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THERMODYNAMIC ANALYSIS OF PRELIMINARY DESIGN OF POWER PLANT UNIT I PATUHA, WEST JAVA, INDONESIA

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ABSTRACT

Patuha geothermal field in west Java, Indonesia has nine production wells and four non-commercial wells that are intended for power plant unit 1. The non-commercial wells can be used as injection wells. The production wells have enthalpies that range between 2400 and 2700 kJ/kg with non-condensable gas contents ranging between 1.10 and 1.77 %-weight of steam. The objective of this study was to determine the optimum output power to be generated using a single-flash system geothermal energy conversion model. The study is based on thermodynamic analyses and uses the Engineering Equation Solver software for its calculations.

The results show that with the current supply from the existing nine wells, the turbine can produce 60,130 kW with a power output of 56,262 kW. This value is generated by a 6 bar separator pressure and a 0.08 bar-a condenser pressure. The steam ejector would consume 6.5 kg/s steam, equivalent to about 2,922 kW of electrical power. The auxiliary power needed for pumping, the cooling tower fan and other applications totals 3,868 kW.

To obtain optimum output power and reduce equipment size, it is suggested that the power plant be designed using 6.5 bar separator pressure and a 0.1 bar-a condenser pressure. With these pressures, the design turbine-generator would generate 58,141 kW and the output power would be 55,417 kW. The steam ejector would consume 5.2 kg/s steam, equivalent to about 2,327 kW of electrical power. The auxiliary power for pumping, cooling tower fan and other applications would equal 2,724 kW.

A double-flash system was also analysed to improve the single-flash plant. By adding a second flash on the separation water, the plant would generate an additional 1,251 kW output power. Also, combined with low-pressure steam from non-commercial wells, the plant could generate an additional 4,661 kW output power. The extra output power would add 8.4% to the single-flash plant output power.

1. INTRODUCTION

1.1 Location

The Patuha geothermal field is located in the western part of the island of Java, Indonesia, approximately 35 km southwest of the city Bandung, and 130 km southeast of Jakarta. Located in the volcanic highlands of western Java, the land in the Patuha area is used primarily for growing tea. Volcanoes extend along the long axis of the island of Java, which is located near the boundary of two major lithospheric plates: the Indian Ocean - Australia plate on the south; and the Eurasian plate to the north. The heat source for many geothermal systems in Indonesia is molten rock, cooling at relatively shallow levels in the earth's crust. Volcanic eruptions bring some of the molten rock to the surface, expressed in a series of lava eruptions or pyroclastic flows. Within the project area are several volcanic eruption centres, including Gunung Patuha, Gunung Patuha Selatan, and Gunung Urug. The volcanic peaks range in elevation from 2,200 to 2,400 m above mean sea level; the wells are drilled from the surrounding terrain, which lies at an elevation of +1,900 to +2,050 m (msl.). There are numerous hydrothermal features (hot spring and fumaroles) in the project area. From northwest to southeast, these are the Rancawalini and Cimanggu hot springs, Kawah Putih – a large acid lake, Cibunggok hot spring and lakes Kawah Tiis and Kawah Ciwidey, located at the eastern margin of the drilled area. Figure 1 shows a satellite image of Indonesia (A), West Java province (B), Patuha area (C) and the location of unit 1(D).

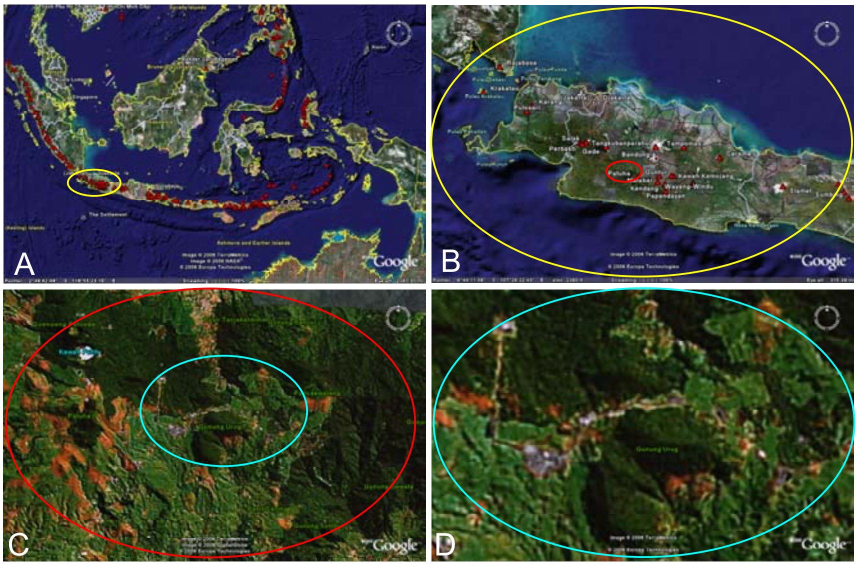

FIGURE 1: Satellite image of Indonesia showing the location of the Patuha field (courtesy of Google Earth)

1.2 Potential reserves of the Patuha field

Drilled core holes and full-diameter wells at Patuha occupy an area exceeding 35 km^2; the initial development focused on a 15 km^2 area centred around three young volcanoes (Gunung Patuha, Gunung Patuha Selatan and Gunung Urug). Of this total area, the commercial productivity is considered to be proven for a 6 km^2 area and probable for a 2 km^2 area. Combining these areas and using a range of productivity of 15 – 25 MW/ km^2 (appropriate for high-enthalpy resources like Patuha) yields a reserve estimate ranging from 120 to 200 MW for 30 years.

As of September 1997, the former developer of the Patuha field identified drilling pad locations to the north, northwest, and southwest of the initial development area and planned to drill "step out" wells in those areas. Another developer drilled commercially productive geothermal wells to the west. The existence of an extensive, shallow thermal aquifer with temperatures exceeding 150°C along the northern side of the volcanic massif presents another exploration target and may indicate the presence of deeper, higher temperature fluids in the area.

Taken together, the available data suggest that the Patuha field is very extensive and that further drilling will double or triple both the proven and probable areas. This would increase the overall field reserves to a level approaching or exceeding the maximum permissible generation level of 400 MW. This field, in the short-term plan, will be developed for three 60 MW units. The reserves already demonstrated may be sufficient to support this level of development. Further drilling will confirm the viability of this level of development in the next few years (GeothermEx, 1997).

1.3 Well characteristics

As of January 1998, the developer had completed thirteen full-sized wells, nine productive wells and four non-commercial wells. Figure 2 shows the locations of Patuha unit 1 wells. The nine production wells were flow tested for two weeks or less and demonstrated a combined initial wellhead capacity of 81 MW, using a conversion factor of 2 kg/s per MW. In September 2003, four Patuha wells (W8, W6, W9 and W2) were flow tested again for a period of approximately two months. Comparing the results of the earlier, shorter tests with the recent, longer tests (all reflecting productivity at a wellhead pressure of 10.3 bar) suggests that the productivity of wells W2, W3, W8 and W9, after a period of 2 months is 40-50% lower than it is after a period of a few hours or days.

FIGURE 2: Patuha unit 1 layout (courtesy of Google Earth)

Well W6, for which both data sets are available, suggests no decline at all. While some decline during the first few weeks of production is typical for steam-dominated wells like those at Patuha, a 40-50% decline over a test period of 2 months is considered to be anomalous. Since there had been no sustained production from any well in the field between the two test periods, the observed decline may be the result of phenomena other than reservoir pressure decline (e.g., scaling, mechanical problem with the well, measurement error, etc.).

Estimation of the current average Patuha production capacity is about 6.5 MW. This average includes the four wells for which the cause of decline is uncertain (W2, W3, W8 and W9) and another well not tested in 2003 for which a 40% capacity reduction from its initial productivity was conservatively assumed (W3). Some of the non-commercial Patuha wells could be used for injection of the small amount of water separated from the steam and/or the excess condensate from the power plant. If production wells continue to tap the steam zone at Patuha, the injection requirement will be minimal. Therefore, any needed injection capacity could be easily realized. If a larger, liquid-dominated resource is developed, intended injection wells will be required (GeothermEx, 1997).

1.4 Non-condensable gases

The term, 'non-condensable gas', is used to describe gas that must be removed from the condenser in order to maintain pressure. Non-condensable gas consists of gas originating from the geothermal field (geothermal gas) and air. Geothermal steam contains a mixture of gases obtained from dry geothermal fields, or if obtained from wet geothermal fields, a steam/water mixture. The relative proportions and quantities of each gas vary from field to field and sometimes from well to well within the same field. The gas content of geothermal steam is usually expressed in %-weight in steam.

Table 1 shows the composition of dry gas by %-weight in steam from three production wells in Patuha geothermal field. The weight fraction of the total gas in the steam from wells W8, W6 and W9 is 1.10, 1.77 and 1.75%, respectively.

TABLE 1: Non-condensable gases in three wells in Patuha

Parameter	Composition of dry gas (% weight)		
	W8	W6	W9
CO_2	95.052	95.922	96.863
H_2S	2.946	1.745	1.298
NH_3	< 0.0001	0.008	0.007
SO_2	< 0.0001	< 0.0001	< 0.0001
HCl	0.184	0.048	0.037
HF	< 0.0001	< 0.0001	< 0.0001
H_2	0.020	0.020	0.020
O_2+Ar	0.010	0.008	0.009
N_2	1.772	2.223	1.750
CO	< 0.0001	< 0.0001	< 0.0001
CH_4	0.020	0.024	0.017
C_2H_6	0.000	0.000	0.000
C_3H_8	< 0.0001	< 0.0001	0.000
C_4H_{10}	< 0.0001	< 0.0001	0.000
C_5H_{12}	< 0.0001	< 0.0001	< 0.0001

2. STUDY DESCRIPTION

2.1 General overview

There are nine production wells dedicated to plant unit 1. Enthalpy of the wells ranges between 2400 and 2700 kJ/kg. For the study it is assumed that the content of non-condensable gas in the wells is 1.75% weight of steam. The wells were tested in 1996 and 2003. The steam will be utilized for electricity generation at the optimum output power. A single-flash power plant will be used for conversion energy. The wells will choke at the optimum well head pressure. A cyclone separator will be used to separate the fluid from the wells. The steam will flow to a demister before going to the turbine. The direct contact of barometric leg type condenser is used for the plant. For a gas extraction system, the plant will use a hybrid steam ejector, an integrated steam ejector and a vacuum pump. A wet cooling tower with a mechanical induction system will be used as the heat rejection system of the plant. This will be the first power plant in the area.

2.2 Objective of this study

The objective of this study is to make a thermodynamic analysis to determine the optimum output power from a single-flash power plant. To achieve the objective, the study will determine the following:

- Turbine power;
- Motive steam mass flowrate for the steam ejector;
- Auxiliary power: circulation water pumping power, fan power, vacuum pump power and others;
- The optimum separator and condenser pressure which gives the maximum output power.

A double-flash system will be analysed for improvement on the single-flash power plant. Separation water from the separator will be flashed at the low-pressure separator. The low-pressure steam from the second flash will combine with high-pressure steam at the low-pressure turbine.

Low-pressure steam from one of four non-commercial wells will be used to increase the mass flowrate to the low-pressure turbine. The study will give the total additional output power from the plant.

3. METHODOLOGY

3.1 General approach

The study covers calculations based on theory which can be found in engineering textbooks. The power plant design is provided by the calculations steps below:

1. Well head pressure chosen to maximize the output power;
2. Total mass flowrate and enthalpy mixture of the production wells' system;
3. Total mass flowrate of fluid and the quality of fluid after the separation process;
4. Turbine power;
5. Condenser heat load and cooling water flowrate;
6. Gas extraction mass flowrate and motive steam required for the steam ejector;
7. Inter condenser heat and mass flowrate of cooling water required for the inter condenser;
8. Electric motor power required for driving the vacuum pump;
9. Electric motor power for driving the hot well pump;
10. Mass of circulating water per unit mass of dry air;
11. Electric motor power required for the cooling tower fan;
12. Electrical motor power required for inter condenser cooling water circulation.

3.2 Scope and limitations

The scope of the study is to make calculations for a preliminary design of a single-flash system. The single-flash system will use direct contact with a barometric leg condenser, a hybrid steam ejector, and a wet cooling tower mechanically induced draft. The economic calculations are not included for the power plant equipment. Double-flash calculations were only made for one value of low-pressure separation, due to limitations of available data.

3.3 Sources of data

The data used in this analysis was obtained from the following sources:

1. Well production tests in 1996 and 2003;
2. Enthalpy and dryness measurement of three wells in 2004;
3. Preliminary assessment of Patuha project in 1998;
4. Downhole pressure and temperature survey in 2003.

3.4 General assumptions

1. W3 was not tested in 2003, but a 40% capacity reduction from its initial productivity was conservatively assumed;
2. For calculation of molecular weight, four major gases were assumed to represent the non-condensable gases;
3. A hybrid-steam ejection, using a two-stage steam ejector series combined by a liquid ring vacuum pump is assumed for Patuha field;
4. Direct contact condensers are generally preferred in direct geothermal steam cycle applications;
5. The efficiency of the pump, electric motor and fan are 0.8, 0.85 and 0.75;
6. The efficiency of the turbine is 0.75;
7. Pressure drop from wells to separator is assumed to be maximized. The allowed drop pressure is 0.05 bar-a per 100 m;
8. Pressure drop in the demister is 0.01 bar-a;

9. The temperature difference between cooling water entering the cooling tower and hot air leaving the cooling tower is 6ºC;
10. The temperature drop of the hot water from the wells to the cooling tower is 3ºC; and
11. The wet bulb temperature for Patuha field is 25ºC.

3.5 Analysis and treatment of data

The data was compiled, organized and prepared for processing using Engineering Equation Solving (EES) software. The regression formula for production wells was determined using Curve Expert software. The graphs in Section 6 were plotted from EES software. Standard international (SI) units were used for the calculations.

3.6 Accuracy and general sources of error

An iteration process was used to make corrections to the determined steam consumption for the steam ejector. The same iteration was used for calculating the value of corrected condenser pressure. One iteration was sufficiently accurate.

