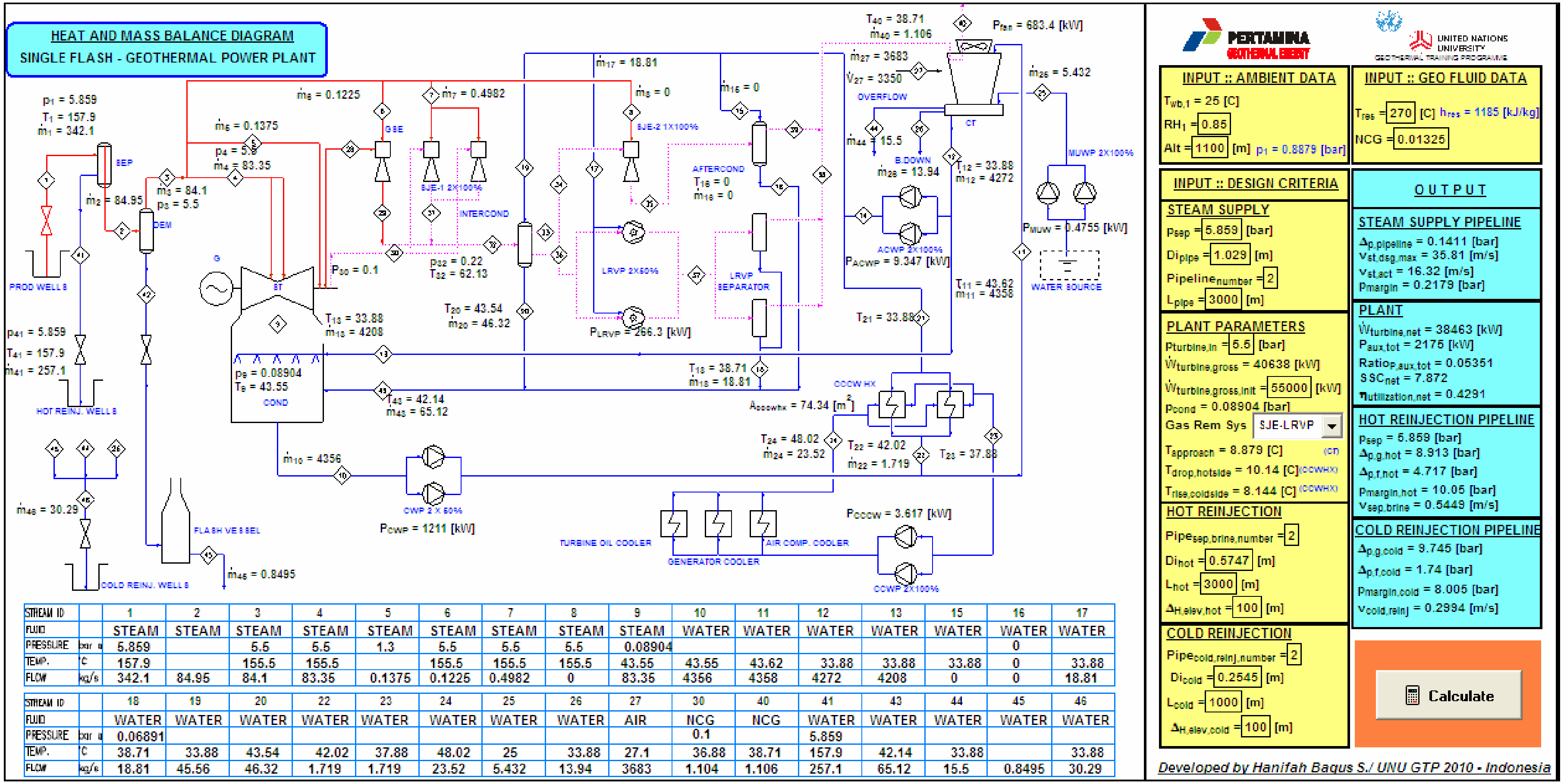


FIGURE 4: Heat and mass balance for partial load operation; one 50% capacity liquid ring vacuum pump is out of service at wet bulb temperature 25°C, NCG content 1.325%, with hybrid gas removal system

