

Air-Cooled Binary Rankine Cycle Performance With Varying Ambient Temperature

Josephine Varney¹ and Nigel Bean²

¹South Australian Centre for Geothermal Energy Research & School of Mathematical Sciences, University of Adelaide, Adelaide, SA

²School of Mathematical Sciences, University of Adelaide, Adelaide, SA
josephine.varney@adelaide.edu.au • nigel.bean@adelaide.edu.au

Keywords

Air-cooling, ambient temperature, binary Rankine cycle, supercritical heat exchanger

ABSTRACT

Air-cooling is necessary for geothermal plays in dry areas and ambient air temperature significantly affects the power output of air-cooled thermal power plants. Hence, a method for determining the effect of ambient air temperature on subcritical and supercritical, air-cooled binary Rankine cycles using moderate temperature geothermal fluid and various working fluids is presented. Part of this method, includes a method for maximizing working fluid flow from a supercritical heat exchanger. In the example presented isobutane is used as the working fluid, while the geothermal fluid temperature and flowrate are set at 150°C and 126kg/s. Results of this analysis show that for every 14°C increase in ambient air temperature, above the ambient temperature used for design purposes, there is ~20% loss in brine efficiency; while conversely, there is no gain in brine efficiency for any drop in ambient air temperature below the ambient air temperature used for design purposes. Using the ambient air temperature distribution from Leigh Creek, Australia, this analysis shows that an optimally designed plant produces 6% more energy annually than a plant designed using the mean ambient temperature.

Introduction

In Australia, Geodynamics, Petrathern and Panax are the three companies closest to producing electricity from enhanced geothermal systems (EGSs). All three companies have their EGS sites located in very dry areas of South Australia (Australia's driest state). Due to this lack of water, all three companies are planning to use *air-cooling* in their power plants to produce electricity.

Operators of air-cooled power plants [7] are already aware of the significant impact that ambient air temperature has on the power output of their plants. This was re-inforced recently in a

study by Wendt and Mines [8] which showed that ambient air temperature could reduce the maximum power output by up to 35% and 60%, for theoretical air-cooled binary Rankine cycle plants located in Grand Junction, Colorado and Houston, Texas respectively.

Created by Entingh (with help from Mines et al) [4] GETEM (Geothermal Electricity Technology Evaluation Model) is the most well known and widely referenced geothermal modeling system. GETEM is exceptionally broad ranging and allows for the inclusion of many different variables; however, it currently assumes water cooling to 10°C, so doesn't allow for air-cooling and the associated effect of ambient air temperature.

Since ambient air temperature has a significant impact on the power output of air-cooled power plants, we want to understand, more specifically, how this will affect the performance of EGS and hot sedimentary aquifer (HSA) plays in Australia. From this, we also gain some insights into optimal design for air-cooled Rankine cycles.

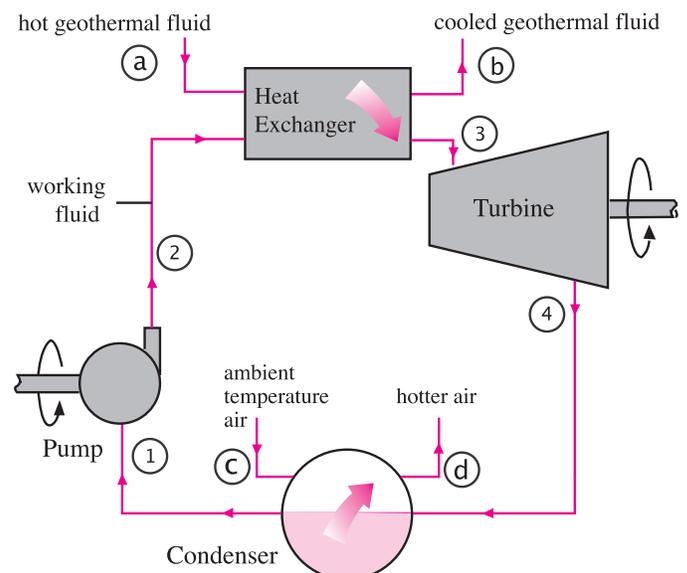


Figure 1. Schematic for an air-cooled binary Rankine cycle.

