EVALUATION OF WASTE BRINE UTILIZATION FROM LHD UNIT III FOR ELECTRICITY GENERATION IN LAHENDONG GEOTHERMAL FIELD, INDONESIA

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ABSTRACT

The geothermal power plant (PLTP) LHD unit III in Lahendong geothermal field is scheduled to begin operation in 2008. The power plant and the piping are under construction. This plant is supplied from the wells in cluster-5 of Lahendong geothermal field. More than 600 tons/hr of separated water at a temperature of 180°C will be reinjected directly into the reinjection well. The separated water comes from the separator which is supplied from 4 production wells in this cluster. These wells produce two-phase fluid with a wetness of 80%. The dry steam from the separator will be used to generate 20 MWe from a condensing turbine single-flash electric power plant which is to be operated by the state electricity company. The separated water will be reinjected. The paper discusses alternatives for possible additional electricity production from the separated water, which still contains high enthalpy, by using direct flashing of hot brine or using hot brine to heat the secondary fluid with a binary or Kalina cycle. These alternatives are tested by building a power plant model and doing model simulations. The rejected brine can generate 4.8 MWe using a flash system with a condensing turbine, and 4.57 MWe by using a binary cycle. The design considerations, utilization limits due to chemical problems, and the optimum parameters to get maximum electricity generation are also discussed.

1. INTRODUCTION

1.1 Field description

Lahendong geothermal field is located in the northern part of Sulawesi Island, Indonesia, 30 km south of Manado, the capital city of North Sulawesi province (Figure 1). This field is categorized as a two-phase geothermal field where geothermal fluid produced from all of the wells is a mixture of steam and brine. The field is operated by Pertamina Geothermal Energy and the power plant is owned and operated by the state electricity company (PLN). By the year 2006, the number of wells was 23, and the total installed capacity in the field 40 MWe. The geothermal system in this area is divided into two different categories based on temperature and type of fluid in the reservoir. In the south (clusters-13 and 4), the temperature of the reservoir is very high (350°C) with the dryness of the steam valued at more than 80%; in the north (cluster-5), the temperature of the reservoir is about 250°C, with a large

FIGURE 1: Location of the Lahendong geothermal field

fraction of water (dryness about 20%). Exploration drilling in this field was carried out from 1984 to 1987 with 7 exploration wells drilled throughout the area to identify the boundaries of the reservoir. Between 1988 and 1998, nine standard wells were drilled in cluster-4. Maximum steam availability at the wellhead from 4 production wells in this cluster is 217 tons/hour. In 2001, the first 20 MWe single-flash condensing turbine was installed and put into commercial operation. This power plant was supplied from production wells in cluster-4 with 147 tons/hour steam consumption.

The developmental drilling continued with 7 wells in clusters 13 and 5. Steam gained from two wells in cluster-13 was added to the excess steam from cluster-4 to generate the second PLTP LHD unit-2 (20 MWe), commissioned in 2007. The location of the wells at each cluster and the power plant can be seen in Figure 2.

The third 20 MW (PLTP unit III) is scheduled to be commercially operational by 2008. The plant will have an installed capacity of 20 MW and will be supplied with geothermal fluid from 4 production wells in cluster-5 (wells LHD-5, LHD-9, LHD-21 and LHD-23).
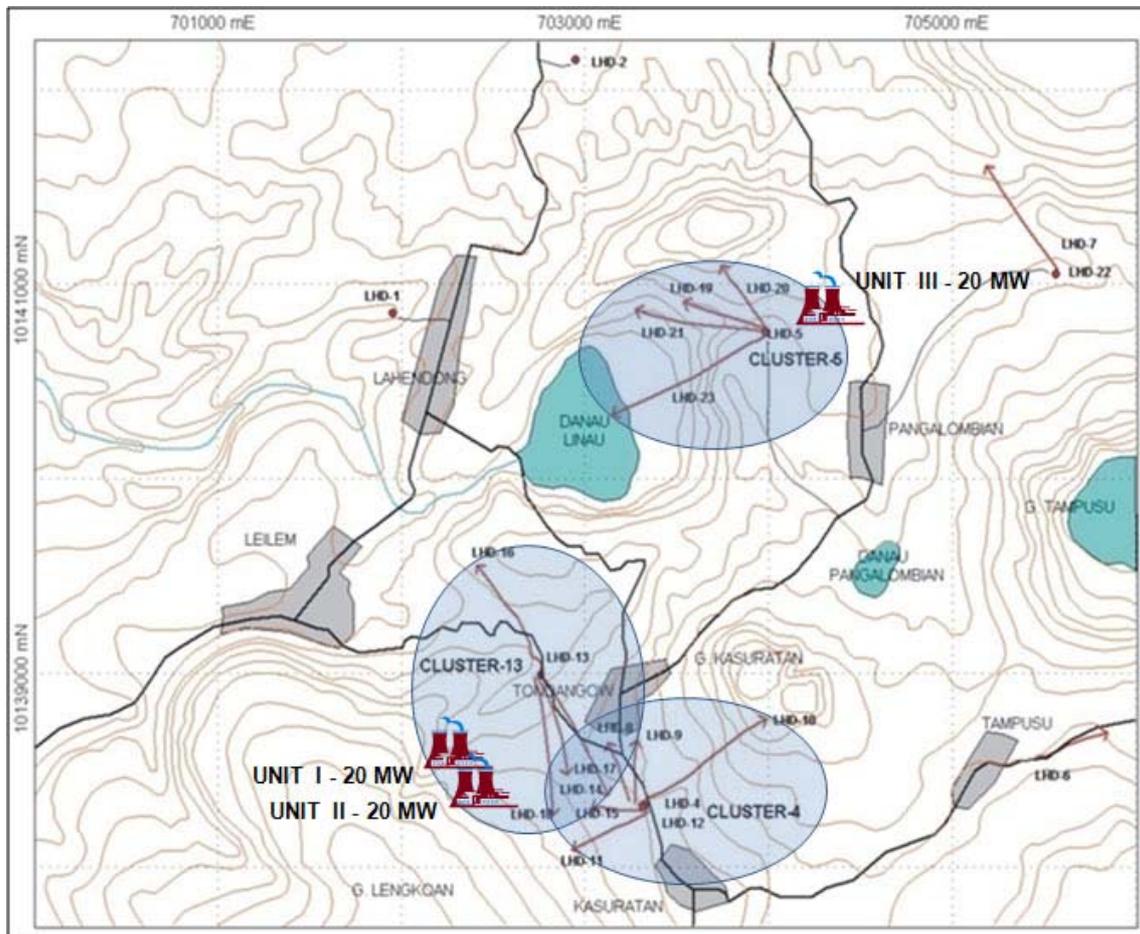

FIGURE 2: Lahendong geothermal field, wells and power plant locations

2. STUDY DESCRIPTION

2.1 General overview

LHD unit III plant has been designed in such a way that 800 tons/hr of two-phase brine-steam mixture (dryness 20%) from 4 production wells in cluster-5 (wells LHD-5, LHD-9, LHD-21 and LHD-23) at wellhead pressure 13.4 bar can be throttled down to a flashing pressure of 10.5 bar and separated in the two separators to produce total steam of 175 tons/hr in the steam manifold. The separated steam will be piped to the power plant through a demister to make the steam dryer before entering turbine power plant unit III. The scheduled 20 MWe may consume total steam of about 147 tons/hr. The separated brine, on the other hand, will be collected in the brine manifold and reinjected directly back into the formation (Figure 3). It is the rejected brine that is being considered for use as the source for the proposed smaller electricity generation that will be discussed in this paper. Data of brine at brine manifold is:

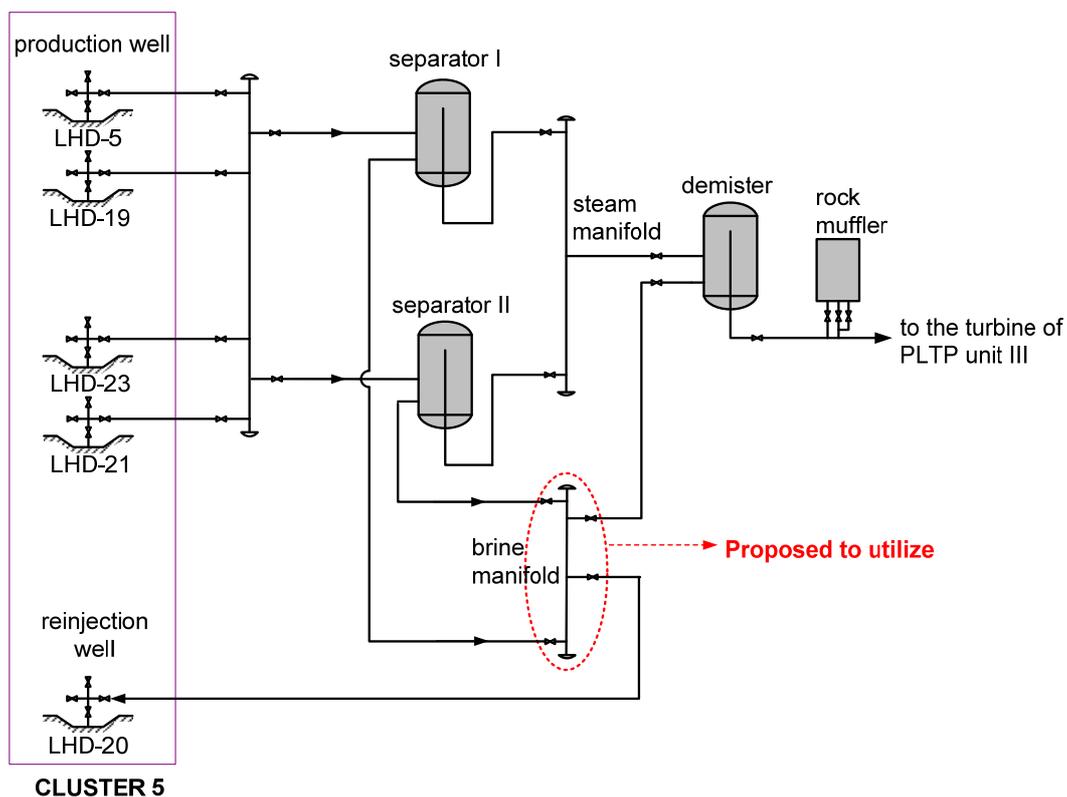|	
Temperature:	180.8° C
Pressure:	10.23 bar
Mass flowrate:	624.82 ton/hr (173.56 kg/s)
Mass enthalpy:	768.41 kJ/kg

FIGURE 3: Simplified piping design for the 20 MWe Lahendong unit III

2.2 Objective of the study

In this paper, the author makes a preliminary thermodynamic model of the waste brine utilization using five different plant configurations. These are:

- Flash system with back pressure turbine;
- Flash system with condensing turbine;
- Binary cycle without regeneration;

- Binary cycle with regeneration;
- Kalina cycle.

The model will be simulated to compare the optimum power outputs generated by each type of configuration and to determine the parameters of those systems. The limitations of the rejected brine's temperature, due to chemical problems, will also be included to determine maximum possible power output. Power outputs from each configuration will be compared in order to decide on the most suitable plant based on the input conditions of the separated brine in the brine manifold.

3. METHODOLOGY

3.1 General information and assumptions

The software used to perform the fluid thermodynamic property calculations and to run the models for each operating condition is the Engineering Equation Solver (EES) software (F-Chart Software, 2007). The Kalina model to be considered is based on Kalina model KSC 34. Economic calculations are not included in this paper due to limited information about the price of the power plant equipment.

General assumptions are:

1. The ambient air temperature (t_{dry}) in Lahendong field is 28°C with a relative humidity of 0.76;
2. Atmospheric pressure in Lahendong field is 0.96 bar;
3. Pressure drop in the system is neglected;
4. The isentropic efficiency of the turbine is 0.85;
5. The isentropic efficiency of the pump and the fan is 0.75 and 0.65, respectively;
6. Temperature increase of the cooling water in the condenser is 10˚C;
7. The overall heat transfer coefficient, *U*, used to calculate the area required for the heat exchanger was assumed:
   a. $U = 0.9$ for the evaporator;
   b. $U = 1$ for the regenerator;
   c. $U = 1.1$ for the condenser.
8. Due to a lack of final chemical data of the wells in cluster 5, the silica concentration of the brine will be determined using quartz concentration based on the reservoir temperature in cluster 5 (250°C).

