GEOTHERMAL BINARY PLANTS: WATER OR AIR COOLED ?

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Abstract

Cooling geothermal power plants is necessary in order to condense the vapour feeding the turbine, lower the heat rejection temperature, raise power output and increase heat to power conversion efficiency. Three main cooling options are used: a) surface water (once-through systems), b) wet type cooling towers, and c) dry type cooling towers. Cooling with surface water yields the lowest condensing pressure and temperature and the highest conversion efficiency, followed by wet cooling towers, and then by dry cooling towers. Regarding the need for cold water supply, the order is reversed. Typical values are 970 t/h, 30 t/h and zero t/h respectively per MWe of installed power. In terms of costs, once through cooling may require both high capital costs and electricity consumption for transporting water. Dry cooling is the most expensive option due to the much higher heat capacity and heat transfer coefficient of water compared with ambient air. A dry cooling tower for a binary power plant of high conversion efficiency may cost 10 times more than its wet counterpart, which may result in raising overall power plant costs by 50%. In flash plants, where there is plenty of steam condensate to use as make up water, the standard technology adopted almost exclusively is cost effective direct contact condensers coupled with wet cooling towers. In binary plants, where the more expensive shell-and-tube or plate heat exchangers are used as surface condensers, the selection of the cooling system type is governed by water availability, local water use regulations and economics.

1. Introduction

The goal of this paper is to examine and compare the different cooling options adopted in geothermal power plants. For our analysis we divide geothermal power plants into the following main types:

- Back pressure flash plants where the steam is discharged from the turbine at 1 bar(a) pressure or 100℃.
- Condensing flash plants which use a condenser in order to condense the steam at the turbine discharge at lower pressure and temperature.
- Binary plants, which use the geothermal hot water or steam to boil a closed loop of a secondary working fluid, which drives a turbine and is condensed at the turbine discharge, and then is conveyed by a pump to the geothermal source and follows the same cycle again and again.

Present technology for cooling the power plant condensers may use either or both water or air and includes:

- Cooling with surface water in once through systems
- Cooling with water evaporation in air draft by wet type cooling towers
- Cooling with air by dry type cooling towers

2. Need for cooling

The Carnot efficiency of a geothermal power plant, either flash or binary, is:

$$n = \frac{Tg - Tc}{Tg} \tag{1}$$

where:

"Tg" is the temperature of the geothermal source in degrees Kelvin, and

"Tc" is the cooling water temperature in degrees Kelvin

Given the temperature of the geothermal source "Tg", from equation (1) it is evident that the Carnot efficiency of the plant is increased if we decrease "Tc", which can be achieved by condensing the steam or fluid vapour to the lowest possible temperature. This can be achieved by introducing a condenser at the turbine discharge, which should discharge the latent heat of the condensate to a cooling fluid.

If we consider the turbine alone, for a given mass flow rate and specific enthalpy of the geothermal steam the delivered mechanical work to the power generator "w" equals to:

$$w = n_t \cdot m \cdot (H_s - H_o) \tag{2}$$

where:

"m" is the steam mass flow rate,

"H_s" is the specific enthalpy of the steam or fluid vapour entering the turbine,

"H_o" is the specific enthalpy of the steam or fluid vapour at the turbine discharge, and

"n_t" is the turbine efficiency, usually within the range 60%-90%

The power "N" generated at the generator equals to:

$$N = n_g \cdot w \tag{3}$$

where:

"n_g" is the efficiency of the generator, usually around 95%

Overall heat to power conversion efficiency of the plant "n" equals to:

$$n = \frac{N}{m \cdot H_s} = \frac{n_g \cdot n_t \cdot (H_s - H_o)}{H_s} \tag{4}$$

From above equations (2), (3) and (4) it becomes evident that in order to increase the power generated by a given quantity of geothermal steam or working fluid vapour, and hence the heat to power conversion efficiency, we should minimize "H_o".This can be achieved by discharging the steam at the turbine exit to the minimum possible temperature, which, according to the thermodynamic properties of the fluids, corresponds to the lowest possible pressure. Such low pressure can be generated by condensing the steam or fluid vapour to the lowest possible temperature with a cooling fluid. We arrived therefore to the same conclusion with the Carnot efficiency analysis of equation (1).

