

Improving the Annual Net Power Output of Geothermal Binary Power Plants

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ABSTRACT

Current research activities in the field of geothermal binary power plants mainly aim for an enhanced performance of the power plant cycle at the design point. Experience from running power plants, however, shows that improving the design point performance is not the only issue since plant operation can deviate significantly from the design conditions. This contribution will therefore discuss general aspects of improving the annual net power output with focus on the cooling system. By means of numerical modelling and example case studies assuming a basic Organic Rankin Cycle (ORC) using dry cooling and wet cooling, main improvement aspects will be discussed for the design and operation of the cooling system.

1. INTRODUCTION

Geothermal binary power plants that use hot fluid with temperatures between 100 and 200 °C can be applied for both, the extended use of high-enthalpy geothermal resources as well as the exploitation of low-enthalpy geothermal systems even far away from volcanically active areas. Even though geothermal low temperature power plants offer a huge potential and have been installed at more than 100 sites, techno-economic challenges still prevent a broader use.

Current research activities in the field of low temperature power plant technology mainly aim for an enhanced performance of the power plant cycle at the design point by means of improving conversion cycle set-up (e.g. trans-critical cycles or multi-pressure cycles), selecting more suitable working fluids (e.g. new pure fluids or working fluid mixtures) or developing plant components with higher performance (e.g. turbines or heat exchanging equipment). Experience from running power plants, however, shows that improving the design point performance is not the only issue since plant operation can deviate significantly from the design conditions, esp. due to variable ambient conditions. In comparison to site- and plant-specific investigations, which have been carried out for existing power plants, this contribution will discuss general aspects of improving the annual net power output. The cooling system thereby plays the central role in this paper since geothermal binary plants typically need to remove relatively large amounts of waste heat by means of power consuming cooling equipment (e.g. forced-draft wet cooling towers or air-cooled condensers).

By means of numerical modelling and example case studies assuming a basic Organic Rankin Cycle (ORC) using dry cooling and wet cooling, main improvement aspects will be discussed for the design and operation of the cooling system. The presented work is based on current research activities and the experiences from planning the geothermal research power plant in Gross Schoenebeck (Germany).

2. MODEL DESCRIPTION

In order to evaluate the influence of the cooling system design on the net power output of geothermal binary power plants, numerical models of a conversion cycle with air-cooled condenser and a wet cooling tower, respectively, have been developed in the Simulation Software EES (EES Engineering Equation Solver, <http://www.fchart.com/ees/>). The net power output thereby considers the electrical power produced in the conversion cycle and the electrical power consumed by components in the conversion cycle and the cooling system. Based on the numerical models it is possible to calculate the installed electrical capacity and the size of the main components of a binary power plant at the design point as well as to perform quasi-stationary simulations of a defined plant set-up during operation. The following paragraphs give a brief description of the models. More details on parameters and assumptions are given with the case studies.

The **conversion cycle** is modeled as subcritical, saturated steam cycle ORC which uses a dry working fluid. The ORC is fed by hot geothermal fluid which is assumed to be pure water. It contains the feed-pump to pressurize the working fluid, an evaporator for heating and evaporation, the turbine for the conversion of thermal to kinetic energy, and the recuperator for using the residual heat in the working fluid after expansion. Even though the recuperator will introduce additional pressure losses in the ORC which increases the feed pump power, on the one side, and decreases the turbine output, on the other, it should be used with cooling systems such as discussed below since it will reduce the auxiliary power for the waste heat removal. The condensation process, which completes the conversion cycle, will be assigned to the cooling system (see Fig. 1).

All heat exchangers are represented by shell-and-tube heat exchangers. The heat transfer is calculated based on corresponding Nusselt- and α - correlations taken from VDI Heat Atlas 2010. Pressure drops are factored in as design parameters and off-design correlations based on the working fluid mass flow. In order to estimate the power consumption of the feed pump based on a given efficiency, also the pressure drop of the condensation process is accounted for. The turbine is modeled with a fixed laval-nozzle inlet, a global isentropic efficiency and a mechanical efficiency. Carrying out design point calculations, the heat exchanger specifications (heat exchanger area and number of tubes) and the flow area of the turbine nozzles are results from the thermodynamic cycle design which determines the evaporation temperature yielding the maximum power output (turbine power – feed pump consumption). For the simulation of the ORC operation, evaporation temperature and working fluid mass flow are calculated using the heat exchanger and turbine design specifications.

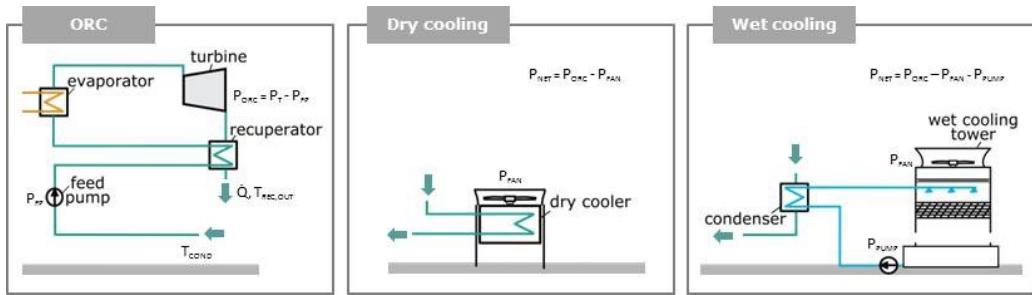


Figure 1: Set-up of ORC, dry cooling with air-cooled condenser and wet cooling with cooling tower.