Method

The majority of Australia's 366 existing geothermal exploration licences are located in arid to semi-arid areas of the continent, targeting relatively low enthalpy EGS and HSA targets. In this context, it is likely that binary Rankine cycles and air-cooling will be the most viable technologies for electricity production from many projects. Hence, we chose, in this paper, to model an air-cooled binary Rankine cycle plant.

A binary Rankine cycle plant has two separate circulating fluids: the *geothermal fluid* which brings the heat from deep in the earth to the surface, and the *working fluid* which takes heat from the geothermal fluid and uses this heat to generate electricity (see Figure 1).

Although not commonly mentioned, all Rankine cycles have another fluid, the *cooling fluid*; this is the fluid which removes heat from the vaporized working fluid, allowing it to condense and then be pumped back up to pressure. Generally, this cooling fluid is water because it has excellent thermodynamic properties for cooling, it is stable, abundant and cheap (which explains why 99% of the power plants in the USA use water cooling [5, p. 12]). However, where water is scarce, ambient air is used for cooling because it is also stable, abundant and cheap (although its thermodynamic properties, for cooling purposes, are not as good as water).

The working fluid in a Rankine cycle goes through four separate *processes*, changing the fluid into four different *states*. At State 1 the working fluid is a low pressure, low temperature saturated liquid, it is then pumped up to high pressure liquid (State 2), and then heated to become a high pressure vapor (State 3). Pressure and temperature, of the working fluid, drop across the turbine (to produce mechanical energy) to leave a low temperature, low pressure vapor in State 4. This vapor is then condensed to become the low pressure, low temperature saturated liquid of State 1, and the cycle starts again.

An *ideal* Rankine cycle assumes that the pump and the turbine operate isentropically, and that the condenser and the heat exchanger operate at constant pressure. Determining the power output from an ideal Rankine cycle is well known and widely covered in textbooks [1, 9, 2] so we will not go into it in detail here. Simply, if the following are known:

- (i) temperature of the saturated liquid at State 1,
- (ii) temperature and pressure of the vapor at State 3,
- (iii) working fluid mass flowrate,

the net-power generated by the ideal Rankine cycle can be determined.

Determining the Temperature at State 1

To maximize the power output from a Rankine cycle plant, it is necessary to have the minimum possible temperature at State 1. For an air-cooled Rankine cycle plant the minimum temperature at State 1, and hence the chosen temperature for State 1, is given by

$$\begin{aligned} T_1^{\text{WF}} &= T_c^{\text{CF}} + \Delta T_{\text{PP-C}}, \\ &= T_{\text{Amb}} + \Delta T_{\text{PP-C}}. \end{aligned}$$

This equation assumes there is no restriction on the mass flowrate of air or size of the condenser. Given the abundance of air and the remote location of the Australian plants this is a reasonable assumption.

Determining the Temperature and Pressure at State 3

For a given T_1 there are many feasible turbine-inlet (or State 3) temperatures and pressures. Determining the turbine-inlet temperature and pressure which generates the maximum net-power is not a trivial exercise and is discussed later. However, as a first step in the optimization process, a turbine-inlet temperature and pressure are chosen from a feasible range. The feasible range ensures: the fluid is completely vaporised (or a supercritical fluid), that T_3^{WF} and P_3^{WF} are within the working fluids operating range and that both T_3^{WF} and P_3^{WF} are greater than T_4^{WF} and P_4^{WF} respectively. Further, we required the turbine to operate completely in the 'dry' region.

Determining the Working Fluid Mass Flowrate

To generate maximum power using a Rankine cycle with a given turbine-inlet temperature and pressure, the maximum working fluid flowrate must be used. In a binary Rankine cycle the working fluid flowrate is limited by the heat exchanger, so this step must be maximized to generate maximum power.

In order to function, a heat exchanger needs two things:

1. Heat Balance

In an ideal heat exchanger, all the heat from the hot fluid is absorbed by the cold fluid. When a heat exchanger operates at constant pressure, the heat balance equation simplifies to

$$\dot{m}^{\text{hot}} \Delta h^{\text{hot}} = \dot{m}^{\text{cold}} \Delta h^{\text{cold}}, \quad (1)$$

for any section of the heat exchanger.