3.3 Calculation steps

In this study, calculations were performed in the steps listed below.

For a flash system with an atmospheric exhaust turbine:
- steam mass flowrate resulting from the flashing process;
- turbine power output;
- utilization efficiency (exergy efficiency).

For a flash system with a condensing turbine:
- steam mass flowrate resulting from the flashing process;
- steam mass flowrate needed for non-condensible gases (NCG) removal using two-stage steam-jet ejector;
- turbine power output;
- condenser heat load and mass flowrate of the cooling water;
- power needed for the condensate pump;

- power needed for the cooling water pump;
- power needed for the cooling tower fan;
- total power output;
- utilization efficiency (exergy efficiency);
- silica scaling index;
- every calculation is performed with different flashing and condenser pressures to get optimal values.

For a binary cycle:
- maximum working fluid mass flowrate;
- thermodynamic properties at the inlet and outlet states of the plant components using mass balance and energy balance equations;
- turbine power output;
- required heat exchanger area;
- condenser heat load and mass flowrate of the cooling water;
- power needed for the cooling water pump;
- power needed for the cooling tower fan;
- total power output;
- utilization efficiency (exergy efficiency);
- silica scaling index;
- every calculation is performed with a different pinch in the heat exchanger and evaporation temperature to get optimal values.

For the Kalina cycle:
- maximum working fluid mass flowrate;
- thermodynamic properties at the inlet and outlet states of the plant components using mass balance and energy balance equations;
- turbine power output;
- condenser heat load and mass flowrate of the cooling water;
- power needed for the cooling water pump;
- power needed for the cooling tower fan;
- total power output;
- utilization efficiency (exergy efficiency);
- silica scaling index;
- every calculation is performed with different evaporator pressures and composition of ammonia-water mixtures to get optimal values.

The thermodynamic models for the different plant configurations are shown in Appendix I.

4. THEORY OVERVIEW

4.1 Plant configuration

The waste hot brine geothermal resource can be utilized by two fundamentally different schemes:

1. *Direct system:* the hot brine is flashed and the steam is sent directly into a steam turbine, called a flash system. According to the turbine exhaust pressure, there are two types of this system: a flash system with an atmospheric-exhaust pressure turbine, and a flash system with a condensing turbine.
2. *Indirect system*: the hot brine is used as a heat source for heating a working fluid in a separate closed conversion cycle. There are two types of indirect systems to be discussed in this paper, the *binary cycle* and the *Kalina cycle*.

4.1.1 Flash system

Process description.
The throttle valve creates a flow restriction which maintains a pressure drop from state 1 to state 2 (Figure 4) therefore producing a steam-water mixture. The steam is then separated in the flash vessel which is constructed so that the two phases are separated by centrifugal and gravity effects, producing a stream of saturated liquid and saturated vapour. We assume that the vessel is perfectly insulated (adiabatic) and that the throttle operates isenthalpically. The saturated steam is sent directly to the turbine which is coupled with a generator to produce power. The steam mass flowrate is determined by using the following equation :

$$h_1 = h_2$$

$$\dot{m}_{steam} = x_2 . \dot{m}_{brine\ in}$$

where x_2 = The quality of the steam-water mixture entering the flash vessel, defined by:

$$x_2 = \frac{h_2 - h_5}{h_3 - h_5}$$

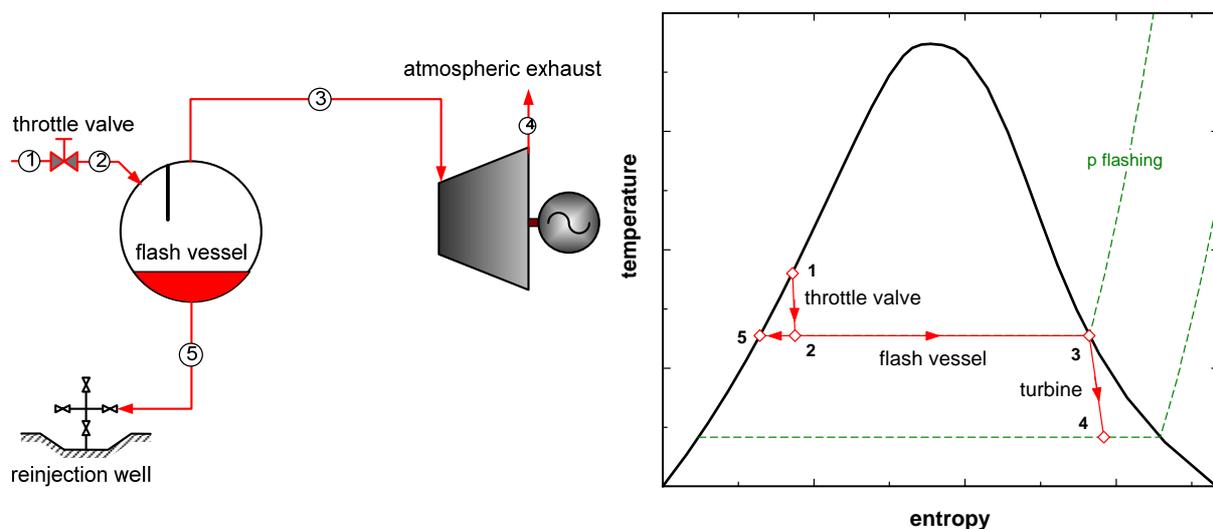According to the type of turbine (exhaust condition of the turbine), this system can be divided into 2 types:

FIGURE 4: Schematic diagram and t-s diagram for a flash system with a back-pressure turbine

Back-pressure / atmospheric exhaust steam turbine
In a back-pressure unit, steam from the flasher/separator, after passing the turbine, is exhausted directly to the atmosphere. These units are simplest, lowest in capital cost, can be constructed and installed very quickly and put in operation in around 13-14 months (Hudson, 1988). The steam consumption per each power produced is almost double from the condensing type at the same inlet pressure. A simplified schematic of an atmospheric exhaust plant is shown in Figure 4. The altitude that the unit is operated at has an important effect on the power that is produced from a given inlet pressure and steam mass flow. At higher altitudes the lower exhaust pressure, corresponding to the lower atmospheric pressure, results in greater power generation (Hudson, 1995).

Condensing steam turbine
Adding a condenser allows the steam-gas mixture to expand down to a sub-atmospheric pressure (vacuum pressure) and produces more power with respect to atmospheric discharge (Figure 5).
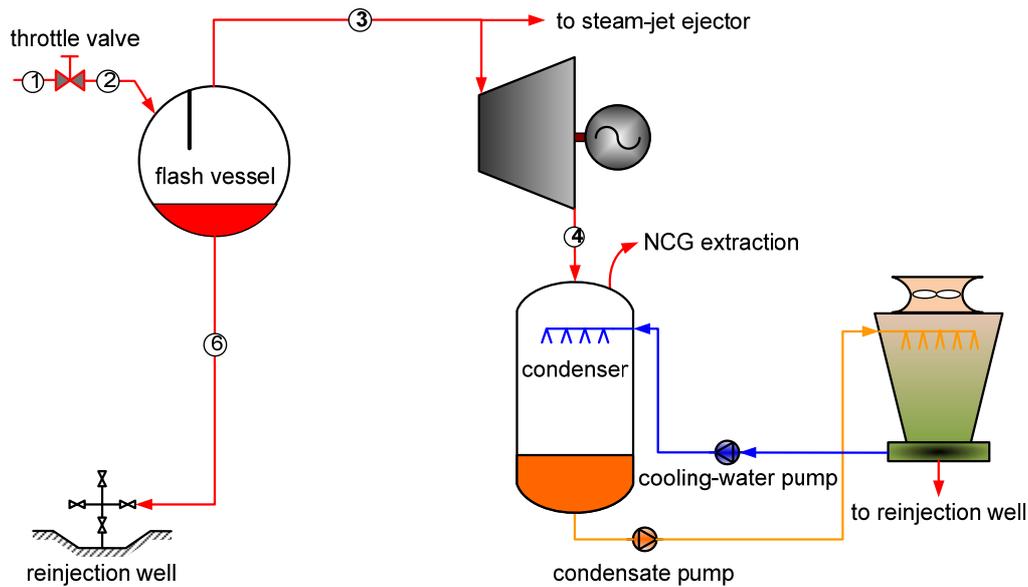
FIGURE 5: Schematic diagram for a flash system with a condensing turbine

If the condensation pressure reaches sub-atmospheric values, it is then necessary to employ a device (steam-jet ejector or a vacuum pump) in order to extract the non-condensable gases from the condenser and finally release them into the ambient air. A single stage steam jet ejector can operate at condenser pressures down to 0.13-0.26 bar, but two stages or more are needed for lower pressures. This model uses a two stage steam jet-ejector for condenser pressure down to 0.08 bar.

4.1.2 Binary cycle

Process description.
The working fluid absorbs heat from a heat source, in this case the hot brine via shell-and-tube heat exchangers. This heat causes the working fluid to evaporate, producing the high-pressure vapour that is then expanded through a turbine that is coupled to the generator. The low-pressure turbine exhaust vapour is then condensed, using either air-cooled heat exchangers or a water-cooled, shell-and-tube condenser. In this case, a water-cooled condenser is coupled with a wet cooling tower. From the condenser, the liquid working fluid is pumped to a high pressure and returned to the pre-heater to close the cycle (Figure 6).
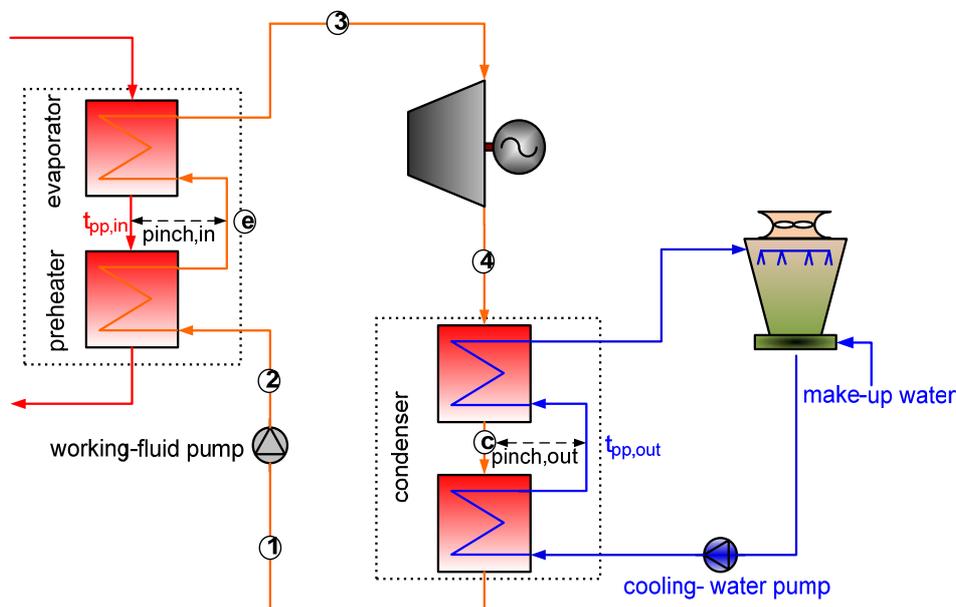