The **dry cooling** is represented by an air-cooled condenser which consists of a horizontal finned-tube bank and fans generating forced-draft. Heat transfer and tube-sided pressure drop are calculated based on corresponding Nusselt- and α - correlations taken from VDI Heat Atlas 2010. Since the pressure drop of the condensation process is also defined as design parameter in the ORC, such as previously described, the tube flow velocity is chosen to meet this parameter. The air-sided pressure drop is derived based on Brockmann (2000). The fan power is calculated with the static and dynamic pressure difference. The design point calculations are used to specify the geometry of the air-cooled condenser and the total design capacity of the fans based on a given thermal load and ambient conditions. During operation, the condensation temperature resulting from the specified dry-cooler design is derived.

The **wet cooling** consists of a surface condenser, a cooling water pump and a once-cell forced-draft counter flow wet cooling tower. Heat transfer and tube-sided pressure drop are calculated based on corresponding Nusselt- and α - correlations taken from VDI Heat Atlas 2010. Such as for the heat exchangers of the ORC, the shell-side pressure drop is assigned as design parameter. The thermal cooling tower performance is calculated from the Merkel number (Kröger, 2005). This means that for a specific cooling tower design, the thermal performance of the spray and rain zone and hence the necessary liquid-gas-ratio is calculated. The fan power is derived from the static pressure difference, the dynamic pressure difference and the pressure gain at the diffusor. The power consumption of the cooling water pump is calculated based on the tube-side pressure loss in the condenser, the pressure head to overcome the height difference between ground level and inlet to the water distribution system and the pressure at the water distribution nozzles.

The models have been validated based on the design data for the geothermal research power plant Gross Schoenebeck. Operation simulation results have been checked regarding plausibility. For future work, a validation with real operating data is planned.

3. CASE STUDIES

3.1 Cooling System Design

3.1.1 Boundary Conditions

Besides the heat input to the ORC the condensation temperature is determining for the power produced in the conversion cycle. To reach a desired condensation temperature at given ambient conditions, the cooling system needs to be specified accordingly. Designing the cooling system, it needs to be considered that a lower condensation temperature results in an increase of the turbine power but also in a higher power consumption of the cooling system. Referring to dry cooling and wet cooling, such as described in chapter 2, it is also important to keep in mind that low fan power consumption is usually accompanied with a large size of the cooling system.

In the following paragraphs, the design parameters of the cooling system which have a significant opposed effect on both, either condensation temperature and auxiliary power consumption or auxiliary power consumption and cooling system size will be analyzed for an example case using the following assumptions:

- Heat input: temperature 150°C, flow rate 30 kg/s
- ORC: working fluid n-butane, pinch point evaporator 7K, effectiveness recuperator 70%, isentropic pump efficiency 80%, mechanical pump efficiency 80%, isentropic turbine efficiency 75%, pressure losses preheating 0.2 bar (tube-side) and 0.05 bar (shell-side), pressure losses evaporation 0.2 bar (tube-side) and 0.15 bar (shell-side), pressure losses recuperator 0.02 bar (tube-side) and 0.05 bar (shell-side), pressure loss condenser 0.01 bar, diameter evaporator and recuperator 1.4e-3 m
- Ambient conditions: ambient temperature T_{air} 10°C, relative humidity r_{air} 75%, atmospheric pressure 1.001 bar

Dry cooling and wet cooling will be discussed separately. In order to evaluate the size of the cooling equipment, the base area of the air-cooled condenser and the wet cooling tower, respectively, will be the variable of interest.

3.1.2 Dry Cooling

The main parameters to specify the thermal performance of an air-cooled condenser are the thermal capacity \dot{Q} , the condensation temperature T_{COND} , and the initial temperature difference ITD at the inlet of the dry cooler ($=T_{AIR,IN} - T_{BUTANE,IN}$). Even though the working fluid is superheated at the inlet of the air-cooled condenser, the condensation temperature is decisive for the condensation process within the tubes so that this effective initial temperature difference ITD_{eff} ($=T_{AIR,IN} - T_{COND}$) will be used in the following. A lower ITD_{eff} will lead to a lower condensation temperature, but in return to a higher fan power and a larger dry cooler base area.

Besides a given ITD_{eff} , the geometry of the air-cooled condenser has also influence on the dry cooler size and the fan power. However, there is no significant opposing effect on fan power and dry cooler size if any. Previous parameter studies showed that the fin thickness, for example, has a strong effect on the fan power due its influence on the air-side pressure loss. The effect on the

heat exchange area and hence the size of the dry cooler is much smaller. A higher fin height, as another example, leads to a decrease in tube length. The fan power, however, decreases only until a certain height before it increases again.

The parameter of interest for this case study is hence the ITD_{eff} . The geometry parameters are assumed as follows:

- Finned-tube bank: fin height $1.3e-3$ m, fin spacing $1.6e-3$ m, fin thickness $0.25e-3$ m, longitudinal tube pitch $30e-3$ m, transversal tube pitch $60e-3$ m, heat conductivity fin 300 W/(mK), heat conductivity tube 50 W/(mK), inner tube diameter $1.4e-3$ m, tube thickness, $0.43e-3$ m, number of tube rows 4
- Fan: efficiency 70% , flow area equals $0.4 \times$ base area dry cooler

Based on these parameters and the assumptions described in section 3.1.1, the maximum net power that can be achieved with a given dry cooler base area of has been calculated. In Fig. 2 (top, left) it can be seen that the design net power P_{NET} increases with a larger dry cooler base area A_{DRYC} . However, the increase in net power is becoming smaller with larger area. It is also important to note that the dry cooler base area increase is limited due to the decrease in inlet air flow velocity w_{AIR} (Fig. 2, bottom right) which should be above 2 m/s in order to prevent recirculation of warm air.