2. Driving Force

The place in a heat exchanger where the two fluids have the minimum temperature difference is called the pinch point [2, p.162]. When designing a heat exchanger the minimum temperature difference at the pinch point is set (usually between 5-10°C). The hot fluid must then always be hotter than the cold fluid plus the minimum temperature difference at the pinch point, throughout the entire length of the heat exchanger. So, for any point x along the length of the heat exchanger,

$$T^{\text{hot}}(x) \geq T^{\text{cold}}(x) + \Delta T_{\text{PP-HX}} \quad (2)$$

To achieve the maximum flowrate in a heat exchanger, the position of the pinch point (along the length of the heat exchanger) must be chosen optimally. The maximum working fluid flowrate is then calculated using this optimal pinch point, T_3^{WF} , T_a^{GThF} and \dot{m}^{GThF} as inputs into equation (1).

It is well known that the optimal position of the pinch point, in a heat exchanger in a *subcritical* binary Rankine cycle, must be at either the working fluid vaporization point or at either end

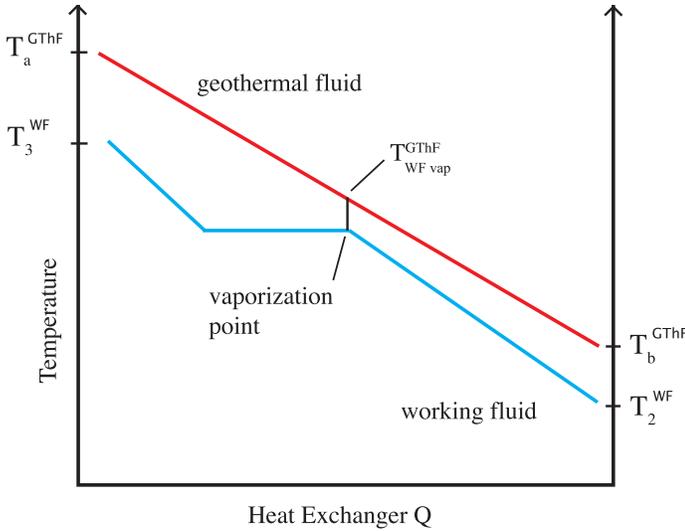


Figure 2. Subcritical heat exchanger schematic.

point of the heat exchanger [2, p.162] (see Figure 2). Hence, determining the maximum working fluid flowrate is fairly straight forward in this case.

In a heat exchanger in a *supercritical* binary Rankine cycle, the working fluid (as shown in Figure 3) has a gentle curve, reflecting a constantly changing heat capacity. This means that there is no obvious choice for the optimal position for the pinch point, along the length of the heat exchanger

We choose to address this problem in the following way.

1. Given the temperature and pressure information for States 2, 3 and a, and \dot{m}^{GThF} , it is possible, using equation (1), to write the working fluid flowrate as a function simply of the cold geothermal fluid temperature,

$$\dot{m}^{WF} = f(T_b^{GThF}).$$

2. However, calculating the working fluid flowrate using equation (1), without knowing (or using) the pinch point, means that we cannot be sure that equation (2) holds for the entire heat exchanger. So, for any given T_b^{GThF} , to ensure that equation (2) holds for the entire heat exchanger, the following method is used:

- a) Use equation (1) to calculate the working fluid flowrate, as follows

$$\dot{m}^{WF} = \dot{m}^{GThF} \frac{(h_a^{GThF} - h_b^{GThF})}{(h_3^{WF} - h_2^{WF})}. \quad (3)$$

- b) Divide the heat exchanger into i segments of equal heat balance. Given that the \dot{m}^{WF} was calculated using equation (3), we know that the heat balance equation (equation (1)) holds for each segment with

$$\Delta h_{\text{segment}}^{GThF} = \frac{h_a^{GThF} - h_b^{GThF}}{i},$$

$$\Delta h_{\text{segment}}^{WF} = \frac{h_3^{WF} - h_2^{WF}}{i}.$$

- c) Using the fixed working fluid pressures in the heat exchanger (P_a^{GThF} and P_3^{WF}), and the enthalpy at the beginning and end of each segment, create segmented approximations of the temperature profiles of the geothermal and working fluids.

- d) Using these temperature profiles as inputs into equation (2), determine if equation (2) holds for all x , and hence if the heat exchanger is feasible.