The enthalpy discharge data was available for only three wells. For the others, the enthalpy was determined using the enthalpy from the one of the three wells which had similar downhole characteristics in pressure and temperature. Non-condensable gas data was available for only three wells. Non-condensable gas of the other wells was determined using the non-condensable gas data from a well at the same pad. The data available for the non-commercial wells has low accuracy, but was used in the double-flash calculations.

4. THEORETICAL ANALYSIS

4.1 Gathering system design considerations

The single-flash steam plant is the mainstay of the geothermal power industry. In July 2004 there were 135 units of this kind in operation in 18 countries around the world. In September 2006, 12 units were in operation in Indonesia with a single-flash steam system, with a total capacity of 660 MW.

The single-flash plant is a relatively simple way to convert geothermal energy into electricity when the geothermal wells produce a mixture of steam and liquid. The mixture is separated into distinct steam and liquid phases with a minimum loss of pressure. This is done in a cylindrical cyclone pressure vessel, usually oriented with its axis vertical, where the two phases disengage owing to their inherently large density differences. The siting of the separator is a part of the general design of the plant. Separators can be located at the power house, at satellite stations in the field or at the well head. The separators in this single-flash system model were located at the power house. A typical 60 MW single-flash power plant needs 10-12 production wells and 3-4 injection wells (Dipippo, 2005).

4.2 Thermodynamics of the conversion process

4.2.1 Temperature-entropy process diagram

The processes undergone by the geo-fluid are best viewed in a thermodynamic state diagram in which the fluid temperature is plotted on the ordinate and the fluid specific entropy is plotted on the abscissa. A temperature-entropy diagram for the single-flash plant is shown in Figure 3.

4.2.2 Flashing process

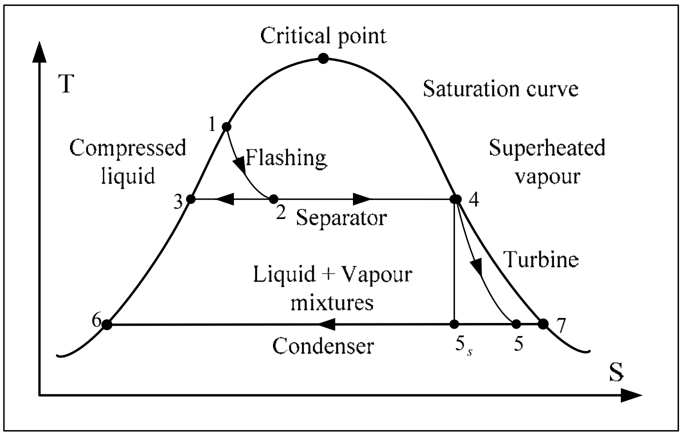

FIGURE 3: T-S diagram for a single-flash plant

The sequence of processes begins with geo-fluid under pressure at state 1, close to the saturation curve. The flashing process is modelled as one at constant enthalpy, an isenthalpic process, because it occurs steadily, spontaneously, essentially adiabatically, and with no work involvement. Any change in the kinetic or potential energy of the fluid as it undergoes the flash, is also neglected. Thus it can be written as:

$$h_1 = h_2 \tag{1}$$

where h = Enthalpy (kJ/kg).

4.2.3 Separation process

The separation process is condensable as one at constant pressure, an isobaric process, once the flash has taken place. The quality of dryness fraction, *X* of the mixture that forms after the flash, state 2, can be found from:

$$X_2 = \frac{h_2 - h_3}{h_4 - h_3} \tag{2}$$

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 separator.

4.2.4 Turbine expansion process

The work produced by the turbine per unit mass of steam flowing through it, w_t (kJ/kg), is given by:

$$w_t = h_4 - h_5 \tag{3}$$

For a turbine under steady operation, the inlet state of the working fluid and the exhaust pressure are fixed. Therefore, the ideal process for an adiabatic turbine is an isentropic process between the inlet state and the exhaust pressure.

What is desired of a turbine is the ratio of the actual work output of the turbine to the work output that would be condensable if the process between the inlet state and the exit pressure were isentropic:

$$\eta_t = \frac{Actual\ turbine\ work}{Isentropic\ turbine\ work} = \frac{W_a}{W_s} \tag{4}$$

where η_t = Isentropic efficiency of a turbine.

Usually the changes in kinetic and potential energies, associated with a fluid stream flowing through a turbine, are small relative to the change in enthalpy and can be neglected. Then the work output of an adiabatic turbine simply becomes the change in enthalpy, and the equation becomes:

$$\eta_t = \left(\frac{h_4 - h_{5a}}{h_4 - h_{5a,s}} \right) \quad (5)$$

h_{5a} and $h_{5a,s}$ are the enthalpy values at the exit state for the actual process and the isentropic process, respectively (Figure 4). The value of η_t greatly depends on the design of the individual components that make up the turbine. A well designed, large turbine has isentropic efficiency above 90%. For a small turbine, however, it may drop below 70%. The value of isentropic condensability of a turbine is determined by measuring the actual work output for the measured inlet conditions and the exit pressure (Cengel and Boles, 2006). This value can then be used conveniently in the design of power plants. The power developed by the turbine is given by:

$$\dot{W}_t = \dot{m}_s\, w_t = X_2\, \dot{m}_{total}\, w_t \quad (6)$$

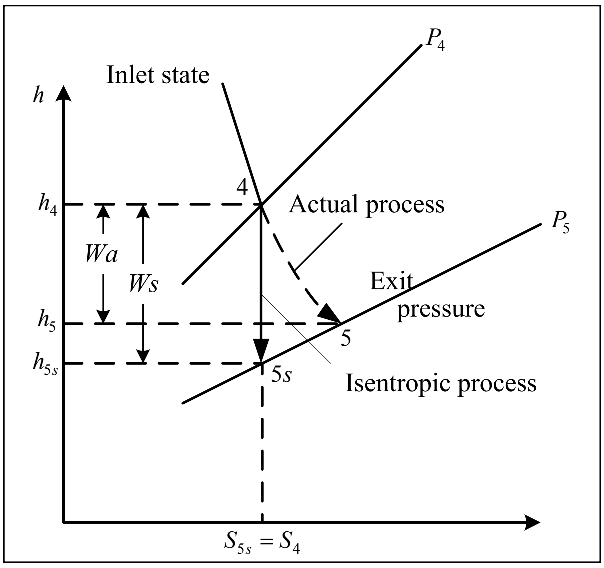

FIGURE 4: *h-S* diagram for the actual process and isentropic process of an adiabatic turbine

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:

$$\dot{W}_e = \eta_g\, \dot{W}_t \quad (7)$$

All auxiliary power requirements for the plant must be subtracted from this to obtain the net, sellable power. These parasitic loads include all pumping power, the cooling tower fan and station lighting.

4.2.5 Condensing process

The function of a condenser is to condense the incoming exhaust steam and thereby create a sub-atmospheric condition in the condenser. The condenser pressure is dependent only on the quantity of non-condensable gas present in the condenser and the maximum temperature that the cooling water attains. During the passage through the condenser, pressure will drop, thereby increasing the turbo-generator output.

General condenser design ensures that the temperature of the non-condensable gas (and associated water vapour) at the condenser outlet will be below the maximum temperature that the cooling water attains. This is achieved by having a gas cooling part within the main condenser or by having a completely separate gas cooler, external to the condenser.

In a conventional thermal power station all the steam which is exhausted into the surface condenser is returned into the cycle as a highly demineralised condensate. In a geothermal power station such a requirement does not exist. While it may be possible to exhaust the steam and geothermal gases from the turbine at atmospheric pressure, it is economical to have a vacuum exhaust.

In direct geothermal steam cycle applications, direct contact condensers are generally preferred. Direct contact condensers have appreciable lower initial costs and tend to be simpler in design than surface condensers. Unlike the latter, the direct contact condenser requires little maintenance or cleaning, and the head transfer performance does not deteriorate with time. Direct contact condensers may occupy about one third the space of a surface condenser for the same duty, and there will be a corresponding reduction in costs of turbine pedestals and other concrete work. The main disadvantage of direct contact condensers is associated with the fact that the condensate and dissolved geothermal gases are mixed with the cooling water (Geothermal Institute, 1992a). The First Law of thermodynamics leads to the following equation on the required flowrate of cooling water:

$$\dot{m}_{cw} = X_2\, \dot{m}_{total}\left[\frac{h_5 - h_6}{c_p(T_6 - T_{cw})}\right] \tag{8}$$

Because of the gases mixed with the cooling water, condenser pressure should be corrected. The volume flowrate of gas extraction in the condenser is higher than the volume flowrate of non-condensable gas. Gas extraction consists of non-condensable gas and steam (Pall Valdimarsson, personal communication), therefore:

$$\dot{V}_{gas,extraction} = \dot{V}_{ncg} + \dot{V}_{steam} \tag{9}$$

$$\frac{M_{ncg}}{M_{steam} + M_{ncg}} = \frac{\dot{V}_{ncg}}{\dot{V}_{ncg} + \dot{V}_{steam}} \tag{10}$$

$$\frac{P_{ncg}}{P_{cond}} = \frac{\dot{V}_{ncg}}{\dot{V}_{ncg} + \dot{V}_{steam}} \tag{11}$$

Then the new condenser pressure is determined by the following equation:

$$P_{con,x} = P_{con}\left(1 + \frac{P_{ncg}}{P_{ncg} + P_{steam}}\right) \tag{12}$$

where M_{ncg} = Molecular weight of non-condensable gas;
M_{steam} = Molecular weight of steam;
P_{con} = Condenser pressure (bar-a);
$P_{con,\,x}$ = New condenser pressure after correction (bar-a);
P_{ncg} = Pressure of non-condensible gas (N/m^2);
P_{steam} = Pressure of steam (N/m^2);
$\dot{V}_{ncg}$ = Volume flowrate (of non-condensible gas) (m^3/s)
$\dot{V}_{steam}$ = Volume flowrate of steam (m^3/s)

In the barometric condenser, the cooling water is made to cascade down a series of baffles in the form of water curtains or sheets of a high surface-to-volume ratio to mix thoroughly with the turbine exhaust that is trying to rise from a lower inlet. The steam condenses and the mixture goes down a tail pipe to the hot well.

The tail pipe compresses the mixture to atmospheric pressure at the hot well by virtue of its static head and, thus, replaces the pump used in the spray-type condenser. The pressure differential created by the tail pipe must overcome the pressure difference between the atmosphere (P_{atm}) and the condenser pressure, plus the friction pressure drop caused by the mixture flow (ΔP_f) in the tail pipe itself, thus:

$$\rho H \frac{g}{g_c} = P_{atm} - P_{con,x} + \Delta P_f \tag{13}$$

and

$$\Delta P_f = \frac{V^2}{2g} \tag{14}$$

where ρ = Density (kg/m^3);
H = Height of tail pipe (m);
g = Gravitational acceleration (m/s^2);
g_c = Conversion factor, 1 kgm/Ns^2;
V = Velocity of mixture in tail pipe (m/s)°.

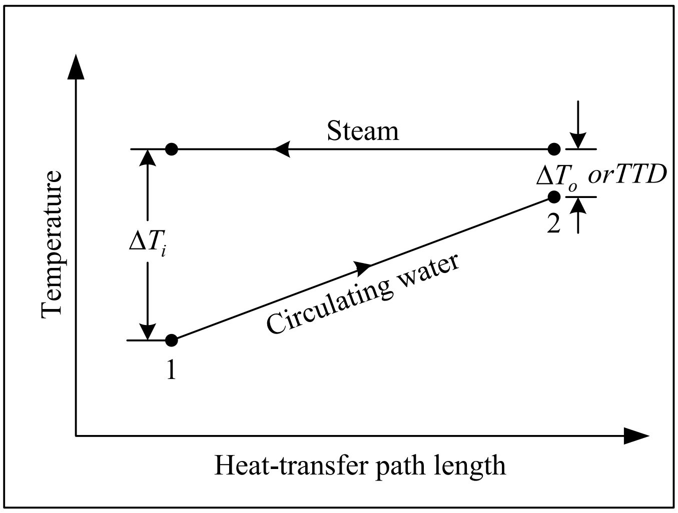

FIGURE 5: Condenser temperature distribution; ΔT_i = Difference between saturation steam temperature and inlet circulating water temperature (°C) and ΔT_o = Difference between saturation steam temperature and outlet circulating water temperature (°C)

Figure 5 shows the condenser temperature distribution. ΔT_o is also called the terminal temperature difference, *TTD*. A large *TTD* results in a small condenser but increasing water flow, as the water temperature rises (T_2-T_1), is reduced. A small *TTD* results in a larger condenser, reduced water flow and higher exit-water temperature.

An oversized condenser, bigger than the design, will increase the temperature difference in the condenser and lower condenser pressures. Proper design, therefore, depends upon many factors, such as capital costs, operating costs, water availability and environmental concerns.

The circulating water inlet temperature should be sufficiently lower than the steam saturation temperature to result in reasonable values of ΔT_o. It is usually recommended that ΔT_i should be between about 11 and 17°C and that the ΔT_o, or *TTD* should not be less than 2.8°C (El-Wakil, 1984).

4.2.6 Gas extraction system

Steam ejectors

Steam jet ejectors are used extensively in conventional thermal and geothermal power stations for the extraction of non-condensable gas (and associated water vapour). Two-stage steam jet ejector systems are generally used in geothermal power stations.

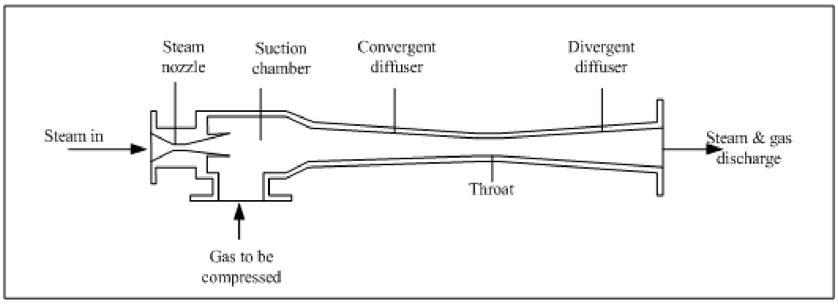

FIGURE 6: Single-stage steam ejector

Figure 6 shows a diagram of a single-stage steam jet ejector. A steam jet ejector operates on the venture principle. The motive steam is expanded through the nozzle to the design suction pressure. The pressure energy of the steam is converted to velocity energy and on leaving the nozzle at high supersonic velocities the steam passes through the suction chamber and enters the converging diffuser or entrainment, as gas and associated water vapour.