From Fig. 2 (top, right) it can be observed that the optimum ITD_{eff} to reach the maximum net power output decreases with the available dry cooler area also leading to a decreasing T_{COND} . The optimum fan power (Fig. 2, bottom left) increases up to a dry cooler area of about 270 m^2 and then decreases. The reason for this maximum is the opposed course of volumetric flow \dot{V}_{AIR} and inlet flow velocity of the air w_{AIR} (Fig. 2, bottom right).

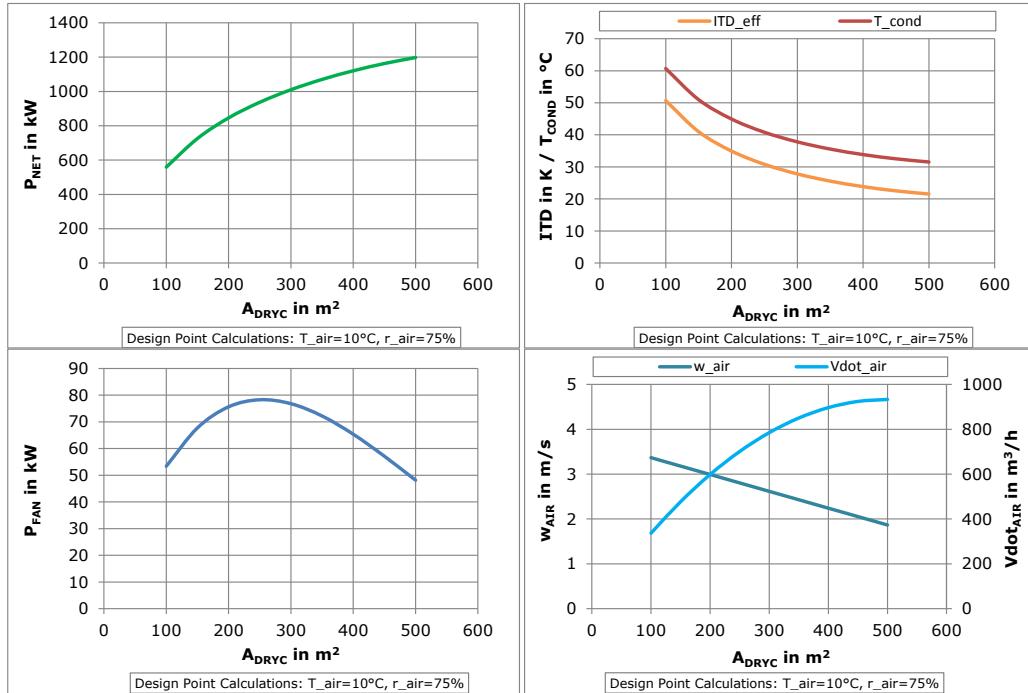


Figure 2: Design values vs. dry cooler base area of a dry-cooled ORC with maximum net power at given ambient conditions.

3.1.3 Wet Cooling

The wet cooling includes two main components, the cooling tower and the condenser. The size of the condenser heat exchange area is decisive for the condensation temperature that can be achieved from a defined cooling water flow. Regarding a mechanical draft counter-flow wet cooling tower, the main design parameter which are determining for the condensation temperature, the auxiliary power demand but also the size of the cooling tower, are cooling tower fill type and height, wet bulb temperature approach, cooling water spread and cooling water load. Following opposing effects can be seen:

- Cooling tower fill type & height: The cooling tower fill is the key component in the cooling tower. Assuming that the fill type is selected depending on the quality of the cooling water, the thermodynamic cooling performance increases with increasing fill height. A higher fill height, however, leads to a larger air-sided pressure loss coefficient. Additionally, cooling tower height and pumping power increase. Previous parameter studies have analyzed different fill types and different fill heights. Regarding the fill types it was observed that a similar net power output could be achieved with cross-fluted film fills of different sheet spacing and vertical flow film fills in case an optimized set-up (maximum net power) was chosen. For each fill type the optimum fill height did not change much within the scope of the parameter study. For the case study in this paper, only one fill type (cross-fluted film fill, sheet-spacing 19mm) and one fill height (1.8 m) will be considered. The fill performance data have been taken from manufacturer data (Accu-Pak CF1900, <http://www.brentwoodprocess.com>).
- Wet bulb temperature approach ΔT_{WBT} : A lower approach to the wet bulb temperature ($T_{CW,cold} - T_{WBT}$) decreases the cold water temperature and also the warm water temperature in case the cooling water temperature spread is set. At a given cooling tower fill height and cooling water mass flow, a lower approach is realized by a higher air mass flow and hence at the cost of a larger power demand by the cooling tower fans.

- Cooling water temperature spread ΔT_{CW} : In general, a lower cooling water temperature spread ($T_{CW,warm} - T_{CW,cold}$) increases the cooling water mass flow and hence the auxiliary power for the cooling water pumps. A higher cooling water mass flow furthermore increases the condenser size and the necessary ground area of the cooling tower, in case a certain cooling water load is realized. At a given approach, a lower temperature spread also decreases the condensation temperature and increases the required air mass flow as well as then fan power consumption.
- Cooling water load or rain density L_{RAIN} : A low cooling water load (cooling water mass flow per cooling tower base area) mainly decreases the air-sided pressure losses in the spray and rain zone and leads to a lower power demand of the cooling tower fans. A low cooling water load, however, also results in a larger ground area for the cooling tower.

