THE EFFECT OF HEAT EXCHANGER DESIGN ON THE RETURN ON INVESTMENT OF A GEOTHERMAL POWER PLANT

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ABSTRACT

In this study, a model of a binary geothermal power plant has been developed in Aspen software and validated with the real data from Chena binary geothermal power plant. The validated model is used to investigate the effect of heat exchanger design on Return on Investment (ROI) of the plant. The analyses include type selection and sizing of heat exchangers as well as the possibility of using a recuperator in the system.

The choice of heat exchanger type was found to significantly influence the ROI of the plant. The base case studied here uses shell and tube (S&T) type. The highest ROI of the plant was obtained with plate (PL) type for both heat exchangers where the ROI increases from 0.737 to 1.107.

The possible sizes of heat exchanger design have been analyzed. The existing heat exchangers were found to already be sized to achieve an optimal ROI of 0.737 with net design power output at 210 kW. Reducing heat exchanger size increases the ROI by only about 2.59%, while reducing the net power output from 210kW to 203.9 kW. The design with maximum size of heat exchangers increases the net power output to 220.3 kW, but the ROI drops significantly by about 17%.

Investigation into the addition of a recuperator to the system indicates that it could increase the net power output by about 1 kW. The system with a recuperator was found to have a superior thermal efficiency of 8.41%, but a low ROI of 0.409.

1. INTRODUCTION

In recent years, the Organic Rankine Cycle (ORC) system has been proposed as an efficient technology for converting low and medium temperature heat sources to electricity. There are some benefits to using an ORC when compared with conventional steam power cycles, including efficient utilization of low energy temperature resources, smaller systems and better economic performance Yamamoto, Furuhata [1].

Geothermal heat energy is a renewable heat energy from underneath the earth's surface with temperatures varying from 50 to 350^{0}C [2]. Recently, some geothermal wells characterized by low temperature liquid, that were not considered in the past for energy generation, have begun to be utilized for power generation. The range of low and medium geothermal source temperatures are 70 - 100^{0}C and 100 - 150^{0}C, respectively.

The application of ORC technology is considered technically and economically feasible. The technology utilizes an organic fluid that has the higher molecular weight, lower evaporation heat, positive slope of the saturated vapor curve in the T-s diagram and lower critical and boiling temperatures when compared to steam. These features make the ORC technology very attractive for implementation of low and medium temperature sources such as solar energy, geothermal energy, biomass products and waste heat.

The selection of the appropriate organic fluid is the first step to designing ORC system that closely matches the heat resources. Modelling of steady-state ORC system performance has been reported in the scientific literature with the majority of studies investigating working fluid selection especially for specific applications [3-12]. Masheiti, Agnew [13] evaluated and compared two working fluids: R-134a and R-245fa for implementation in a low temperature energy source at 73^{0}C coupled with a cooling water supply at 25^{0}C. Saleh, Koglbauer [11] and Kuo, Hsu [6] performed analyses of different kind of working fluids and proposed solutions for fluid screening. Saleh, Koglbauer indicated that 31 pure component working fluids are suitable for ORC systems. In their analysis, the critical temperature, normal boiling temperature, and critical pressure for the working fluids were arranged in specific order to give an indication of their suitability. Huo, Hsu proposed a dimensionless parameter combining the Jacob number, condensing temperature and evaporating temperature as very effective tool for fluid selection.