A two-stage steam jet ejector system consists of two single-stage ejectors operating in series, with each discharging into a condenser. Figure 7 shows a diagram of a two-stage system. For given suction and discharge pressure, the steam consumption of a single stage ejector is dependent on the mass flowrate (and molecular weight) of the gas to be handled.

The function of an inter-condenser is to condense the first stage motive steam and to remove part of the water vapour that initially saturated the non-condensable gas. This ensures that the second stage handles only non-condensable gas saturated with water vapour. Therefore, an inter-condenser minimises the required duty in that it condenses the second stage motive steam and ensures that a minimum of water vapour is discharged with the non-condensable gas. In addition, the after-condenser acts as a noise suppressor.

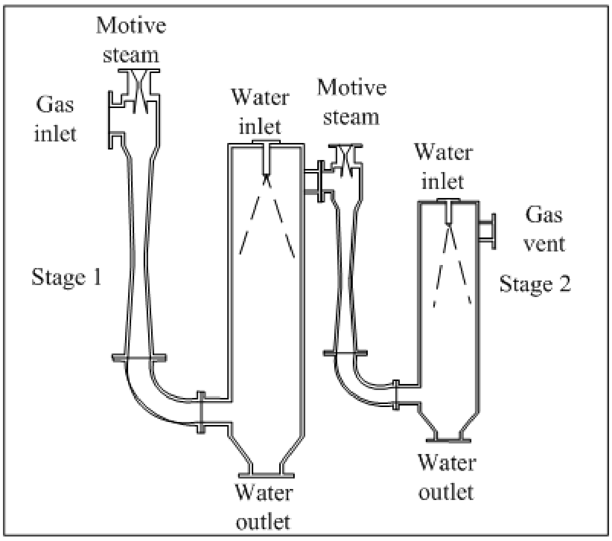

FIGURE 7: Two-stage steam ejector system

In order to reduce the motive steam consumed by the ejector, the second stage can be replaced by a vacuum pump. Integration of a steam jet ejector with a vacuum pump is commonly referred to as a hybrid system. It is one of the more efficient methods for producing a process vacuum. In addition, a two-stage ejector system is also installed for redundancy. Next to the first-stage ejector, liquid ring vacuum pumps are the most used vacuum producing devices in the industry.
To calculate the required power consumption or steam consumption necessary to remove a given quantity of water vapour saturated non-condensable gas from a condenser, it is necessary to know the following (Geothermal Institute, 1992b):

a. Mass flowrate and molecular weight of non-condensable gas;
b. Suction pressure;
c. Discharge pressure;
d. Gas suction temperature (for this analysis it was assumed that the second stage gas suction temperature equalled the first);
e. Mass flowrate of the water vapour; and
f. Steam pressure.

In general, the capacity of a steam jet ejector is stated in terms of air removal ability, i.e. kg/s. To calculate the steam consumption, the following procedure is used:

a. Treat the non-condensable gas and water vapour independently. Little error is introduced if the non-condensable gas, although of several constituent gases, is treated as a single gas of calculated molecular weight, as long as the bulk of the non-condensable gas is of one gas;
b. Find the entrainment ratio for the non-condensable gas and water vapour. This ratio is defined as the ratio of the weight of gas to the equivalent of the non-condensable gas and water vapour.
c. Determine the total air equivalent for the non-condensable gas and water vapour;
d. Calculate the compression ratio, the ratio of discharge pressure to suction pressure at each stage;
e. Calculate the expansion ratio of the steam, i.e. steam pressure/suction pressure;
f. From the intersection of the compression ratio and expansion ratio, obtain the air-to-steam ratio;
g. Calculate the required steam flowrate by dividing the total air equivalent by the air-to-steam ratio.

To calculate the gas volume flowrate and the mass flowrate of the saturated water vapour, the Gas law is used.

1. *Dalton's laws of partial pressure*
   a. The saturation pressure of a vapour depends only upon its temperature. The quantity of vapour required to saturate a given space is independent of any other gas or vapour which may be present.
   b. The pressure of a mixture of gases and vapour is the sum of the pressure each would exert separately if it were alone in the space occupied by the mixture.

2. *Ideal gas equation:*

$$P\dot{V} = \dot{m}RT \tag{15}$$

where $\dot{V}$ = Gas volume flowrate (m^3/s);
P = Pressure (N/m^2);
$\dot{m}$ = Gas mass flowrate (kg/s);
R = R_o/M_{wt} = 8,314/ M_{wt} (J/kg K);
M_{wt} = Molecular weight of the gas;
T = Temperature of non-condensable gas (°C) saturated with water vapour at 25ºC or 298.1K.

In the situation of water vapour saturated non-condensable gas:

$$P_t = P_{ncg} + P_{wv} \tag{16}$$

Therefore:

$$P_{ncg} = P_t - P_{wv} \tag{17}$$

and

$$\dot{V}_t = \dot{V}_{wv} + \dot{V}_{ncg} = \dot{V} \tag{18}$$

$$\dot{V} = \left(\frac{\dot{m}RT}{P} \right)_{twvncg} \tag{19}$$

$$\dot{V} = \frac{\left(\dot{m}RT \right)_{ncg}}{\left(P_t - P_{wv} \right)} \tag{20}$$

The specific volume of water vapour or steam in a saturated condition is dependent only on its temperature, therefore:

$$\dot{m}_{wv} = \frac{\dot{V}}{n_{wv}} \tag{21}$$

where n_{wv} = The specific volume of the water vapour (m^3/kg)

Liquid ring vacuum pump

A liquid ring pump system is often used in geothermal, nuclear and conventional power stations for the maintenance of condenser vacuums. A system usually consists of two liquid ring pumps operating in series. The first case of a liquid ring pump being used in a geothermal power station for the removal of non-condensable gas was in the Onikobe power station, Japan (Geothermal Institute, 1992b). More recently, a liquid ring pump has been used in many geothermal power stations for the removal of non-condensable gas. In a geothermal power plant, a liquid ring pump is used in place of a second stage steam jet ejector. The compression action is performed by a rotating ring of liquid, usually water. Liquid ring pump performance data is best obtained from manufacturers' published data.

4.2.7 Cooling tower process

Power plants generate large quantities of waste heat that is often discarded through cooling water in nearby lakes or rivers. In some cases, however, the cooling water supply is limited or thermal pollution is a serious concern. In such cases the waste heat must be rejected to the atmosphere, with cooling water re-circulating and serving as a transport medium for heat transport between the source and the sink (the atmosphere). One way of achieving this is through the use of wet cooling towers.

A wet cooling tower is essentially a semi enclosed evaporative cooler. Air is drawn into the tower from the bottom and leaves through the top. Warm water from a condenser is pumped to the top of the tower and is sprayed into the air-stream. The purpose of spraying is to expose a large surface area of water to the air. As the water droplets fall under the influence of gravity, a small fraction of water, usually a few percent, evaporates and cools the remaining water. The temperature and the moisture content of the air increases during this process.

The cooled water is collected at the bottom of the tower and pumped back to the condenser to absorb additional waste heat. Make-up water must be added to the cycle to replace the water loss due to evaporation and air draft. To minimize the water carried away by the air, drift eliminators are installed in the wet cooling tower above the spray section. The air circulation in the cooling tower is provided by fans; therefore, it is classified as a forced-draft cooling tower (Cengel and Boles, 2006).

The amount of water vapour in the air can be specified in various ways. Probably the most logical way is to specify directly the mass of water vapour present in a unit mass of dry air. This is called the absolute or specific humidity but is also called the humidity ratio and is denoted by ω.
The first law steady-state steady-flow equation with three fluids will now be written for the tower fill, see the system shown in Figure 8. It applies to all types of wet towers. Changes in potential and kinetic energies and heat transfer are all negligible. No mechanical work is done. Thus, only enthalpies of the three fluids appear. Following psychometric practice, the equation is written for a unit mass of dry air (El-Wakil, 1984).

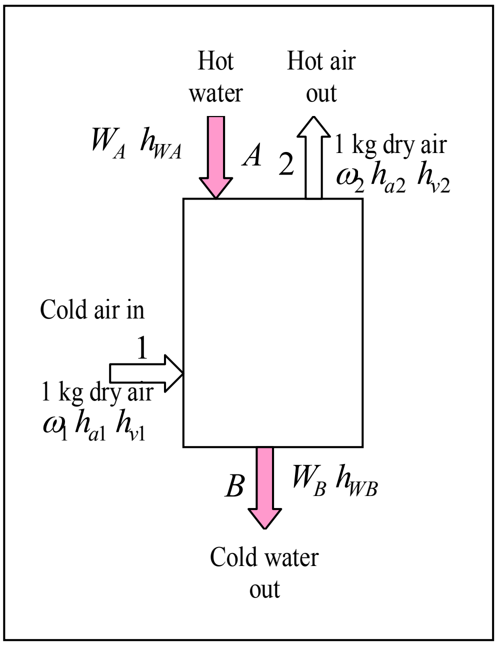

FIGURE 8: The tower fill as a steady-state steady-flow system

$$h_{a1} + \omega_1 h_{v1} + W_A h_{WA} = h_{a2} + \omega_2 h_{v2} + W_B h_{WB} \tag{22}$$

where h_a = Enthalpy of dry air (kJ/kg);
ω = Mass of water vapour per unit mass of dry air, absolute humidity (dimensionless);
h_v = Enthalpy of water vapour (kJ/kg);
W = Mass of circulating water per unit mass of dry air (dimensionless);
h_W = Enthalpy of circulating water (kJ/kg).

Because of the low pressure and temperatures commonly encountered in towers, the above equation can be simplified with little error by following approximations.

$$h_{a2} - h_{a1} = c_p(T_2 - T_1) \tag{23}$$

where c_p = Specific heat of air kJ/kg
T = Temperature (°C)

The dry air goes through the tower unchanged. The circulating water loses mass by evaporation. The water vapour in the air gains mass due to the evaporated water. Thus, based on a unit mass of dry air, and with the subscripts 1 and 2 referring to air inlet and exit, and the subscripts A and B to circulating-water inlet and exit, respectively (the air leaving the system at 2 is often saturated):

$$\omega_2 - \omega_1 = W_A - W_B \tag{24}$$

Equation 22 can now be written in the form:

$$\omega_1 h_{v1} + W_A h_{WA} = c_p(T_2 - T_1) + \omega_2 h_{v2} + (W_A - (\omega_2 - \omega_1) h_{WB}) \tag{25}$$

From the cooling water calculation in the condenser section, it is known that the volume flowrate of hot cooling water entering the cooling tower is $\dot{m}_{cw}$ (m^3/s), so dry air mass flowrate can be found from:

$$\dot{m}_{dry,air} = \frac{\dot{m}_{cw}}{W_A} \tag{26}$$

Then, exhaust mass flowrate from the cooling tower can be calculated using the following equation:

$$\dot{m}_{exhaust} = \dot{m}_{dry,air}(\omega_2 - \omega_1) + (0.03(\dot{m}_{dry,air}(\omega_2 - \omega_1)) \tag{27}$$

The equation assumes that drift losses are 3% of total evaporation losses.

The dry air volume flowrate is then defined as follows:

$$\dot{V}_{dry,air} = \frac{\dot{m}_{dry,air}}{\rho_{air,out}} \tag{28}$$

4.2.8 Pumps

In a power plant, pumps play an important part in cooling water circulation. In general, centrifugal pumps are commonly used for this purpose. The important operating characteristics of a pump are capacity Q, head H, power P, and efficiency η. The pump capacity Q is the volume of liquid per unit time delivered by the pump. In SI units, the corresponding units are litres per second (l/s) and cubic meters per second (m^3/s).

The pump head H represents the net work done on a unit weight of liquid in passing from the inlet or suction flange 's' to the discharge flange 'd'. It is given by:

$$H = \left(\frac{p}{\gamma} + \frac{V^2}{2g} + Z\right)_d - \left(\frac{p}{\gamma} + \frac{V^2}{2g} + Z\right)_s \tag{29}$$

The term p/γ, called the pressure head or flow work, represents the work required to move a unit weight of liquid across an arbitrary plane perpendicular to the velocity vector V against the pressure p.

The term $V^2/2g$, called the velocity head, represents the kinetic energy of a unit weight of liquid moving with velocity V. The term Z, called the elevation head or potential head, represents the potential energy of a unit weight of liquid with respect to the chosen site (Karassik, 1985).

The power (kW) is given by:

$$P = 9.797QH \tag{30}$$

5. ANALYSIS AND CALCULATIONS

The analysis starts by designing the flow diagram of the system. Figure 9 shows the single-flash flow diagram of a preliminary design for plant unit I, Patuha. The plant consists of nine production wells for steam supply and an injection well for separated water. A separator and a demister are used for the separation process. A turbine powers the generator, producing electricity. A direct contact condenser with barometric leg is used to condense the turbine exhaust. Two-stage gas ejectors are used, combined with a vacuum pump, as a hybrid system. Centrifugal pumps are used for water circulation from the condenser to the cooling tower. Appendix I shows the flow diagram for the single-flash power plant unit 1 at Patuha with necessary specifications, and Appendix II lists the specifications for the plant. Nomenclature with abbreviations used in the figures is given at the end of the report.