Following further cooling tower design parameter will be assumed to be constant:

- Cooling tower: spray zone height 0.8 m, rain zone height equals air inlet height, height of the air inlet, air inlet cross section equals $0.5 \times$ base area wet cooling tower, ratio of inlet to outlet diffusor cross-section of 0.85, pressure loss coefficient water distribution system 1.5, pressure loss coefficient drift eliminator 2.5, design-coverage spray nozzles 0.8 m^2 , design pressure gauge spray nozzles 0.1 bar
- Fan: efficiency 70%, flow area equals $0.35 \times$ base area wet cooling tower
- Pump: efficiency 80%

Based on these parameters and the assumptions described in section 3.1.1, the maximum net power that can be achieved with a given wet cooling tower base area and condenser heat exchange area has been calculated. In Fig. 3 (top, left) that the net power increases with both, wet cooling tower base area A_{WETC} and condenser heat exchange area A_{COND} . However, the increase in net power is in both cases decreasing for larger areas. The optimum design parameters (maximum net power) are shown in Fig. 3 (top, right) for the example of a condenser heat exchange area of 700 m^2 . Wet bulb temperature approach ΔT_{WBT} , cooling water temperature spread ΔT_{CW} , and water load L_{RAIN} decrease with increasing A_{WETC} leading to a decreasing condensation temperature T_{COND} . It is important to note that the optimum pinch point in the condenser ($\Delta T_{PPC} = T_{COND} - T_{CW,warm}$) increases with larger A_{WETC} . Looking at the power consumption of fans and pump (Fig. 3, bottom left) it can be seen that the fan power decreases whereas the pump power increases with increasing A_{WETC} .

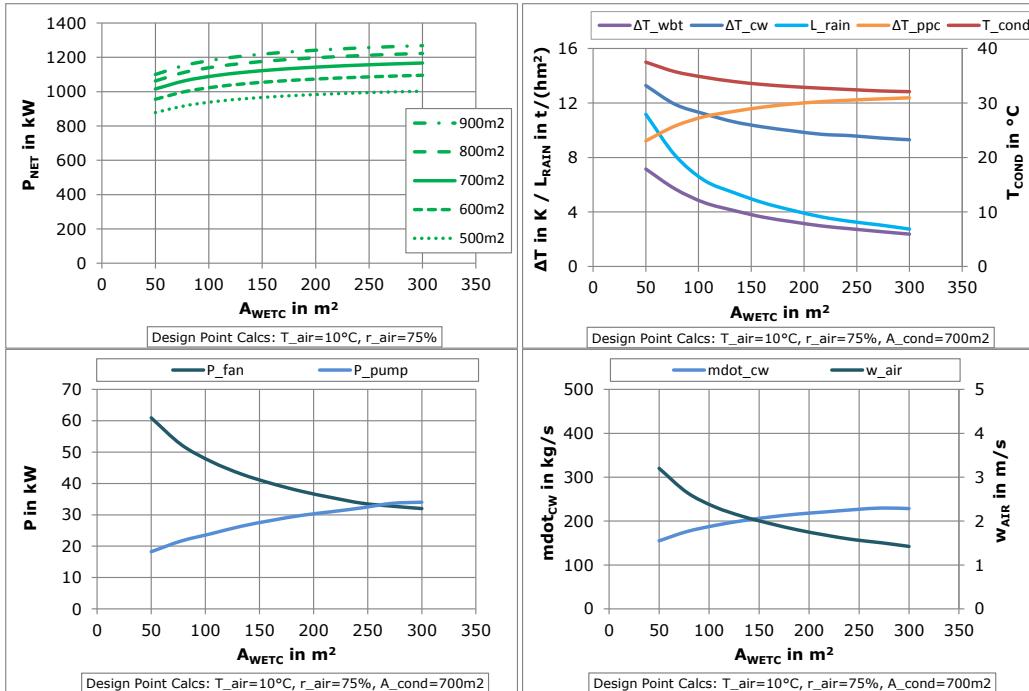


Figure 3: Design values vs. wet cooling tower base area of a wet-cooled ORC with maximum net power at given ambient conditions and given condenser heat exchange area (practical design limits are not considered).

Looking at the inlet air velocity w_{AIR} (Fig. 3, bottom right), it has to be noted that the minimum cooling tower size is limited due to the increase in inlet velocity since too high air velocities prevent an even water distribution in the cooling tower. This is also the case for too low rain densities which should hence be limited to $5 \text{ t/(h m}^2\text{)}$. For the example shown in Fig. 3, this minimum rain density is reached at a cooling tower base area of about 170 m^2 . The effect of this limitation on the optimum design can be seen in Fig. 4.

The limitation of the rain density L_{RAIN} , first of all, leads to larger cooling water mass flows for A_{WETC} above 170 m^2 which causes a significant increase in pump power (Fig. 4, left). Due to the higher auxiliary power, the net power is not increasing with increasing A_{WETC} . Depending on A_{COND} , even a decrease in net power with higher A_{WETC} can be observed.

Based on these results it is important to note that for a given condenser heat exchange area, the optimum condenser design (number of tubes, tube length) is influenced by the cooling tower design which is concluded from the changes in cooling water mass flow and temperature spread.