A limited number of researchers have reported their experience with performance optimization of the ORC plant design by focusing on heat exchanger design. This approach can be used to simplify the search for a cost effective design, as heat exchanger cost contributes largely to the total ORC power plant cost especially for low temperature geothermal plants. Calise, Capuozzo [14] investigated the geometrical features of shell and tube exchangers by determining the best of design parameters and examining different working fluids on ORC plant performance. They studied the performance of heat exchangers in off design conditions of the system by varying heat source temperature from 120 ^{0}C to 300 ^{0}C. Madhawa Hettiarachchi, Golubovic [2] performed optimization of a cost-effective optimum design criterion of heat exchangers for a low temperature of geothermal plant. They used a ratio of total heat transfer area to total net power as the objective function with the steepest descent method. Evaporation and condensation temperatures, velocity of geothermal water in the evaporator and cooling water velocity of condenser are used as varying parameters in order to get the minimum of objective function. Shengjun, Huaixin [15] introduced an optimization procedure with a simulation program written in Matlab using five parameters: thermal efficiency, exergy efficiency, recovery efficiency, heat exchanger area per unit power output (APR) and levelized energy cost (LEC). They also considered the advantages of the two cycle types of ORC: subcritical and transcritical and used shell-tube heat exchangers in their analysis. Quoilin et.al have done a lot of research on ORCs especially in designing and building small ORC plants. Quoilin, Orosz [16] designed a small solar organic rankine cycle with net capacity of 3 kWe installed in Lesotho for rural electrification purposes. Using models, they compared and analyzed the sizing of different components in an ORC cycle and evaluated the overall

NOMENCLATURE

ACC	Air Cooled Condenser
EDR	Exchanger Design & Rating
EOS	Equation of state
F	Correction factor
LMTD	Log-mean temperature difference method
N_p	The annual net profit
ORC	Organic rankine cycle
PL	Plate heat exchanger
Q	Heat
ROI	Return on investment expressed as a fraction per year
S&T	Shell and tube heat exchanger
T	The total capital investment
Temp	Temperature
W	Work
ΔT	The temperature difference at an end of the exchanger

Greek symbols

η	Efficiency

Subscripts

1,2,3,4	State points
In	Input
Out	Output

system. Another paper by Quoilin, Lemort [17] developed a numerical model and carried out an experimental study in their prototype of ORC system with HCFC-123 as the working fluid. The models were used to investigate potential improvements on the system. Both papers from Quoilin et.al used the plate heat exchangers in their ORC system design.

Most of the studies on ORC design above focused on selection of working fluids and parametric analysis for ORC plant optimizations. Although some of them discussed optimization of the heat exchanger design, none of the studies discussed about the effect of the heat exchanger design on ROI of ORC geothermal power generation plant while considering the selection of heat exchanger types.

The objective of this study is to develop models of a geothermal power plant by using Aspen Process Simulation Software. The models are validated with real data from the Chena binary geothermal power plant and they are used to investigate the effect of heat exchanger design on ROI. The type and sizing of heat exchangers are considered as well as analysis of the inclusion of a recuperator. The results are intended to be used as a useful reference for component selection during geothermal power plant design.

2. THE GEOTHERMAL POWER PLANT

2.1 Process diagram of ORC

The ORC system consists of an expander, condenser, pump and vaporizer. Figure 1 shows how these components are connected in a process flow diagram. The cycle process is started by expanding vapor in a turbine connected a generator to generate electricity. The low pressure organic working fluid vapor is condensed and heat is released into the environment. The liquid fluid of condensation result is pressurized by a pump and vaporized by vaporizer to complete the cycle and the whole process restarts again. Q_{41} and Q_{23} are input and output heats to the system, respectively. W_{12} and W_{34} are works of expander and pump, respectively.

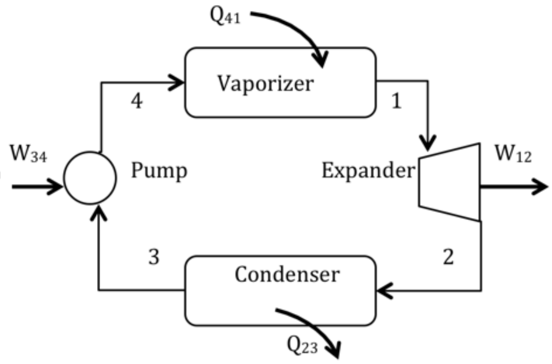


Figure 1: Process flow diagram of ORC system

2.2 The Chena geothermal power plant

The geothermal source is located in Chena in Alaska, USA approximately 96.6 kilometers east-northeast of Fairbanks, at an elevation of 367 meters. The plant has been run at the following operating conditions [18]:

- Water design points
  Heat source: $Temp_{in}$ = 73.33^0 C $Temp_{out}$ = 54.44^0 C mass flow rate = 12.17 kg/s
  Heat sink: $Temp_{in}$ = 4.44^0 C $Temp_{out}$ = 10^0 C mass flow rate = 101.68 kg/s
- Refrigerant design points

Mass flow rate:	12.16 kg/s
Evaporator/turbine inlet pressure:	16 bar
Condenser/turbine exit pressure:	4.38 bar
Turbine gross power:	250 kW
Pump power:	40 kW
Net power output:	210 kW
Vaporizer heat transfer rate:	2580 kW
Condenser heat transfer rate:	2360 kW

The working fluid used in the real plant is R134a. Figure 2 shows a representation of thermodynamic cycle of the system under investigation in the pressure-enthalpy diagram.

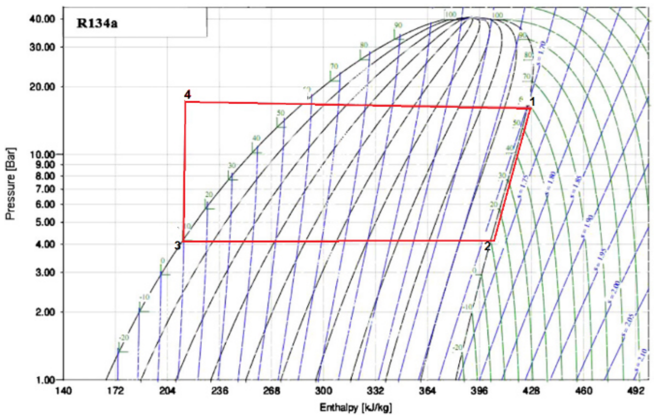


Figure 2: Pressure–enthalpy diagram of thermodynamic cycle (R134A)

3. MODELLING USING ASPEN

The simulation models of the geothermal ORC power plant have been developed by an integration between aspen plus process modelling and Aspen Exchanger Design and Rating (EDR). The Aspen Plus is used to simulate overall system while heat exchangers are modelled by Aspen EDR.

The typical simplifying assumptions of steady-state models are used in these numerical models:

- Thermodynamic equilibrium at inlet and outlet sections of each component
- Negligibility of kinetic and gravitational terms in energy balances
- Negligibility of heat losses toward the environment in each component such as expander, condenser, pump and vaporizer
- One-dimensional flow

3.1 Primary Equations

The models apply mass and energy balances to each of the four ORC cycle components mentioned in section 2.1 and use the primary equations of energy balance and isentropic/transfer efficiency listed in the table 1. The heat exchanger calculations are based on the Log-mean temperature difference (LMTD) method. The pump and expander are modelled from a thermodynamic point of view. They are considered adiabatic and their isentropic efficiencies are calculated using isentropic efficiency equations in table 1. It is assumed that isentropic efficiencies of the pump and expander remain constant at the values found in table 2.

Table 1: The equations of energy balance and efficiency

ORC Component	Energy Balance	Isentropic or Transfer Efficiency
Expander	$W_{12} = \dot{m}h_1 - h_2)$	$\eta_{12} = \frac{h_1 - h_2}{h_1 - h_{2s}}$
Condenser	$Q_{23} = \dot{m}(h_2 - h_3)$	$Q_{23} = U_c A_c F \Delta T_{lm23}$
Pump	$W_{34} = \dot{m}(h_4 - h_3)$	$\eta_{34} = \frac{h_3 - h_{4s}}{h_3 - h_4}$
Vaporizer	$Q_{41} = \dot{m}(h_4 - h_1)$	$Q_{34} = U_v A_v F \Delta T_{lm41}$

