Study & Design of Binary ORC Using Wet Cooling Tower (Existing) of Unit V & VI in Lahendong Geothermal Field, Indonesia

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ABSTRACT

Utilization of hot brine from production separator has a great potential to generate electricity as a secondary turbine in Lahendong and the other PT PGE fields. Currently, the separated water from separator will move directly into reinjection wells with gravity due differential elevation of about 60 meters.

In order to utilize hot brine from production separator, the existing line should be modified by making pipe branch 6" schedule 40 toward secondary turbine, from main line 16" schedule 40. The pressurized water from upstream (7.83 barg) has a pressure drop 0.13 barg and heat loss 0.019°C along 30 meters of new pipeline.

In this study, pentane was chosen as the best selection of binary working fluids due to had power output greater than others and the surface condenser with pressure 1.32 bara is connected to the existing wet cooling tower as the cooling system. The calculation using Scilab and CoolProp as numerical computation, with vaporizer pressure 8.63 bara or 7.70 barg resulting net power 1795.32 kW and SSI value 2.02, while using vaporizer pressure 18.7 bara resulting net power 626.1 kW with SSI value 0.99.

1. INTRODUCTION

Lahendong geothermal field is located in North Sulawesi of Indonesia and as a part of PT Pertamina Geothermal Energy working areas shown in Figure 1. In 2018, total capacity of geothermal production in Lahendong field is 120 MW (PT PGE, 2018a), which is supplied by 6 power plant units. Unit I to IV and unit V to VI have a different location, where units I to IV with total production capacity 80 MW had location in Lahendong and unit V to VI in Tompaso, the overall distance between them is about 18 km.

Figure 1: Location of Lahendong geothermal field 6 x 20 MW, North Sulawesi, Indonesia (PT PGE, 2018a)

1.1 General Background

The Lahendong unit V and VI geothermal field has an elevation of 789 m a.s.l. for general and 768.8 m a.s.l. for switch yard with the average site atmospheric pressure is 0.924 bara. The production wells are at an elevation of about 790 m a.s.l. and the reinjection wells is at an elevation around 723 m a.s.l. This field have a two-phase dominated in geothermal system classification and consist of one cluster of production wells as well as two clusters of reinjection wells, there are:

1. Production wells (Cluster 27)

   LHD-27, LHD-31, LHD-34, LHD-42, LHD-43.

   The three production wells from LHD-27, LHD-31, and LHD-34 supplies the steam to generating power plant unit V, and the two of rest has delivered the steam into power plant unit VI.

2. Reinjection wells (Cluster R)

   Cluster R-1 & R-2: LHD-41, LHD-46 & LHD-40 & LHD-44.

Two clusters in reinjection wells are to maintain production reservoir continuously from five production wells and from cold brine pump. Cold brine pump from thermal ponds had pipe connection to killing line valve on reinjection wellhead and/or to main reinjection line before entering to the reinjection wellhead. The thermal pond's function is to collect steam condensate discharge that was occurred in the pipeline systems due to pressure drop and water discharge from the separator if water reach high level (overflow) through pneumatic emergency dump valve for safety purpose.

1.2 Tracer flow test result in unit V and VI wells

The steam and water flow has been measured during operation with tracer flow test method two to three times between 2016 and 2017 as shown in Table 1. The setting of wellhead pressure based on well testing previously after drilling has completed.

Table 1: Tracer flow test result of unit V and VI wells (PT PGE, 2017)

Well	Date	Steam	Water	Total
		ton/hr		
Unit V wells				
LHD-27	Oct. 3, 2016	45.72	192.96	236.68
	Apr. 21, 2017	26.9	190.13	217.03
	Oct.23, 2017	39.85	176.8	216.65
LHD-31	Oct. 4, 2017	45.72	344.52	390.24
	Apr. 21, 2017	35.4	223.43	258.83
	Oct. 23, 2017	40.82	248.95	289.77
LHD-34	Oct. 3, 2016	57.6	241.92	299.52
	Apr. 21, 2017	94.8	299.61	394.41
	Oct.23, 2017	69.82	252.45	322.27

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2. STUDY DESCRIPTION

The hot water inside pipe from separator outlet have potential to utilize, it means using hot brine to heat the secondary fluids to generate electricity. In outline, the hot water and it flow rate before entering into reinjection wells will be planned to branch out toward to new location, where the turbine will be placed. The hot water would discharge from heat exchanger and pre heat exchanger and goes to thermal pond mixed with steam condensate and emergency dump valve discharge then pumped to reinjection wells with continuously to maintain the reservoir.

This study will focus on designing binary ORC using wet cooling tower existing in Unit V and VI with suitable secondary fluids against to the temperature source. Additionally, the amount of electricity that can be generated related to silica saturation index and minimum temperature from pre heat exchanger to the pond will be quantified. Furthermore, due to the limitation in utilizing a whole of brine from separator outlet, the flow rate that will flow to heat exchanger in order to heat secondary fluids will also be studied.

