PUBLISHED VERSION

J.J. Varney and N. Bean

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Proceedings of the 2012 Australian Geothermal Energy Conference, 2012 / C. Huddlestone-Holmes, E. Gerner (eds.): pp.218-228

Building a model to investigate the effect of varying ambient air temperature on air-cooled organic Rankine cycle plant performance

Varney, J.J [1] and Bean, N [1]

[1]The University of Adelaide

Josephine.Varney@adelaide.edu.au

Keywords: air-cooling, ambient temperature, binary Rankine Cycle, super-critical 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

Air-cooling is necessary for geothermal plays in the South Australian desert and other dry areas. Ambient air temperature significantly affects the power output of air-cooled thermal power plants, and so a method for quantifying and predicting this effect is needed. This paper presents a method for determining the effect of ambient air temperature on subcritical and supercritical, air-cooled binary Rankine cycle plants. This model is built using basic thermodynamic principals only and does not use or rely on industry standard models such as GETEM or ASPEN. This significantly reduces the number of inherent assumptions and the subsequent complexity, making cause and effect clearer.

Since each site can have only one plant, it can only be optimally designed for one ambient air temperature. Therefore, the plant must run in off-design conditions when the current ambient air temperature is higher or lower than the design ambient temperature. Assuming a geothermal fluid temperature of 150°C, the results of this analysis show that for every 1°C increase in immediate ambient air temperature, above the design ambient temperature, there is ~1.5% loss in brine efficiency. While conversely, there is no gain in brine efficiency if the current ambient air temperature drops below the design ambient air temperature.

Using the ambient air temperature distribution from Leigh Creek, South Australia, further analysis shows that an optimally designed plant produces 6% more energy, annually, than a plant designed for the mean ambient temperature. Similar results are obtained for geothermal fluid temperatures up to 250°C, using temperature distributions from Moomba, Roxby Downs and the Coonawarra.

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).

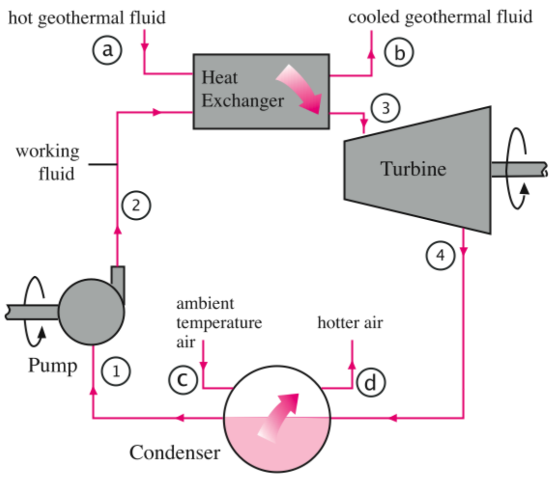


Figure 1: *Schematic for an air-cooled binary Rankine cycle*

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 vapour (State 3). Pressure and temperature, of the working fluid, drop across the turbine (to produce mechanical energy) to leave a low temperature, low pressure vapour in State 4. This vapour 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:

(iv) temperature of the saturated liquid at State 1,

(v) temperature and pressure of the vapour at State 3,

(vi) 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

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

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 optimisation 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} are p_3^{WF} are within the working fluids operating range and that both T_3^{WF} are p_3^{WF} are greater than T_4^{WF} are 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

   $$m^{hot}\Delta h^{hot} = m^{cold}\Delta h^{cold} \qquad (1)$$

   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,

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

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 m^{GThF} as inputs into equation (1).

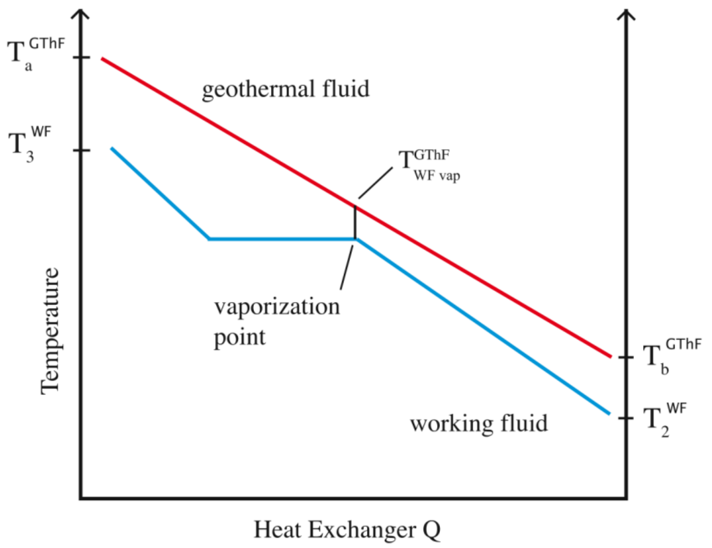


Figure 2: *Subcritical heat exchanger schematic*

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 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 m^{GThF} , it is possible, using equation (1), to write the working fluid flowrate as a function simply of the cold geothermal fluid temperature,