Here a correction factor (F) for multiple tube-side and/or shell side passes, derived through the work of Nagle (1933) and Underwood (1934), can be calculated as [19]:

$$F = \frac{\sqrt{R^2+1}ln[(1-S)/(1-RS)]}{(R-1)Ln\frac{2-S\left(R+1-\sqrt{R^2+1}\right)}{2-S\left(R+1+\sqrt{R^2+1}\right)}} \tag{1}$$

where

$$R = \frac{T_{in}^{hot} - T_{out}^{hot}}{T_{out}^{cold} - T_{in}^{cold}} \tag{2}$$

$$S = \frac{T_{out}^{cold} - T_{in}^{cold}}{T_{in}^{hot} - T_{in}^{cold}} \tag{3}$$

ΔT_{lm} can be calculated as

$$\Delta T_{lm} = \frac{\Delta T_1 - \Delta T_2}{Ln\left(\Delta T_1 / \Delta T_2\right)} \tag{4}$$

The ROI is defined as the ratio of profit to investment [20]. This can be expressed as

$$ROI = \frac{N_p}{T} \tag{5}$$

The term T makes reference to the capital cost of power plant and heat exchanger cost is assumed as the capital cost. Aspen EDR generates the costing calculation once all the geometry of each component part of the heat exchanger has been calculated. The cost is calculated according to the three elements of the exchanger cost: the material cost, the labor cost and the mark-ups on material and labor. The default cost database included in Aspen has been used in the analysis [21].

The thermal efficiency of the ORC is defined on the basis of the first law of thermodynamics as the ratio of the net power output to the heat addition referring to figure 1.

$$\eta_{th} = \frac{W_{12} - W_{34}}{Q_{41}} \tag{6}$$

3.2 Property Methods

The accuracy of the model results depends strongly on a suitable prediction of the working fluid's thermodynamic properties. The cubic Peng-Robinson equation of state (EOS) has been adopted to calculate the thermodynamic and thermo physical characteristics of R134 working fluid and geothermal brine. The geothermal brine has been assumed equal to thermodynamic and thermo physical characteristic of pure water. The validity of this EOS for simulation has been confirmed by comparing the data obtained from software with those available from the website of NIST [22]. For every case that has been compared, the errors concerning the most important thermodynamic and thermo physical data between the simulated and the actual data resulted lower than 2%.

4. RESULTS AND DISCUSSION

4.1 Model validation

The simulation results were validated with the real operational data from Chena Geothermal power plant that has been discussed in section 2.2. Table 2 shows that the comparison between both data is very good with a maximum deviation of 3.56%. Thus, this model can be used to represent the real plant. In addition, figure. 3 shows the model of the plant at the nominal design points.