94% of the total generated power globally is derived by steam flash plants, the majority of which use condensing turbines, which in practice yield twice as much power output than the atmospheric exhaust ones. Due to the thermodynamics involved, all geothermal binary plants, which correspond to 6% of global geothermal power output, use working fluid condensing turbines.

3. Water vs. air cooled condensers

In nature, available cooling fluids are either water (from sea, lakes, rivers, or subsurface) or air. In terms of heat transfer, water has more favourable properties than air, as follows:

- Water has over 4 times higher specific heat "$c_{pw} = 4{,}19 kJ / kg^{o}C$" than ambient air "$c_{pa} = 1{,}00 kJ / kg^{o}C$".
- Water is 830 times more dense than air. For example for cooling fluid temperature of 15 °C, the water's density is 999 kg/m³ whereas the air's density is 1,2 kg/m³. This results in higher volumetric heat capacity and heat transfer coefficient.
- Water has volumetric heat capacity "$VHC_w = 4182 kJ / m^{3o}C$", approximately 3450 times the one of ambient air, which is "$VHC_a = 1{,}21 kJ / m^{3o}C$". This implies that in order to have the same heat transfer effect, 3450 more volume of air has to be moved than in the case of water, resulting in the need for bulky and expensive equipment for air-handling, plus higher electricity consumption for the air fans than the water pumps.
- In condensers water yields typical heat transfer coefficient 58 times higher than the one of air, which we have estimated as "$h_w = 4{,}84 kW / m^{2o}C$" for water and "$h_a = 0{,}084 kW / m^{2o}C$" for air. This implies that the surface of the condenser and the corresponding costs will be accordingly higher if air is used as cooling fluid rather than water.

The heat exchange surface has a direct impact on the weight and size of the condenser, which are the most important economic variables defining the corresponding costs.

4. Surface water (once-through systems)

In this category the cooling fluid is water, which is transported to the power plant through pipes from a river, a lake or the sea. The temperature of the cooling water in this case varies in proportion to season's temperature. It can be 5℃ - 25℃. This is why surface water yields the lowest condensing pressure and temperature in compare to the other two types. A typical value of water supply is 970 t per h and per MWe of installed power for approximately 10℃ temperature gain of the cooling water across the condenser. As far as it concerns the plant's cost, electricity consumption for transporting water (pipes, pumps etc.) may not be at all negligible, depending on the location and distance of the water source.

Binary plants normally use horizontal double pass shell-and-tube heat exchangers as surface condensers, with the cooling water flowing inside the tubes and the steam and condensate in cross flow within the shell. Although not a standard practice, use of plate heat exchangers instead, may be a tempting option due to their compact size, their mass production and easy to dismantle/mantle and clean capabilities and their high overall heat transfer coefficient, typical values of which are 10-20 kW/m^2.

In cases where sea, lakes or rivers are located close to the power plant, and in cases where local regulations for water use allow it, cooling with surface water should be considered as one of the available options. Its main advantage is that it can yield the lowest possible condensing temperature, and hence the maximum conversion efficiency as:

- Surface waters tend to have lower temperature than ambient air during the summer period; for example in South Europe sea water has temperature of ~25℃ during the summer, while ambient temperatures around 35℃ are common.
- In most European countries surface waters do not froze when ambient temperature drops below 0℃.
- No cooling towers of either wet or dry type are necessary.
- The heat delivered to the cooling water can be utilized for downstream heating applications, resulting in a geothermal heat and power cogeneration plant and further increasing overall energy efficiency.

Main drawbacks include:

- Need for large water quantity.
- Fooling or corrosion in the condenser in cases of adverse chemistry, or organisms present in the cooling water
- High capital costs for piping and pumping stations or electricity consumption in case the water has to be transported from large distances.

5. Wet type cooling towers

The cooling water that is used in the condenser is conveyed to the cooling tower in order to reduce its temperature so that it will be recycled and looped through the system. An important reduction of its temperature is accomplished in the tower. In small or medium size plants, such as geothermal power plants, cooling towers usually use mechanical ventilation (fan) for the advection of the air stream. In these plants cooling towers that are mostly used are cross-flow and traverse-flow. The typical temperature difference between the inlet and outlet cooling water is 10℃. As far as it concerns the temperature of the cooling water that comes out of the cooling tower, it reaches at least 25℃, resulting in condensing temperatures around 40℃, depending on the ambient temperature.