   $$m^{WF} = f\left(T_b^{GThF}\right)$$

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

   $$m^{WF} = m^{GThF}\frac{\left(h_a^{GThF} - h_b^{GThF}\right)}{\left(h_3^{WF} - h_2^{WF}\right)} \qquad (1)$$

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

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

   $$\Delta w_{segment}^{WF} = \frac{h_3^{WF} - h_2^{WF}}{i}$$

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:

$$m^{WF} = f\left(T_b^{GThF}\right) \quad \text{If heat exchanger is feasible,}$$

$$-1 \text{ otherwise.} \qquad (2)$$

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

$$T_b^{GThF} \in \left[T_2^{WF} + \Delta T_{PP\_HX}, T_a^{GThF}\right]$$

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 optimisation routine.

However, great care must be taken with the precision of the optimisation step, because small inaccuracies in m^{WF} are amplified in the power surface and results in significantly jagged (non-smooth) surface, which is inaccurate and very difficult to optimise.

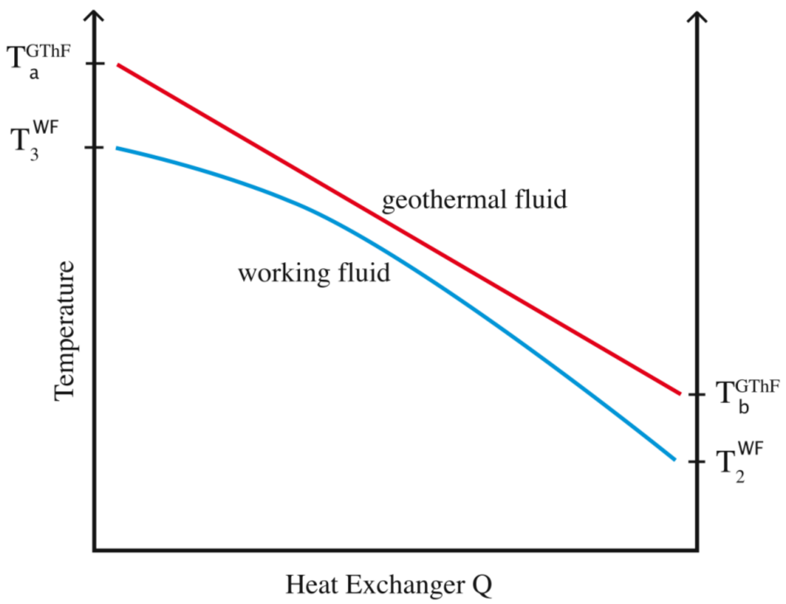


Figure 3: *Supercritical heat exchanger schematic*

The optimisation process

Figure 4 outlines the optimisation 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 entropy as the only variables, $Power = g\left(T_3^{WF}, s_3^{WF}\right)$, this means to find the maximum net-power we need to solve a 2D optimisation problem, in which is embedded the maximum working fluid flowrate calculation. In order to do this using any standard constrained optimisation routine, we transformed the feasible region from a non-linear region to a linear region.

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.