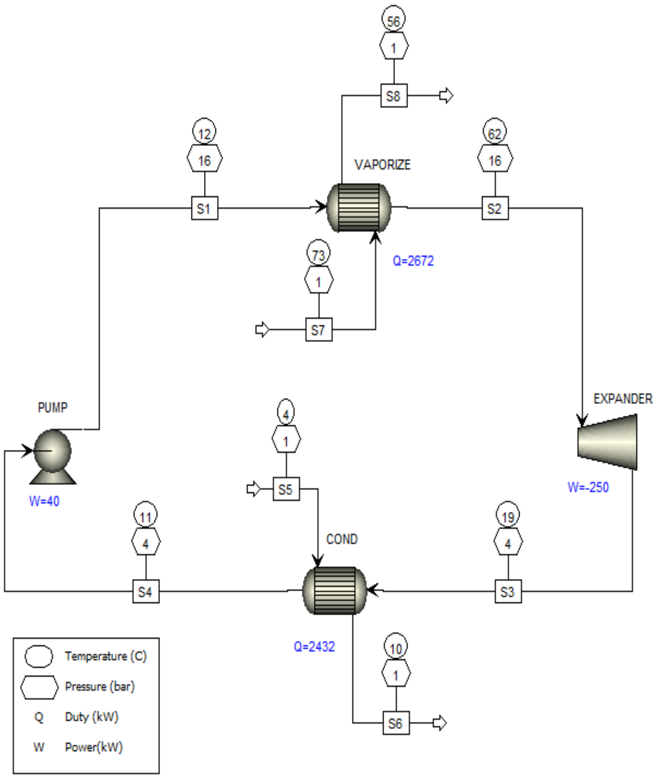


Figure 3: Aspen model of the Chena Power Plant at the real plant design condition

Table 2: Validation of the numerical model with real plant design data

Parameters	Real power plant data	Simulation Result	% Relative error (│ΔX│ *100)/X
Geothermal fluid masss flowrate [kg/s]	33.39	33.39 [a]	0.00
Geothermal fluid temperature [^{0}C]	73.33	73.33 [a]	0.00
Cooling water mass flowrate [kg/s]	101.68	101.68 [a]	0.00
Cooling water source temperature [^{0}C]	4.44	4.44 [a]	0.00
Working fluid type	R 134	R 134 [a]	0.00
Expander effciency	0.80	0.8 [a]	0.00
Expander mechanical efficiency	-	0.958 [b]	0.00
Expander inlet pressure [bar]	16.00	16.00 [a]	0.06
Expander outlet pressure [bar]	4.39	4.39 [a]	0.00
Gross power output [kW]	250.00	250 [b]	0.00
Pump Power [kW]	40.00	40.00 [b]	0.00
Pump efficiency	-	0.56 [b]	0.00
Driver efficiency	-	0.51 [b]	0.00
Geothermal exit temperature [^{0}C]	54.44	55.70 [b]	0.48
Cooling water exit temperature [^{0}C]	10.00	9.70 [b]	3.00
Working fluid mass flowrate [kg/s]	12.17	12.17 [a]	0.00
Net plant power [kW]	210.00	210.00 [b]	0.00
Thermal efficiency	0.08	0.08 [b]	0.00
Vaporizer heat transfer rate [kW]	2580.00	2680.00 [b]	3.56
Condenser heat transfer rate [kW]	2360.00	2432.00 [b]	3.05

[a] Set Variables
[b] Calculated variables from the Aspen simulation

4.2 Design analysis of the Chena geothermal ORC power plant using the validated models

This study analyzes the influence of heat exchanger design to ROI of plant investment. In order to conduct this case study so that it will represent a real plant condition, some assumptions used in simulation analyses are:

- The turbine and pump are simulated with fixed values of mass flow rate, working pressures and efficiency according to the real plant design data in table 2 in order to avoid failure in the real operation.
- The heat duty for each type of the heat exchanger is equal to heat duty from the real plant design data.
- The sizing heat exchanger analysis includes three limitation parameters of the plant operation:
  a. Chena geothermal plant, like most other geothermal power plants, makes use of re-injection of the used geothermal fluid into the re-injection wells in order to improve the pressure on the production wells. Table 2 displays that the geothermal exit temperature from real plant data and the simulation model is 54.44 ^{0}C and 55.70 ^{0}C, respectively. The temperature has to be maintained in order to avoid salt precipitation and the cooling of the geothermal fluid. Based on this argument, the maximum reduction of exit temperature of the geothermal fluid is assumed around 1 ^{0}C in order to get more heat into the system with larger size the heat exchangers.
  b. Minimization of heat exchanger size has to consider the moist condition. Moisture inside the expander can cause severe mechanical damage to the rotor and stator, that have been designed for dry steam [23].
  c. Furthermore, the sizing of heat exchanger has to avoid temperature crossovers of hot and cold streams through the exchanger unit during heat transfer process (the basic of pinch point principle).