Wet cooling towers combine the use of water as a cooling media to the condenser and benefit from its favourable heat capacity and heat transfer properties compared with air, while they do not need the large volumes of surface water needed in once through cooling systems. Instead, they evaporate water at a cooling tower, and need a much smaller quantity of make up water to compensate the evaporation losses plus the water blowdown necessary to maintain water quality. Typical needs for make up water of a geothermal binary plant have been estimated as 30 t/h per MWe of installed power capacity.

In geothermal flash plants, there are usually enough water quantities available for the make up water of the cooling towers, as the much less make up water needed per MWe of delivered power, corresponds to a fraction only of the available steam condensate. There, wet cooling towers are usually coupled with direct contact condensers, where the cooling water is sprayed and mixed with the steam condensate, and which are simpler in design and much more cost effective than surface condensers used in binary plants. For this reason, direct contact condensers and wet type cooling towers are the standard technology in geothermal flash plants. Exceptions are encountered in cases where large quantities of surface water are available locally, and in extremely cold climates in order to avoid frosting water droplets precipitating in the plant neighbourhood.

6. Dry type cooling towers

In dry type cooling towers, the temperature of the air that comes out of the tower in order to cool the fluid in the condenser is higher than 25℃. Typical values are 25-30℃ resulting in condensing temperatures around 40-50℃. In a dry type cooling tower no water supply is necessary. Regarding auxiliary power consumption, they usually consume twice as much electricity than wet cooling towers.

Due to the need for many times higher heat exchange surface and the large volume of air that has to be moved through them, dry type cooling towers are the most expensive option. A dry type cooling tower costs 5-10 times as much as a wet type one depending on the condensing temperature of the turbine. If low condensing temperatures are considered, an air source geothermal binary plant may have 50% higher capital costs than one with a wet type cooling tower of the same efficiency. In practice, air source geothermal binary plants are designed with considerably less conversion efficiency and costs 10-20% higher.

However, in cases of lack of water, strict local water use regulations, extremely low ambient temperatures during winter which cause water droplets from wet type cooling towers to freeze onto nearby vegetation, dry type cooling towers may be the only available option.

7. Rankine cycle optimisation

In the framework of the LOW-BIN (Efficient Low Temperature Geothermal Binary Power) project, which is co-financed by the 6th European framework programme (FP6), we have been modelling Rankine cycles in order to optimise costs and conversion efficiency. Optimisation has been performed using the EASY software code (Evolutionary Algorithm System by National Technical University of Athens, ref. http://velos0.ltt.mech.ntua.gr/EASY).

For the above optimisation a low temperature binary plant has been considered, supplied by 65℃ geothermal water, with R134a as working fluid. The cooling water temperature is 10℃. The plant is schematically presented in Figure 1.