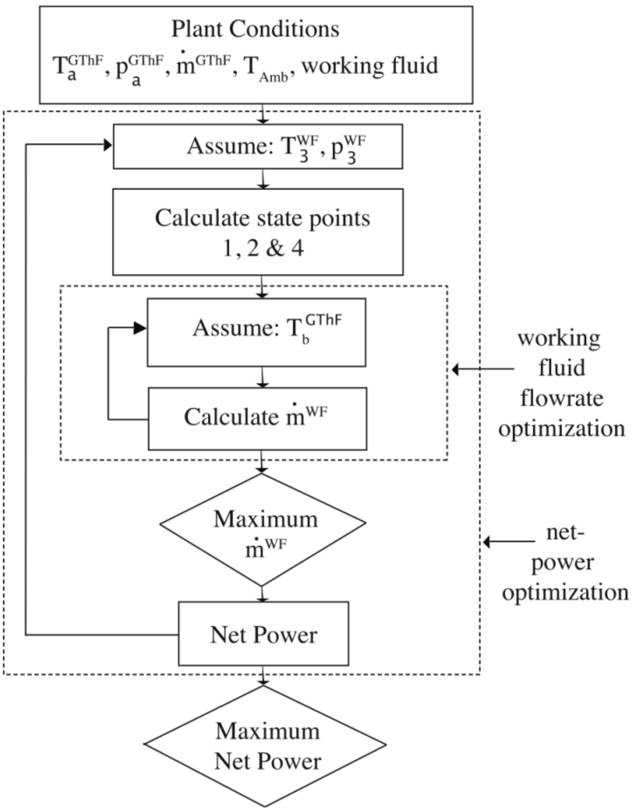


Figure 4: *Flowchart of the optimisation procedure*

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 modelling, 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.

As outlined in Figure 5, to determine the effect of ambient temperature, we first set the plant conditions: the temperature, pressure and flowrate of the geothermal fluid, the design ambient temperature (this is what we are calling ambient temperature used for plant design purposes), and the type of working fluid. From this, we calculate the optimal plant design conditions, in particular, turbine inlet temperature and pressure and the working fluid flowrate. The plant design conditions, together with the off-design assumptions are then used to calculate the power (or energy) produced for varying daily ambient air temperatures.

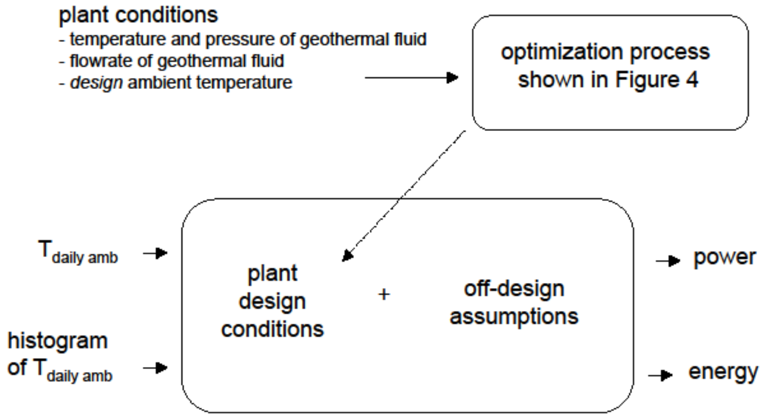


Figure 5: *Flowchart for the off-design calculations*

Performance Measures

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

$$\eta_{th} = \frac{P_{turbine} - P_{pump}}{Q_{in}} \qquad (3)$$

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, 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

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

In our opinion, this is a more useful measure of performance for EGS and HSA plays for five 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.
4. The results scale linearly with geothermal fluid flowrate. So, in relative terms brine efficiency results will be the same as power results which have assumed a specific geothermal fluid flowrate.
5. Power can easily be calculated given flowrate, as follows:

   $$Power = \eta_{brine} \times m^{GThf}$$

   For example, in SI units,

   $$kW = \frac{kJ}{kg} \times \frac{kg}{s}$$

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.

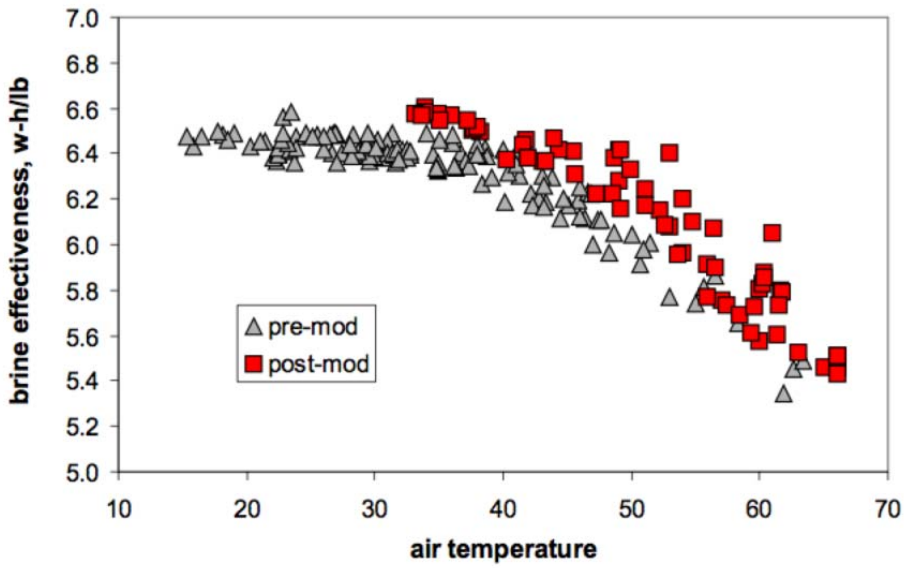
At Mammoth, the geothermal fluid temperature ranges from 300-350°F, and their air-cooled binary system operates using isobutane as its working fluid. Figure 5a shows brine efficiency versus ambient temperature. The grey triangles show results from the plant's normal operating conditions (in 2000), the 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 6, that on a plot of brine efficiency versus ambient air temperature, our data qualitatively agrees with the real-world Mammoth data. Quantitatively, our data some-what 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 modelling, Wendt and Mines [8] show brine efficiency dropping ~20% (over 25°F) above the plant design point. However, their modelling with temperatures colder than the design point differs significantly from our modelling, 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.