3. By setting all the infeasible working fluid flowrates to a negative number (say -1), we create a new function, equation (4), and the maximum working fluid flowrate is the maximum of this function:

$$\dot{m}^{WF} = \begin{cases} f(T_b^{GThF}), & \text{if heat exchanger is feasible,} \\ -1, & \text{otherwise.} \end{cases} \quad (4)$$

4. We can also infer that the maximum working fluid flowrate must lie somewhere in the range mapped by,

$$T_b^{GThF} \in [T_2^{WF} + \Delta T_{PP_HX}, T_a^{GThF}].$$

5. Finding the maximum of this function, is in fact quite simple, as it is one dimensional and unimodal, and the domain of the function is bounded. This can be done using any 1-dimensional constrained optimization routine.

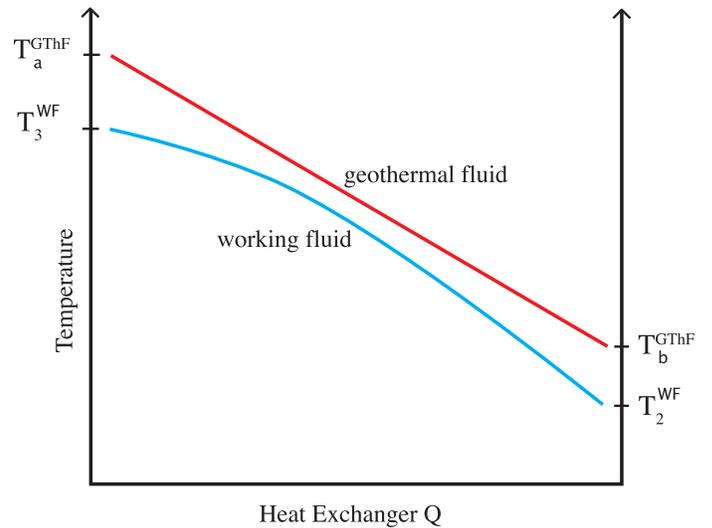


Figure 3. Supercritical heat exchanger schematic.

The Optimization Process

Figure 4 outlines the optimization process we use to maximize the net-power from an air-cooled binary Rankine cycle plant. For a given set of, what we have called, *plant conditions* (geothermal fluid temperature, pressure and flowrate, ambient temperature and choice of working fluid) we iteratively, found the State 3 temperature and pressure (from within a specified feasible range) that produced the maximum net-power.

In essence, we have created a function for net-power using State 3 temperature and pressure as the only variables,

$Power = g(T_3^{WF}, p_3^{WF})$. This means to find the maximum net-power we need to solve a 2D optimization problem, in which is embedded the maximum working fluid flowrate calculation. This can be done using any standard constrained optimization routine.

In order to calculate the power generated from an ideal Rankine cycle it is necessary to make a number of assumptions related to design efficiencies, pinch points etc. The values we use are listed here:

- (i) Isentropic turbine efficiency - 85%
- (ii) Mechanical turbine efficiency - 95%
- (iii) Pump efficiency - 70%
- (iv) Condenser pinch point - 7.5°C
- (v) Heat exchanger pinch point - 5°C

We also choose to require all cycles to be *dry*, that is that no expansion (in the turbine) occurs in the two-phase region.