The condenser heat exchange area A_{COND} , in return, has also an effect on the cooling tower design. For a given cooling tower base area of 150 m^2 it is shown in Fig. 5 that pump power P_{PUMP} , cooling water temperature spread ΔT_{CW} and wet bulb temperature approach ΔT_{WBT} are only slightly influenced by a changing A_{COND} (also because of the limit of L_{RAIN}). Nevertheless, a slightly higher warm water temperature which is cooled down to a slightly lower cold water temperature is still causing an increase in fan power P_{FAN} . Looking at the pinch point in the condenser ΔT_{PPC} , which results from the optimization, much lower pinch points and hence significantly lower condensation temperatures T_{COND} can be realized with larger condenser area which will lead to a higher design net power.

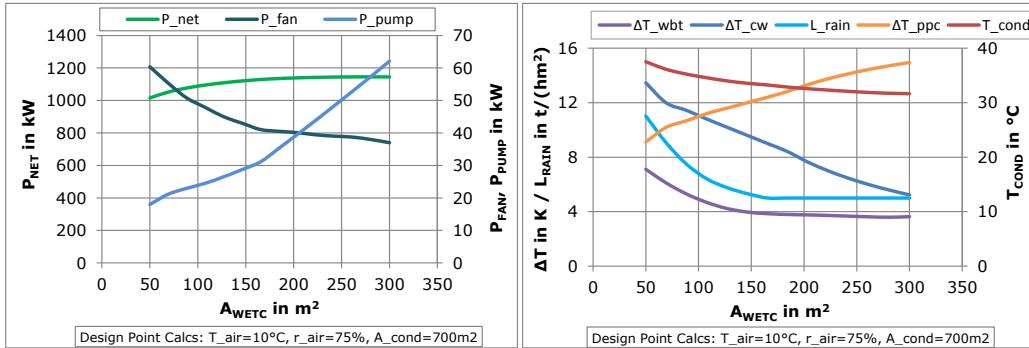


Figure 4: Design values vs. wet cooling tower base area of a wet-cooled ORC with maximum net power at given ambient conditions and given condenser heat exchange area (practical design limits are considered).

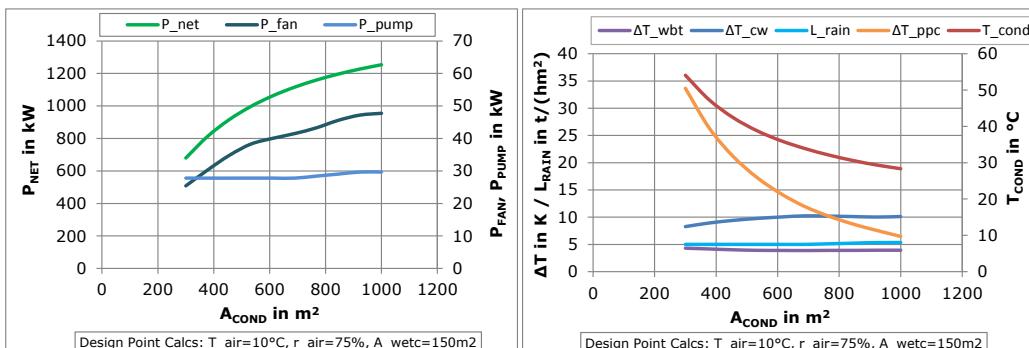


Figure 5: Design values vs. condenser heat exchange area of a wet-cooled ORC with maximum net power at given ambient conditions and given wet cooling tower base area (practical design limits are considered).

3.2 Cooling System Operation

3.2.1 Boundary conditions

In the following paragraphs, main aspects of annual cooling system operation will be discussed for the example of a specified ORC using the boundary conditions defined in section 3.1.1 and a design condensation temperature of 30°C . The ORC data are shown in Tab. 1. A theoretically possible operation performance of this ORC for changing condensation temperatures is shown in Fig. 6 assuming that feed pump and especially turbine would operate with constant efficiencies.

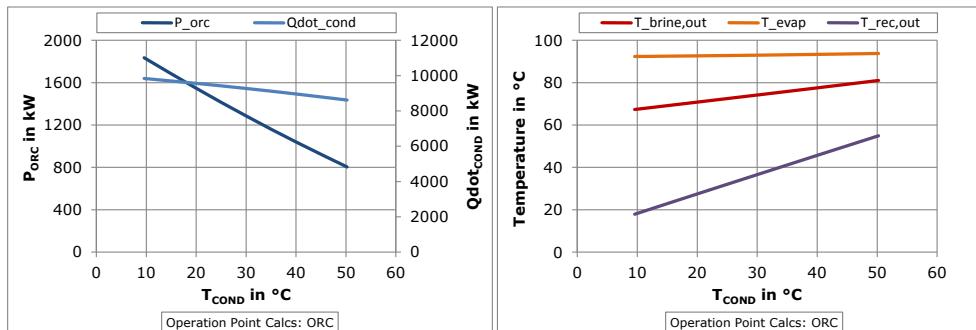
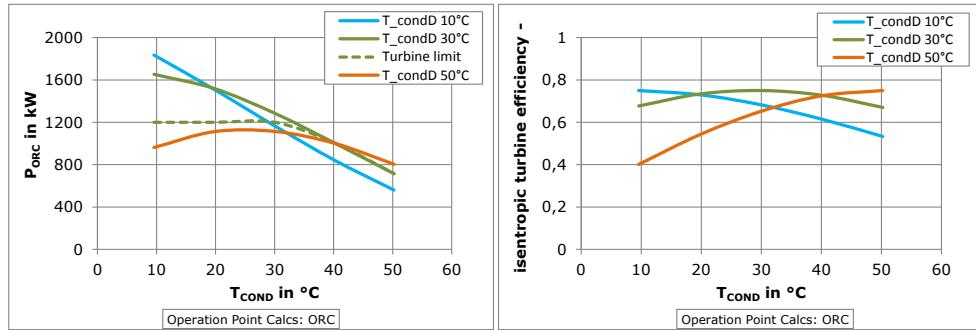
It can be seen that power output of the ORC decreases linearly with about 27 kW/K for ascending condensation temperature T_{COND} (Fig. 6, left). Whereas the evaporation temperature only negligibly increases with increasing T_{COND} , it should be noted that the brine outlet temperature is changing with varying condensation temperature. Depending on the chemical composition of the brine this aspect might be important since scaling might occur at lower temperatures.