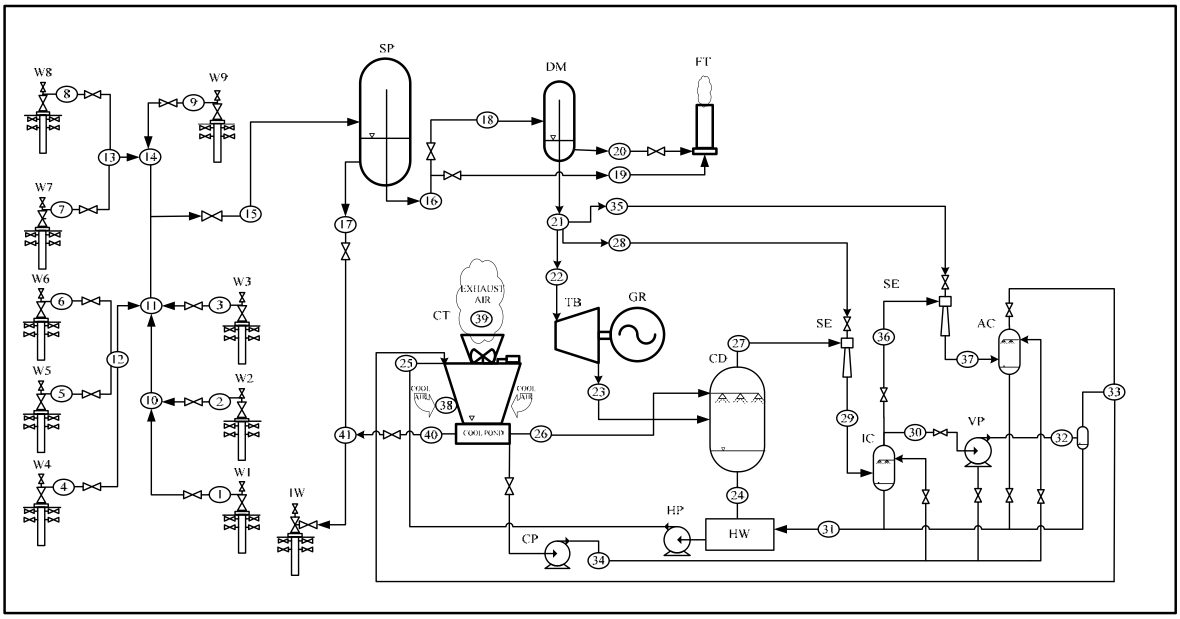

FIGURE 9: Single-flash flow diagram of a preliminary design of the power plant for unit I, Patuha

5.1 Wells

5.1.1 Productivity data

When drilling a geothermal well, it is important to record its productivity data, the total geo-fluid mass flowrate measured in kg/s as a function of the well head pressure (WHP) measured in bar. The flow increases as the well is opened and the pressure lowers. The first step in designing a plant is to decide which well head pressure should be chosen to maximize the output power from a single-flash plant connected to these wells. For this purpose it is convenient to correlate the productivity curve with the best least-squares fit. The well production correlations can be found in Appendix III. In the Patuha field there are nine production wells intended for plant unit 1. The nine production wells are feeding two-phase fluid to a cyclone separator at the power house. The separated steam enters the turbine via a short pipeline and the separated liquid is sent to an injection well. Table 2 shows well capacity estimations, enthalpy and distances from the separator pad. The distances are used for pressure drop calculations. The maximum design pressure drop is 0.05 bar per 100 m.

TABLE 2: Estimated capacity of wells in Patuha

No.	Wells	Estimated capacity (MW)	Enthalpy (kJ/kg)	Distance (km)
1	W1	12.50	2400	3.00
2	W2	2.90	2400	2.00
3	W3	8.20	2400	0.75
4	W4	3.20	2480	1.50
5	W5	13.90	2480	1.50
6	W6	7.70	2480	1.50
7	W7	4.50	2700	0.50
8	W8	5.00	2700	0.50
9	W9	2.00	2580	1.50
Total		59.90		

5.1.2 Heat and mass balance of a well system

The second step in plant design is to calculate the total mass flowrate and enthalpy mixture of the production wells. It can be determined by a heat and mass balance analysis as follows:

Figure 10 shows the well flow process system, consisting of nine production wells (W1, W2, W3, W4, W5, W6, W7, W8 and W9) and one injection well (IW). Point 15 represents the total mass of fluid flowing into the separator. Point 17 represents water separation from the separator to the injection well.

Point 15 is the sum of the mass flowrates at points 14 and 11; the enthalpy of the mixture can be calculated using the following equations:

$$\dot{m}_{15} = \dot{m}_{14} + \dot{m}_{11} \tag{31}$$

and

$$h_{15} = \left[\frac{h_{14}\,\dot{m}_{14} + h_{11}\,\dot{m}_{11}}{\dot{m}_{14} + \dot{m}_{11}}\right] \tag{32}$$

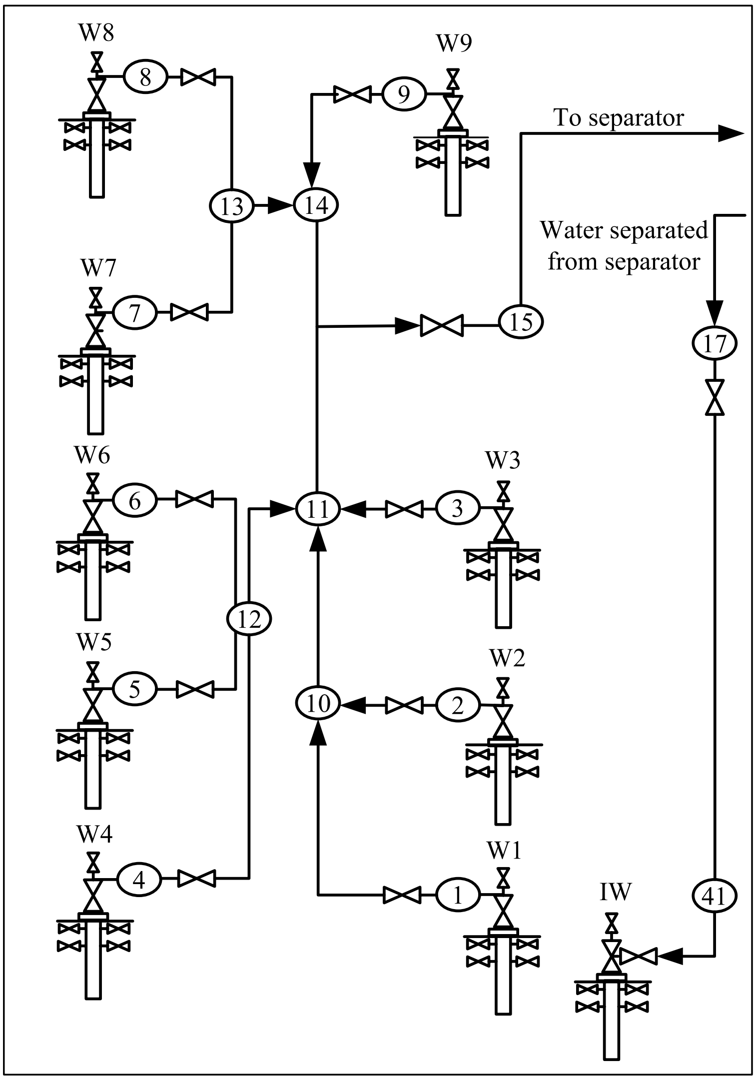

FIGURE 10: Well flow plan

5.2 Separator and demister

The third step in plant design is to analyse and calculate the total mass flowrate of the fluid and its quality after the separation process. Figure 11 shows the flow process of the separator, demister and flash tank (Siregar, 2004). The pressures at points 15, 16 and 17 are equivalent and the same as the separator pressure:

$$P_{15} = P_{16} = P_{17} = P_{separator} \tag{33}$$

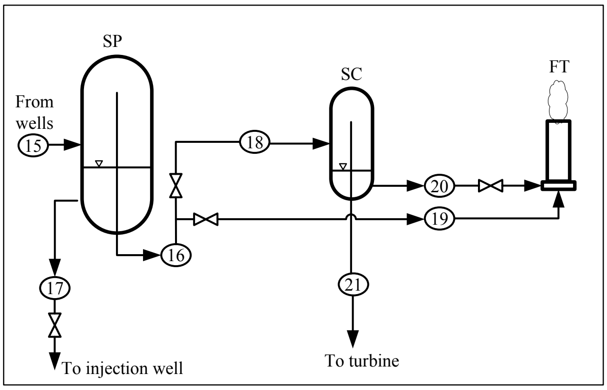

FIGURE 11: Separator, demister and flash tank

The quality of the fluid at the separator outlet is:

$$X_{15} = \left[\frac{h_{15} - h_{17}}{h_{16} - h_{17}}\right] \tag{34}$$

The mass flowrate of fluid coming from the separator is given by:

$$\dot{m}_{16} = X_{15}\,\dot{m}_{15} \tag{35}$$

Then, the mass flowrate of the separated water from the separator to the re-injection well may be written as:

$$\dot{m}_{17} = (1 - X_{15})\,\dot{m}_{15} \tag{36}$$

The mass flowrate of the fluid at point 18, coming from the separator and going into the demister, can be determined by the following equation:

$$\dot{m}_{18} = \dot{m}_{16} - \dot{m}_{19} \tag{37}$$

Then the mass flowrate of the fluid at point 21 going to the turbine can be found from:

$$\dot{m}_{21} = \dot{m}_{18} - \dot{m}_{20} \tag{38}$$

The steam quality at point 21 is given by the following equation:

$$X_{21} = \left[\frac{h_{18} - h_{20}}{h_{18} - h_{20}}\right] \tag{39}$$

5.3 Turbine

The fourth step in plant design is to analyse and calculate the turbine work. The isentropic efficiency of the turbine is given by the following equation:

$$\eta_t = \left[\frac{h_{22} - h_{23}}{h_{22} - h_{23,s}}\right] \tag{40}$$

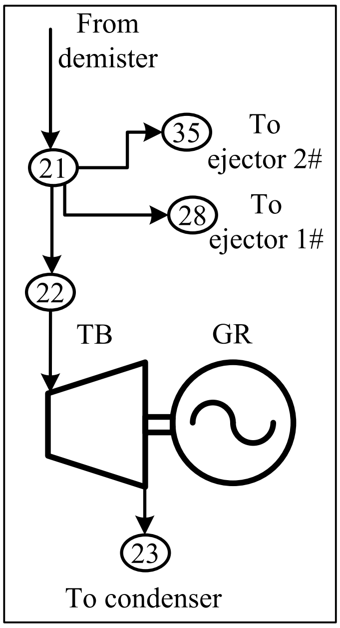

FIGURE 12: Turbine

Figure 12 shows that steam from the demister (point 21) is used to supply the turbine (point 22), the first stage steam ejector (point 28) and the second stage steam ejector (point 35). In normal operation, the second stage ejector is replaced by a liquid ring vacuum pump, allowing the second stage ejector to be operated as a redundancy.

So, in normal operation, steam supplied from the demister is used for the turbine and first stage steam ejector only. The steam mass flowrate supplied to the turbine is given by the following equation:

$$\dot{m}_{22} = \dot{m}_{21} - \dot{m}_{28} \tag{41}$$

The steam quality at point 22 is the same as at point 21, so the turbine power is given by the following equation:

$$\dot{W}_t = X_{22}\,\dot{m}_{22}\,(h_{22} - h_{23}) \tag{42}$$

The generator efficiency is η_g , hence the turbine-generator power is defined by the following equation:

$$\dot{W}_g = (\eta_g / \eta_t)\dot{W}_t \tag{43}$$

The turbine efficiency is assumed to be 0.85 and the generator efficiency is assumed to be 0.75.

5.4 Condenser

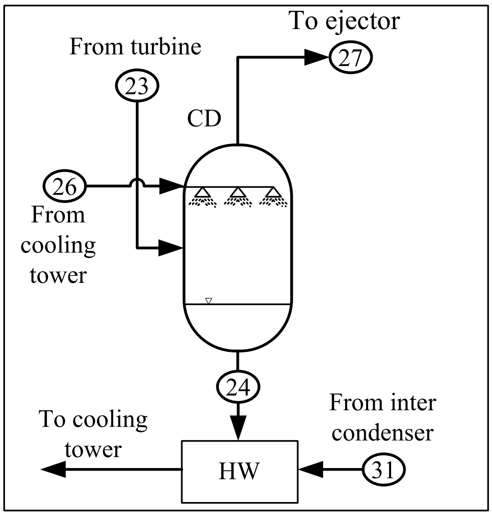

FIGURE 13: Condenser

The fifth step in plant design is to analyse and calculate the condenser heat load and the cooling water mass flowrate. Figure 13 shows the flow diagram of the condenser. The condenser heat load can be calculated using the following equation:

$$Q_{con} = (h_{231} - h_{24})\dot{m}_{221} \tag{44}$$

where Q_{con} = Condenser heat load (kJ);
h_{231} = Enthalpy at point 23 after iteration using the $P_{con,x}$ (kJ/kg).
$\dot{m}_{221}$ = Mass flowrate at point 22 found by subtracting the motive steam of the first stage steam ejector, kg/s.

The mass flowrate of the cooling water for the condenser can be found by this equation:

$$\dot{m}_{cw} = \frac{Q_{con}}{h_{24} - h_{26}} \tag{45}$$

The correction pressure for the condenser can be found from Equation 12.

The height of the barometric leg of the condenser can be calculated using Equations 13 and 14, where the density of the mixture is ρ_{24} and its velocity 0.2 m/s. Knowing the height of the barometric leg is necessary for the design condenser and the hot well pump head calculation.

5.5 Hybrid steam ejector

The sixth step in plant design is to analyse and calculate the gas extraction mass flowrate and the motive steam required for the steam ejector. The following is the procedure for calculating the motive steam flow, as discussed in a theoretical analysis:

1. Calculate the gas volume flowrate and mass flowrate of vapour-water using Equations 16-21. The non-condensable gas mass flowrate which needs to be removed from the condenser is determined from the mass flow of fluid which entered the condenser. Patuha non-condensable gas is 1.75% by weight of steam.

TABLE 3: Molecular weight of non-condensable gases

Gas	Composition of gases (% weight)				Molecular weight	Gas contribution
	W8	W6	W9	Average		
CO_2	95.05	95.92	96.85	95.94	44.00	42.21
H_2S	2.95	1.75	1.30	2.00	34.08	0.68
HCl	0.18	0.05	0.04	0.09	36.46	0.03
N_2	1.77	2.22	1.75	1.91	28.01	0.54
Total						43.46

Table 3 is the summary of non-condensable gas molecular weight calculations done by calculating the weighted average of the 4 major gases representing non-condensable gases in the Patuha field. For example, CO_2 has a molecular weight of 44. The total weight of CO_2 is 95.95% (average) of the molecular weight of non-condensable gases. Thus, the contribution of CO_2 to the molecular weight of non-condensable gases is 42.21. Similarly, the contribution of H_2S is 0.68, HCl 0.03 and N_2 0.54. The total molecular weight of non-condensable gases in the Patuha field is 43.46.