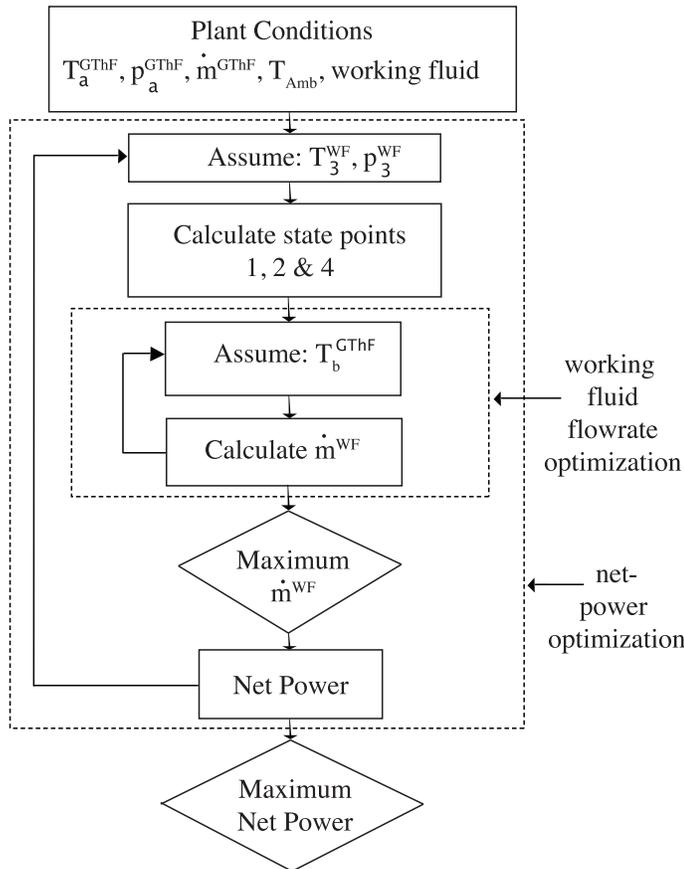


Figure 4. Flowchart of the optimization procedure.

Determining the Effect of Ambient Air Temperature

In order to determine the effect of ambient air temperature, it is important to realize that any site will have only one power plant. The power plant will be built to run optimally for a *given set of plant conditions*. This means that when the ambient air temperature varies, the plant will run *off-design*.

In our modeling, we vary the ambient air temperature but keep all other plant conditions constant. In response to the varying ambient air temperature, we make the following assumptions for a plant running in these off-design conditions:

- (i) State 3 temperature and pressure, and working fluid flowrate remain at design conditions.
- (ii) If the actual ambient air temperature is greater than the design ambient air temperature, then the turbine back-pressure (p_4^{WF}) is increased to ensure that the working fluid is a saturated liquid at State 1. The net-power is then recalculated. This is required in practice because the fluid entering the pump must be a liquid for the pump to work properly.
- (iii) If the actual ambient temperature is lower than the design ambient temperature, then the turbine back-pressure (p_4^{WF}) is kept at the design back-pressure. This is because lowering the turbine back-pressure at State 4, would result in a lower temperature at State 2, which, given the design of the heat exchanger, would make it impossible to achieve the design temperature at State 3.

Performance Measures

Power plant performance is often judged using thermal efficiency, where thermal efficiency is

$$\eta_{th} = \frac{P_{turbine} - P_{pump}}{\dot{Q}_{in}} \quad (5)$$

For traditional coal-fired power plants this is a useful measure of performance, as the top line reflects revenue and the bottom line reflects the cost of coal, which is the largest portion of variable operations and maintenance costs in a coal fired power plant [3, p. 75].

In EGS and HSA power plants, \dot{Q}_{in} is the amount of heat withdrawn from the geothermal fluid to generate electricity. However, most of the costs in EGS and HSA plays are directly linked to the flowrate of the geothermal fluid, not how much heat can subsequently be removed from it to generate electricity.

Mines [6] often uses, what he terms, *brine efficiency* to reflect the performance of geothermal power plants, where brine efficiency is defined as

$$\eta_{brine} = \frac{P_{turbine} - P_{pump}}{\dot{m}^{GThF}}$$

In our opinion, this is a more useful measure of performance for EGS and HSA plays for three reasons:

1. Capital cost is directly linked to the number of wells drilled, and each well-pair drilled generates a geothermal fluid flowrate.
2. The parasitic power required to run these plants, is predicted to be the largest portion of variable operations and maintenance costs, which again, is linked to geothermal fluid flowrate.
3. It removes the need to assume a geothermal fluid flowrate, one of the largest unknowns in these plays at the moment.

Validation

To check the assumptions we made for a plant running in off-design conditions, we compared our calculated data to real data from the Mammoth Pacific Plant, California, USA [6]. The Mammoth plant data was used because it was the only publicly available data linking ambient air temperature to plant power output, from an air-cooled binary geothermal power plant using isobutane that we could find. Also, please note, we have used non-SI units, in this section only, for ease of comparison with Mammoth plant data.

At Mammoth, the geothermal fluid temperature ranges from 300-350°F, and their air-cooled binary system operates using isobutane as its working fluid. Figure 5a shows brine efficiency versus ambient temperature. The grey triangles show results from the plant's normal operating conditions (in 2000), the red squares show results from, what we will call, *Mines wet-cycle trial* [6]. In his wet-cycle trial Mines changed the state at State 3 (i.e. the T_3^{WF} and P_3^{WF}), so that the isobutane was no longer completely dry in the turbine. Since we have assumed a dry turbine, the grey triangles are the data we need to compare with.