However, constant turbine efficiency during operation is not a realistic scenario. Different turbine operational behavior has been reported in the recent years (e.g. Ghasemi 2013, Erhart 2013, Manente 2013). Since a detailed discussion of this matter would go beyond the scope of this paper, the potential influence of real turbine behavior is estimated using the approach presented by Ghasemi (2013), in which the isentropic turbine efficiency is calculated as function of maximum specific enthalpy drop and maximum volumetric flow rate. The performance of the example ORC assuming different turbine behavior is shown in Fig. 7.

For the discussion of the cooling system operation, the performance of the ORC with a turbine design for a condensation temperature of 30°C and limited turbine output (Fig. 7, left) will be considered.

Table 1: Design values of the example ORC

Variable	Design value
Evaporation temperature in °C	92.97
Working fluid mass in kg/s	23.45
Brine outlet temperature in °C	74.16
Turbine power in kW _{el}	1341.1
Feed pump power in kW _{el}	54.9
Waste heat after recuperator in kW _{th}	9270.8
Temperature after recuperator in °C	36.58
Heat exchange area evaporator in m ²	255.9
Number of tubes in the evaporator -	212
Heat exchange area recuperator in m ²	408.4
Number of tubes in the recuperator	536
Total flow area turbine nozzles in mm ²	6653
P_{ORC} in kW_{el}	1286.2

**Figure 6: Operating values vs. condensation temperature of the example ORC assuming constant pump and turbine efficiency.****Figure 7: Operating values vs. condensation temperature of the example ORC assuming variable turbine efficiency and different turbine design points.**

3.3.2 Dry Cooling

In the following, the dry cooler specified according to Tab. 2 and in section 3.2.2 is used. In Fig. 8, the annual net power output of the ORC operated with this dry cooler is shown assuming constant speed and variable speed fan operation.

Table 2: Design values of the example dry cooler

Variable	Design value
Dry cooler base area in m ²	300
Number of tubes	3735
Fan power in kW	90.8

For constant fan operation, it can be seen that the net power increases almost linearly with about 40 kW/K for decreasing ambient temperatures until the maximum turbine capacity is reached (Fig. 8, top left). With constant fan operation, T_{COND} behaves proportionally with T_{AIR} (Fig. 8, bottom right).

If then fan speed is varied in order to reach the maximum net power, a higher net power can be achieved especially for colder ambient temperatures (Fig. 8, top right) due to the reduction of the fan speed and hence fan power (Fig. 8, bottom left). A slightly

higher power output could also be achieved for warmer ambient temperatures by increasing the fan speed. The improvement potential using variable fan speed hence mainly depends on the turbine behavior and limitation of the power output in case colder condensation temperatures cannot be converted to a higher output of the turbine. For a reliable estimation of the improvement potential a more detailed model reflecting the turbine operation and also real fan efficiencies would be needed.

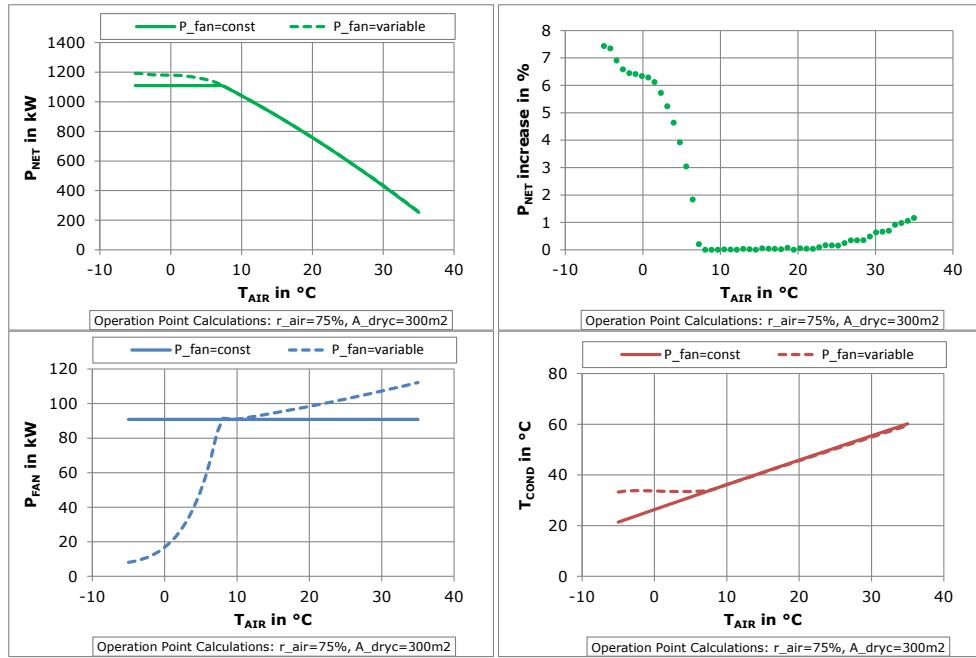


Figure 8: Operating values vs. ambient temperature of the example dry-cooled ORC for constant and variable fan-speed operation.