3. THEORETICAL BACKGROUND

3.1 Pipe design

The cross-sectional area inside of pipe can be obtained by the following equation:

$$A = \frac{1}{4} \cdot \pi \cdot D_{in}^2 \tag{1}$$

Fluid velocity inside of pipe:

$$V = \frac{m}{A \cdot \rho} \tag{2}$$

The pipe thickness by pressure design according to power piping standard:

$$t_r = \frac{P \cdot D}{2 \cdot (S \cdot E + P \cdot Y)} \tag{3}$$

$$t_m = t_r + CA \tag{4}$$

3.2 Pressure drop analysis

The Reynolds number should be calculated using equation:

$$Re = \frac{\rho \cdot V \cdot D_{in}}{\mu} \tag{5}$$

The end based on the amount of Reynolds number, friction factor should be calculated from the one of equations below, which is equation 6 called Darcy friction factor and equation 7 called Swanee-Jain equation that used to solve the Darcy-Weisbach friction factor for a full-flowing circular pipe as approximation of the implicit Colebrook-White equation.

$$Re \leq 2100, \quad \mathrm{f} = \frac{64}{Re} \tag{6}$$

$$Re > 2100, \mathrm{f} = \frac{0.25}{\left(log_{10} \left[\frac{\frac{\epsilon}{D_{in}}}{3.7} + \frac{5.74}{Re^{0.9}} \right] \right)^2} \tag{7}$$

The friction head can be calculated by:

$$H_f = \frac{f \cdot V^2 \cdot Le}{2 \cdot g \cdot D_{in}} \tag{8}$$

Then the pressure drop due to friction along of pipeline and the elevation difference can be explained by:

$$\Delta P_f = \rho \cdot g \cdot H_f \tag{9}$$

$$\Delta P_H = \rho \cdot g \cdot (Z_s - Z_e) \tag{10}$$

Furthermore, the total of pressure drop along of the pipeline systems can be summarized:

$$\Delta P_t = \Delta P_f + \Delta P_H \tag{11}$$

3.3 Heat loss analysis

To calculate the overall heat transfer on a cylinder plane inside and outside of pipe, the temperature difference is divided by the total thermal resistance between two surfaces (Ohm's law):

$$q = \frac{\Delta T}{R} \tag{12}$$

$$q = \frac{\Delta T}{R_{conv,1} + R_1 + R_2 + R_3 + R_{conv,2}}$$

$$R = \frac{1}{hi \cdot 2 \cdot \pi \cdot r_1 \cdot L} + \frac{\ln r_2/r_1}{2 \cdot \pi \cdot k_1 \cdot L} + \frac{\ln r_3/r_2}{2 \cdot \pi \cdot k_2 \cdot L} + \frac{\ln r_4/r_3}{2 \cdot \pi \cdot k_3 \cdot L} + \frac{1}{ho \cdot 2 \cdot \pi \cdot r_4 \cdot L} \tag{13}$$

$$\frac{q}{L} = \frac{2 \cdot \pi \cdot (T_{in} - T_{out})}{\frac{1}{hi \cdot r_1} + \frac{\ln r_2/r_1}{k_1} + \frac{\ln r_3/r_2}{k_2} + \frac{\ln r_4/r_3}{k_3} + \frac{1}{ho \cdot r_4}} \tag{14}$$

$$\Delta T = \frac{q}{m \cdot Cp_l} \tag{15}$$

3.4 Organic rankine cycle turbine (binary)

The hot brine from the production separator would enter the evaporator and preheater to heated the working fluid to aim reach boiling point and change into vapor, then it vapors flows into turbine in order to generate electricity. This scenario is no need the recuperator as the other source to heated of working fluids, the heat source only obtains directly from the hot brine. Meanwhile, the working principle of the ORC process in Figure 2 can be defined in detail as follows:

- Process 4 - 5 : Isentropic compression in the working fluid pump;
- Process 5 - 1 : Constant pressure heat addition in preheater and evaporator;
- Process 1 - 2 : Isentropic expansion in the turbine;
- Process 2 - 4 : Constant pressure heat rejection in the condenser.

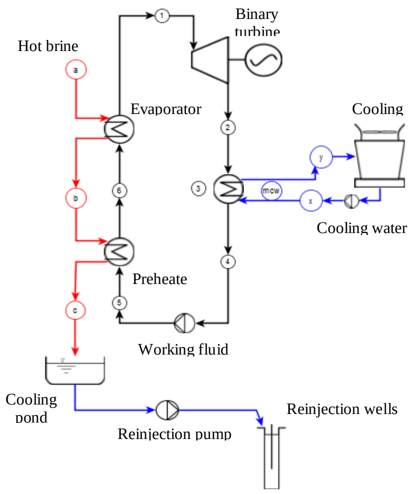

Figure 2: Working principle of binary turbine cycle

According to DiPippo (2016), we can calculate the binary power and the other process using parameters as shown in Table 2 below.