FIGURE 6: Simplified schematic of a basic binary cycle

It is also possible to incorporate an additional heat exchanger into the cycle, normally known as a regenerator. In this heat exchanger, residual sensible heat in the low-pressure turbine exhaust stream is used for initial preheating of the cold liquid from the motive fluid pump, thus increasing the cycle efficiency. Figures 7 and 8 show the T-s and P-h diagrams for the binary plant.
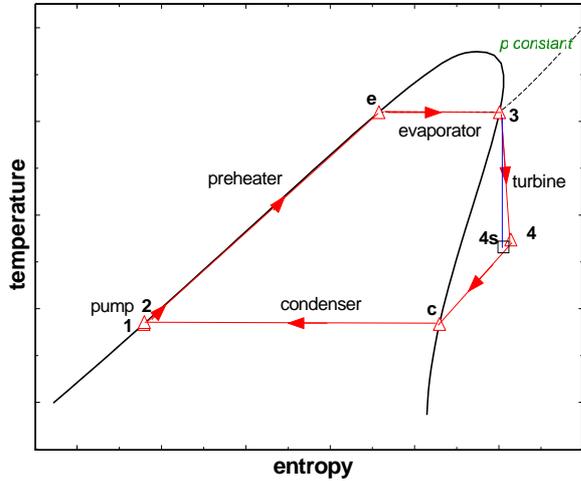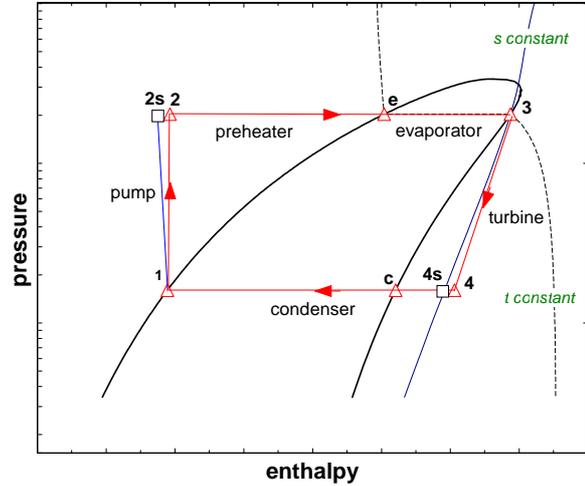

FIGURE 7: T-s diagram for a binary cycle

FIGURE 8: P-h diagram for a binary cycle

Heat exchanger

Heat exchangers are required to heat and evaporate the binary fluid, as well as to de-superheat and condense it. The binary fluid would normally be heated and evaporated in two separate units, a pre-heater and an evaporator. The pre-heater provides sensible heat transfer in order to raise the working fluid to its boiling point (state e). The evaporation occurs from e-3 along an isotherm for a pure working fluid. The place in the heat exchanger where th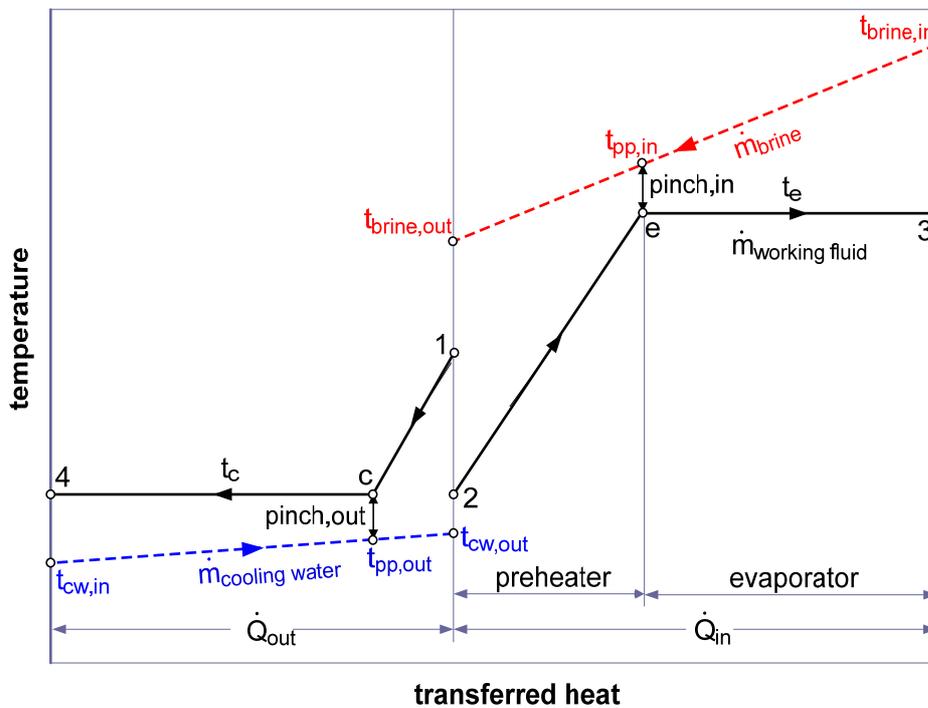e brine and the working fluid experience the minimum temperature difference is called "the pinch". State 2 is a compressed liquid, the outlet of the feed pump, state e is a saturated liquid at the evaporator pressure, and state 3 is a saturated vapour. The condenser absorbs the sensible heat of the working fluid with cooling water to de-superheat the working fluid to its dew point (state c). The condensation occurs from c-4 along an isotherm line (Figure 9).

FIGURE 9: Q-t diagram for a binary cycle

The size of the heat exchanger is physically large and the equipment represents a great portion of the capital cost of a binary plant. The heat exchanger size can be calculated using the following equation:

$$q = UA\Delta T_{LM} \text{ or } A = \frac{q}{UA\Delta T_{LM}}$$

where q = The heat transfer rate;
U = The overall heat transfer coefficient; and
ΔT_{LM} = The logarithmic-mean-temperature-difference, which is found from:

$$\Delta T_{LM} = \frac{GTTD - LTTD}{ln\left[\frac{GTTD}{LTTD}\right]}$$

where $GTTD$ = Greater terminal temperature difference;
$LTTD$ = Lower terminal temperature difference.

Referring to Figure 9, the logarithmic-mean-temperature-difference in the pre-heater is:

$$\Delta T_{LM} = \frac{\left(t_{brine,out} - t_2\right) - \left(t_{pp,in} - t_e\right)}{ln\left[\frac{t_{brine,out} - t_2}{t_{pp,in} - t_e}\right]}$$

The overall heat transfer coefficient should be determined by experiment with the appropriate fluids to be used in the plant.

4.1.3 Kalina cycle

A Kalina cycle is principally a modified Rankine cycle using a "mixture" of ammonia and water as the working fluid instead of a pure component (typically water).

The Kalina PP on the block diagram shown in Figure 10 is based on the Kalina Cycle System 34 (KCS34) license, which applies the same flow scheme as that in the Húsavík power plant in Northern
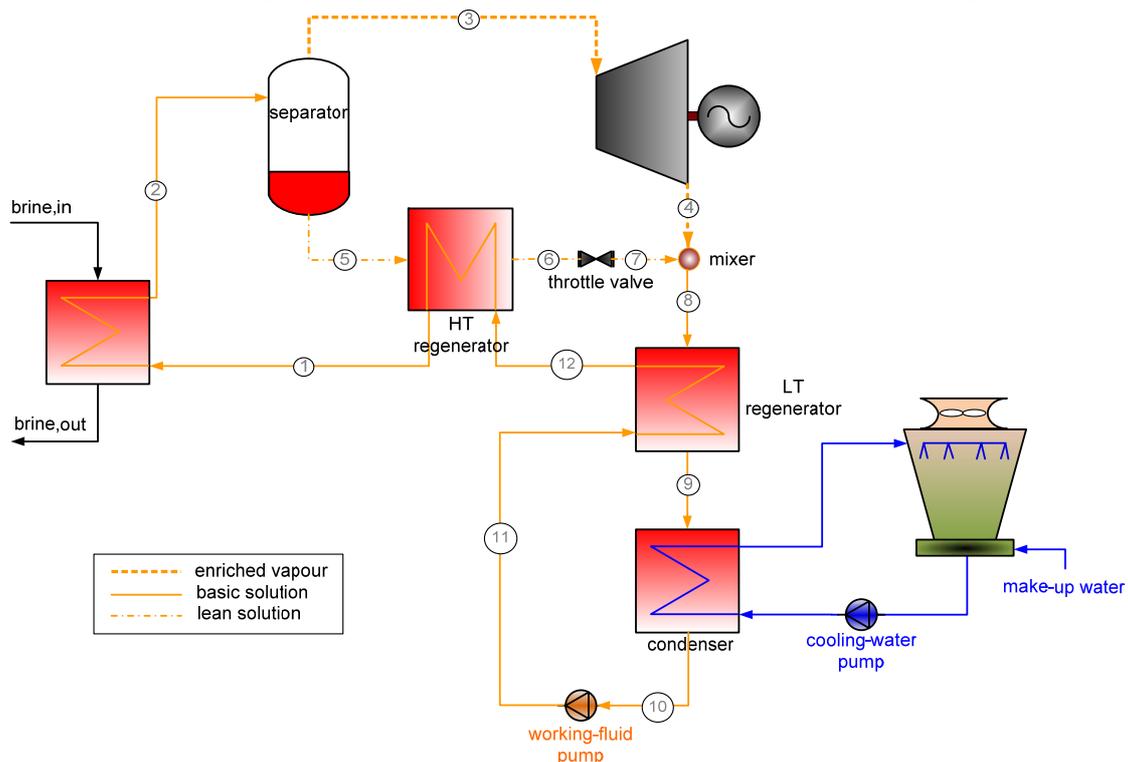

FIGURE 10: Schematic diagram for the Kalina cycle

Iceland. The basic ammonia-water stream is vaporized in the vaporizer and then separated into a saturated rich vapour ammonia-water mixture stream and a saturated lean liquid ammonia-water mixture stream in the separator. The basic stream reappears when the saturated liquid stream is cooled in the HT regenerator, throttled in the throttle valve, and then mixed with the saturated vapour stream which is passed through the turbine. After that the basic stream is cooled in the LT recuperator and condensed in the condenser. The condensed basic stream is pressurized before re-entering the LT and HT regenerator. Here the basic stream returns to its initial state and the processes are repeated.

The main benefit of the Kalina cycle is that heat addition to the process happens at a variable temperature, and can thus be fitted to the falling temperature of a heat source with a finite heat capacity, reducing the generation of entropy in the heat exchange with the primary fluid (Valdimarsson and Elíasson, 2003).

The composition of the fluid is changed in the cycle at different points. The ammonia in the mixture begins to vaporize first; as it boils off, the liquid mixture ammonia concentration decreases, and the boiling temperature of the liquid mixture increases. This reduces the temperature mismatch between the topper's waste heat and the fluid in the evaporator as shown in Figure 11 and allows an efficiency rise in the bottoming cycle.
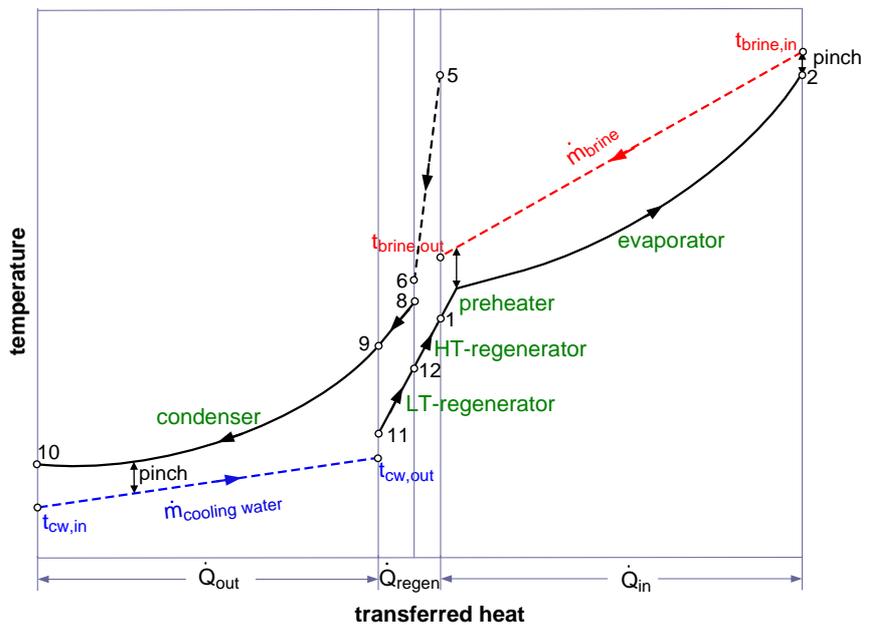

FIGURE 11: Q-t diagram for the Kalina cycle

4.2 Wet cooling tower

In a wet cooling tower, air and water are intimately mixed to provide heat transfer. Therefore, psychometry is the basis for analysis of heat transfer in a wet cooling tower. Heat transfer in cooling towers occurs via two major mechanisms: transfer of sensible heat from water to air by convection, and transfer of latent heat by the evaporation of water.