4.3 Effect of heat exchanger type and sizing on ROI

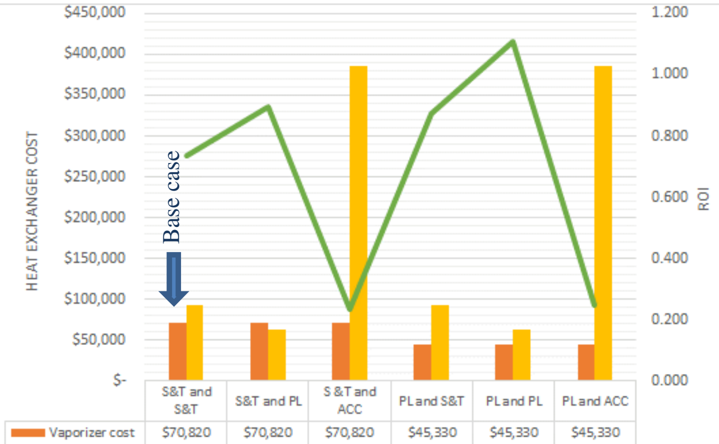
In this study, the evaluation of investment is analyzed by ROI method where this profitability measure is defined in equation 5. The annual net profit is calculated with data given in Table 3 and the total capital investment is counted according to the total cost of heat exchangers and it neglects the cost of the pump and turbine because their costs are set to be fixed.

Table 3: Data for calculating annual net profit

DATA	VALUE
Price of electricity in Alaska 2007 [24]	$ 0.13/kWH
Capacity factor of the plant [25]	0.9
Operating hour per year	8760 hours
Cost of production [18]	$ 0.05 / kWH
Cost of maintenance [18]	$ 0.01 / kWH

4.3.1 Heat exchanger type on ROI

The costs of heat exchangers dominate the total cost of ORC plants especially in plants driven by a low temperature geothermal resource. The brine of geothermal resources has known fouling and unique characteristics that influence the plant design. According to Holdmann [18], Chena water analysis results show both geothermal water quality and the surface water are soft and have low ammonium, so all possible types of heat exchanger may be selected as long as they meet the thermal and hydraulic requirements. The base case of analyses used shell and tube (S&T) type since the type is installed in the real plant for the vaporizer and condenser [18].



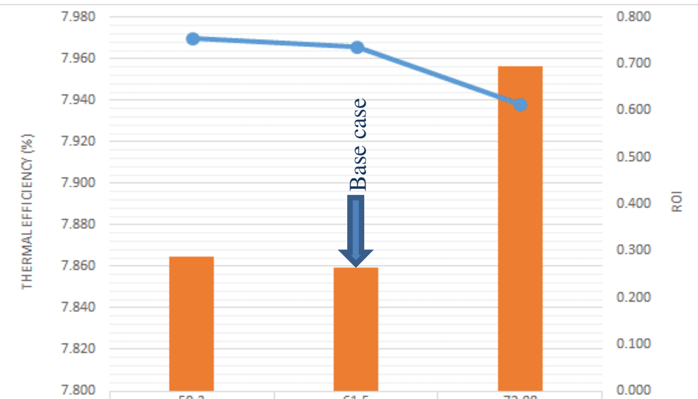
	S&T and S&T	S&T and PL	S&T and ACC	PL and S&T	PL and PL	PL and ACC
Vaporizer cost	$70,820	$70,820	$70,820	$45,330	$45,330	$45,330
Condenser cost	$92,766	$63,589	$384,880	$92,766	$63,589	$384,880
ROI	0.737	0.897	0.234	0.873	1.107	0.248