Table 2: Thermodynamics, environmental and health properties of working fluids (DiPippo, 2016)

No.	Fluid	Critical temp. (°C)	Critical pressure (bar)	Molar mass (kg/kmol)
1.	Propane	96.95	42.36	44.09
2.	i-Butane	135.92	36.85	58.12
3.	n-Butane	150.8	37.18	58.12
4.	i-Pentane	187.8	34.09	72.15
5.	n-Pentane	193.9	32.40	72.15

3.4.1 Heat exchanger analysis: preheater and evaporator

The hot brine source is indicated with point a, it means represent of geothermal fluids properties from production separator. Then point c is an outlet of preheater, this outlet temperature should be kept as possible to avoid scaling in the heat exchangers side due to temperature.

Furthermore, in Figure 3, point 5 is a working fluid entrance from condensing process and pumped to preheater. The working fluids in point 6 have heated before entering the evaporator, the point 1 is the vapor that toward into turbine after getting process in the evaporator.

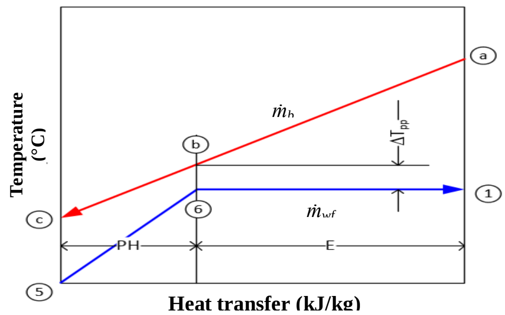

Figure 3: Temperature-heat transfer diagram for preheater and evaporator

Considering the entire package as a thermodynamic system, the governing equation is:

$$\dot{m}_b \cdot (h_a - h_c) = \dot{m}_{wf} \cdot (h_1 - h_5) \quad (16)$$

If the heat capacity of the geothermal fluid is known, the left hand side of the equation would change to:

$$\dot{m}_b \cdot \bar{c}_b \cdot (T_a - T_c) = \dot{m}_{wf} \cdot (h_1 - h_5) \quad (17)$$

Figure 3 describes the heat transfer relationship between geothermal fluid and working fluid. Points a to c explain how hot brine temperature will decrease after it passes through the evaporator and preheater. The working fluid that enters into preheater will receive the initial stage of heat transfer and after it passes through the evaporator will change to vapor phase. The minimum temperature difference in the preheater and evaporator, between the geothermal fluid and working fluid, is called the pinch-point. The value of that difference is designated the pinch-point temperature difference (ΔT_{pp}).

Points 5, 6, and 1 of working fluid in this diagram should be known in the cycle. Point 5 is compressed liquid, point 6 is saturated liquid at the boiler pressure, and point 1 is a saturated vapor.

Preheater:

$$\dot{m}_b \cdot \bar{c}_b \cdot (T_b - T_c) = \dot{m}_{wf} \cdot (h_6 - h_5) \quad (18)$$

Evaporator:

$$\dot{m}_b \cdot \bar{c}_b \cdot (T_a - T_b) = \dot{m}_{wf} \cdot (h_1 - h_6) \quad (19)$$

The pinch-point temperature difference is selected after an economic analysis. High price of electricity will allow large, efficient and expensive heat exchangers which translates into small pinch (Dr. Páll Valdirmarsson, Adjunct Professor, Reykjavík University, personal communication, August 28, 2018). This allows T_b to be found from the known value for T_6 as well as found the T_c.

$$T_b = T_6 + \Delta T_{pp} \quad (20)$$

$$T_c = T_a - (T_a - T_b)\left[\frac{h_1 - h_5}{h_1 - h_6}\right] \quad (21)$$

$$q_{in} = h_1 - h_5 \quad (22)$$

3.4.2 Binary turbine power

Figure 4 explain the state point 1 is the vapor phase from the evaporator and toward into binary turbine inlet to generate the electricity. State point 2 as the turbine outlet and have an isentropic process, it means the inlet entropy is equal to outlet entropy ($S_1 = S_2$). In the real turbine, the vapor enthalpy is the enthalpy change in the ideal turbine and multiplied by the turbine isentropic efficiency. For a given working fluid, the thermodynamic properties can be found from Table 7.