Wet cooling tower calculation
These calculations were performed based on the following assumptions:
- Steady operating conditions; the mass flowrate of dry air remains constant during the entire process;
- The kinetic and potential energy changes are negligible;
- The cooling tower is adiabatic.

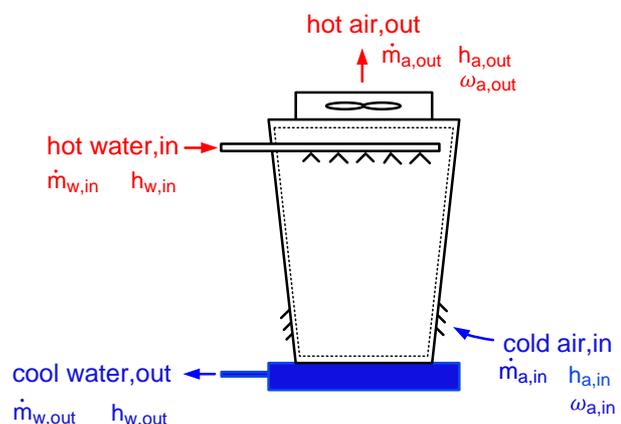Applying the mass and energy balances on the cooling tower (Figure 12) gives:

FIGURE 12: Energy balances in a cooling

Dry mass air balance:

$$\dot{m}_{a,in} = \dot{m}_{a,out} = \dot{m}_a$$

Water mass balance:

$$\dot{m}_{w,in} + \dot{m}_{a,in}.\omega_{a,in} = \dot{m}_{w,out} + \dot{m}_{a,out}.\omega_{a,out}$$

Energy balance:

$$\dot{m}_{a,in}.h_{a,in} + \dot{m}_{w,in}.h_{w,in} = \dot{m}_{a,out}.h_{a,out} + \dot{m}_{w,out}.h_{w,out}$$

Solving for $\dot{m}_a$ gives:

$$\dot{m}_a = \frac{\dot{m}_{w,in}(h_{w,in} - h_{w,out})}{(h_{a,out} - h_{a,in}) - (\omega_{a,out} - \omega_{a,in}).h_{w,out}}$$

The volume flowrate of air into the cooling tower is:

$$\dot{V}_{air,in} = \frac{\dot{m}_a}{\rho_{a,in}}$$

The mass flowrate of the required makeup water is determined from:

$$\dot{m}_{make-up} = \dot{m}_a(\omega_{a,out} - \omega_{a,in})$$

where $h_{a,in}$ = Enthalpy of the cold dry air (kJ/kg dry air), from the psychometric chart;
  $h_{a,out}$ = Enthalpy of the hot dry air (kJ/kg dry air), from the psychometric chart;
  ω = Ratio of water vapour to dry air in a particular height of air, expressed as a ratio of
      kilograms of water vapour per kilogram of dry air, from the psychometric chart;
  $\dot{m}_{a,in}.\omega_{a,in}$ = Water content in the incoming air stream;
  $\dot{m}_{a,out}.\omega_{a,out}$ = Water content in the leaving air stream; and
  h_w = Enthalpy of the saturated liquid water, from the steam table.

The power of the cooling tower fan is:

$$P_{fan} = \frac{\dot{V}_{air}.\Delta p}{\eta_{fan}}$$

4.3 Total power output

The total power output of a plant is the power produced by the turbine from which the auxiliary power consumption for a pump and a fan is subtracted.

$$P_{net}\text{ (flashing with atm.-exhaust turbine) } = P_{turbine}$$

$$P_{net}\text{ (flashing with condensing turbine) } = P_{turbine} - (P_{condensate\ pump} + P_{cooling\text{-}water\ pump} + P_{fan})$$

$$P_{net}\text{ (binary and Kalina cycle) } = P_{turbine} - (P_{feed\ pump} + P_{cooling\text{-}water\ pump} + P_{fan})$$

4.4 Overall cycle analysis

Plant performance can be obtained using the second law in the form of the utilization efficiency (exergy efficiency) η_{EX}, which is defined as the ratio of the actual net plant power to the maximum

theoretical power obtained from the geofluid in the reservoir state, in this case the hot brine in the brine manifold. The utilization efficiency η_{EX} measures how well a plant converts the exergy (or available work) of the resource into useful output. For a geothermal plant, it is found as follows:

$$\eta_{EX} = \frac{P_{net}}{\dot{Ex}} = \frac{P_{net}}{\dot{m}_{brine}\left[(h_{brine,in} - h_0) - T_0(s_{brine,in} - s_0)\right]}$$

where P_{net} = The net electric power generated by the plant, ;
  T_0 = The dead-state temperature (K), in this case, the ambient wet-bulb temperature because of using a wet cooling tower;
  h_0 = The enthalpy value at the dead-state pressure and temperature; and
  s_0 = The entropy value at the dead-state pressure and temperature.

5. DESIGN CALCULATIONS

5.1 Design calculations for a flash system

Determining the temperature of the cooling fluid ($t_{cw,in}$)
In Lahendong field, the highest ambient temperature throughout the year reaches 28°C with an average relative air humidity of 78%. The wet bulb temperature calculated from the psychometric chart is 25°C. Because of a limited sou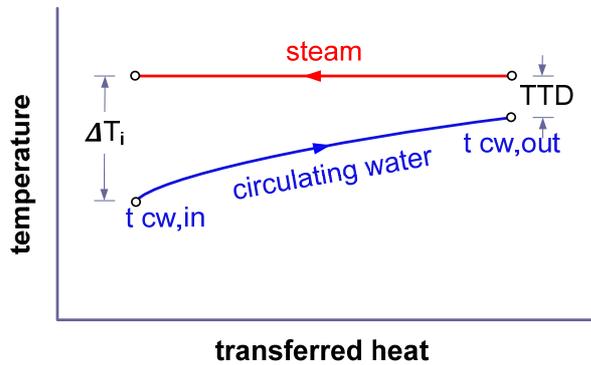rce of cooling water, the wet cooling tower was chosen. With a wet cooling tower, the "cold water" temperature approaches the "wet bulb" temperature. Realistically the best approach temperature is about 3°C. Therefore, in this calculation a "cold water" temperature of 28°C was used.

FIGURE 13: Condenser temperature distribution

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

By using the possible cooling water input ($t_{cw,in}$) from the wet cooling tower at 28°C and assuming a minimum approach in condenser (ΔT_i) by 12°C (Figure 13), temperature of condensation to be $t_{cw,in} + \Delta T_i = 41$°C. The condenser pressure is, then, the saturated pressure at this temperature. From the steam table or using EES minimum p_{cond}, based on this criteria, the condenser pressure is 0.08 bar (see Table 1).

TABLE 1: The saturation temperature at different condensing pressure

P cond. [bar]	t sat [°C]
0.1	45.8
0.09	43.7
0.08	41.5
0.07	39
0.06	36.2
0.05	32.9
0.04	29
0.03	24.1
0.02	17.5
0.01	7

5.2 Design calculations for a binary cycle

Choosing the secondary fluid based on the resource temperature of the brine
The working fluid must be selected according to its critical temperature, which must be suitable for the

temperature level of the geothermal source. Critical temperatures for some common binary fluids are in Table 2.

Based on this table, the possible commercial secondary fluids for a source temperature of 180°C are only isopentane and n-pentane which have a critical temperature above 180°C.

Determining the condensing temperature and corresponding saturated condensing pressure

The condensing temperature is dependent on the available temperature of the cooling fluid. By assuming a temperature increase of the cooling water by 10˚C and taking a certain value a TTD (Terminal Temperature Difference) between the temperature of the water at the exit of the condenser and the temperature of the turbine exhaust, the temperature of the condensers can be found. The turbine exit pressure (condenser pressure) is then the working fluid vapour pressure at this temperature.

TABLE 2: Critical temperature of different secondary fluid for binary cycle

ORC fluid	Critical temp. [°C]
Isopentane	187.2
N-pentane	196.5
Isobutane	134.7
N-butane	152

Determining the mass flowrate of the secondary fluid

Assume that the exit condition of the evaporator is saturated steam. A superheated region is not needed because the addition of superheat produces only small increases in the power output over the saturated vapour cycle due to the lack of significant divergence of the lines of constant entropy (see the P-h diagram in Figure 7). Therefore, superheat cycles are not being considered for geothermal power cycles with the currently popular working fluids. Supercritical cycles are also not being considered though may offer thermodynamic advantages when used with brine temperatures above 233C (Kestin et al., 1980).

The maximum mass flowrate of the calculated secondary fluid is based on the equation:

$$\dot{m}_{brine} \cdot \left(h_{brine,in} - h_{brine,pp} \right) = \dot{m}_{wf} \cdot h_{latent@te}$$

By determining a certain evaporation temperature and a certain pinch in the evaporator, the maximum mass flowrate of the working fluid can be calculated.

5.3 Design calculations for a Kalina cycle

In the Kalina calculation a special procedure from EES called "NH3H2O procedure" is used to determine the thermodynamic properties of ammonia-water at the inlet and outlet states of every plant component.

5.4 Limitations of the brine outlet temperature

Final chemical data from the production wells in cluster-5 Lahendong and also from the main separator are not available, but we can estimate the quartz concentration based on the reservoir temperature using the equation (DiPippo, 2005):

$$Qc(t) = 41.598 + 0.23932 t_{res} - 0.011172 t_{res}^2 + 1.1713 \times 10^{-4} t_{res}^3 - 1.9708 \times 10^{-7} t_{res}^4$$

In the main separator, by assuming that silica remains in the liquid phase, the concentration of silica in the brine will increase according to:

$$S_I = \frac{Qc(t)}{1-x}$$

where t_{res} = Temperature of the reservoir in cluster-5 (250°C);

x = Quality of the geothermal fluid in the main separator (0.2)

From the calculation, S_I equals 579.4 ppm

For utilizing the brine by reflashing the brine in the flash vessel, the concentration of silica in the brine will increase according to:

$$S_{II} = \frac{S_I}{1 - x_1}$$

where x_1 = Quality of the geothermal fluid entering the flash vessel (second separator), the value of x is dependent on the flashing pressure;

S_I = Brine silica concentration from the main separator;

S_{II} = Brine silica concentration from the flash vessel.

Using the Fournier and Marshall correlation for amorphous silica solubility gives:

$$log_{10}s = -6.116 + 0.01625T - 1.758 \times 10^{-5}T^2 + 5.257 \times 10^{-9}T^3$$

where T = Brine outlet temperature (K);

s = Silica solubility, must be multiplied by 58,400 to obtain ppm.

The *SSI* index is the ratio of silica concentration in the brine and amorphous silica solubility, if *SSI* > 1, the separated brine is supersaturated with respect to amorphous silica (silica scaling occurs).

For a flash system:

$$SSI = \frac{S_{II}}{s}$$

For a binary and a Kalina cycle:

$$SSI = \frac{S_I}{s}$$

Table 3 shows *SSI* at different flashing pressures to determine the lowest limit of hot brine utilization. Using the equations above, the lowest brine outlet temperature that we can reach without a scaling problem is:

- For a flash system, 152.9°C corresponding to a flashing pressure of 5.15 bar;
- For binary and Kalina cycles, 146.7°C.