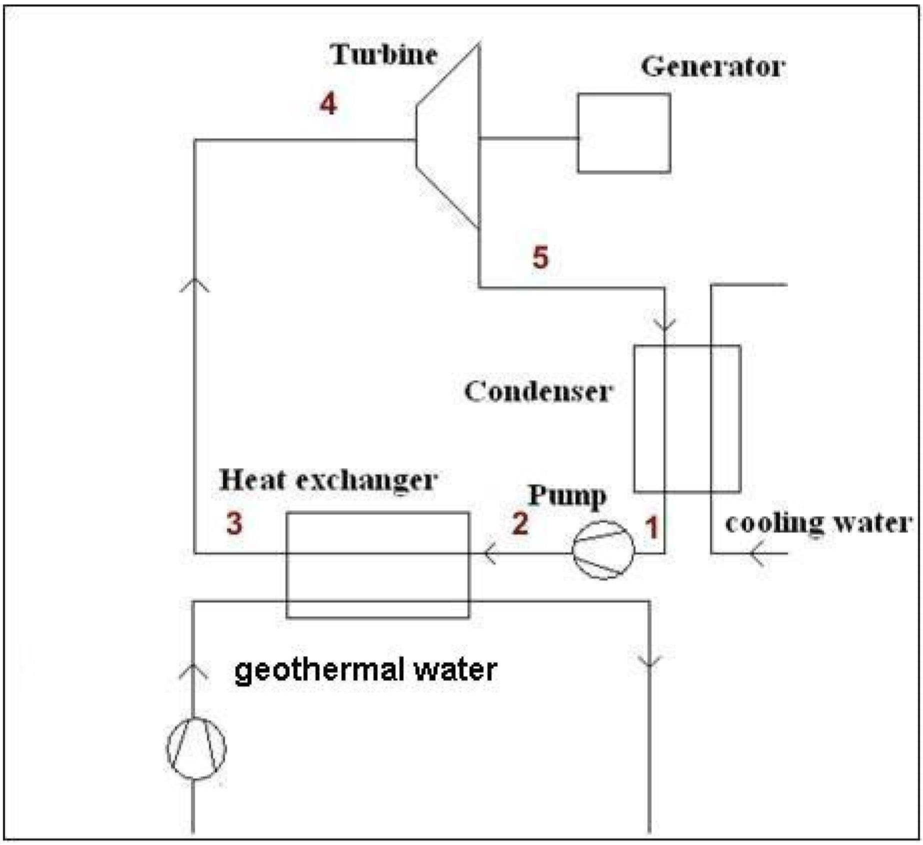

Figure 1: Binary plant layout.

For the initial phase of the project we have modelled the Rankine cycle using certain assumptions, such as simplified geometry for the condenser, which will be relaxed in later stages. The overall heat-transfer coefficient used for modelling the heat transfer at the condenser is based on the following formula:

$$U_o = \frac{1}{\dfrac{A_o}{A_i}\dfrac{1}{h_i} + \dfrac{A_o \ln(r_o / r_i)}{2\pi kL} + \dfrac{1}{h_o}} \qquad (5)$$

where A_o and A_i represents the out and in surface areas respectively of the inner tubes, L is the length of the tubes, h_i *is the heat transfer coefficient* inside the tubes where the cooling fluid flows, h_o *is the heat transfer coefficient* outside the tubes where the working fluid flows and *k represents the thermal conductivity* of the tube's material.

The heat transfer coefficient outside the tubes (R-134a) is based on the following formula for laminar film condensation on horizontal tubes:

$$h_o = 0.725\left[\frac{\rho(\rho-\rho_v)gh_{fg}k_f^{\ 3}}{\mu_f d(T_g - T_w)}\right]^{0.25} \tag{6}$$

where ρ and ρ_v represent the density of R-134a in liquid and vapour forms respectively, h_{fg} is the latent heat, μ_f is the dynamic viscosity, k_f is the thermal conductivity, d the outside diameter of the tube, T_g is the saturation temperature of the fluid condensate and T_w the temperature of the tube's wall.

The heat transfer coefficient inside the tube for turbulent flow is based on the following formulas :

$$h_i = \frac{Nuk}{D}$$
$$Nu = 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \tag{7}$$

The R134a Rankine cycle was optimised defining the following variables:

- the outlet pressure of the pump,p_2
- the hot ground water supply, m_{gr}
- the supply of the R-134a in the cycle, m_{R134a}
- the temperature difference of the ground water in the heat exchanger,ΔT_H
- the temperature difference of the cooling fluid in the condenser,ΔT_C

The corresponding upper and lower limits are listed in table 1.

Table 1. Upper and lower limits of the optimisation variables

Variable	Lower limit	Upper Limit
p_2 (kPa)	750	1200
m_{gr} (kg/sec)	45	55
m_{R134a} (kg/sec)	10	20
ΔT_H (°C)	10	30
ΔT_C (°C)	7,5	12,5

The objectives of the optimization are:

- Maximization of the <u>total efficiency of the plant</u>

$$\eta_{cycle} = \frac{w_{turbine}}{q_{heatexch}} = \frac{h_4 - h_5}{h_3 - h_2}$$

- Minimization of the <u>cost of the plant</u>. Since the cost of the heat exchanger and the condenser constitute a major part of the plant cost, for our optimizing purposes we can substitute the plant cost by their cost. So the new goal is to minimize the cost of the heat exchanger and the condenser which is proportional to their surface $\rightarrow$ Minimization of the <u>exchangers' surface</u>.