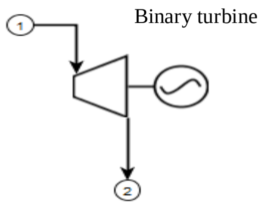

Figure 4: Process diagram of binary

Moreover, the desired power output will determine the required working fluid mass flow rate.

$$\dot{W}_T = \dot{m}_{wf} \cdot (h_1 - h_2) = \dot{m}_{wf} \cdot \eta_t \cdot (h_1 - h_{2S}) \quad (23)$$

$$w_t = h_1 - h_2 \quad (24)$$

Also, we should find the vapor fraction in isentropic state; the isentropic enthalpy of turbine and the actual enthalpy of turbine outlet from working fluid properties as described below:

$$s_{2s} = s_1 \quad (25)$$

$$x_2 = \frac{s_2 - s_{f2}}{s_{fg2}} \quad (26)$$

$$h_{2s} = h_{f2} + x \cdot \left(h_{g2} - hf_2\right) \quad (27)$$

$$h_2 = h_1 - \eta_T \, (h_1 - h_{2s}) \quad (28)$$

Note, if *x* value is larger than 1, then the steam is still superheated and the solution does not involve *x*. We can find the isentropic enthalpy of working fluid using equation 29 as follow:

$$h_{2s} = h_{f2} + \left(\frac{h_{g2} - h_{f2}}{s_{g2} - s_{f2}}\right) \cdot \left(s_{2s} - s_{f2}\right) \quad (29)$$

3.4.3 Condenser and cooling tower type

As shown in Figure 5, the vapor phase of working fluid at point 2 will change to condensate at point 3 after receiving the condensate from the condenser, either the process in the condenser with air cooled, water cooled or shell and tube condenser (Nugroho, 2007).

In this paper, the surface condenser with wet cooling tower (existing) has been chosen as binary process cycle due to have advantage of its output being less sensitive to wet bulb temperature variations. On the other hand, a majority to using air cooled condenser due to the lack of suitable quality make-up water available in location. Then in air cooled condenser, during summer season will affect significantly to the power output because the ambient temperature rises and it cannot cool the working fluid properly.

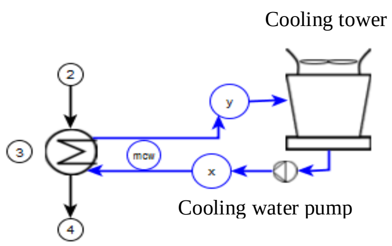

Figure 5: Process diagram of water cooling system

The cooling water to condenser has supplied from cooling tower or air cooler at point x, and the heated water will move towards point y cooling the water supply after getting processed in the condenser. Furthermore, the relationship between the flow rates of the working fluid and the cooling water is:

$$\dot{m}_{cw} \cdot \left(h_y - h_x\right) = \dot{m}_{wf} \cdot (h_2 - h_4) \quad (30)$$

$$\dot{m}_{cw} \cdot \bar{c} \cdot \left(T_y - T_x\right) = \dot{m}_{wf} \cdot (h_2 - h_4) \quad (31)$$

To dissipate the required amount of waste heat, a cooling tower with a specified range, $T_y - T_x$, will need a mass flow rate determined by equation (31). This manner is acceptable if the cooling water having a constant specific heat $\bar{c}$ for the small temperature range from the inlet to outlet. Furthermore, we should calculate the other parameters in cooling system in order to obtain approach temperature; minimum approach in condenser; terminal temperature difference, condensation temperature, hot water temperature and heat rejected to the cooling tower as expressed below:

$$TTD = \geq 2.8°C \quad (32)$$

$$T_{cond.} = T_{cw} + \Delta T_i \quad (33)$$

$$T_{hw} = T_{cw} + Temp.increase\ of\ cooling\ water \quad (34)$$

$$q_c = h_2 - h_4 \quad (35)$$

During uses water cooled condenser, some water will be lost due to evaporation, drift and blow down then we should calculate the make-up water as needed (Mwagomba, 2016). There are water losses due to drift losses in droplets carried out of the cooling tower with exhaust air as seen in Figure 6, although the inflow of dry air is unchanged.

In general, the evaporation loss rate is 1-1.5% of the total circulating water, blow down is normally 20% of evaporation loss, and the drift loss is 0.03% of the total circulating water flow rate.

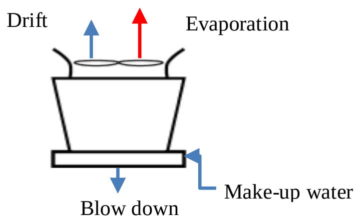

Figure 6: Losses in wet cooling systems

Water mass balance:

$$\dot{m}_y + \dot{m}_{a,in} \cdot \omega_{a,in} = \dot{m}_x + \dot{m}_{a,out} \cdot \omega_{a,out} \quad (36)$$

The energy balance:

$$\dot{m}_y \cdot h_y + \dot{m}_{a,in} \cdot h_{a,in} = \dot{m}_x \cdot h_x + \dot{m}_{a,out} \cdot h_{a,out} \quad (37)$$

To simplify the equation, solving for $\dot{m}_a$:

$$\dot{m}_a = \frac{\dot{m}_y \cdot \left(h_y - h_x\right)}{\left(h_{a,out} - h_{a,in}\right) - \left(\omega_{a,out} - \omega_{a,in}\right) \cdot h_x} \quad (38)$$