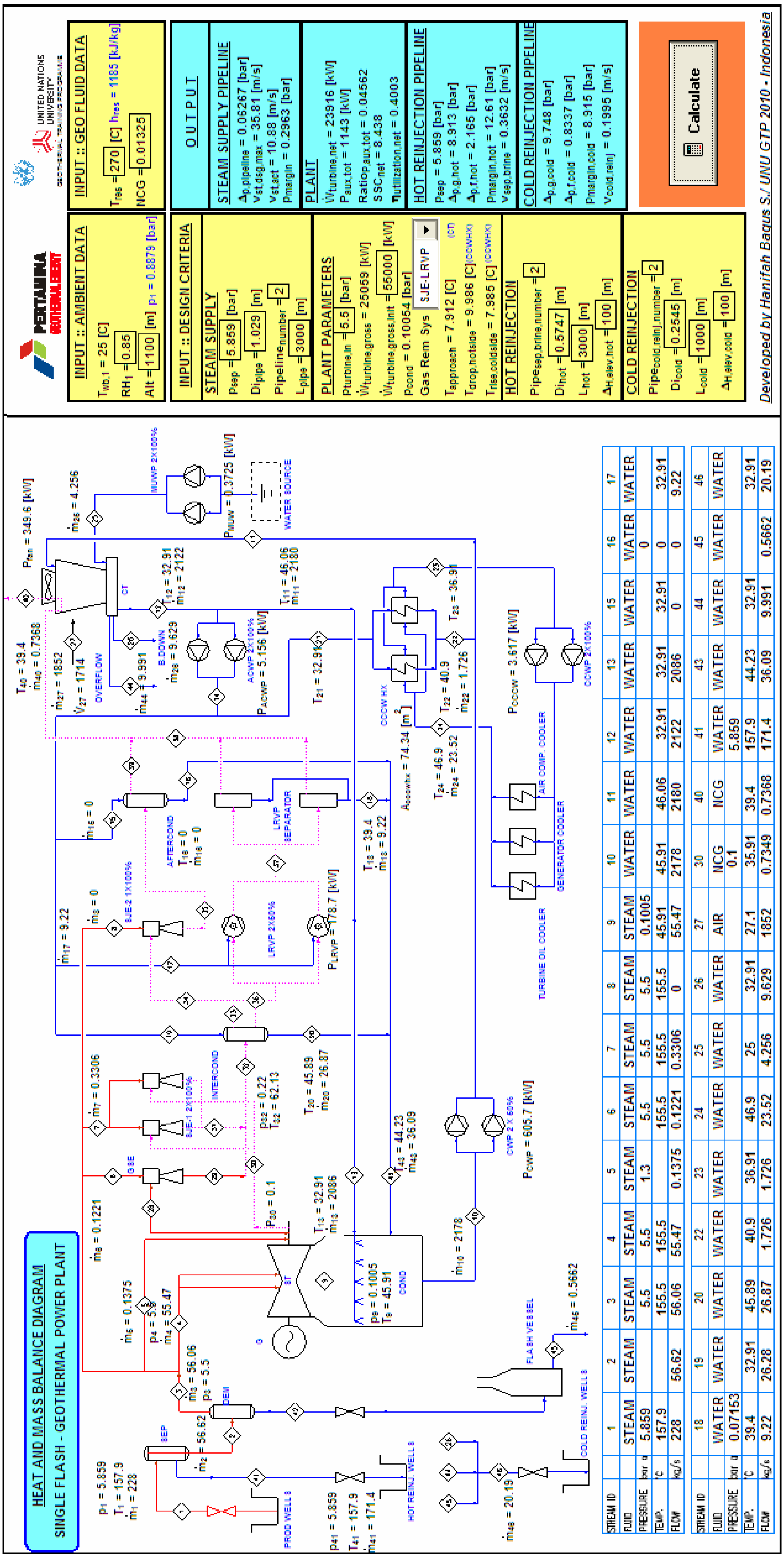
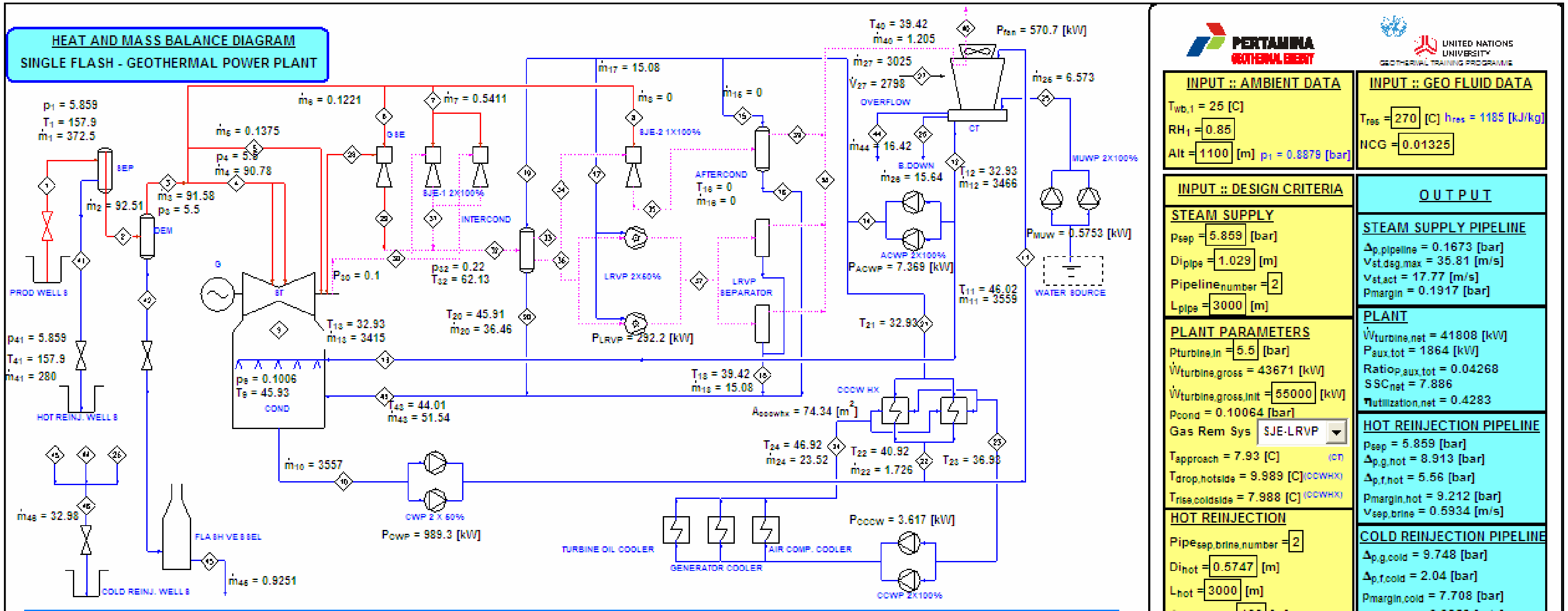