It can be seen from Figure 5, that on a plot of brine efficiency versus ambient air temperature, our data qualitatively agrees with

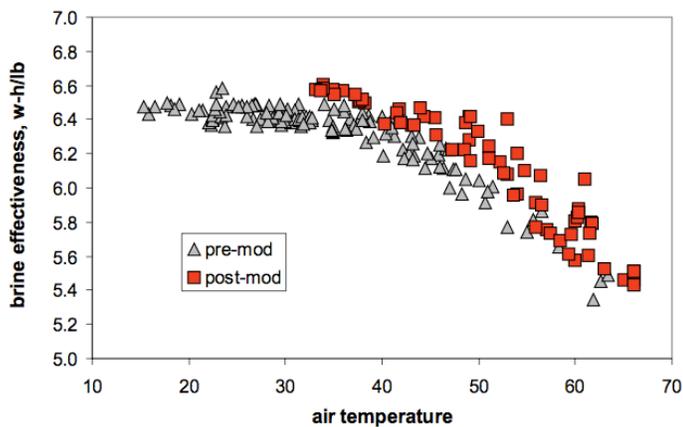


Figure 5a. Mammoth Plant Data [6].

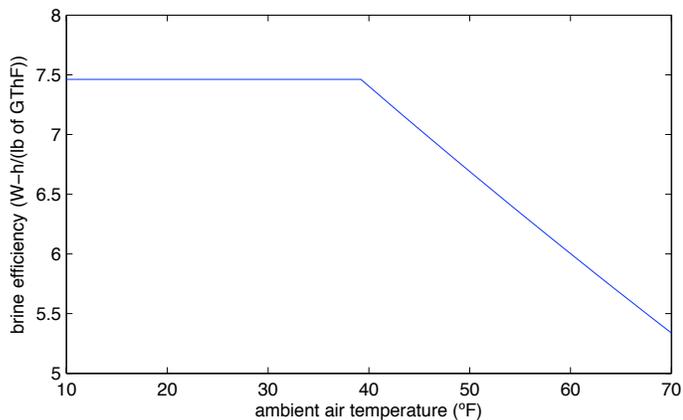


Figure 5b. Effect of ambient air temperature on brine efficiency (using: $T_a^{GThF} = 300^\circ\text{F}$, ambient air temperature for design purposes of 38°F , and isobutane as a working fluid).

the real-world Mammoth data. Quantitatively, our data somewhat over-estimates the Mammoth data, but this is to be expected given that we have assumed an ideal plant. Our data also over estimates the effect of ambient temperature; the Mammoth plant loses ~17% over 25°F (from 38°F to 63°F), where our data loses 23% over the same range. This indicates that our off design assumptions are a little too harsh and/or that the Mammoth plant use some strategies, in hot weather, to mitigate the loss of power generation. For example, turning up the fans and/or spraying cooling water to aid fan cooling. These methods have the effect of reducing ΔT_{PP-C} (where we assumed this value is constant).

In their modeling, Wendt and Mines [8] show brine efficiency dropping ~20% (over 25°F) above the plant design point. However, their modeling with temperatures colder than the design point differs significantly from our modeling, due to their inclusion of a variable nozzle design in the turbine.

We believe our model is sufficiently close, both qualitatively and quantitatively, to real world data to provide meaningful insights into the effect of ambient air temperature on the performance of air-cooled, binary Rankine cycle power plants.

Results

The flowchart in Figure 4 shows that for a given set of plant conditions (\dot{m}^{GThF} , T_a^{GThF} , P_a^{GThF} and T_{Amb}) it is possible to determine the maximum power output for an air-cooled binary Rankine cycle. To examine the effect of ambient air temperature, the plant conditions of \dot{m}^{GThF} , T_a^{GThF} and P_a^{GThF} are considered constant in this section. However, ambient air temperature is considered to vary, not only in range but also in frequency, as ambient air temperature does in the real world (see Figure 6).

Since power plants are built to run optimally for a given set of plant conditions, we will call these conditions the *plant design conditions*. In particular, if a plant is designed to run optimally for a given ambient temperature, we will call this temperature, the 'design ambient air temperature'.