a)



b)

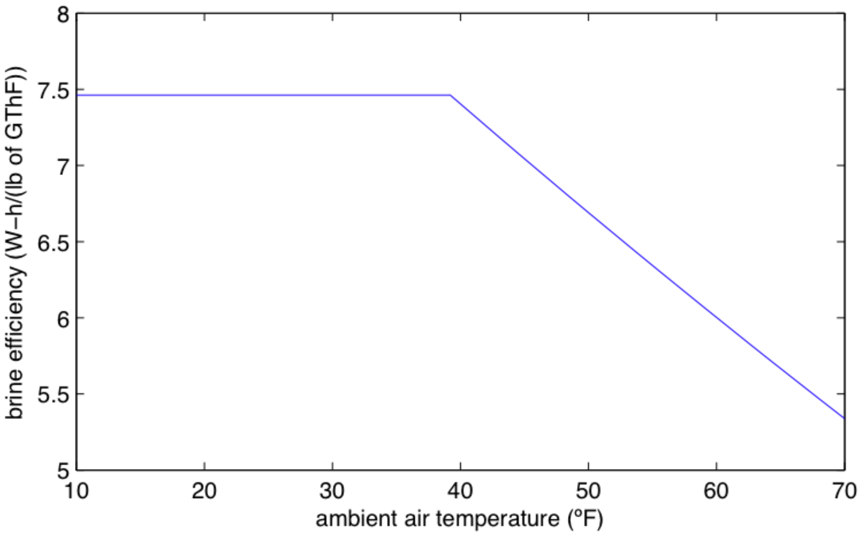


Figure 6: *Validation Results. (a) Mammoth Plant Data [6]. (b)Effect of ambient air temperature on brine efficiency (using:* T_G^{GThF} *=300°F, ambient air temperature for design purposes of 38°F, and isobutane as a working fluid)*

Results

The results of this work are presented in the paper 'Performance of air-cooled organic Rankine cycle plants using temperature distributions from arid parts of South Australia' also presented at AGEC.

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.

By designing for a temperature which is colder than the mean site temperature a plant's output can be increased by 5-8%. Since, designing for a lower ambient temperature will increase initial plant costs (due to the requirement for a larger heat exchanger), in the final analysis, the capital costs will need to be weighed against ongoing improved production capabilities.

List of abbreviations

h	enthalpy (J/kg)
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)
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

Cenegel, Y.A., and A.B. Boles, 1989. “Thermodynamics an Engineering Approach.’ McGraw-Hill, 5th Edition.

DiPippo, R., 2008. “Geothermal Power Plants: Principals, Applications, Case Studies and Environmental Impact.” Elsevier, 2nd Edition.

U.S. Energy Information Administration (EIA), April 2011. “Annual energy outlook 2011 with projections to 2035.” U.S. Energy Information Administration (EIA).

Entingh, D.J., July 2006. “DOE Geothermal Electricity Technology Evaluation Model (GETEM): Volume 1 – Technical Reference Manual.” Department of Energy.

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).

Mines, G., 2000. “Summary of investigations of the modified turbine inlet conditions in a binary power plant.” Proceedings of GRC 2000 Annual Meeting.

Morris, C., April 2012, Personal correspondence.

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.

Van Wylen, G.J., and R.E. Sonntag, 1958. “Fundamentals of Classical Thermodynamics.” John Wiley and Sons, Inc, 3rd Edition.