FIGURE 5: Heat and mass balance for partial load operation; one 50% capacity circulating water pump is out of service at wet bulb temperature 25°C, NCG content 1.325%, with hybrid gas removal system



STREAM ID		1	2	3	4	5	6	7	8	9	10	11	12	13	15	16	17
FLUID		STEAM	STEAM	STEAM	STEAM	STEAM	STEAM	STEAM	STEAM	STEAM	WATER	WATER	WATER	WATER	WATER	WATER	WATER
PRESSURE	bar a	5.859		5.5	5.5	1.3	5.5	5.5	5.5	0.1006						0	
TEMP.	°C	157.9		155.5	155.5		155.5	155.5	155.5	45.93	45.93	46.02	32.93	32.93	32.93	0	32.93
FLOW	kg/s	372.5	92.51	91.58	90.78	0.1375	0.1221	0.5411	0	90.78	3557	3559	3466	3415	0	0	15.08

STREAM ID		18	19	20	22	23	24	25	26	27	30	40	41	43	44	45	46
FLUID		WATER	WATER	WATER	WATER	WATER	WATER	WATER	WATER	AIR	NCG	NCG	WATER	WATER	WATER	WATER	WATER
PRESSURE	bar a	0.0716									0.1		5.859				
TEMP.	°C	39.42	32.93	45.91	40.92	36.93	46.92	25	32.93	27.1	35.93	39.42	157.9	44.01	32.93		32.93
FLOW	kg/s	15.08	35.66	36.46	1.726	1.726	23.52	6.573	15.64	3025	1.203	1.205	280	51.54	16.42	0.9251	32.98

FIGURE 6: Heat and mass balance for partial load operation; one of five cooling tower cells is out of service at wet bulb temperature 25°C, NCG content 1.325%, with hybrid gas removal system

APPENDIX II: Technical specifications of the main equipment for a 2×55 MW geothermal power plant