The saturated pressure P_t of the mixture in the system, with a gas and water-vapour temperature of 25 ºC (shown in Figure 14) is represented by P_{27} for the first stage ejector and by P_{37} for the second stage. The saturated pressure of water-vapour at 25 ºC (P_{wv}) can be found by EES. Using Equation 20, the volume flowrate of non-condensable gas in the first and second stages can be found, and through Equation 21, the water-vapour mass flowrate in both.

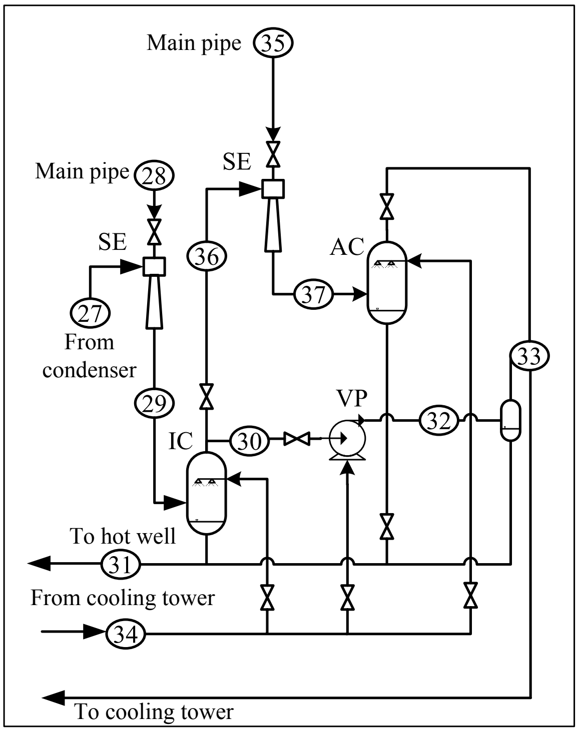

FIGURE 14: Hybrid steam ejector

The suction and discharge pressure of each stage is determined by the following calculations (Geothermal Institute, 1992b):

Each stage uses equal pressure ratios based on a system suction and discharge pressure of 90% condenser pressure and 1.05 bar-a, respectively, while allowing for a pressure loss of 0.019 bar-a in each inter-cooler and after-cooler. The following formula decides the suction and discharge pressures for each stage through equal ratios:

$$\frac{P_{29}}{P_{27}} = \frac{P_{37}}{P_{36}} \tag{46}$$

where P_{27} = 90% of condenser pressure
P_{36} = 0.9683 P_{29}
P_{37} = 1.05 bar-a

2. Calculate the entrainment ratio using the equation which was determined from the entrainment ratio curve. The entrainment ratio for a non-condensable gas can be determined by the equation:

$$E_{ncg} = \left[(5.73 \cdot 10^{-4} + 18.36) + \frac{(2.01 \cdot (M_{ncg}^{0.86})}{(18.36 + (M_{ncg}^{0.86})} \right] \tag{47}$$

The entrainment ratio for water-vapour can be determined by the equation:

$$E_{wv} = \left[(5.73 \cdot 10^{-4} + 18.36) + \frac{(2.01 \cdot (M_{wv}^{0.86})}{(18.36 + (M_{wv}^{0.86})} \right] \tag{48}$$

3. Calculate total air equivalent (*TAE*):

$$TAE = \left[\frac{M_{ncg}}{E_{ncg}} + \frac{M_{wv}}{E_{wv}} \right] \tag{49}$$

For the second stage ejector, the gas content of the motive steam used in the first stage must be added to the initial non-condensable gas mass flowrate for the stage under consideration. The non-condensable gas mass flowrate for the second stage is greater than that of the first.

4. The compression ratio is defined as the ratio of discharge to suction as expressed in Equation 46.
5. The expansion ratio for the first and second stages is defined as the ratio of motive steam pressure to suction pressure:

$$Er = \frac{P_{28}}{P_{27}} \tag{50}$$

$$Er = \frac{P_{35}}{P_{36}} \tag{51}$$

6. The air steam ratio can be found by a curve that has been transformed into a small program in EES called procedure ratio_1 and procedure ratio_2. Inputs required for this program are the expansion ratio and the compression ratio.

7. Finally, the motive steam mass flowrate for both stages can be found from:

$$\dot{m}_{28} = \frac{TAE_1}{AS_1} \tag{52}$$

and

$$\dot{m}_{35} = \frac{TAE_2}{AS_2} \tag{53}$$

5.6 Inter condenser

The seventh step in plant design is to make calculations for the inter condenser heat load and the mass flowrate of cooling water required for the inter condenser. The inter-condenser heat load is given by the following equation:

$$Q_{i-con} = (h_{291} - h_{311})\dot{m}_{291} \tag{54}$$

where $\dot{m}_{291}$, h_{291} and h_{311} are derived from $\dot{m}_{29}$, h_{29} and h_{31} through an iteration process.

Then, the cooling water flowrate for the inter-condenser can be found from:

$$\dot{m}_{cw-i} = \left[\frac{Q_{con-i}}{h_{25} - h_{34}}\right] \tag{55}$$

5.7 Vacuum pump

The eighth step in plant design is to calculate the electric motor power required for driving the vacuum pump. The vacuum pump power can be found from the volume flowrate of gas extraction and the suction pressure data. A vacuum pump is used in place of the second stage steam ejector. So the second stage volume flowrate of gas extraction and the second stage suction pressure should be known. The vacuum pump manufacture data is used to obtain the power of the pump and the motor required. The manufacturer's data has been transformed to a program for this calculation.

$$P_{v\_pump\_motor} = \left[f(\dot{V}_{\_gas\_st\_21}, P_{35}) \right] \tag{56}$$

The power of the vacuum pump is a function of the second stage volume flowrate of gas extraction and the second stage suction pressure.

5.8 Hot well pump

The ninth step in plant design is to calculate the electric motor power for driving the hot well pump. So, the power of the pump for the cooling water must be found. The volume flowrate of the total cooling water is equal to the volume flowrate of the cooling water required for the condenser added to the volume flowrate of cooling water for the inter-condenser and the volume flowrate of the turbine exhaust. The total volume rate of cooling water must be pumped from the condenser and inter-condenser to the cooling tower by the hot well pump, defined by the following equation:

$$\dot{V}_{total} = \dot{V}_{cw} + \dot{V}_{cw-i} + \dot{V}_{turbine\_exhaust} \tag{57}$$

The volume flowrate of cooling water of the condenser and the inter-condenser are given by:

$$\dot{V}_{cw} = \frac{\dot{m}_{cw}}{\rho_{24}} \tag{58}$$

and,

$$\dot{V}_{cw-i} = \frac{\dot{m}_{cw-i}}{\rho_{34}} \tag{59}$$

where ρ_{24} and ρ_{34}, are the density of water at point 24 and point 34 at the condenser and inter condenser, respectively.

Figure 15 shows the layout of the hot well system. Based on it, the head, *H*, of the hot well pump can be calculated:

Static pressure on discharge, $P_d = 0$;
Static pressure on suction, $P_s = 0$;
Density of fluid on discharge, ρ_d = Density of condensate which enters the cooling tower;
The density of the fluid on suction, ρ_d = Density of the condensate at condenser temperature;
$g = 9.8\ m/s^2$;
Velocity of discharge fluid, $V_d = 0.4$ m/s;
Velocity of suction fluid, $V_s = 0.2$ m/s;
Elevation of discharge, $Z_d = 16$ m;
Elevation of discharge, $Z_s = 0$ m;
Constant = 9.797;
$\eta_{pump} = 0.8$;
$\eta_{pump} = 0.85$;

The volume flowrate, $Q = \dot{V}_{total}$.

So, with the *H* and *Q* known, by using Equation 30, the pump power and motor can be determined.

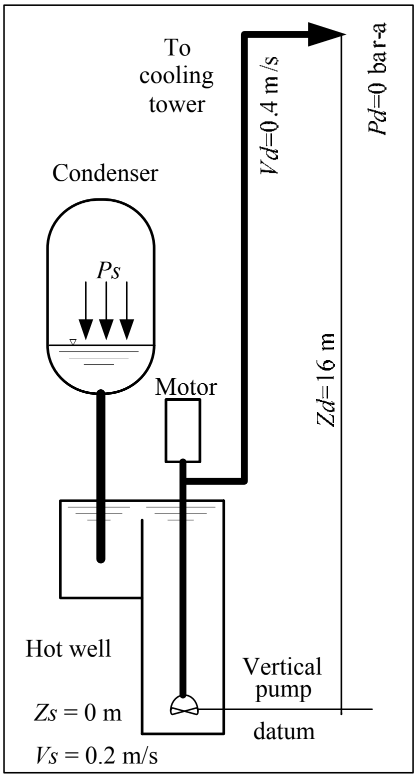

FIGURE 15: Hot well pump

5.9 Cooling tower

The tenth step in plant design is to calculate the mass of circulating water per unit mass of dry air, so the mass flowrate of dry air and the volume flowrate of dry air which exits the cooling tower can be determined. Figure 16 shows the flow diagram of the cooling tower. The temperature design is determined by:

Cooling water from the cooling tower, $T_{cw,1} = 30°C$
Wet bulb temperature, $T_{wb} = 25°C$
Cold air temperature, $T_{cold\_air} = 25°C$
Drop of hot water temperature from condenser to cooling tower, $\Delta T_{con\_cw2} = 3°C$
Drop of temperature from hot water to hot air, $\Delta T_{cw2\_hot\_air} = 6°C$

The temperature of the cooling water entering the cooling tower is given by:

$$T_{cw,2} = T_{con\_sat} - \Delta T_{con\_cw2} \tag{60}$$

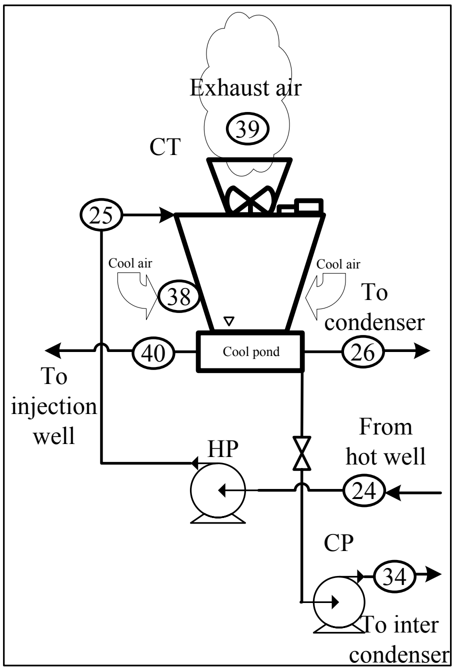

FIGURE 16: Cooling tower

T_{con_sat} = Saturation temperature at condenser pressure.

With the help of EES the following was determined:

a) Enthalpy of the cooling water exiting the cooling tower, h_{26};
b) Enthalpy of the cooling water entering the cooling tower, h_{25};
c) Specific heat of hot air, c_p;
d) Enthalpy of vapour at cold air temperature, h_{g1};
e) Enthalpy of vapour at hot air temperature, h_{g2};
f) Humidity ratio at cold air temperature, ω_1;
g) Humidity ratio at hot air temperature, ω_2.

Using Equation 25, we can find the mass of circulating water per unit mass of dry air which is entering the cooling tower W_A. Using Equations 26 and 27, we can find the mass flowrate of dry air and the exhaust mass flowrate from the cooling tower.

5.10 Power of motor fan in cooling tower

The eleventh step in plant design is to calculate the electric motor power required for the cooling tower fan. By assuming the fan efficiency η_{fan} and the drop pressure suction and discharge of the fan to be ΔP_{fan}, the fan power can be calculated as (El-Wakil, 1984):

$$P_{fan} = \left[\frac{\dot{V}_{dry,air}\, \Delta p_{fan}}{\eta_{fan} \cdot 10^3} \right] \tag{61}$$

If the electric motor efficiency η_{motor}, is known, it is possible to find the electric motor power requirement for the cooling tower from:

$$P_{motor,fan} = \frac{P_{fan}}{\eta_{motor}} \tag{62}$$

Δp_{fan} is defined as the driving pressure, which should equal the air pressure losses in the tower. It is also used to calculate the height for a natural-draft cooling towers. For fan calculation, the driving pressure design is 0.00120 bar-a.

5.11 Motor power of inter condenser circulation water pump

The twelfth and final step in plant design is to calculate the electric motor power required for the inter-condenser cooling water circulation. Figure 17 shows the flow diagram of the cooling water circulation pump. The volume flowrate of the inter-condenser's cooling water is known from Equation 59. Design of head, H, for the inter condenser pump is given through the following:

Static pressure on discharge, P_d, is P_{291} and static pressure on suction, P_d, is 0.
The density of the fluid on suction and discharge (ρ_s and ρ_d) is equal to the density of cooling water from the cooling tower.
Discharge fluid velocity is 2 m/s and the suction fluid velocity is 1 m/s.

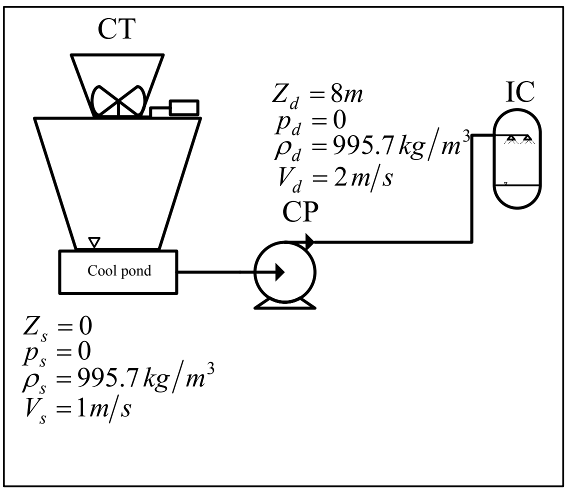

FIGURE 17: Circulation pump

The elevation of the discharge from the site is 8 m and that of the suction is 0 m.
The pump's head can be found by putting the design values into Equation 29.
The efficiency of the pump is 0.8 and the motor's efficiency is 0.85.

By using Equation 30, the power of the pump and the electric motor required can be found.