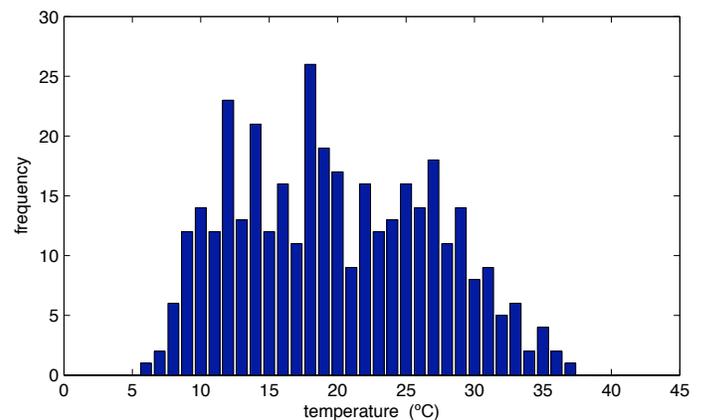


Figure 6. Histogram of 2011 Leigh Creek daily maximum temperature data.

Figure 5 confirms that ambient air temperature has a significant impact on the power output of air-cooled binary Rankine cycle plants. More specifically, according to our modeling, these plants lose around 20% brine efficiency for every 14°C increase

in actual ambient air temperature, above the design ambient air temperature. Further, when the actual ambient air temperature is colder than the design ambient air temperature, there is no increase in brine efficiency.

Figure 7 shows, what we call here the ‘optimal line’, the maximum brine efficiency for all ambient air temperatures (i.e. this line assumes a new plant was designed for each point on this line). The optimal line is compared, in this figure, to two individual plant designs, with design ambient air temperatures of 4°C and 20°C. Clearly, the brine efficiency for a single plant, will meet the ‘optimal line’ at the design ambient air temperature. From Figure 7, it is also easy to see that a plant loses significantly more, in comparison to the optimal line, at temperatures which are colder than the design ambient air temperature, than it does from temperatures which are hotter than this temperature.

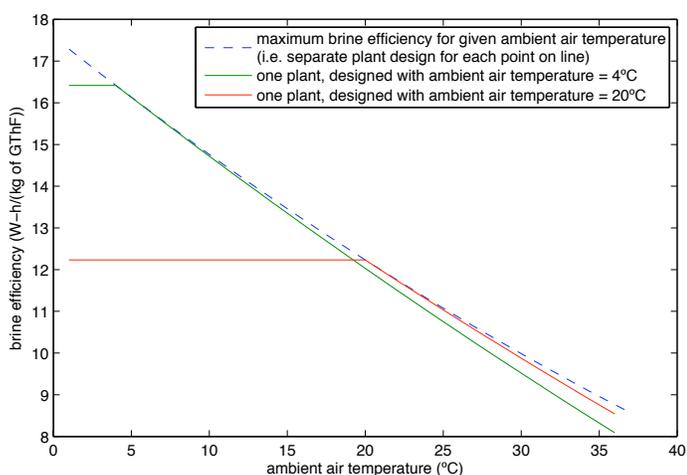


Figure 7. Effect of ambient air temperature and ambient air temperature used for design purposes, on brine efficiency (using: $T_a^{GThF} = 150^\circ\text{C}$ and isobutane as a working fluid).

However, Figure 7 doesn't answer the question, ‘What is the best plant design for a specific temperature distribution?’ It is easy to assume, from a quick look at Figure 7, that the best design for a plant, considering its ambient temperature distribution, is to choose the coldest ambient air temperature from the distribution and use this as the design ambient air temperature for the plant. However, while this is close, it is not actually the best answer.

For a range of different *design ambient air temperatures*, we calculated the energy produced per day using the maximum daily temperature data from Leigh Creek, then summed this data to give the annual energy production (see Figure 8). From Figure 8 we can see the maximum annual energy production occurs at ~9.5°C, not at the minimum ambient air temperature for Leigh Creek in 2011, 3.5°C.

This can be explained by looking more closely at Figure 7, where the plant with the coldest design ambient air temperature is represented by the green line and the plant with a hotter design ambient air temperature is represented by the red line. It is easy to see that the green line does significantly better at all ambient air temperatures which are colder than the design ambient air temperature of the red line; however, the red line does slightly better than the green line for all temperatures which are hotter than the

design ambient air temperature of the red line. Hence, the optimal design ambient air temperature is unlikely to be at the minimum of the ambient temperature distribution because this value is unlikely to occur very often, and it means that the plant will do worse for all other temperatures in the distribution, compared to a plant with a slightly higher design ambient air temperature.