Figure. 4: Heat exchanger cost and ROI of six different systems

Figure 4 shows the comparison of heat exchanger cost and ROI in six systems with different types of heat exchangers in the vaporizers and condensers. The ROI of base case is 0.737. The usage of plate (PL) type increases the ROI and reaches a maximum ROI at 1.107 when PL type is used in both vaporizer and condenser. However, the air cooled condenser (ACC) type decreases the ROI significantly, since the price is 4 – 6 times that of S&T and PL type and it needs an additional load of electricity in order to operate air condenser fans. The ROI reaches the lowest value of 0.234 when the system is designed to use S&T vaporizer and ACC. The main characteristics of the different types of vaporizer and condenser are given in Table 4. The values of heat duty in each type of heat exchangers are not be exactly the same value as the heat duty of real design for the vaporizer and condenser, because the calculation of heat duty takes into consideration of the heat exchanger geometry of different types. But the plant model must produce the same net power output of the plant design about 210 kW. Comparison of the required heat transfer area between PL and S&T types shows the PL type needs larger area than S&T type, but the price of PL type is cheaper than S&T, so it means that price per unit area of PL type is lower than S&T type.

In addition, ACC type requires the largest area of condenser options. This is because air has significantly less favorable properties of heat transfer than water such as water has over 4 times higher specific heat ($c_{p,water}$ = 4.19 kJ/kg^{0}C and $c_{p,air}$ = 1,0 kJ/kg^{0}C) and water is 830 times more dense than air (density water and air at 15 ^{0}C is 999kg/m^3 and 1,2 kg/m^3). Table 4 shows that heat transfer coefficient of air is 1018 W/m^{2}K which is significantly lower than the heat transfer coefficient of water in S&T and PL type at 5135 W/m^{2}K and 2098 W/m^{2}K, respectively. The poor heat transfer of air necessitates a significantly higher surface area for heat transfer, so the equipment is very expensive. However, in case no cooling water is available on the site, ACC must to be selected. In addition, the fans of ACC consume a lot of electricity that causes a reduction of the net power plant output.

For a special case where the ambient temperature is usually subzero during several months in the winter causing a freeze, like in Chena, ACC type is the solution of the operational problem. The Chena geothermal power plant experienced a failure of the operation due to a frozen cold water supply during the late winter and early spring months, because the temperature dropped to -10 ^{0}C and hampered the operations of the plant [18]. The ACC becomes a good solution during the winter in order to maintain a sustainable plant operation. Seasonal temperature variation is not taken in consideration of the analysis



	59.3	61.5	72.08
Efficiency	7.864	7.859	7.956
ROI	0.756	0.737	0.614

Figure 5: Thermal efficiency and ROI of possible sizing of heat exchangers in the plant

Vaporizer Parameters	Heat Exchanger Type	
	S&T	PL
Cold side temperature (in/out) [^{0}C]	12.46 / 62.77	10.63 / 62.87
Hot side temperature (in/out) [^{0}C]	73.33 / 55.65	73.33 / 55.84
Cold/hot sides pressure drop [bar]	0.087 / 0.047	0.00247 / 0.00463
Cold/hot side heat transfer coeff. (mean) [W/m^{2}K]	1695 / 2934	857.5 / 1853.6
Overall heat transfer coefficient [W/m^{2}K]	1023	569.6
Heat duty [kW]	2680	2710.5
Required exchanger area [m^2]	278.7	503

Condenser Parameters	Heat Exchanger Type		
	S&T	PL	ACC
Cold side temperature (in/out) [^{0}C]	4.44 / 9.71	4.44 / 9.77	4.44 / 10.64
Hot side temperature (in/out) [^{0}C]	19.52 / 11.31	19.38 / 9.49	20 / 8.1
Cold/hot sides pressure drop [bar]	0.173 / 0.0553	0.031 / 0.0061	0.00192 / 0.07429
Cold/hot side heat transfer coeff. (mean) [W/m^{2}K]	5135 / 2071	2098 / 1227	1018 / 2240
Overall heat transfer coefficient [W/m^{2}K]	1382	761.6	686.2
Heat duty [kW]	2436	2465	2500
Required exchanger area [m^2]	429	761.6	26623.2