6. RESULTS AND DISCUSSION

6.1 Optimum separator pressure

The optimum separator pressure is defined as the pressure at which the power plant's output is maximized. To find the optimum pressure of the separator, the condenser pressure is kept constant and the power plant output is calculated for different separator pressures. Similarly, the condenser pressure is given various values. The separator pressures were varied between 4 and 10 bar. The condenser pressures were varied between 0.05 and 0.2 bar-a.

Figure 18 shows the results of the calculations with plant output power versus separator pressures for different condenser pressures. The uppermost curve gives the highest power output. The curve is based on 0.08 bar-a condenser pressure. The results show that the 6-6.5 bar separator pressure gives the maximum output power. Using 6 bar pressure gave a marginally higher result than the 6.5 bar, but the difference was not significant. However, the difference in steam consumption of the two pressures was significant. The plant using a 6 bar separator pressure consumed 2 kg/s more steam than the plant using a 6.5 bar separator pressure. So the 6.5 bar separator pressure was selected as the optimum separator pressure, giving an output power of 56,211 kW. By deducting the separator pressure by 0.01 bar of lost pressure in the demister, the optimum inlet pressure turbine could be obtained. Hence, the optimum inlet pressure turbine is 6.49 bar.

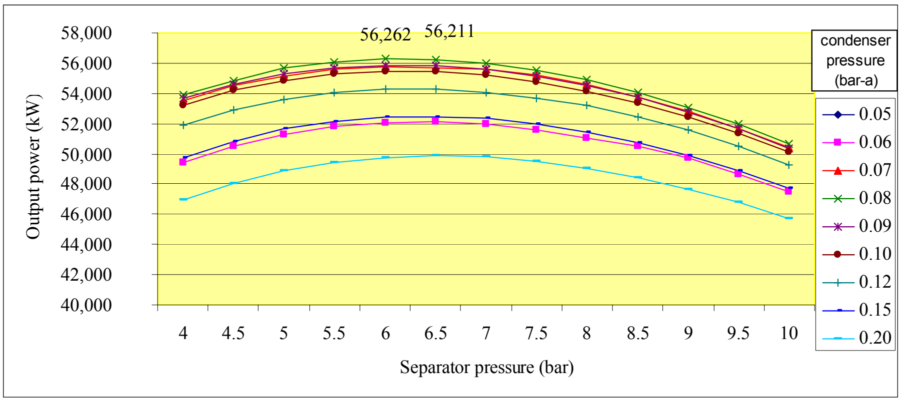

FIGURE 18: Separator pressures vs. power output curve

6.2 Condenser pressure design

6.2.1 Auxiliary power

Figure 19 shows the condenser pressure versus the auxiliary power for a 6.5 bar separator pressure. Auxiliary power consists of the pumping power, the cooling tower motor fan, the motor vacuum pump and others. Low condenser pressure leads to more auxiliary power as shown in Figure 19. The auxiliary power reached the highest value at 0.06 bar-a condenser pressure. The auxiliary power

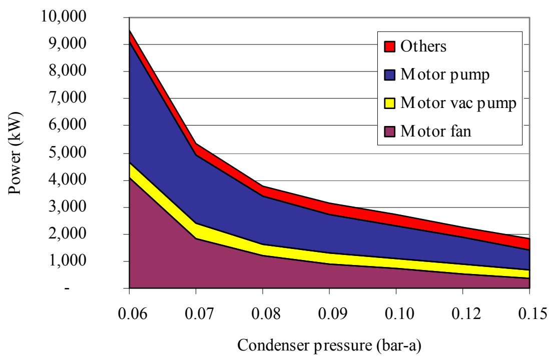

FIGURE 19: Condenser pressure vs. auxiliary power

decreases as the condenser pressure becomes lower, and correlates to the sizing and spacing of the equipment.

TABLE 4: Auxiliary power (kW) for different condenser pressures at 6.5 bar separator pressure

Condenser pressure (bar-a)	Motor fan (kW)	Motor vac pump (kW)	Motor pump (kW)	Others (kW)	Total (kW)
0.06	4,059	600	4,459	400	9,518
0.07	1,833	600	2,482	400	5,315
0.08	1,199	400	1,792	400	3,791
0.09	893	400	1,433	400	3,126
0.10	713	400	1,210	400	2,723
0.12	510	400	952	400	2,262
0.15	358	300	754	400	1,812

Higher auxiliary power allows for larger equipment. Low condenser pressure leads to high initial and maintenance costs and requires more space than higher pressure. The calculation of the total auxiliary power for different condenser pressures with 6.5 bar separator pressure is summarised in Table 4.

6.2.2 Turbine power and output power

Figure 20 shows the condenser pressure versus the turbine power, output power and auxiliary power. The figure shows condenser pressures ranging between 0.06 and 0.15 bar-a at a separator pressure of 6.5 bar. The turbine power decreases as the condenser pressure lowers. The power output will increase and is at maximum value at 0.07 bar-a condenser pressure. Below that value, the power output decreases as the condenser pressure becomes lower.

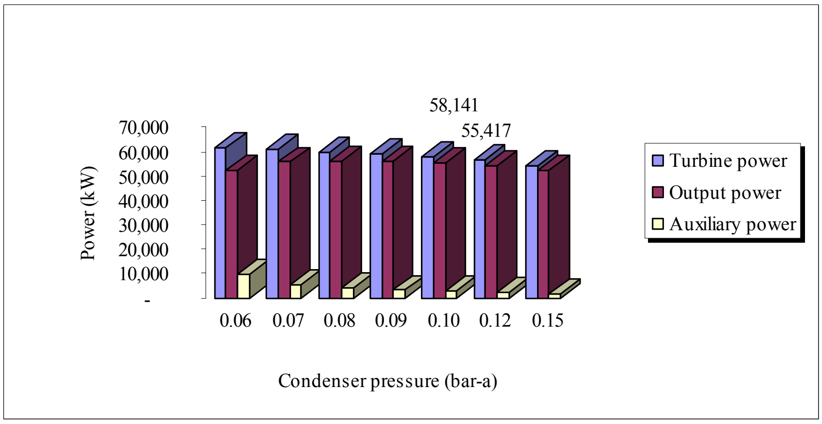

FIGURE 20: Condenser pressure vs. turbine output and auxiliary power

The calculations of turbine output and auxiliary power (kW) for different condenser pressures at the 6.5 bar separator pressure are summarised in Table 5.

TABLE 5: Turbine, output and auxiliary power for 6.5 bar separator pressure

Condenser pressure (bar-a)	Turbine power (kW)	Output power (kW)	Auxiliary power (kW)
0.06	61,620	52,102	9,518
0.07	61,014	55,698	5,316
0.08	60,002	56,211	3,791
0.09	58,955	55,830	3,125
0.10	58,141	55,417	2,724
0.12	56,556	54,294	2,262
0.15	54,284	52,472	1,812

6.2.3 Temperature design

As mentioned in the theoretical analysis, the circulating water inlet temperature should be sufficiently lower than the steam saturation temperature to result in reasonable values of ΔT_o. It is usually recommended that ΔT_i should be between about 11 and 17°C and that ΔT_o, the TTD, should not be less than 2.8°C. In this study, the *TTD* design value used was 3°C and the circulating water inlet temperature was 30°C. Calculations were made to find which condenser pressure resulted in the recommended ΔT_i. Figure 21 shows the results of the calculations with 0.09, 0.1 and 0.12 bar-a condenser pressures giving values in the range of the recommended ΔT_i. The 0.1 bar-a condenser pressure was selected as the ΔT_i value was in the middle of the range.

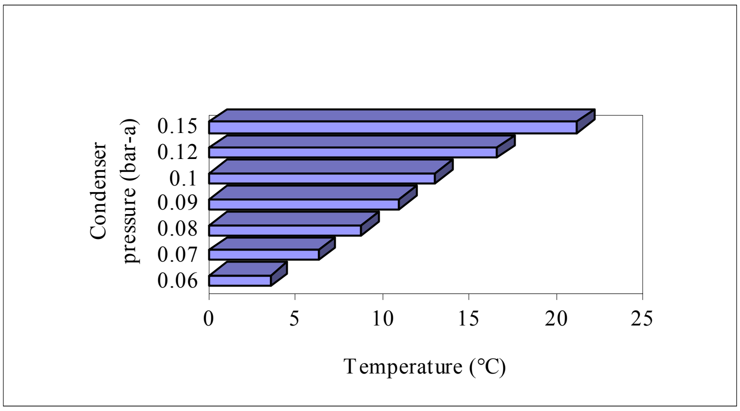

FIGURE 21: Condenser pressure vs. ΔT_i.

6.3 Output power to turbine power ratio

Figure 22 shows the ratio of output power to turbine power operating at 6.5 bar separator pressure with a condenser pressure ranging between 0.06 and 0.15 bar-a. The ratio determines the optimum utility of the turbine power. The optimum utility of turbine power is defined as the maximum ratio of output power to turbine power. The highest ratio was reached at a 0.15 bar-a condenser pressure. The lowest ratio was at a 0.06 bar-a condenser pressure. This means that the utility of the turbine power will increase by operating at a higher condenser pressure.

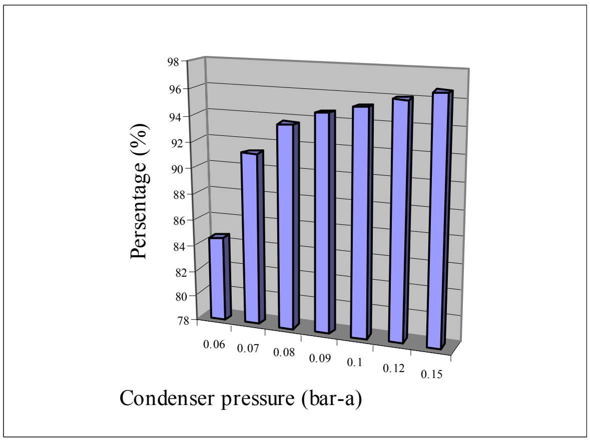

FIGURE 22: Condenser pressure vs. output to turbine power ratio

6.4 Second stage ejector versus vacuum pump performance

Low condenser pressure leads to more motive steam and power for the hybrid steam ejector. The hybrid steam ejector is a heavy duty one and impacts negatively on operation and maintenance. A second stage ejector was installed for redundancy and will operate automatically if the vacuum pump has troubles or shuts down. The steam supply for the turbine was then reduced by the motive steam of the second stage ejector. The power output would be reduced by about 2,327 kW. Figure 23 compares the power consumption of the motor pump versus that of the steam ejector at a 6.5 bar separator pressure and 0.1 bar-a condenser pressure.

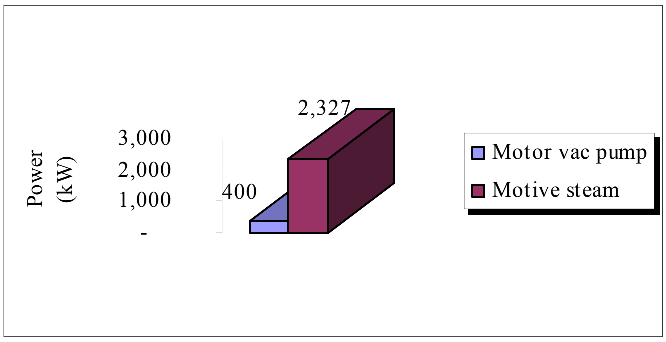

FIGURE 23: Vacuum pump motor power vs. motive steam, stage 2

7. USING DOUBLE FLASH FOR PLANT IMPROVEMENT

7.1 Theory analysis

The double-flash steam plant is an improvement on the single-flash design in that it can produce 15-25% more output power for the same geothermal fluid conditions. The plant is more complex, more costly and requires more maintenance but the extra power output often justifies the installation of such plants. Double-flash plants are fairly numerous and are in operation in nine countries. As of the middle of 2004, there were 70 units of this kind in operation, 15% of all geothermal plants (Dipippo, 2005). In Indonesia there is no plant of this kind. The processes for the double-flash plant are shown in Figure 24.

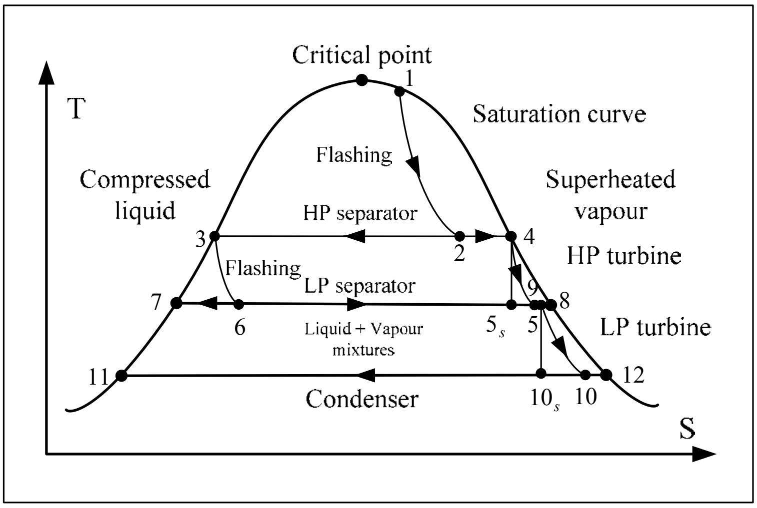

FIGURE 24: *T-S* diagram for the double-flash plant

7.2 Analysis and calculations

A second flash process is imposed on the separated liquid leaving the primary separator in order to generate additional steam, albeit at a lower pressure than the primary steam. The single-flash calculation gave 19 kg/s of separation water in result. The separation water has an enthalpy of 678 kJ/kg and it is supposed to be injected into an injection well. The separation water is flashed by the low-pressure separator. The steam from the second flash is used for extra generation by a low-pressure turbine.

The schematic diagram for a double-flash plant is shown in Figure 25. The design differs from the single-flash plant in Figure 9 in that a low-pressure separator as a second flasher has been added and there is a low-pressure steam line from it to the turbine in addition to the high-pressure one from the separator. Appendix IV shows the same flow diagram for a double-flash power plant for Patuha with necessary specifications, and Appendix V lists the specifications for the plant.