The mass flow of evaporation can be defined as follows (El-Wakil, 1984):

$$\dot{m}_e = \dot{m}_{air} \cdot (\omega_{out} - \omega_{in}) \tag{39}$$

Furthermore, referring to Perry and Green (2008) formulas, the drift losses as well as blow down for mass flow and make-up water required can be considered by the following equation:

$$\dot{m}_{drift} = 0.0002 \cdot \dot{m}_{cw} \tag{40}$$

$$\dot{m}_{bl} = \frac{\dot{m}_e - (Cycle - 1) \cdot \dot{m}_{drift}}{Cycle - 1} \tag{41}$$

$$\dot{m}_{mu} = \dot{m}_e + \dot{m}_{drift} + \dot{m}_{bl} \tag{42}$$

or referring to McDonald (2009), the blow down can be reached using:

$$\dot{m}_{bl} = \frac{\dot{m}_e}{Cycle - 1} \tag{43}$$

The air exit temperature for the cooling tower is therefore provided in equation (Leeper, 1981):

$$T_{c2} = \frac{(T_y + T_x)}{2} \tag{44}$$

or

$$T_{c2} = T_{db\ or\ c1} + T_{range} \tag{45}$$

$$T_{range} = T_y - T_x \tag{46}$$

Most geothermal power plants use mechanical draft type in cooling tower system, either wet or dry cooling. To calculate the power of the cooling tower fan using equation:

$$\Delta P_f = \rho_{air} \cdot H_{ct} \cdot g \tag{47}$$

$$\dot{V}_{air} = \frac{\dot{m}_{air}}{\rho_{air,out}} \tag{48}$$

$$\dot{W}_{fan} = \frac{\dot{V}_{air} \cdot \Delta P_{fan}}{\eta_{fan} \cdot \eta_{motor}} \tag{49}$$

3.4.4 Feed pump and cooling water pump

The state point of feed pump is between point 4 and 5, and the liquid is having a constant density. Then we can find the isentropic enthalpy, actual enthalpy of pump as well as cycle thermal efficiency using equation:

$$h_{5s} = h_4 + v_4 \cdot (P_{5s} - P_4) \cdot 100 \tag{50}$$

$$h_5 = h_4 + \frac{(h_{5s} - h_4)}{\eta_p} \tag{51}$$

$$w_p = h_5 - h_4 \tag{52}$$

$$\eta_{th} = \frac{(w_t - w_p)}{q_{in}} = 1 - \left(\frac{q_c}{q_{in}}\right) \tag{53}$$

$$\dot{W}_p = \frac{m_{wf} \cdot (h_{5s} - h_4)}{\eta_p} \tag{54}$$

Furthermore, we need to install the cooling water pump between cooling tower outlet and condenser inlet in order to transfer a fresh water to the condenser and cooling tower continuously. Hence the power needed to the pump will be expressed in equation:

$$Q_{water} = \frac{\dot{m}_{water}}{\rho_{water}} \tag{55}$$

$$WHP = \frac{Q \cdot H}{3960 \cdot \eta_{pump}} \tag{56}$$

For pump calculation, Leeper (1981) suggest that the head for the cooling tower pump as follows:

$$H_{ct} = Z_{dot} + 10 \tag{57}$$

3.5 Silica saturation index (SSI)

The scaling can be occurred in the brine if *SSI* more than 1 and if *SSI* equal to 1 is mean the silica in an equilibrium state. According to Fournier and Rowe (1977), the solubility of amorphous silica in the water can be obtained:

$$\log s_o = 4.52 - \left(\frac{731}{T\ (Kelvin)}\right) \tag{58}$$

Then we can calculate of correcting the silica solubility in the presence of fluid salinity through equation (Setschenow, Chen and Marshall):

$$\log D_{(t)} = -1.0569 - 1.573 \cdot 10^{-3} \cdot t \tag{59}$$

$$Salinity\ (m_{Cl}) = \frac{[Cl^-]}{35.5 \cdot 1000} \tag{60}$$

$$s_{eq} = s_o \cdot 10^{-m.D_{(t)}} \tag{61}$$

Lastly, we can determine the silica saturation index as follows:

$$SSI = \frac{SiO_{2\ brine}}{s_{eq}} \tag{62}$$

4. STUDY & DESIGN OF BINARY ORC

4.1 Piping

4.1.1 Pipe size selection

Based on operation/production data and tracer flow test result of unit V, would consist of used parameters as initial data to design and analysis within this research project:

- Atmospheric pressure = 0.924 bara;
- Separator pressure = 7.676 barg = 8.6 bara;
- Total flow rate of brine = 779.4 ton/hr or 216.5 kg/s (LHD-27, LHD-31, LHD-34).

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The new pipeline that will attach in the main line (16" sch. 20) would be chosen size of 6" schedule 40 with inlet diameter 0.15406 m. Using Bernoulli's principle, it can be defined the upstream pressure in the new pipe as a liquid transport to binary turbine.

Then from pressure above, we can find brine velocity 2.44 m/s, minimum thickness of pipe 6" is 3.43 mm according to process piping standard and other physical properties.