TABLE 3: *SSI* at different flashing pressures

P flashing [bar]	T brine outlet [C]	S_{II} [ppm]	s [ppm]	SSI
9	175.4	586.7	752.4	0.78
8	170.4	593	721.3	0.82
7	165	600	687.5	0.87
6	158.8	607.9	650.4	0.93
5	151.8	616.8	609	1.01
4	143.6	627.5	561.9	1.12
3	133.5	640.6	506.8	1.26
2	120.2	658.2	438.6	1.50
1	99.61	686	344.4	1.99

6. RESULTS AND DISCUSSION

From the model simulation with different parameter inputs, the optimum parameters can be found for obtaining the optimum power output and make an optimization of different plant configurations.

6.1 Optimization of the flash system

Changing the flashing and condensing pressures
The relationship between the flashing pressure and the power output gained is shown in Figure 14. By changing the separator pressure at a certain condensing pressure, it was found that by lowering the separator pressure more steam was produced and also the quality of the steam from the turbine so more power was obtained (as shown in Figure 14 for the condensing turbine, and Figure 15 for the back pressure turbine). Increasing the power output, the optimum flashing pressure value will be reached that yields a maximum value of 2.19 bar for the condensing turbine and 3.5 bar for the back pressure turbine. Below this flashing pressure, the power output will decrease, due to lower value of enthalpy drop in the turbine. In the condensing turbine, decreasing power outputs are also caused by increasing steam mass flowrate consumptions for the steam jet ejector. The steam mass flowrate needed for NCG removal will increase very rapidly at flashing pressures below 2.19 bar (Figure 14).
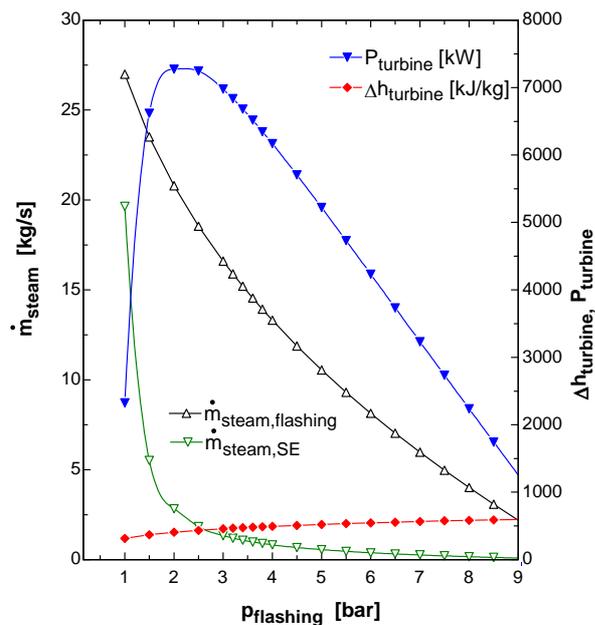

FIGURE 14: Steam mass flow rate from the flashing process and for the steam-jet ejector, enthalpy drop in the turbine and power output vs. flashing pressure in the condensing turbine
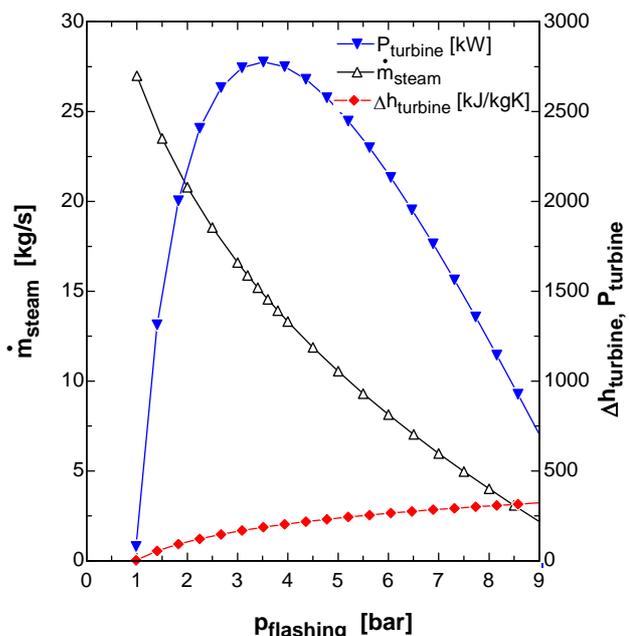
FIGURE 15: Steam mass flow rate, enthalpy drop in the turbine and power output vs. flashing pressure in the back pressure turbine

For the condensing turbine, the optimum flashing pressure of 2.19 bar produced 6768 kW, but for the back pressure turbine, the optimum flashing pressure of 3.5 bar produced 2776 kW. Lowering the condenser pressure at a certain flashing pressure tends to increase power output from the system, as shown in Figure 16. In the condensing turbine, the condensing pressure minimum is limited to 0.08 bar due to the conditions of the available cold sink.Decreasing flashing pressure will decrease the brine outlet temperature. It will limit the maximum power output by lowering the flashing pressure because of an increasing possibility of silica scaling with a decreasing brine outlet temperature. Figure 17 shows increasing *SSI* (Silica Scaling Index) with decreasing flashing pressure.
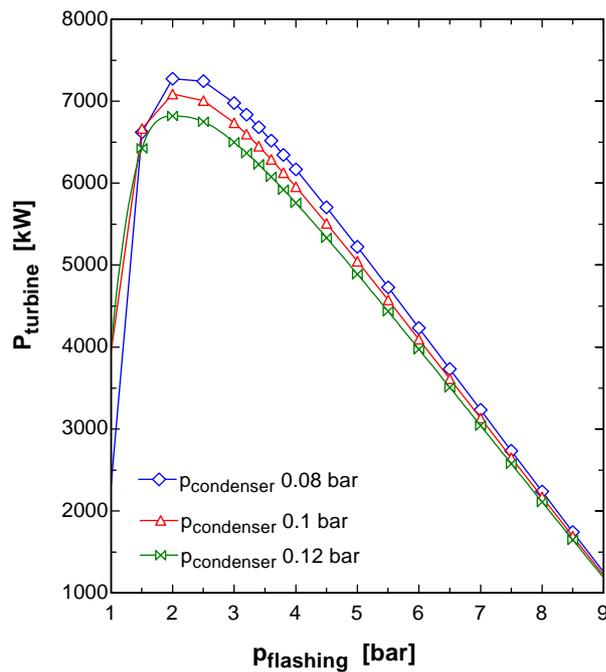
FIGURE 16: Power output at different
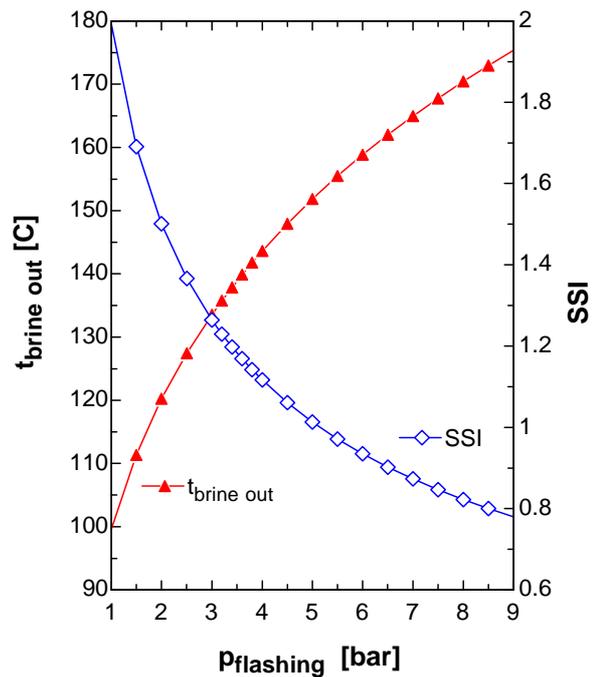condensing pressure and brine outlets

FIGURE 17: Brine outlet temperature and
SSI vs. flashing pressure

6.2 Optimization of the binary cycle

Choosing the suitable working fluid for the binary cycle
Figure 18 shows a comparison of power outputs using different possible working fluids (isopentane
and n-pentane). Isopentane will produce more power at the same evaporator pressure (turbine inlet
pressure). That also means, that for the same
pressure inlet turbine more power is produced.
Therefore, in further calculations in this paper
isopentane is used as a working fluid.

Adding the regenerator
A regeneration heat exchanger is added between
the organic system turbine and the condenser
(Figure 19). Since the organic fluid has a
retrograde dew point - or saturation curve -
organic vapour tends to superheat when steam is
expanded through the turbine. The regenerator
is used to recover the superheated steam for the
preheating of the organic fluid prior to further
heating in the preheater. The effects of the
regeneration are shown in Figure 20. By
maximizing the regeneration, more power is
produced for the same temperature extraction of
the brine. But a cycle without regeneration will
have more power at the lower temperature of the
return brine. Figure 20 shows the maximum
power output for the binary cycle with a
maximum regeneration of 9176 kW by
extracting the temperature of the brine in the
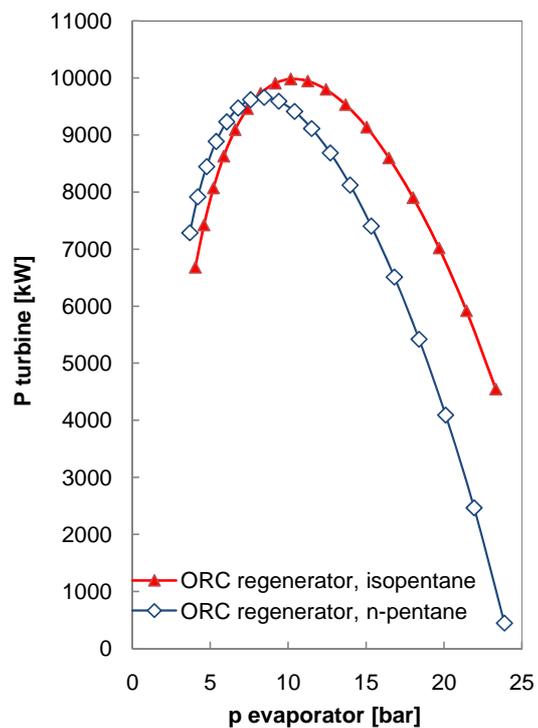evaporator at 88.64°C. The maximum power

FIGURE 18: Power output of a binary cycle
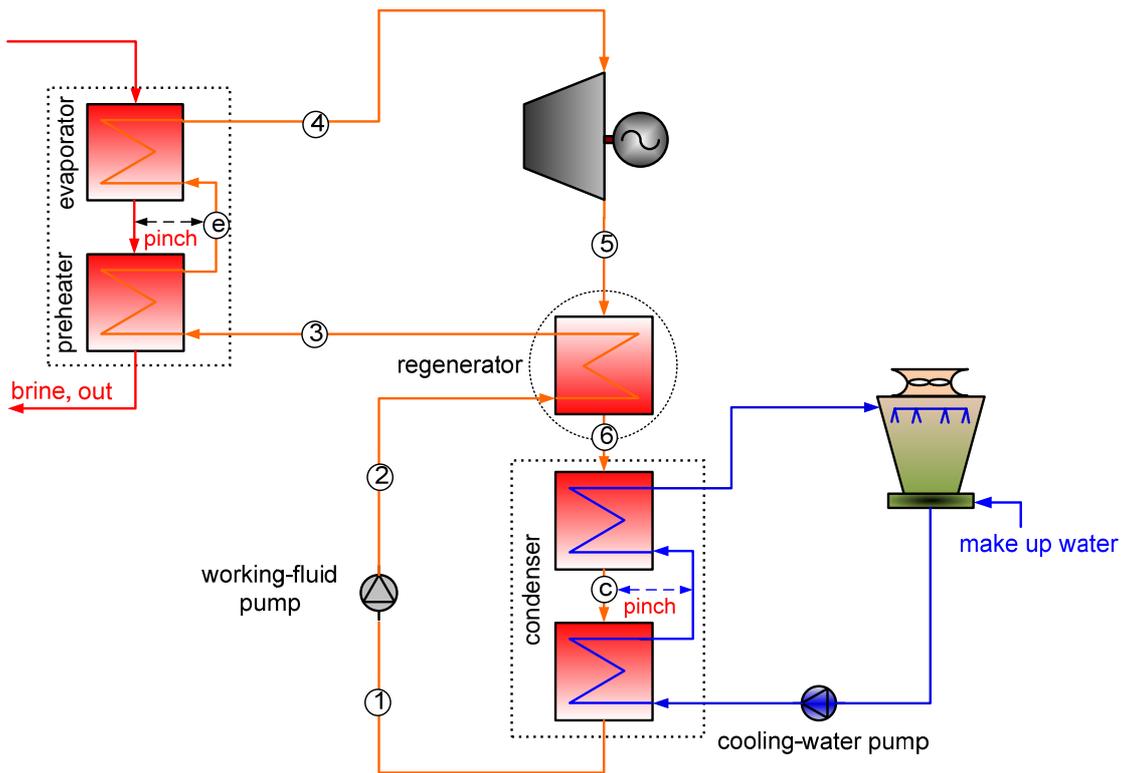using isopentane and n-pentane

FIGURE 19: A schematic diagram for a binary cycle with regeneration

output for the binary cycle without regeneration is 9299 kW when extracting the temperature of the brine in the evaporator at 76.84°C.