No.	Equipment	Qty.	Parameter	Unit	Technical specification
1	Steam turbine	2 sets	Type		Single cylinder, double flow, condensing
			Rated output	kW	55,000
			Inlet pressure	bar	5.5
			Inlet temperature	°C	155.46
			Rated flow	kg/s	109.6
			Speed	rpm	3,000
2	Turbo generator	2 sets	Rated output	kW	55,000
			Voltage	kV	13.8
			Frequency	Hz	50
			Power factor		0.8
			Excitation		Brushless
			Cooling system		Air cooled
3	Gland sealing system	2 sets	Steam flow	kg/s	0.01375
4	Condenser	2 sets	Type		Direct contact, spray
			Pressure	bar	0.1
			Temperature	°C	45.81
5	Gas removal system: Steam ejector	2 sets	<u>1st stage steam jet ejector:</u>		
			Configuration		2 x 100%
			Steam flow	kg/s	0.66
			Steam pressure	bar	5.5
			Suction pressure	bar	0.1
			Discharge pressure	bar	0.22
			<u>2nd stage steam jet ejector:</u>		
			Configuration		1 x 100%
			Steam flow	kg/s	4.68
			Steam pressure	bar	5.5
			Suction pressure	bar	0.22
			Discharge pressure	bar	1.1
6	Intercondenser	2 sets	Configuration		1 x 100%
			Gas/steam flow	kg/s	1.37
			Cooling water flow	kg/s	54.22
			Pressure	bar	0.13
			Outlet water temperature	°C	51.16
			Gas outlet temperature	°C	51.15
7	Aftercondenser	2 sets	Configuration		1 x 100%
			Gas/steam flow	kg/s	7.00
			Cooling water flow	kg/s	193.4
			Pressure	bar	0.13
			Outlet water temperature	°C	51.16
			Gas outlet temperature	°C	51.16

No.	Equipment	Qty.	Parameter	Unit	Technical specification
8	Gas removal system: Liquid ring vacuum pump	4 units	Configuration		2 x 50%
			Suction pressure	bar	0.22
			Discharge pressure	bar	1.1
			Discharge temperature	^oC	43.99
			Working fluid		Water
			Working fluid flow	kg/s	30.56
			Motor power/unit	kW	285
9	Liquid ring vacuum pump separator	4 units	Configuration		2 x 50%
			Gas/liquid flow	kg/s	32.87
			Outlet fluid flow	kg/s	30.56
			Outlet water temperature	^oC	45.81
			Gas outlet temperature	^oC	39.4
10	Circulating water pump	4 units	Type		Vertical, canned, wet suction, mixed flow
			Configuration		2 x 50%
			Flow	kg/s	4,356
			Total developed head	m	20
			Motor power	kW	606
11	Cooling tower	2 units	Type		Field erected, mechanical, induced draft, wet
			Cell No./unit cooling tower		5
			Circulation water flow	kg/s	4,356
			Wet bulb temperature	^oC	25
			L/G ratio		1.202
			Approach temperature	^oC	8
			Cooling range	^oC	13
			Hot water temperature	^oC	45.81
			Cold water temperature	^oC	33
			Motor power fan	kW	144
12	Auxiliary cooling water pump	4 units	Type		Horizontal, split casing, centrifugal
			Configuration		2 x 100%
			Flow	kg/s	170.3
			Total developed head	m	10
			Motor power	kW	25
13	Closed circuit cooling water pump	4 units	Type		Horizontal, split casing, centrifugal
			Configuration		2 x 100%
			Flow	kg/s	23.5
			Total developed head	m	10
			Motor power	kW	4

No.	Equipment	Qty.	Parameter	Unit	Technical specification
14	Closed circuit cooling water heat exchanger	4 units	Type		Plate-type
			Configuration		2 x 100%
			Heat transfer surface area	m^2	74.34
			Overall heat transfer coeff.	kW/m^2	5
			Hot side inlet temperature	°C	47
			Hot side outlet temperature	°C	37
			Cold side inlet temperature	°C	33
			Cold side outlet temperature	°C	41
			Hot water flow	kg/s	23.5
			Cold water flow	kg/s	1.73
15	Make up water pump	4 units	Type		Horizontal, split casing, centrifugal
			Configuration		2 x 100%
			Flow	kg/s	7.8
			Total developed head	m	5
			Motor power	kW	0.68
16	Wellpad separator station	3 stations	Two-phase inlet flow/sta.	kg/s	330.53
			Steam outlet flow/sta.	kg/s	82.07
			Brine outlet flow/sta.	kg/s	248.5
17	Steam supply pipeline	1 lot	Material		A106 Gr.B
			Nominal diameter	mm	1,000
			Schedule number		40
			Total length	m	6,000
18	Hot reinjection pipeline	1 lot	Material		A106 Gr.B
			Nominal diameter	mm	600
			Schedule number		40
			Total length	m	6,000
19	Cold reinjection pipeline	1 lot	Material		A106 Gr.B
			Nominal diameter (mm)	mm	250
			Schedule number		40
			Total length (m)	m	2,000