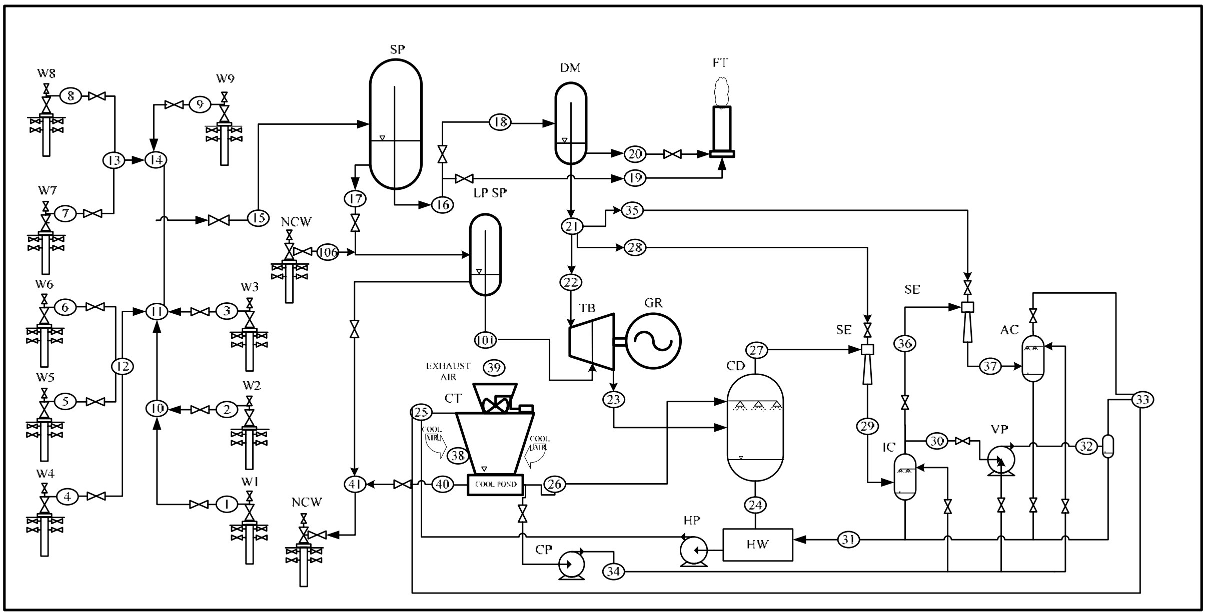

FIGURE 25: Double-flash flow diagram of the preliminary design of Patuha unit 1

The turbine shown is a dual admission, a single-flow machine where the low-pressure steam is admitted to the steam path at an appropriate stage so as to merge smoothly with the partially expanded high-pressure steam. The mass flow rate of the low-pressure steam can be found by the following equations:

$$\dot{m}_{101} = (1 - X_{15})\dot{m}_{15} X_{100} \qquad (63)$$

The enthalpy and quality of the mixture of low-pressure steam and high-pressure steam at the inlet low-pressure turbine can be found by:

$$h_{102} = \left[\frac{(X_{15} h_{231}) + (1 - X_{15}) X_{100} h_{101}}{X_{15} + (1 - X_{15}) X_{100}} \right] \qquad (64)$$

and,

$$X_{102} = \left[\frac{h_{102} + h_{41}}{h_{101} + h_{41}} \right] \qquad (65)$$

Finally the low-pressure turbine power can be found by:

$$\dot{W}_{LP} = \left(\dot{m}_{231} + \dot{m}_{101} \right) (h_{102} - h_{103}) \qquad (66)$$

7.3 Double flash with one non-commercial well

One non-commercial well at Patuha was considered in order to add to the low-pressure steam. Only low-quality data was available, giving the well a mass flowrate of 11 kg/s at 2 bar wellhead pressure. Further production tests are necessary to support the analysis. For this model, a low-pressure separator for the well was added as shown in Figure 25. With the additional low-pressure steam from the well, the enthalpy of the total mixture at the inlet low-pressure turbine could be found by:

$$h_{102} = \left[\frac{(X_{15} h_{231}) + (1 - X_{15}) X_{100} h_{101} + X_{106} h_{106}}{X_{15} + (1 - X_{15}) X_{100} + X1_{106}} \right] \qquad (67)$$

The total mass flow of the low-pressure steam could be found from:

$$\dot{m}_{LP} = \left(\dot{m}_{231} + \dot{m}_{101} + \dot{m}_{106} \right) \qquad (68)$$

7.4 Results

The results of calculations for two designs of double-flash plants are shown in Figure 26. In it the two double-flash designs are compared with the single-flash design. The double-flash plant 1 uses the separation water from the single flash. The double-flash plant 2 combines the separation water with low-pressure steam from one non-commercial well. All

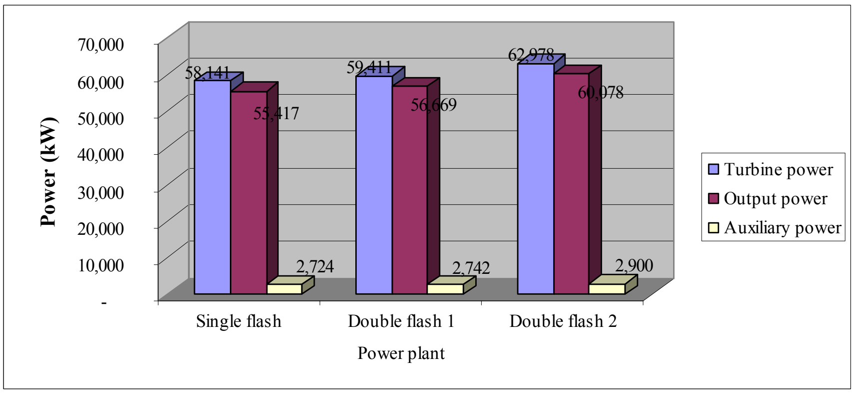

FIGURE 26: Performance for the power plant

the systems operate at 6.5 bar separator pressure and 0.1 bar condenser pressure. The double-flash designs were both operating at 2 bar low-pressure separator.

Figure 26 shows that the double-flash plant 1 can attain turbine power of 59,411 kW and output power of 56,669 kW. It increases the output power by 1,251 kW on the single-flash design. The double-flash plant 2 can attain turbine power of 62,978 kW and output power of 60,078 kW, increasing the output power by 4,661 kW on the single-flash system, about 8.4% more than the single-flash output power.

In the theoretical analysis the double-flash plant was supposed to produce 15-25% more output power. The Patuha field conditions gave a lower result than 15% because the steam, with enthalpy in the range between 2400 and 2700 kJ/kg, is almost dry. In these almost dry steam conditions, the separation water from the primary separator is small.

8. CONCLUSIONS AND SUGGESTIONS

The thermodynamic analysis of the power plant design and calculations gave a maximum output power of the plant by operating the condenser at 0.08 bar-a and the separator pressure at 6.0 bar. At this design pressure, turbine power would attain 60.1 MW and power output of 56.2 MW. The auxiliary power for pumping, cooling tower fan and others is 3.9 MW. A steam ejector consuming 6.5 kg/s steam, is approximately equal to 2.9 MW of electricity.

Considering the recommended condenser temperature distribution design and optimum output power, the condenser pressure suggested for design is 0.1 bar-a and the separator pressure 6.5 bar. The inlet pressure for the turbine is 6.49 bar. With this pressure design the plant will be reduced in size, space, initial cost and maintenance cost of the condenser, hybrid steam ejector, cooling tower and all the auxiliary equipment. This pressure design will generate an output power of 55.4 MW, with a turbine power of 58.1 MW. The auxiliary power needed for pumping, cooling tower fan and others is 2.7 MW. The steam ejector consumes 5.2 kg/s steam which is approximately equal to 2.3 MW of electricity.

The double-flash plant was designed for a 2 bar low-pressure separator. The optimum output power of the plant was obtained when operating the condenser at 0.1 bar-a, with a separator pressure of 6.5 bar. The inlet pressure for the turbine was 6.49 bar. With this design pressure, the turbine power would attain 59.4 MW with a power output of 56.7 MW. The auxiliary power for pumping, cooling tower fan and others totalled 2.7 MW. The steam ejector consumed 5.2 kg/s steam.

Combined with steam from a non-commercial well, the double-flash design gave the optimum power output by operating the condenser at 0.1 bar-a with a separator pressure of 6.5 bar. At this design pressure, the turbine power attained 63 MW with an output power of 60.1 MW. The auxiliary power for pumping, cooling tower fan and others totalled 2.9 MW. The double-flash system would give 4.7 MW of additional output power, 8.4% more than the single-flash system.

The low additional output power that can be attained from the double-flash system is caused by the steam conditions at Patuha field. The enthalpy range of the steam is between 2400 and 2700 kJ/kg, so it is almost dry. This means that very little separation water is available from the primary separator to obtain additional steam for generating additional power.

ACKNOWLEDGEMENTS

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NOMENCLATURE FOR FIGURES

WX	= Well number X	IC	= Inter cooler
SP	= Separator	AC	= After cooler
DM	= Demister	CT	= Cooling tower
FT	= Flash tank	CP	= Circulation pump
TB	= Turbine	VP	= Vacuum pump
GR	= Generator	HP	= Hot well pump
CD	= Condenser	LPSP	= Low pressure separator
SE	= Steam ejector		

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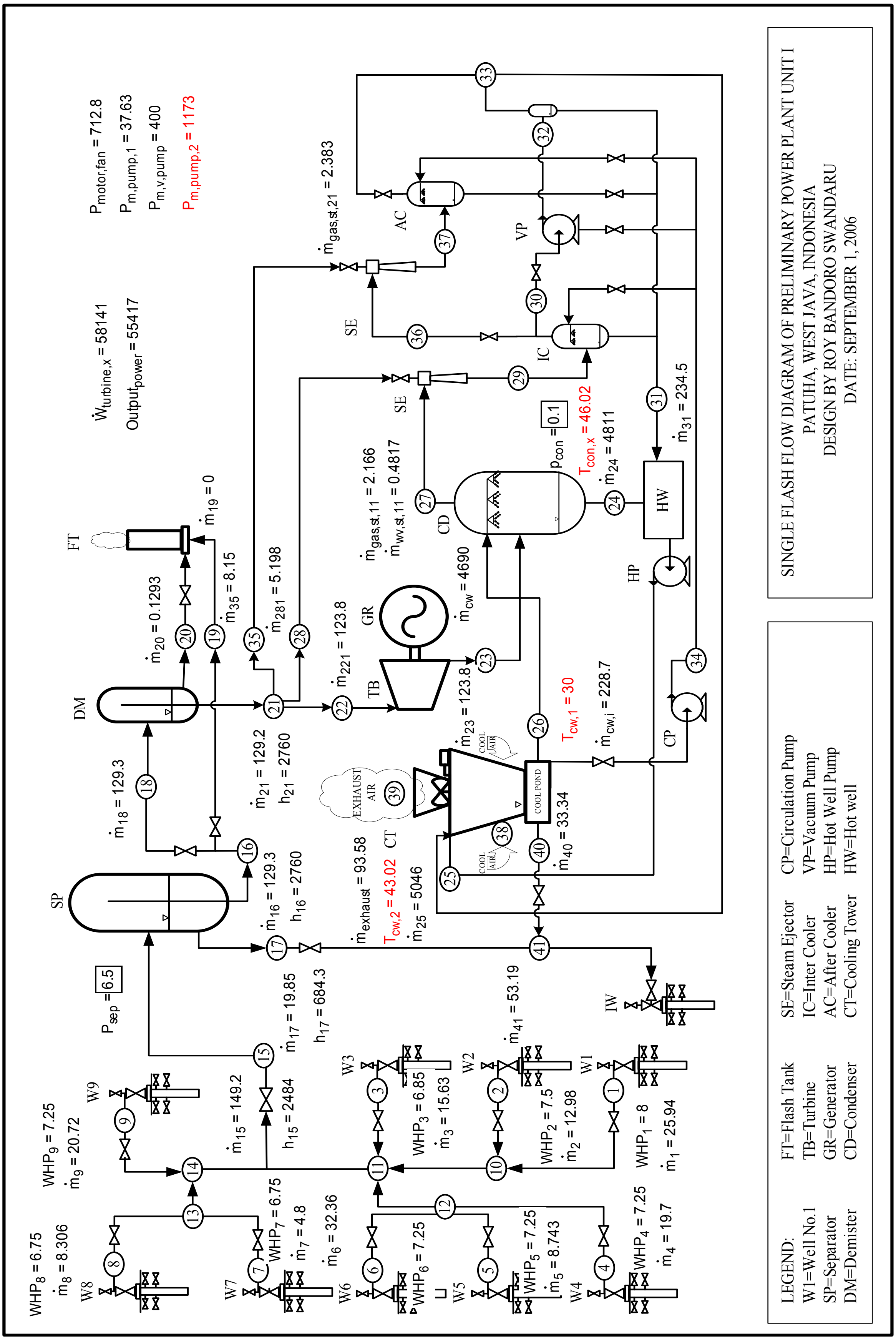
APPENDIX I: Single-flash flow diagram of the preliminary power plant unit 1, Patuha



APPENDIX II: Specifications of a single-flash power plant unit 1, Patuha

No.	Specifications	Value/type
1	*Turbine data*	
	Rated capacity, kW	58,141
	Steam inlet pressure, bar	6.49
	Steam temperature, °C	161.9
	Non condensable gas, % weight	1.75
	Steam flowrate, kg/s	123.8
	Turbine efficiency, %	80
2	*Generator data*	
	Rated capacity, kW	60,000
	Efficiency,%	93
3	*Condenser data*	
	Type	Direct contact
	Pressure design, bar-a	0.1
	Temperature design, °C	46
	Cooling water temperature, °C	30
	Outlet water temperature, °C	43
	Cooling water mass flow, kg/s	4918.7
	Hot well pump power, kW	1173
4	*Gas extractor data*	
	Type	Hybrid steam ejector
	First stage	Steam ejector
	First stage motive steam flowrate, kg/s	5.2
	Second stage type	Liquid ring vacuum pump
	Rated capacity of vacuum pump, kW	400
	Second stage redundancy type	Steam ejector
	Second stage motive steam flowrate, kg/s	8.15
5	*Cooling tower data*	
	Type	Mechanical induced draft
	Designed web-bulb temperature, °C	25
	Fan motor power, kW	712.8

APPENDIX III: Sample of an Engineering Equations Solver calculation model for a single-flash power plant