3.3.3 Wet Cooling

Based on the definitions in section 3.2.3, the remaining design values are summarized in Tab. 3. For this set-up, Fig. 9 shows the annual net power output of the power plant assuming constant speed and variable speed fan operation. Constant speed fan operation thereby refers to ambient temperatures where there is no risk of freezing of the cooling tower. For lower ambient temperatures, the fan speed will be reduced in order to keep a minimum cold water temperature of 5 °C. The cooling water pump is operated with constant speed, since design rain density is already allowed minimum and higher rain density would not be advantageous.

Table 3: Design values of the example wet cooling

Variable	Design value
Wet cooling tower base area in m ²	150
Cooling water load in t/(h m ²)	5
Fan power in kW	51.5
Condenser heat exchange area in m ²	700
Number of tubes in the condenser	559
Pump power in kW	17.8

For constant fan operation, it can be seen that the net power increases almost linearly with about 33 kW/K for decreasing ambient temperature T_{AIR} until the maximum turbine capacity reached (Fig. 9, top left). Below an ambient temperature of 0°C, the fan speed is reduced in order to keep the cold water temperature at 5°C so that the net power output increases a little.

If then fan speed is varied in order to reach the maximum net power, a higher net power can be achieved. For both, low and high ambient temperatures, a reduction of the fan speed and hence fan power would be advisable (Fig. 9, bottom left). Since the fan should be variable speed in any case to prevent freezing, a controlled operation of the fan should be considered to improve the annual net power output. For a reliable estimation of the improvement potential a more detailed model reflecting the turbine operation and also real fan efficiency would be needed.

3.3 Cooling System Design Point Selection

Based on the results presented in the previous sections, the question about the influence of the design point of the cooling system arises. For two example cases based on the definitions in section 3.2 and assuming a given cooling system base area, the ambient temperature will now be varied.

In Fig. 10 it is observed that the design net power decreases linearly with higher design ambient temperatures. The optimum fan power and optimum ITD_{eff} for reaching the maximum net power output also decrease with warmer ambient conditions.

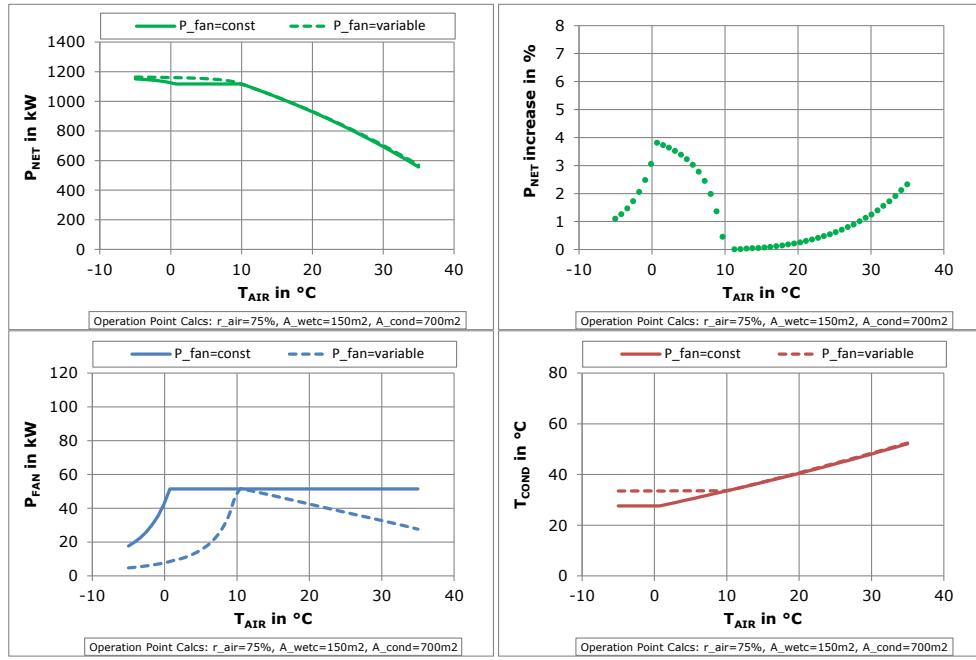


Figure 9: Operating values vs. ambient temperature of the example wet-cooled ORC for constant and variable fan-speed operation.

In Fig. 11 the influence of the ambient temperature on the cooling tower design is shown. Due to the limit of the rain density L_{RAIN} , the pumping power is not affected by changing conditions. The fan power instead decreases with higher ambient temperatures, such as observed for the dry cooler. Comparable to the optimum ITD_{eff} of the dry cooler, the optimum wet bulb temperature approach ΔT_{WBT} also decreases with higher T_{AIR} .

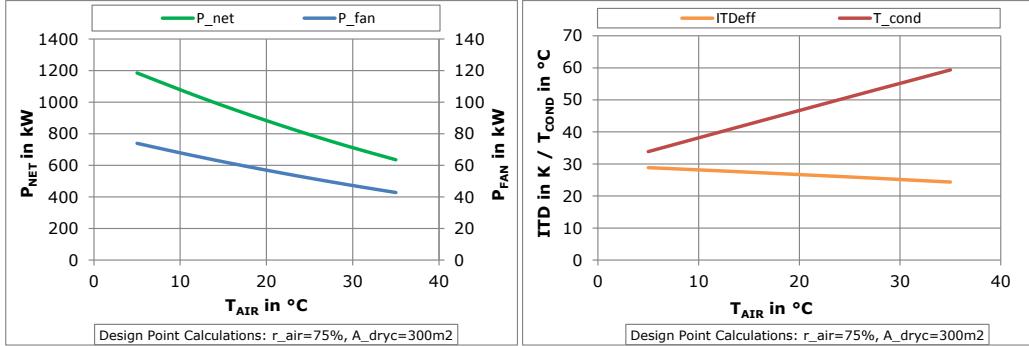


Figure 10: Design values vs. ambient temperature of a dry-cooled ORC with maximum net power at given ambient conditions and dry cooler base area.