The results are presented in figures 2 and 3 for water cooled and air cooled condensers respectively. A comparison of selected representative cases is shown in table 2. In figures 2 and 3 a number of optimal solutions have been plot. Each solution is represented by two numbers which constitute the surface of the heat exchangers (geothermal heat exchanger and R134a condenser) and the overall conversion efficiency. Additionally, each solution resulted from a different set of variables corresponding to an optimal Rankine cycle.

In water cooled systems (figure 2), the values of total efficiency vary between 6,1% and 7,2%, when the total surface of the heat exchangers is between 100 and 250 m². As we can observe from the diagram, after the value of 135 m², very small efficiency rises are associated with large increments in the heat exchangers surface. For this reason we consider a representative solution corresponding to heat exchangers surface and conversion efficiency of 138 m² and 6,96% respectively.

In air cooled systems (figure 3), the values of total efficiency are between 6,0% and 6,9%, when the total surface of the heat exchangers is between 2400 and 4000 m^2. As above, we can select as a representative solution the one that corresponds to heat exchangers surface and conversion efficiency of 3230 m² and 6,78% respectively.

If we compare figures 2 and 3 and the results presented in table 2, we conclude that the surface of a water cooled condenser is approximately 25 times less than an air cooled one. The total heat transfer coefficient of a water cooled condenser is ~5600 W/m^2C when the air cooled condenser is only ~100 W/m^2C. Costs for air-cooled condensers should be higher than a water cooled ones accordingly.