Figure 8 also shows that choosing the optimum design ambient air temperature of 9.5°C (for the design conditions shown) gives around a 6% increase in annual energy production, over choosing the mean value at Leigh Creek in 2011 (21°C) as the design ambient air temperature.

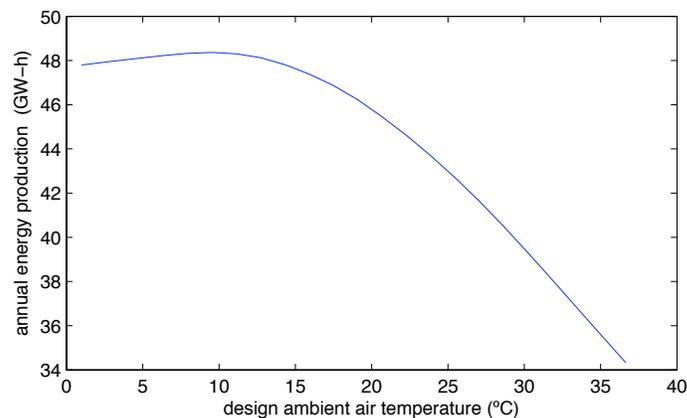


Figure 8. The effect of ambient air temperature used for design purposes on energy output per year (using: $T_a^{GThF} = 150^\circ\text{C}$, $\dot{m}^{GThF} = 125\text{kg/s}$ and Leigh Creek daily temperature data).

Conclusion

Air-cooled binary Rankine cycle plants are significantly and adversely affected by varying ambient air temperature. However, while this loss is fundamentally due to the thermodynamics of the Rankine cycle, considering the temperature distribution and the off-design effects of this distribution can allow for better choice in initial plant design.

This analysis and the data in Figure 5a would suggest that the total energy output from the Mammoth plant could probably be increased if the design ambient air temperature were reduced.

List of Abbreviations

- h enthalpy (J/kg)
- \dot{m} mass flowrate (kg/s)
- p pressure (kPa)
- s entropy (J/(kg·K))
- C_p heat capacity at constant pressure (J/(kg·K))
- P power (W)
- \dot{Q} heat flow per second (J/s)
- T temperature (°C)
- ΔT temperature difference (°C)
- η_{brine} brine efficiency (W-h/kg)
- η_{th} thermal efficiency (dimensionless)

Superscripts

CF	cooling fluid
GthF	Geothermal fluid
cold	cold fluid
hot	hot fluid

Subscripts

1,2,3,4	State 1, 2, 3, or 4
a,b,c,d	State a, b, c, or d
Amb	ambient state
in	heat added to the cycle
PP-C	pinch point in the condenser
PP-HX	pinch point in the heat exchanger

Acknowledgements

Thanks to the South Australian Centre for Geothermal Energy Research for helping to fund this project, and many thanks to Professor Martin Hand and Dr Betina Bendall for their support and input.

References

- [1] Cengel, Y.A., and A.B. Boles, 1989. "Thermodynamics an Engineering Approach." McGraw-Hill, 5th Edition.
- [2] DiPippo, R., 2008. "Geothermal Power Plants: Principals, Applications, Case Studies and Environmental Impact." Elsevier, 2nd Edition.
- [3] U.S. Energy Information Administration (EIA), April 2011. "Annual energy outlook 2011 with projections to 2035." U.S. Energy Information Administration (EIA).
- [4] Entingh, D.J., July 2006. "DOE Geothermal Electricity Technology Evaluation Model (GETEM): Volume 1 – Technical Reference Manual." Department of Energy.
- [5] Stiegel, G.J., A. McNemar, B. Schimmoller, M. Nemeth, J. Murphy, and L. Manfredo, August 2006. "Estimation freshwater needs to meet future thermoelectric generation requirements." National Energy Technology Laboratory, Department of Energy (USA).
- [6] Mines, G., 2000. "Summary of investigations of the modified turbine inlet conditions in a binary power plant." Proceedings of GRC 2000 Annual Meeting.
- [7] Morris, C., April 2012, Personal correspondence.
- [8] Wendt, D., and G. Mines, September 2010. "Interim Report: Air Cooled Condensers for Next Generation Geothermal Power Plants Improved Binary Cycle Performance." Idaho National Laboratory.
- [9] Van Wylen, G.J., and R.E. Sonntag, 1958. "Fundamentals of Classical Thermodynamics." John Wiley and Sons, Inc, 3rd Edition.