Changing the evaporation temperature of the working fluid

If the evaporation temperature decreases, the boiler pressure decreases automatically because the evaporation of the working fluid is at the same temperature. The working fluid exits the evaporator as a saturated mixture when the saturation temperature corresponds to the pressure inside the evaporator. The evaporating line will move down and the heat balance (ratio Q_evap/Q_in) will increase (Figure 21). This would increase the mass flowrate of the working fluid but also lower the enthalpy drop in the turbine. The maximum generating power is balanced between the increasing mass flowrate and the decreasing enthalpy drop in the turbine. Figure 22 shows that the optimal evaporation temperature for the binary cycle with regeneration and pinch 5°C in the evaporator

FIGURE 20: Power output of a binary cycle with and without regeneration vs. brine outlet temperature

is 118.1°C, along with an evaporator pressure of 10.45 bar (Figure 23); the maximum power output based on that parameter is 9167 kW.
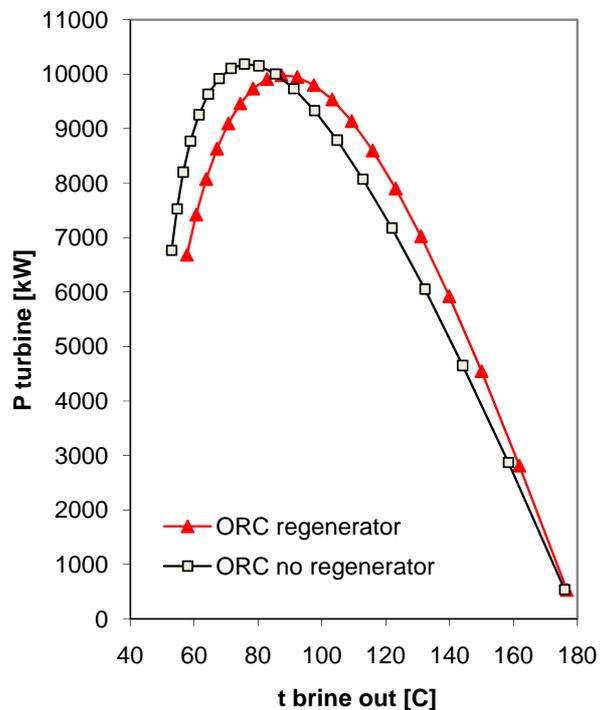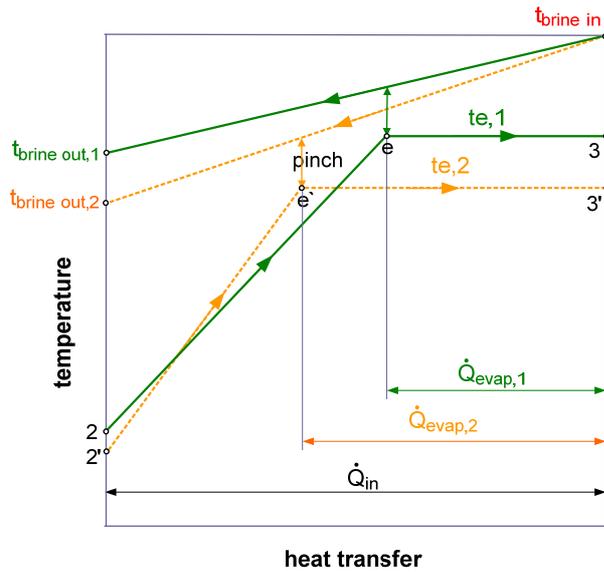
FIGURE 21: T-q diagram for different evaporation temperatures

Decreasing the evaporation temperature with the same pinch in the heat exchanger will also lower the brine outlet temperature (maximizing the utilization of the brine temperature). The exergy efficiency shows that utilization efficiency also increases. Figure 24 shows that the maximum exergy efficiency for the binary cycle is 0.4 at a brine outlet temperature of 88.64°C.

The increased power output with increased utilization of the brine will be limited by the chemical composition of the outlet brine because of the increased possibility of silica scaling with a decreasing brine outlet temperature. Figure 25 shows increasing power output which is followed by increasing SSI.
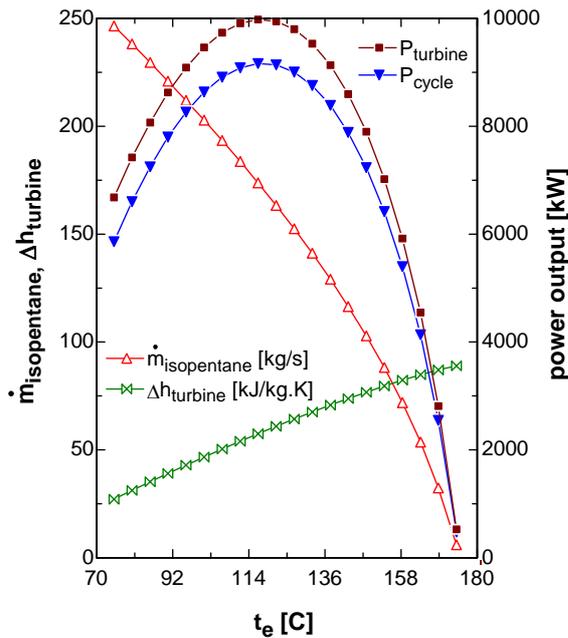

FIGURE 22: Mass flow rate of isopentane, enthalpy drop in the turbine and power output vs. evaporation temperature
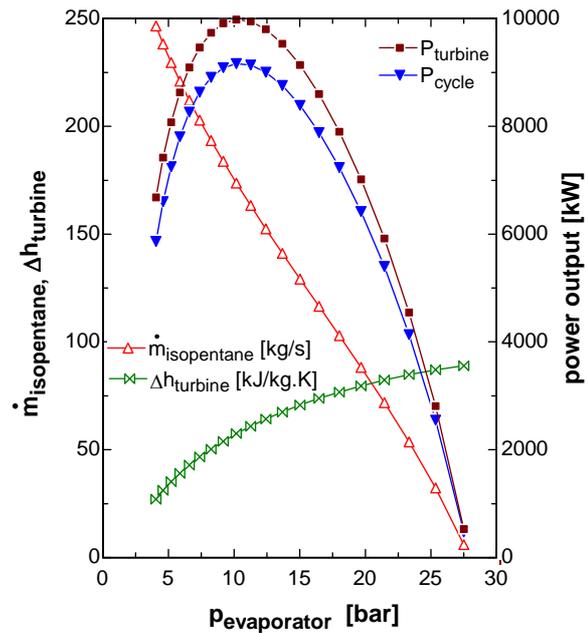

FIGURE 23: Mass flow rate of isopentane, enthalpy drop in the turbine and power output vs. evaporation pressure

Effect of the pinch
By decreasing the pinch while maintaining constant evaporation temperature (Figure 26), the mass flowrate of the working fluid will increase and the power produced by the turbine will also increase (Figures 27 and 28). But it must also be considered that decreasing pinch results in rapidly increasing the required heat exchanger area, as shown in Figure 27. Therefore, in the following calculations in this paper, pinch equal to 5 degrees is used in the binary cycle's heat exchanger.
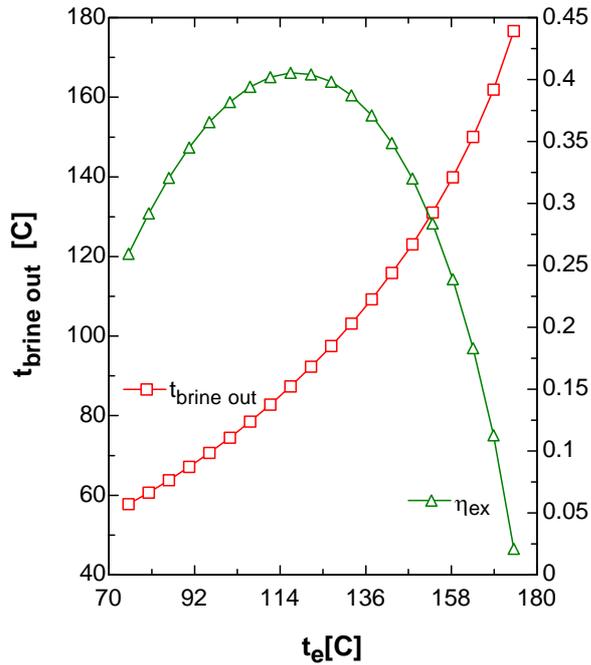
FIGURE 24: Brine outlet temperature and exergy efficiency vs. evaporation temperature
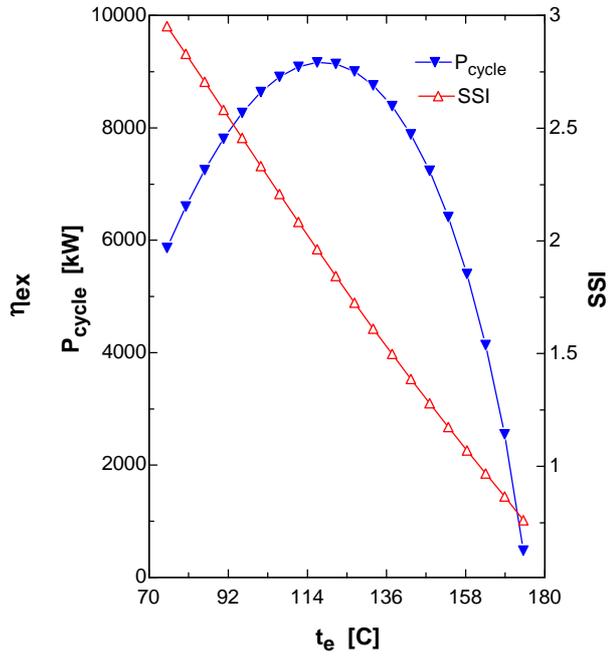