To calculate a flow rate inside pipe 6" schedule 40, using the liquid density from working pressure 8.75 bara, that is at 893.14 kg/m^3. The thickness from pipe 6" schedule 40 is greater than minimum thickness needed 7.11 mm > 3.43 mm, it would prevent the total dissolved solid and acid fluids those carry-out from geothermal fluid wells in 30 years, in the planning of design. Moreover, this design will be planned of flow rate 40.62 kg/s from the total of mass flow 216.5 kg/s according to pipe size capacity has been chosen.

4.1.2 Pressure drop

From working pressure 8.75 bara, so the physical properties of liquid can be determined from steam table to calculate Reynolds number, friction factor, equivalent length and friction head relate to obtain pressure drop as summarized in Table 3 with plan of pipe length 30 meters.

Table 3: Physical properties and pressure drop in pipeline system

No.	Description	Value	Uom
1.	Liquid density (ρ_f)	893.14	kg/m^3
2.	Viscosity (μ_f)	$155.47 \cdot 10^{-6}$	kg/ms
3.	Reynolds number (Re_L)	2159456.9	-
4.	Absolute roughness (€) - commercial steel pipe	$1.5 \cdot 10^{-4}$ $4.57 \cdot 10^{-5}$	ft m
5.	Friction factor (f), Swanee-Jain approach	0.01533	-
6.	Total equivalent length (L_e)	48.59	m
7.	Friction head (H_f)	1.47	m
8.	Pressure drop (ΔP_f)	12857.99 0.13	Pa bar

So, from the results above we can obtain a pressure drop from upstream to downstream along new pipeline is 0.13 bar. In addition, the downstream pressure will be 7.83 barg – 0.13 barg = 7.70 barg or equal to 8.63 bara.

4.1.3 Heat loss along new pipeline

With using working pressure 7.83 bar g or 8.75 bar a, the physical properties of liquid can be obtained from steam table. Moreover, using average air temperature 22.8°C or 295.95K with wind velocity 0.66 m/s (PT PGE, 2015), the physical properties of air can be obtained from interpolation of atmospheric table values.

The pipe insulation consists of two layer materials, there are calcium silicate and aluminum sheet with chosen thickness 50 mm and 0.8 mm each (Nugraha, 2018).

Therefore, we can obtain of heat loss along new pipeline:

$$q = 3455.87\ W\ or\ J/s$$

$$\Delta T = \frac{3455.87}{40.62 \cdot 4383.998} = 0.019\ °C$$

As a result, the end of temperature at downstream side is 174.18°C – 0.019°C = 174.16°C.

4.2 Scenario of designing binary

This study would be calculated the binary power using Scilab and CoolProp. Some parameters also will be determined on best practice way in order to design binary system, and the cycle process have a modification to detailing of each state point. The initial known variables are as follows:

- Fluid = Pentane;
- Vaporizer pressure (P_v) = 8.63 bara (for SSI > 1) and 18.7 bara (for SSI < 1);
- Source temperature (S_1) = 173.57°C;
- Flow rate of brine (m_b) = 40.62 kg/s;
- Condensation temperature ($T_{cond.}$) = 44°C;
- Pinch point of vaporizer ($\Delta T_{pp,in}$) = 5°C;
- Pinch point of condenser ($\Delta T_{pp,out}$) = 10°C;
- Superheat temperature (T_{sh}) = 2°C;
- Boiling margin temperaure (T_{bm}) = 2°C;
- Turbine isentropic efficiency (η_T) = 85% (assumed);
- Feed pump isentropic (η_p) & cooling pump (η_{cwp}) eff. = 75% (assumed);
- Cooling tower fan (η_f) & motor eff (η_{mf}) = 65% & 75% (assumed).

The working pressure is decided 8.63 bara from setting in the feed pump, similar pressure with brine source, but it can decide tentatively just to obtain the initial result of the cycle process.

4.3 Cooling system (existing)

The condenser pressure can be determined from temperature of condensation and will be explained in the cooling system state. As was explained previously, the condenser is using surface type and coupled with wet cooling tower existing and explains the cooling process in the cycle as in Figure 7. On the circulating water inlet temperature should be sufficiently lower than the steam saturation temperature due to obtain a reasonable result of *TTD*, then according to El-Wakil (1984) the recommended of ΔT_i is between 11 to 17°C and *TTD* not less than 2.8°C. However, because Indonesia is a tropical country and relate with the size of condenser (not costly) will choose 17°C for approach temperature in condenser. Furthermore, Table 4 below would describe process condition of state point 3.