Table 4: The main characteristic of vaporizer and condenser

4.3.2 Heat exchanger sizing on ROI

The sizing analysis is conducted in order to evaluate the size of the real heat exchangers in comparison to possible minimum and maximum sizes for Chena system. Figure 5 shows that a size reduction of the heat exchangers with the stream of inlet turbine temperature at 59.3^{0} C increases the ROI by 2.59% (from 0.737 to 0.756), but reduces the net power output from 210 kW to 203.9 kW with an almost constant level of thermal efficiency around 7.8%. However, a maximum size of heat exchangers reduces the ROI dramatically by 17% (from 0.737 to 0.614). The net power output increases from 210 kW to 220.3 kW while increasing the thermal efficiency about 1.2% from 7.859% to 7.956%. The maximum size design reduces the geothermal exit temperature from 55.7 ^{0}C to 54.7 ^{0}C which is still within the limitation range of geothermal outlet temperature that has been explained in section 4.2.

4.4. Effect of recuperator on Rate of Investment

A proposed investment of a recuperator must be evaluated for its economic feasibility. The system with a high remaining energy after expansion of the expander is more reasonable to be analyzed. The design with a maximum size of the heat exchanger that has been discussed in section 4.3.2 is used to analyze the recuperator. The model indicates that installing recuperator increases the net power output from 220.3 kW to 221.3 kW with an incrementally higher thermal efficiency from 7.956% to 8.417%. However, the ROI reduces significantly from 0.614 to 0.409. Although a recuperator has used PL type which is more cost-effective than S&T type.

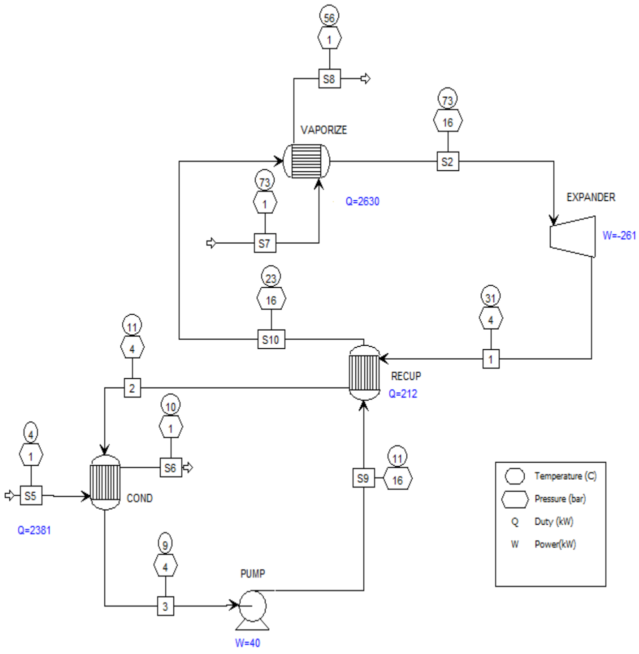


Figure. 6: Process flow of the plant using recuperator

5. CONCLUSION

The effect of heat exchanger design on ROI has been examined by simulation models that have been validated with the real plant design data from the Chena geothermal power generation plant. The simulation results show that the selection of heat exchanger type significantly influences on the ROI of the plant. The model indicates that the ROI of the Chena geothermal power plant is able to be increased from 0.737 to 1.107 by changing from S&T type to PL type for both heat exchangers.

Furthermore, based on the sizing analysis, the real sizes of heat exchangers have already given an optimal ROI of the plant at 0.737 with net power output of 210 kW. The reduction of the heat exchanger size increases the ROI by 2.59%, but reduces the net power output from 210 kW to 203.9 kW and comes with a higher risk of wet fluid inside the expander due to less superheat to buffer fluctuation in the real operation. A design using a maximum size of heat exchangers has a 17% lower ROI at 0.614.

Although an additional recuperator in the maximum design of heat exchanger size increases a net power output and thermal efficiency, but ROI of the plant reduces significantly, therefore the design without a recuperator is preferable.

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