FIGURE 25: Power output and *SSI* vs. evaporation temperature
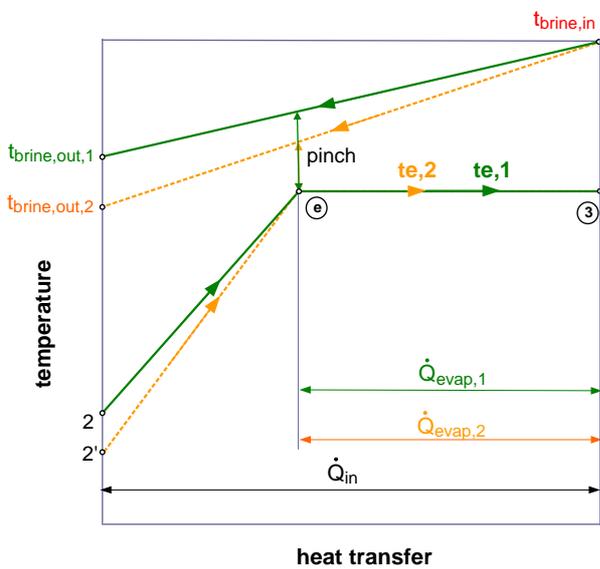
FIGURE 26: T-q diagram for different pinches in the evaporator
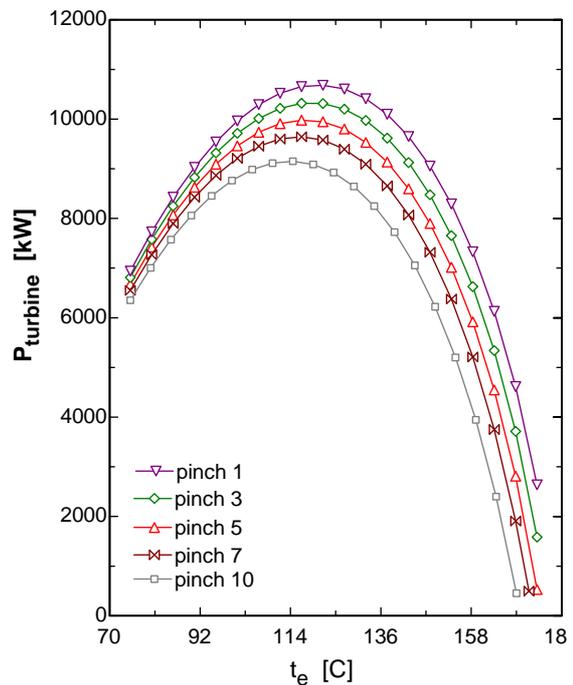
FIGURE 27: Power output and heat exchanger area ratio vs. pinch in the evaporator

6.3 Optimization of the Kalina cycle

Changing the evaporator pressure and the composition of the mixture
The effects of evaporator pressure and the composition of the mixture can be seen in Figure 29. Figure 30 shows that increasing the mixture at the same pressure will increase the power output but also lower the temperature of the brine outlet temperature (increasing the utilization of the brine).

FIGURE 28: Power output at different pinches
vs. evaporation temperature

6.4 Power output for different plant configurations

By simulation of different plant configuration types, we can get the optimum parameters and maximum power outputs (seen in Table 4 and Figure 31).

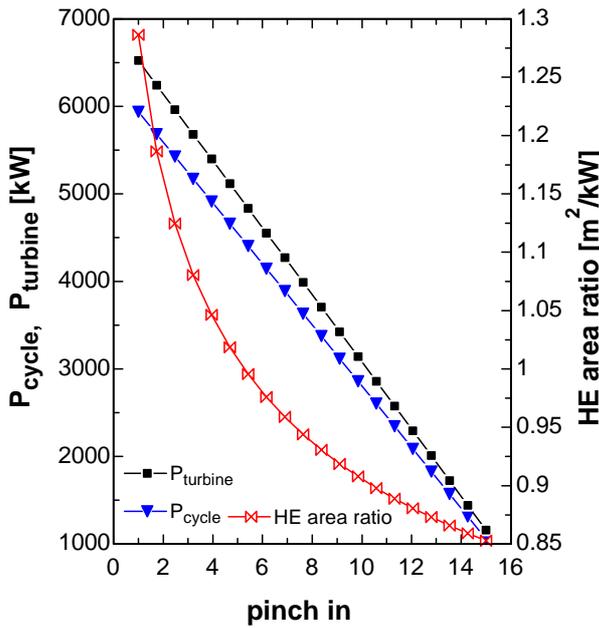Actual maximum power generated will depend on how much the brine outlet temperature can be lowered, and the chemical problems of the rejected brine dealt with. Figure 31 shows that the maximum power generated can be read by a straight line connecting the brine outlet temperature and the power output. If the brine outlet temperature is limited to 127.4°C, flashing by condensing would be the best option, due to higher power output at the same brine outlet temperature. It is followed by a binary cycle with regeneration, a binary cycle without regeneration and finally flashing with atmospheric exhaust. Below 127.4°C, a binary cycle with regeneration produced more power than the others. The Kalina cycle offers higher power output at the rejected brine temperature interval between 118.7 and 111.8°C at very high pressure (60 bar).
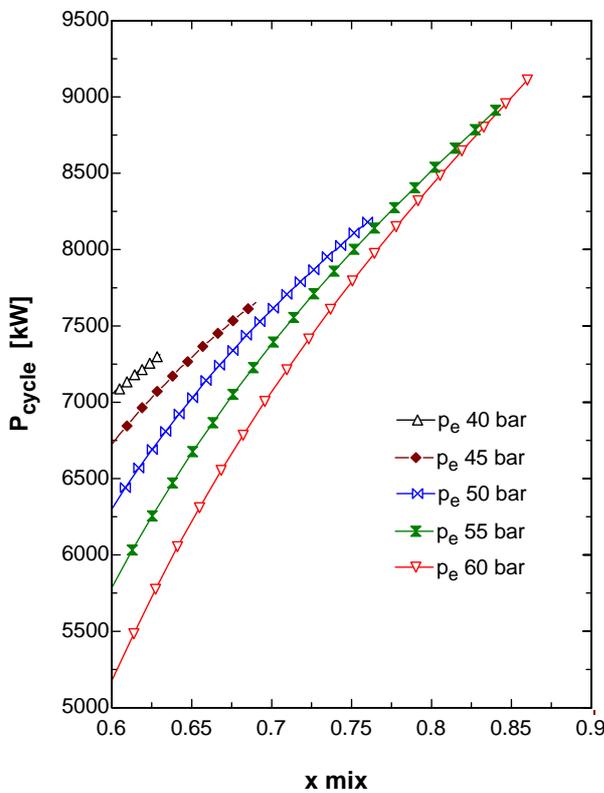

FIGURE 29: Power output at different
evaporator pressures vs. basic mixture
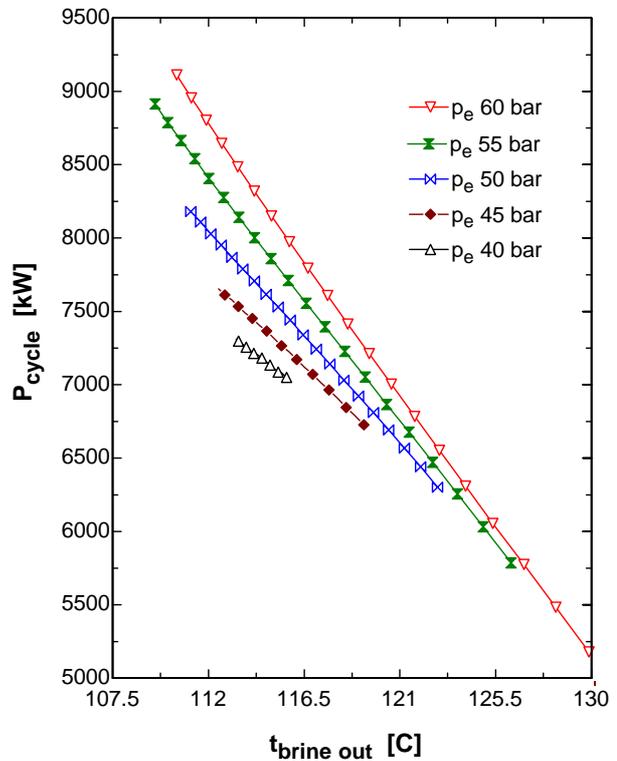composition of ammonia-water

FIGURE 30: Power output at different
evaporator pressures vs. brine outlet
temperatures

Table 4 shows, that a *SSI* value greater than 1 at the maximum power output, indicates a potential for silica scaling. Although the limitation of the temperature for a binary cycle is lower than for a flash system, the power output for a flash system with condensing is higher than a binary cycle. The actual maximum power generation outputs based on the temperature limitation are summarized in Table 5 below.
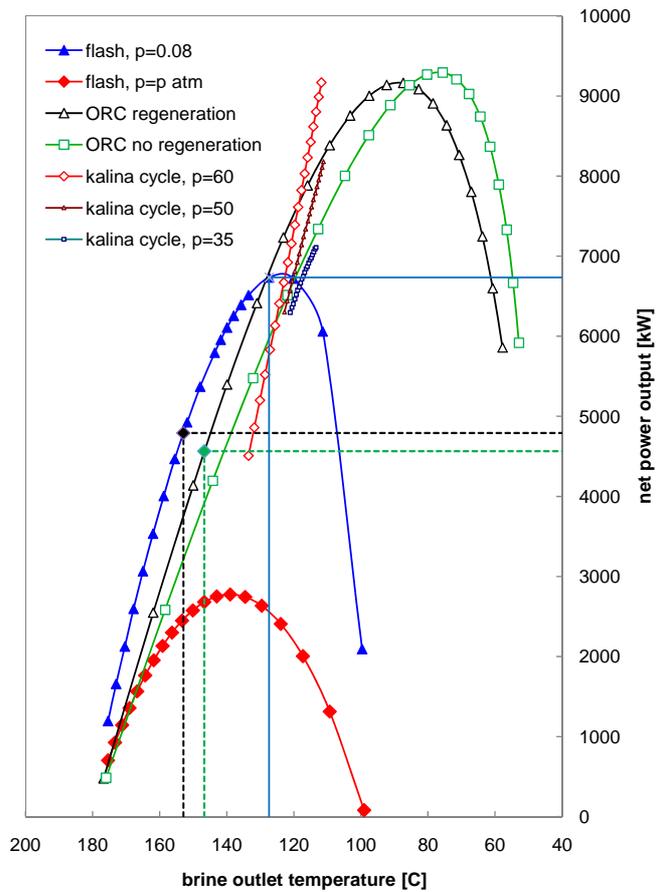
FIGURE 31: Power output for different power plant configurations vs. brine outlet temperature

TABLE 4: Comparison of optimized parameters and power outputs for different plant configurations

Plant configuration	Optimum parameter	Brine outlet temperat. [°C]	Power output		SSI
			P turbine	P net	
			[kW]		
Flash – atm. exhaust turbine	p flashing [bar] = 3.5	139	2776	2776	1.183
Flash –condensing turbine	p flashing [bar] = 2.19 p condenser [bar] = 0.08	123.1	7305	6768	1.445
Binary cycle without regenerator	p evaporator [bar] = 10.45	76.84	10184	9299	2.257
Binary cycle with regenerator	p evaporator [bar] = 10.45	88.64	9982	9167	1.931
Kalina cycle	p evaporator [bar] = 60 ammonia-water ratio (%) = 86	110.5	9874	9111	1.476

TABLE 5: Maximum power outputs for different plant configurations,
if temperature limitation due to silica scaling is taken into account

Plant configuration	Optimum pressure [bar]		Brine outlet temperature [°C]	Power output	
				P turbine	P net
				[kW]	
Flash – atm. exhaust turbine	p flashing	= 5.148	152.9	2465	2465
Flash – condensing turbine	p flashing	= 5.148	152.9	5077	4791
Binary cycle without regenerator	p evaporator	= 23.71	146.7	4341	3922
Binary cycle with regenerator	p evaporator	= 22.72	146.7	5013	4565

7. CONCLUSIONS AND RECOMMENDATIONS

Thermodynamic modelling is very helpful for finding optimal parameters and maximum power outputs from rejected brine. In the model a sensitive analyses can be made by changing one parameter and seeing its effect on the whole system.

Maximum power generation from waste brine utilization depends on how the brine outlet temperature is lowered. In a field that has relatively low silica concentrations, common in low-temperature geothermal fields, indirect systems (binary and Kalina) are more applicable. Temperature limitations due to scaling are lower, therefore it is possible to obtain more power using those cycles.

In the range of brine outlet temperatures down to 127.4°C, flashing with condensing will produce more power. Below 127.4°C, a binary cycle with regeneration produces more power than others. A Kalina cycle offers a higher power output on the interval of rejected brine temperatures between 118.7 and 111.8°C at very high pressure (60 bar).