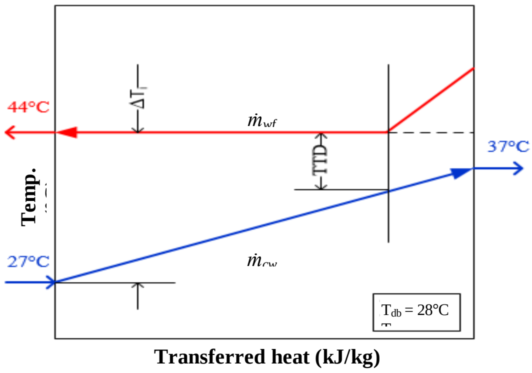

Figure 7: Condenser temperature distribution

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Table 4: The state point of cooling process cycle

No.	Desc.	Val.	Note
Cooling system - existing wet cooling tower in unit V LHD			
1.	Dry bulb temp. (T_{db} or T_{c1})	28°C	PT PGE (2015)
2.	Wet bulb temp. (T_{wb})	24°C	
3.	Relative humidity (RH)	72.03%	Psychrometric calc. online
4.	Max. humidity (RH_{max})	98%	PT PGE (2015)
5.	Approach temp. (A_T)	3°C	The best approach temp. in realistic
6.	Cold water temp. (T_x or T_{cw})	27°C	
7.	Min. approach in condenser (ΔT_i)	17°C	Between 11 ~ 17°C.
8.	Terminal temperature diff. (TTD)	2.8°C	Not less than 2.8°C
9.	Temp. of condensation (T_{cond})	44°C	$T_{cw,in} + \Delta T_i$
10.	Temp. increase of the cooling water	10°C	Assumed
11.	Hot water temp. (T_y or T_{hw})	37°C	
12.	Specific heat cooling fluid ($\bar{c}$)	4.23 kJ/(kg·°C)	Using steam table at P = 1.32 bara

The mass flow of cooling water ($\dot{m}_{cw}$) in cooling system cycle can be obtained by relationship of state point at turbine inlet, turbine outlet, cooling system and working fluid then giving result 81.34 kg/s. The advantage of using wet cooling tower in this study is minimize cost due to using the existing cooling tower in unit V.

4.4 Optimization of binary using numerical computation

The rule of thumb of superheat and boiling margin temperature is having range 1 to 2°C in the common practice and remove the specific heat capacity of source temperature in calculation due to the enthalpy have a more accurately in result (Dr. Páll Valdirmarsson, Adjunct Professor, Reykjavík University, personal communication, September 5, 2018).

After putting the above values into Scilab programme in order to acquire optimization of power output with different working fluids as in Figure 8, results were obtained as describe in Figure 9 is using Pentane with *SSI* > 1 and Figure 10 with *SSI* < 1. The results are summarized in Table 5 and Table 6.

Table 5: Summary of calculated using Scilab and CoolProp (Pentane)

No.	Description	Value: Binary 1	Value: Binary 2	Uom
1.	Inlet pressure	8.63	18.7	bara
2.	Inlet temperature	119.56	161.43	°C
3.	Mass flow of working fluid	31.95	7.73	kg/s
4.	Brine outlet temperature	86.82	150.39	°C
5.	Power output of turbine (gross)	1948.53	682.46	kW
6.	Silica saturation index	> 1	< 1	-

Table 6: Power output in working fluids (SSI < 1)

No.	Working fluid	Net power output (kW)	End temperature (°C)
1.	Isopentane	614.26	150.34
2.	Pentane	626.1	150.39
3.	Butane	1918.12	89.23
4.	Cyclopentane	692.99	150.3

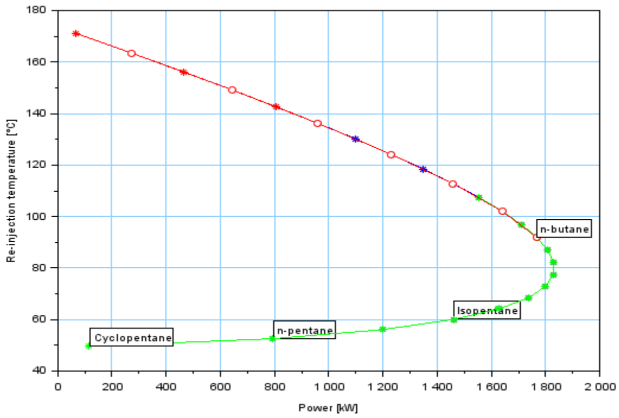

Figure 8: Optimization of binary ORC (drawn and calculated with Scilab)