For dry cooling and wet cooling it can hence be stated that, for a given cooling system base area, a lower design point for the cooling system will lead to a higher design net power which is achieved with a larger fan capacity, a larger turbine and also larger heat exchangers in the ORC. With such a set-up, also a higher annual net power output should be achieved since the plant behavior discussed in section 3.2 (net power decrease for increasing ambient temperatures with 33 and 40 kW/K, respectively) is also applicable. It should, however, be considered that a larger power plant that is designed for lower ambient conditions will result in higher investment for the ORC and the cooling system fans and lower annual full load hours.

4. CONCLUSIONS

By the means of numerical modelling main aspects for the improvement of the annual net power output of an ORC with dry cooling (air-cooled condenser) and wet cooling (water cooled condenser and wet cooling tower) have been discussed. Three main aspects can be summarized:

- **Improving of the design net power by means of an optimized cooling system design.** For both, dry and wet cooling, a higher net power output is first of all achieved by the size of the cooling system. However, it could be seen that the increase in net power decreases with increasing size. In case of the dry cooler, the size was referred to the base area of the air-cooled condenser. In case of wet cooling, component size was referred to the base area of the cooling tower and the heat exchange area of the condenser. Considering practical limits for the design (such as minimum rain density for a wet cooling tower and minimum air inlet flow velocity), the cooling system size has a limit.

The optimum design parameters which influence the condensation temperature and the auxiliary power consumption of the cooling system depend on the size of the cooling system. In case of the dry cooler, the initial temperature difference has been identified as important design parameter. In case of the wet cooling, cooling tower fill type and height, wet bulb temperature approach, cooling water temperature spread and water load should be considered for the optimization.

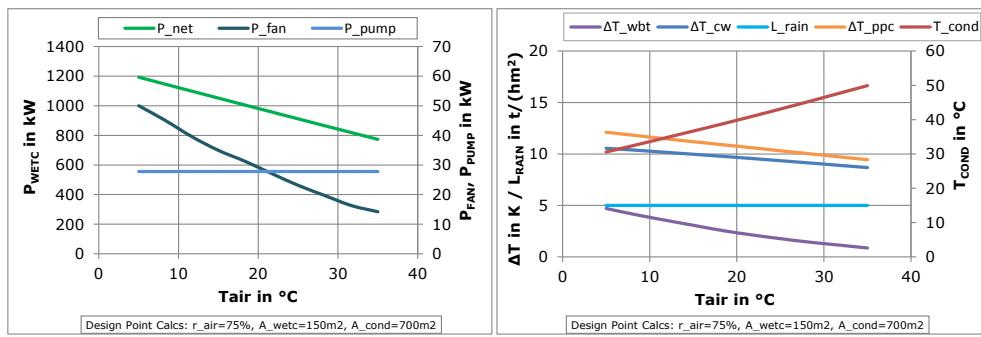


Figure 11: Design values vs. ambient temperature of a wet-cooled ORC with maximum net power at given ambient conditions, wet cooling tower base area and condenser heat exchange area.

- **Improving the annual net power by means of an optimized cooling system operation.** For both, dry and wet cooling, it was shown that a variable speed fan operation can increase the annual net power output. However, a reliable estimation of the improvement potential is site specific and needs a more detailed model reflecting the turbine operation and also real fan efficiencies.

In case of the dry cooler, a higher net power can be achieved for colder ambient temperatures with the reduction of the fan speed. A slightly higher power output could also be achieved for warmer ambient temperatures by increasing the fan speed. The improvement potential, however mainly depends on the turbine behavior and limitation of the power output in case colder condensation temperatures cannot be converted to a higher output of the turbine. The improvement of the annual net power output might hence be limited only to several sites.

In case of the wet cooling, a reduction of the fan speed could increase the net power for both, low and high ambient temperatures. Since the fan should be variable speed in any case to prevent freezing, a controlled operation of the fan should be considered to improve the annual net power output.

- **Improving design and annual net power by means of design point selection.** For dry cooling and wet cooling it can be stated that, for a given cooling system size, a lower design point for the cooling system will lead to both, a higher design net power and a higher annual net power output. The higher net power is thereby achieved with a larger fan capacity, a larger turbine and also larger heat exchangers in the ORC. It should, however, be considered that a larger power plant that is designed for lower ambient conditions will result in higher investment for the ORC and the cooling system fans and lower annual full load hours.

Another important aspect is the design point of the turbine due to the influence of the isentropic efficiency on the plant performance.

In conclusion, power conversion cycle and cooling system should be designed together and not individually especially referring to those design parameters which effect the conversion and the condensation/cooling process. Since the cooling system is in many cases the largest surface component of geothermal binary power plants and cost intensive, a reasonable cooling system size can only be determined by evaluating the resulting net power increase. Reduced numerical models representing the most important design parameters could be used to improve the cooling system design. More detailed models, in contrast, might be helpful for the optimization of the cooling system operation.

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