Actual maximum power generated for utilizing the waste brine in a high-temperature geothermal field like the Lahendong field, will be limited by the lowest temperature that can be reached without encountering problems related to the chemical composition of the rejected brine. Based on a brine outlet lowest temperature of 152.9°C for flashing, the maximum output power from a flash system is 4791 kW; for a binary cycle with a lowest outlet temperature of 146.7°C, the maximum output power is 4565 kW.

The selection of a power plant is very site dependent. Based on a thermodynamic view of fluid conditions limited by a brine outlet temperature, as is the case in Lahendong, flashing with condensing turbine is more suitable than other configurations because it produces more power. The binary or Kalina cycle would probably be more costly. The situation would be different if we used chemical treatments to improve the chemistry of the rejected brine.

The model of the power plant in this paper is based on limited data. The model can be improved and economical analyses made if more data and information is obtained. The final chemical composition of the brine will correct the power output. The latest technology makes it possible to decrease temperature limitations by using chemical treatments to improve the chemistry of the rejected brine but an economical analysis will be needed to compare the benefits of the addition power output with the additional costs of the treatment.

the course. My sincere thanks also go to my supervisors, Dr. Halldór Pálsson and Dr. Gudrún Saevarsdóttir for their guidance and advice throughout the project. Special thanks go to all the lecturers for their comprehensive presentations and willingness to share their knowledge and experience. I would also like to give thanks to the management of PT. Pertamina Geothermal Energy for permission to attend this programme. Finally, my deepest thanks go to my family and friends for their moral and emotional support during these six months.

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APPENDIX I: Thermodynamic models of waste brine utilization using different plant configurations, including flash with atmospheric exhaust, flash with condensing turbine, binary plant with and without regeneration and the Kalina cycle
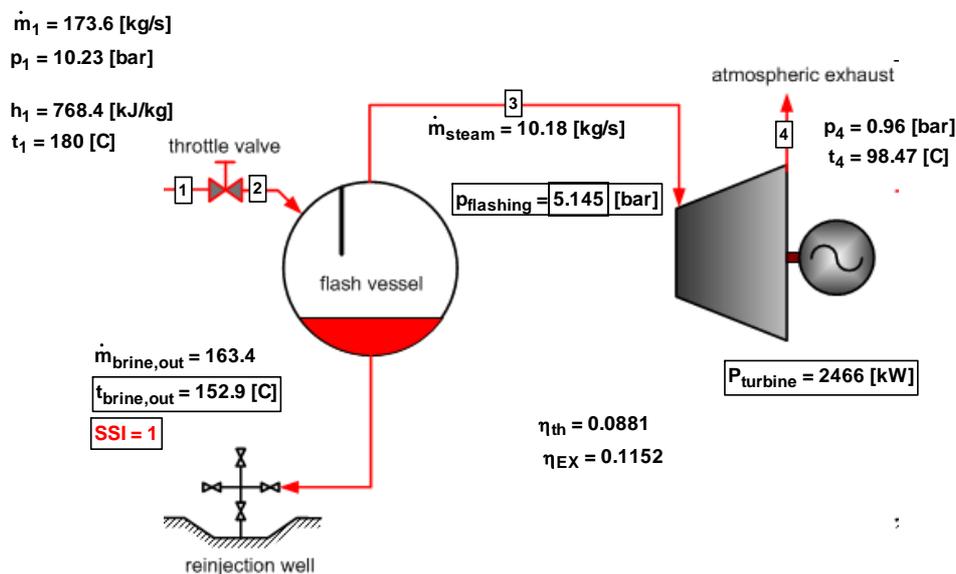

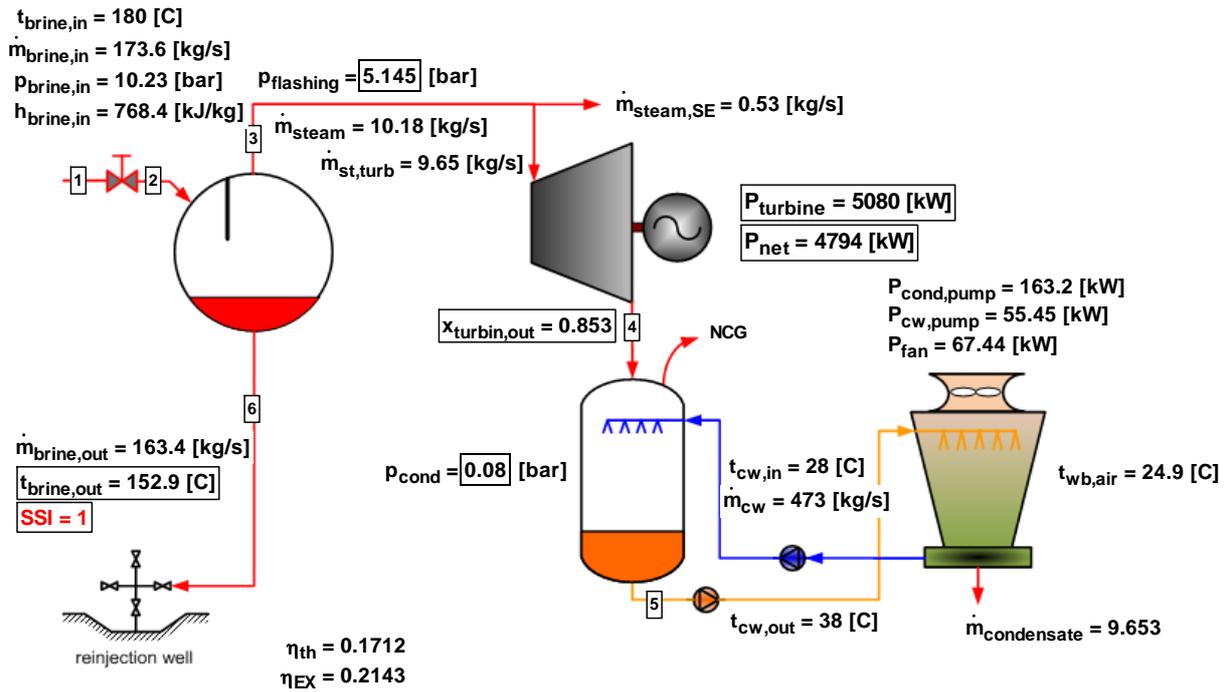FIGURE 1: Flash system with atmospheric exhaust turbine

$t_{brine,in}$ = 180 [C]
$\dot{m}_{brine,in}$ = 173.6 [kg/s]
$p_{brine,in}$ = 10.23 [bar] $p_{flashing}$ = 5.145 [bar]
$h_{brine,in}$ = 768.4 [kJ/kg] $\dot{m}_{steam,SE}$ = 0.53 [kg/s]

$\dot{m}_{steam}$ = 10.18 [kg/s]

3      $\dot{m}_{st,turb}$ = 9.65 [kg/s]

1 2 $P_{turbine}$ = 5080 [kW]

P_{net} = 4794 [kW]

$P_{cond,pump}$ = 163.2 [kW]
$P_{cw,pump}$ = 55.45 [kW]
P_{fan} = 67.44 [kW]

$x_{turbin,out}$ = 0.853   4

NCG

6

$\dot{m}_{brine,out}$ = 163.4 [kg/s] p_{cond} = 0.08 [bar] $t_{cw,in}$ = 28 [C]
$t_{brine,out}$ = 152.9 [C] $\dot{m}_{cw}$ = 473 [kg/s] $t_{wb,air}$ = 24.9 [C]

SSI = 1

5

$t_{cw,out}$ = 38 [C] $\dot{m}_{condensate}$ = 9.653

reinjection well

η_{th} = 0.1712
η_{EX} = 0.2143

FIGURE 2: Flash system with condensing turbine

$\dot{m}_{brine}$ = 173.6 [kg/s] $\dot{m}_{isopentane}$ = 49.75 [kg/s]
$t_{brine,in}$ = 180 [C]

$P_{turbine}$ = 4336 [kW]

3   t_3 = 164.6 [C] P_{cycle} = 3918 [kW]

$P_{pump,cw}$ = 101 [kW]
P_{fan} = 71.74 [kW]
P_{pump} = 245.7 [kW]

p_e = 23.71 [bar]

t_e = 164.6 [C] $t_{pp,in}$ $pinch_{in}$ = 5

4   t_4 = 88.76 [C]

$t_{cw,out}$ = 38 [C] $t_{wb,air}$ = 24.86 [C]

$\dot{m}_{make,up}$ = 9.69 [kg/s]

t_2 = 42.24 [C]
p_2 = 23.71 [bar]

$t_{brine,out}$ = 146.7 [C]

SSI = 1 t_c $t_{pp,out}$ TTD = 2.85
$pinch_{out}$ = 5

η_{ex} = 0.1733 t_1 = 40.85 [C]
η_{th} = 0.1558   1 p_1 = 1.554 [bar]
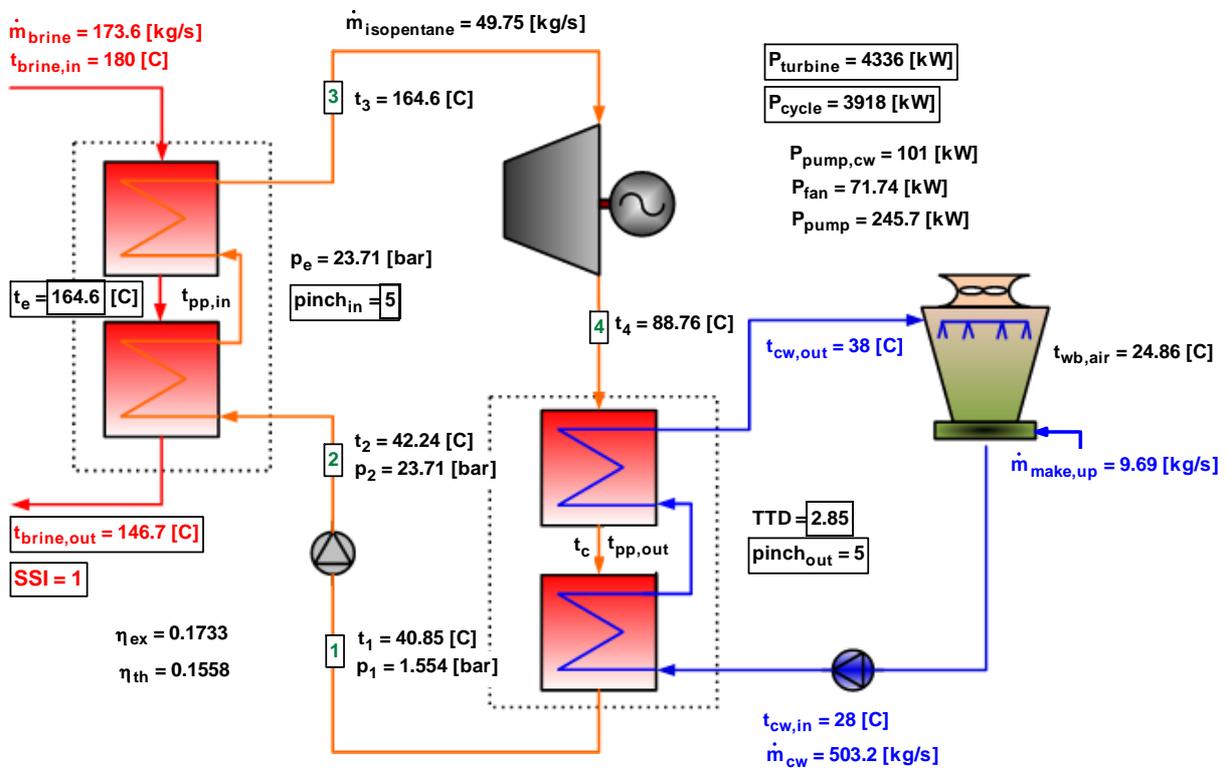
$t_{cw,in}$ = 28 [C]
$\dot{m}_{cw}$ = 503.2 [kg/s]

FIGURE 3: Binary cycle without regeneration

FIGURE 4: Binary cycle with regeneration

FIGURE 5: Kalina cycle