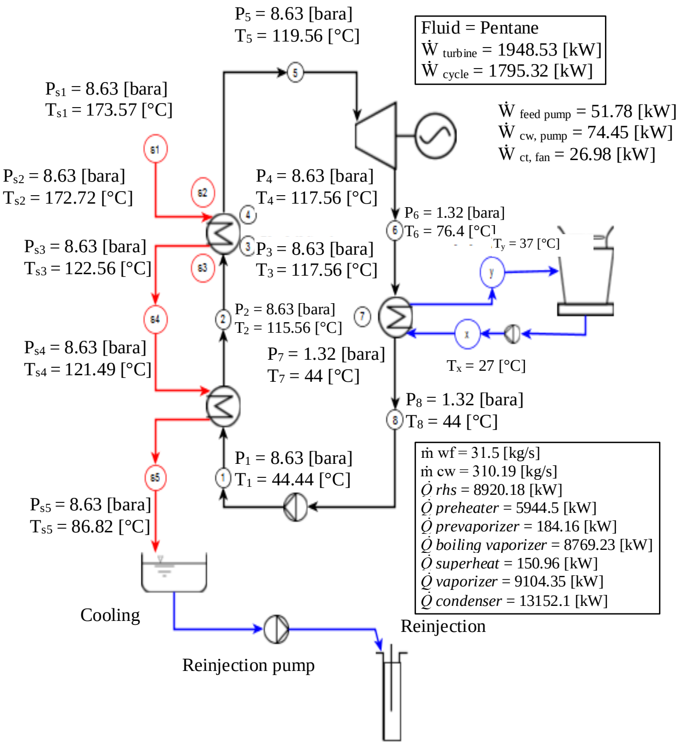

Figure 9: Binary cycle with SSI > 1 and wet cooling type (calculated with Scilab and CoolProp)

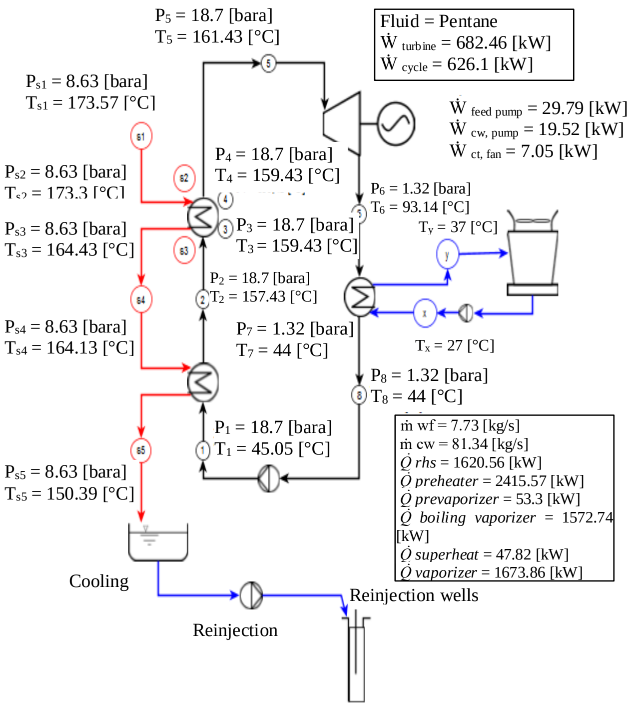

Figure 10: Binary cycle with SSI < 1 and wet cooling type (calculated with Scilab and CoolProp)

4.5 Silica saturation index (SSI)

The laboratory tests result from separator of unit V on June 25, 2018 (PT PGE, 2018b), some of datas had presented:

- pH = 8.6; S_iO_2 = 620 mg/l; Cl^- = 705 mg/l.

The data above explain the potential of silica scaling in the new pipeline system with actual condition parameters, from the tapping line to heat exhanger and pre heat exchanger. Table 7 below shown the *SSI* results with different temperatures.

Table 7: Silica saturation index in binary ORC cycle

Desription	Pressure (bara)	End temp. (°C)	SSI
Binary ORC with wet cooling tower	8.63	86.82	2.02
	18.7	150.39	0.99

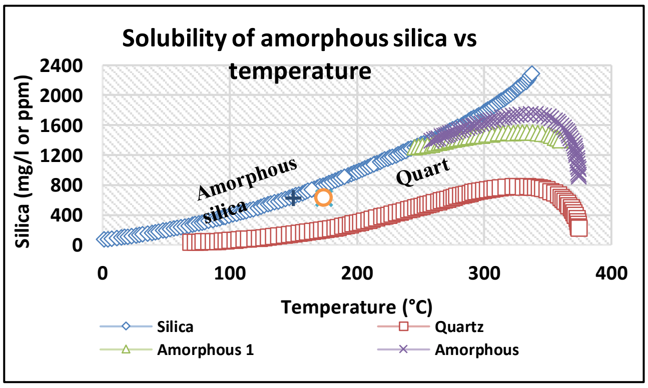

Figure 11: Amorphous silica vs temperature

2. CONCLUSION

From the chemistry compositions of geothermal fluids in unit V Lahendong, the minimum temperature that can be utilized for the system is 150.39°C in order to prevent silica scaling in the evaporator and preheater. By using a suitable working fluid (pentane), the net power of binary that would be able to be exported into the grid is 626.1 kW with existing wet cooling tower as the cooling system. However, the power output has a high result if SSI not considered, about 3 to 4 times what is reported in this study (Nugraha, 2018).

The option of SSI > 1 can accepted if the mitigation of silica scaling have a treatment through silica inhibitor for example. Besides, it should consider between chemical cost and revenue, the duration to clean out facilities (pipeline, pre-heater/heater), reinjection wellhead pressure.

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