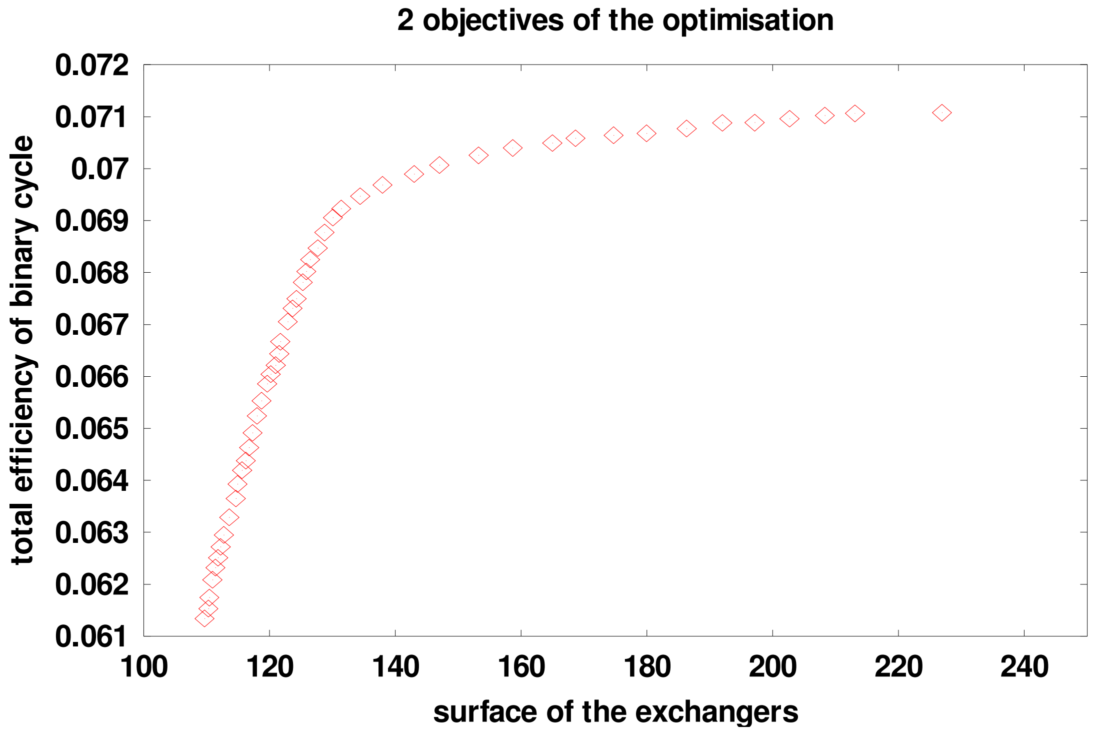

Figure 2: Rankine cycle optimization for water cooled condensers

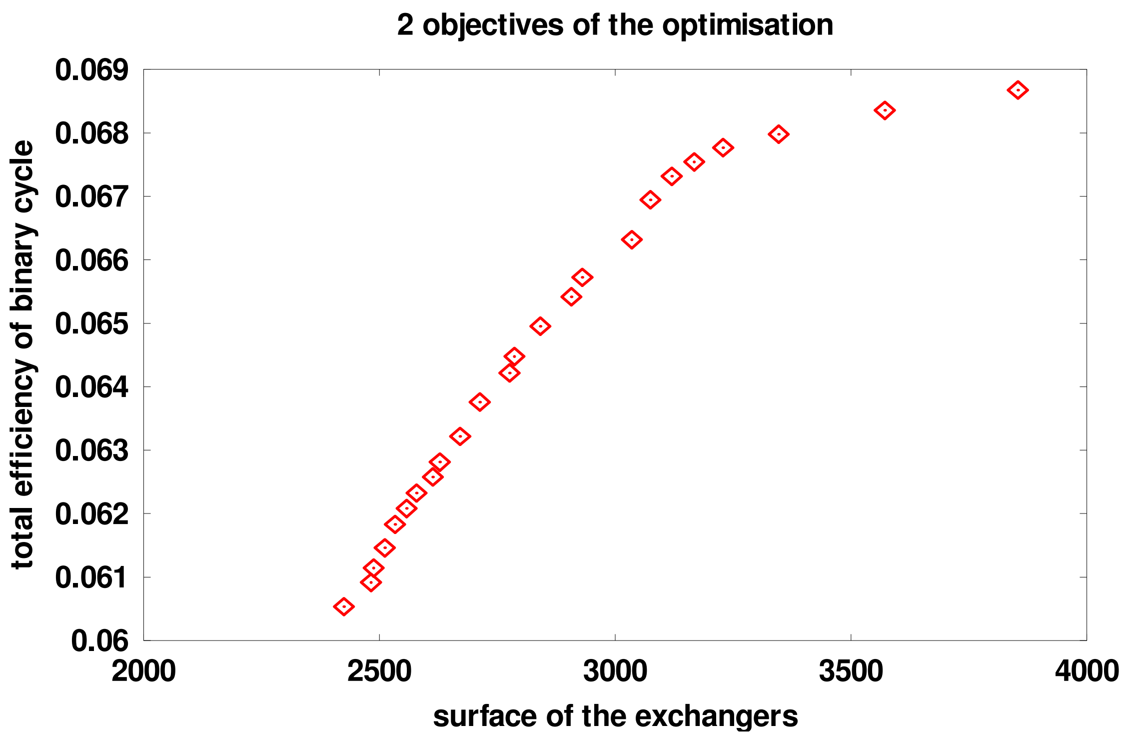

Figure 3: Rankine cycle optimization for air cooled condensers

Table 2. Comparison of representative optimized Rankine cycles for water and air cooled condensers.

Variable	Water Cooled	Air Cooled
p_2 (kPa)	1100	1100
m_{gr} (kg/sec)	52,3	53,0
m_{R134a} (kg/sec)	17,5	17,5
ΔT_H (℃)	17,5	17,8
ΔT_C (℃)	7,5	7,5
R134a pump power (KW)	13	12
cooling fluid flow (m^3/h)	403	$3{,}45*10^5$
Total heat transfer coefficient U	5580	102
Surface of the condenser (m^2)	88	3160
Total H.E. surface (m^2)	138	3230
Conversion efficiency	6,96 %	6,78 %

8. Conclusions

Cooling geothermal power plants is necessary in order to improve conversion efficiency. Water cooling leads to higher conversion efficiencies and lower plant capital and operation costs than air cooling, as has been proved by analysing heat transfer properties of water and air, as well as by optimizing the corresponding Rankine cycle. On the other hand, water cooling needs considerable quantities of cold water supply. Use of water cooled condensers and wet type cooling towers results in drastically reduced, but still considerable water needs. Dry air type cooling towers, despite adverse economics and energy efficiency, may be the only feasible option in cases of water scarcity, or extreme climatic conditions.