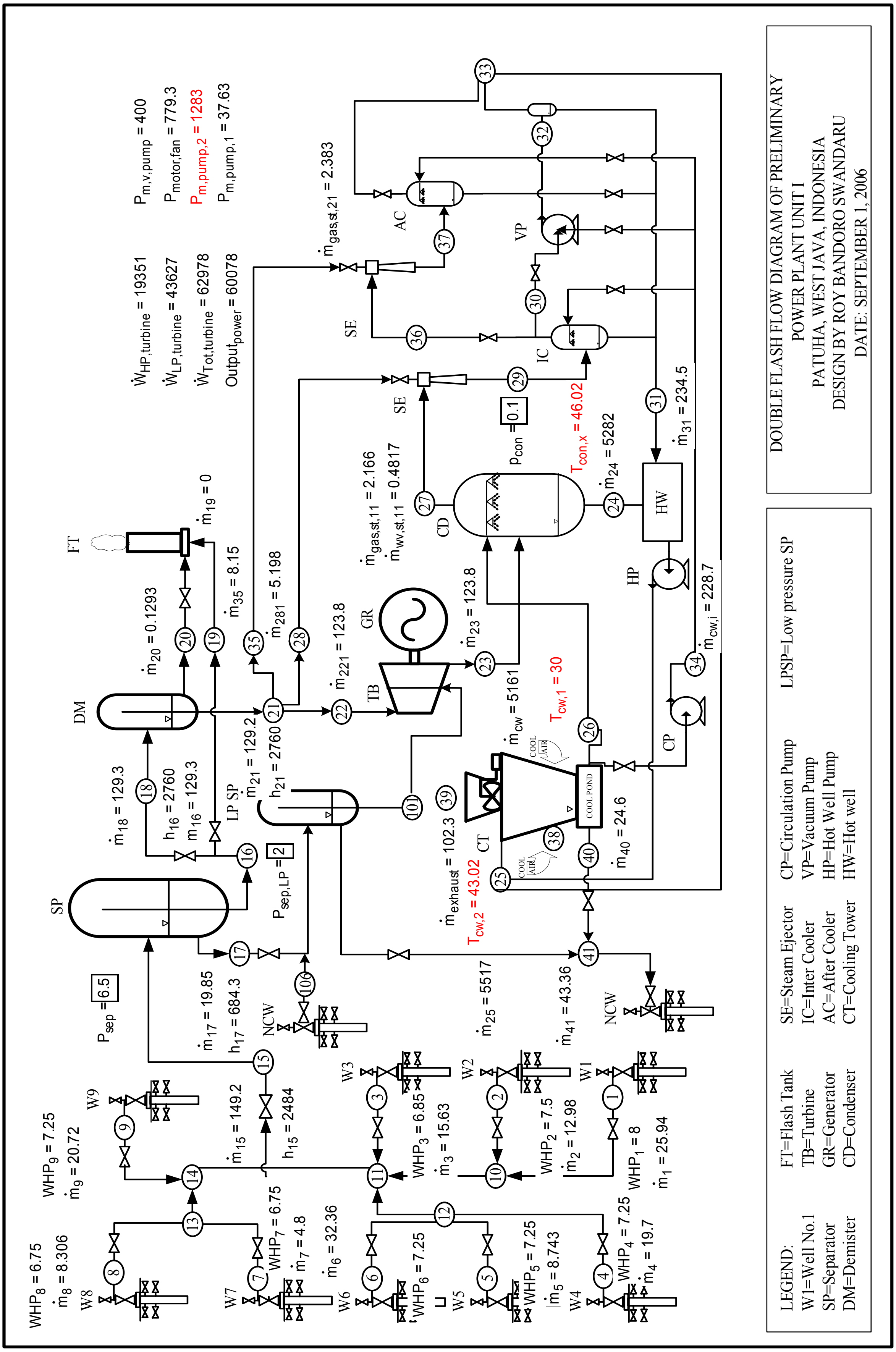
```
{Gaussian Model, mass flow rate of well No.1}
procedure well_1(WHP[1]:m_dot[1])
a =	26.763443
b =	9.363602
c =	5.4686158
m_dot[1]=a*exp((-(b-WHP[1])^2)/(2*c^2))
end

{Gaussian Model, mass flow rate of well No.2}
procedure well_2(WHP[2]:m_dot[2])
a =	19.474808
b =	-4.2554578
c =	13.05509
m_dot[2]=a*exp((-(b-WHP[2])^2)/(2*c^2))
end
```

```
{Rational Function, mass flow rate of well No.3}
procedure well_3(WHP[3]:m_dot[3])
a =       25.565868
b =       -1.5208937
c =       -0.0060485268
d =       0.00022823321
m_dot[3]=(a+b*WHP[3])/(1+c*WHP[3]+d*WHP[3]^2)
end

{MMF Model, mass flow rate of well No.4}
procedure well_4(WHP[4]:m_dot[4])
a =       22.787811
b =       2033.471
c =       -175.56048
d =       1.7511409
m_dot[4]=(a*b+c*WHP[4]^d)/(b+WHP[4]^d)
end

{Sinusoidal Fit,mass flow rate of well No.5}
 procedure well_5(WHP[5]:m_dot[5])
a =       -12.841282
b =       21.584207
c =       0.033641499
d =       0.098160191
m_dot[5]=a+b*cos(c*WHP[5]+d)
end

{Rational Function,mass flow rate of well No.6}
procedure well_6(WHP[6]:m_dot[6])
a =       43.376582
b =       -1.9470723
c =       -0.0045277479
d =       -0.0011981477
m_dot[6]=(a+b*WHP[6])/(1+c*WHP[6]+d*WHP[6]^2)
end

{Hoerl Model,mass flow rate of well No.7}
procedure well_7(WHP[7]:m_dot[7])
a =       6.5291422
b =       0.9514415
c =       0.014838956
m_dot[7]=a*(b^WHP[7])*(WHP[7]^c)
end

{Quadratic Fit,mass flow rate of well No.8}
procedure well_8(WHP[8]:m_dot[8])
a =       14.793796
b =       -1.1494371
c =       0.027885393
m_dot[8]=a+b*WHP[8]+c*WHP[8]^2
end

{Linear Fit,mass flow rate of well No.9}
procedure well_9(WHP[9]:m_dot[9])
a =       31.909056
b =       -1.5437376
m_dot[9]=a+b*WHP[9]
end
```

```
{********************************MAIN PROGRAMME************************************}
{MASS FLOW FROM WELLS}
m_dot[10]=m_dot[1]+m_dot[2]
h[10]=(h[1]*m_dot[1]+h[2]*m_dot[2])/(m_dot[1]+m_dot[2])

m_dot[11]=m_dot[10]+m_dot[3]+m_dot[12]
h[11]=(h[10]*m_dot[10]+h[3]*m_dot[3]+h[12]*m_dot[12])/(m_dot[10]+m_dot[3]+m_dot[12])

m_dot[12]=m_dot[4]+m_dot[5]+m_dot[6]
h[12]=(h[4]*m_dot[4]+h[5]*m_dot[5]+h[6]*m_dot[6])/(m_dot[4]+m_dot[5]+m_dot[6])

m_dot[13]=m_dot[8]+m_dot[7]
h[13]=(h[8]*m_dot[8]+h[7]*m_dot[7])/(m_dot[8]+m_dot[7])

m_dot[14]=m_dot[13]+m_dot[9]
h[14]=(h[13]*m_dot[13]+h[9]*m_dot[9])/(m_dot[13]+m_dot[9])

m_dot[15]=m_dot[14]+m_dot[11]                                  {total mass flow}
h[15]=(h[14]*m_dot[14]+h[11]*m_dot[11])/(m_dot[14]+m_dot[11])  { enthalpy of total mass flow}

m_dot_total=m_dot[1]+m_dot[2]+m_dot[3]+m_dot[4]+m_dot[5]+m_dot[6]+m_dot[7]+m_dot[8]+m_dot[9]

{SEPARATOR}
p[16]=P_sep
h[16]=ENTHALPY(Water,x=1,P=p[16])
p[17]=p[16]
h[17]=ENTHALPY(Water,x=0,P=p[17])
x_15=(h[15]-h[17])/(h[16]-h[17]) {quality of fluid after separator}
m_dot[16]=x_15*m_dot[15]

{SCRUBBER}
m_dot[17]=(1-x_15)*m_dot[15] {mass flow of water to reinjection}
m_dot[18]=(m_dot[16])-m_dot[19] {mass flow to scrubber}
m_dot[19]=0 {mass flow rate to flash tank}
DELTA_p18=0.01  {Drop pressure at point 18}
h[19]=h[16]
h[18]=h[16]
p_scr=P_sep-DELTA_p18
p[21]=p_scr
m_dot[21]=m_dot[18]-m_dot[20]
m_dot[20]=0.001*m_dot[18]
p[22]=p[21]
m_dot[22]=m_dot[21]
h[22]=ENTHALPY(Steam,x=1,P=p[22])
h[21]=h[22]
p[20]=p_scr
h[20]=ENTHALPY(Steam,x=0,P=p[20])
x_21=(h[18]-h[20])/(h[21]-h[20])
x_21=x_22
{**********************************TURBINE / CONDENSER*************************}
{p_con=0.1}                        {condenser pressure}
p[23]=p_con

s_23=ENTROPY(Steam,x=1,P=p[21])                  {turbine enthropy}
h_23_s=ENTHALPY(Steam,s=s_23,P=p[23])                  {enthalpy}

eta_s=(h[22]-h[23])/(h[22]-h_23_s)               {enthropy effisiency}
eta_s=0.8
eta_total=0.75                                   {effisiency total}
```

```
W_dot=x_22*m_dot[22]*(h[22]-h[23])                                    {Turbine power}
W_dot_turbine=(eta_total/eta_s)*W_dot                                 {Turbine power net}

{***********************************Hybrid Steam ejector *********************************}
{Calculate The Gas Volume flow rate}
{First Stage}
T_ncg=25                                        { Temp. of NCG saturated with water-vapour,°C}
T_k_ncg=(273.1+T_ncg)                           { Temp. of NCG saturated with water-vapour,°K}
P_wv=0.032                                 {water vapour pressure saturated  in 25°C , steam table}
R=8314.4/M_gas                                                {Ro/molecular weight of gas}
M_gas=43.49                                            {Molecular weight of total Patuha nc gas}
M_H2O=18
v_dot_gas_st_1=(m_dot_gas_st_1*R*T_k_ncg)/((P[27]*10^5)-(P_wv*10^5)) {Gas volume flow rate of water vapour
saturated at 25°C}
rho_wv=43.40                          {specific volume of water vapour saturated, steam table  25°C}
m_dot_wv_st_1=v_dot_gas_st_1/rho_wv                     {water vapour mass flow at stage 1}

{Second stage}
m_dot_gas_st_2=m_dot_gas_st_1*1.1                            { mass flow rate of gas st stage 1}
v_dot_gas_st_2=(m_dot_gas_st_2*R*T_k_ncg)/((P[36]*10^5)-(P_wv*10^5)){Gas volume flow rate of water vapour
saturated at 25°C}
m_dot_wv_st_2=v_dot_gas_st_2/rho_wv                          {water vapour mass flow at stage 2}

{Calculation of steam flow rate}
P[28]=p[22]                                                 {Steam pressure at ejector stage 1}
P[35]=P[28]                                                 {Steam pressure at ejector stage 2}
P[27]=p_con*0.9                                            {Suction Pressure, ejector stage 1}
P[29]=((P[37]*P[27])/0.9683)^0.5                          {Discharge Pressure, ejector stage 1}
P[36]=P[29]*0.9683                                           {Suction Pressure, ejector stage 2}
P[37]=1.05                                                {Discharge Pressure, ejector stage 2}
m_dot_gas_st_1=m_dot[22]*0.0175                                  { mass flow rate of gas  stage 1}

{Entrainment ratio equetions}
E_gas=((0.00057367806*18.369572)+(2.0104313*(M_gas^0.86384519)))/(18.369572+(M_gas^0.86384519))
E_wv=((0.00057367806*18.369572)+(2.0104313*(M_H2O^0.86384519)))/(18.369572+(M_H2O^0.86384519))

m_dot_air_st_1=(m_dot_gas_st_1/E_gas)+(m_dot_wv_st_1/E_wv)         {Total air equivalent at stage 1}
m_dot_air_st_2=(m_dot_gas_st_2/E_gas)+(m_dot_wv_st_2/E_wv)        {Total air equivalent at stage 2}
C_st_1=(P[29]/P[27])                                                        {Compression ratio}
C_st_2=(P[37]/P[36])                                                        {Compression ratio}
E_st_1=P[28]/P[27]                                                      { Expantion ratio stage 1}
E_st_2=P[35]/P[36]                                                     { Expantion ratio stage 2}
m_dot[28]=m_dot_air_st_1/AS_st_1                                     {steam mass flow, stage 1}
m_dot_35=m_dot_air_st_2/AS_st_2                                      {steam mass flow, stage 2}
```

APPENDIX IV: Double-flash diagram of the preliminary power plant unit 1, Patuha



APPENDIX V: Specifications of a double-flash power plant unit 1, Patuha

No.	Specifications	Value/type
1	*Turbine data*	
	Rated capacity, kW	62,978
	Steam inlet pressure, bar	6.49
	Steam temperature, °C	161.9
	Non condensable gas, % weight	1.75
	HP Steam flowrate, kg/s	123.8
	LP Steam flowrate, kg/s	12.6
	Turbine efficiency, %	80
2	*Generator data*	
	Rated capacity, kW	65,000
	Efficiency,%	93
3	*Condenser data*	
	Type	Direct contact
	Pressure design, bar-a	0.1
	Temperature design, °C	46
	Cooling water temperature, °C	30
	Outlet water temperature, °C	43
	Cooling water mass flow, kg/s	5,389.7
	Hot well pump power, kW	1,283
4	*Gas extractor data*	
	Type	Hybrid steam ejector
	First stage	Steam ejector
	First stage motive steam flowrate, kg/s	5.2
	Second stage type	Liquid ring vacuum pump
	Rated capacity of vacuum pump, kW	400
	Second stage redundancy type	Steam ejector
	Second stage motive steam flowrate, kg/s	8.15
5	*Cooling tower data*	
	Type	Mechanical induced draft
	Designed web-bulb temperature, °C	25
	Fan motor power, kW	779.3