




Hybrid Cooling for Geothermal Power Plants

Final ARRA Project Report

Desikan Bharathan

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Technical Report
NREL/TP-5500-58024
June 2013

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Desikan Bharathan

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Table of Contents

Figures

Tables

Executive Summary

Many binary-cycle geothermal power plants use air as the heat rejection medium. An air-cooled condenser (ACC) system is used to condense the working fluid vapor exhausted from the turbine. Primary reasons for the use of ACC technology are the lack of water at the site and simplicity of the overall plant configuration. However, plants with ACC systems perform poorly during summer hot weather. When the ambient temperature rises above 30^oC, the power output of the plant may decrease by more than 50%. This reduction occurs during times when the electricity demand is high, typically due to air-conditioning loads, and the corresponding price for electricity is also high.

The goal of this project is to identify and analyze advanced cooling strategies that allow air-cooled geothermal power plants to maintain a high electric power output during periods of high ambient dry-bulb temperatures while minimizing water consumption. We used computer models of binary-cycle and flash steam power plants to compare alternatives for boosting hot weather performance. Loss of power production capacity in geothermal power plants during hot summer days has been a subject of many studies reported in the literature. An interim report developed under the current American Recovery and Reinvestment Act (ARRA) study covered the background literature (Ashwood, 2011).

The objective for this study is to arrive at efficient practical means for using low amounts of water with the ACC system to improve the performance of a binary geothermal plant at minimal cost. The use of some water within a dry-cooled system is referred to as a "hybrid" design. Many schemes for using ACCs in hybrid mode have been investigated and reported in the literature. Our work aims to distill those results with a relevant set of goals to identify a subset of the most practical schemes. The results offered in this report are confined to the analysis work carried out during 2012.

Our investigations identified deluge cooling of the condenser tubes in a section that then serves as an evaporative condenser as the least costly option. In addition, we found that handling of the entire air stream to the ACC is generally impractical, and so precooling of the intake air was eliminated from the list of contending options.

With these practical considerations in mind, we came up with the following scheme to implement hybrid cooling. In an ACC, one of the many (eight to ten) bays is "converted" to a deluged, evaporatively cooled condenser. This section of the condenser is isolated such that water sprays do not get carried away by the local wind gusts. Air is directed to flow from bottom to the top. Water sprays deluge the entire set of tubes within this bay. All uncondensed vapor from the purely air-cooled sections is channeled through this section, where the last bit of condensation occurs. During times when this section is used with water deluge, we expect about 30% to 50% of the overall condensation to occur here. Therefore, the air-cooled section sees only a load of 70% to 50%. This level of condensation and corresponding heat load would be typically equivalent to an "effective" reduction in the intake air temperature of about 4^oC to 8^oC. That is, the condenser saturation temperature decreases by that amount. The tubes and manifolds must be properly designed such that they can accommodate this level of condenser duty.

We believe that this type of arrangement for hybrid cooling is most practical for the power plant to adopt. NREL has filed a record of invention for details related to the proposed hybrid cooling approach. We anticipate that patent application(s) related to implementation with geothermal and other power plants or air-cooled condensers used in other industrial systems and applications will follow. Because our pursuit of the intellectual property protection is pending, we have limited our discussion of this concept to simple basic outline at this time.

Abstract

Many binary-cycle geothermal plants use air as the heat rejection medium. Usually this is accomplished by using an air-cooled condenser (ACC) system to condense the vapor of the working fluid in the cycle. Many air-cooled plants suffer a loss of production capacity of up to 50% during times of high ambient temperatures. Use of limited amounts of water to supplement the performance of ACCs is investigated. Deluge cooling is found to be one of the least-cost options. Limiting the use of water in such an application to less than one thousand operating hours per year can boost plant output during critical high-demand periods while minimizing water use in binary-cycle geothermal power plants.

1 Introduction

Many binary-cycle geothermal plants use air as the heat rejection medium. Usually this is accomplished by using an air-cooled condenser (ACC) system to condense the vapor of the working fluid in the cycle. In addition to condensing, desuperheating and subcooling of the liquid also occur within the ACC. During hot summer days, a low condenser temperature cannot be maintained, and the condenser pressure must go up substantially to reject the heat to ambient air. Many air-cooled plants suffer a loss of production capacity of nearly 50%, as documented in the literature. Use of minimal amounts of water to help improve the performance of these binary plants is the subject of this analysis. Many different schemes to use water to improve ACC performance have been reported in the literature. Our aim is to narrow the choices for implementing supplemental cooling with water by assessing the ease of operation, practicality, and cost of the alternatives.

1.1 Objective

The goal of this project is to identify and analyze advanced cooling strategies that utilize low amounts of water to allow ACC-equipped geothermal power plants to maintain a high electric power output during periods of high ambient dry-bulb temperatures.

This project uses computer models of binary-cycle and flash-steam power plants to compare alternatives for boosting hot weather performance. The results of previous works on using evaporative enhancement methods (evaporative media and direct deluge of air-cooled condenser tubes) are used to evaluate hybrid arrangements in which ACCs and water-cooled condensers operate in series. We also investigated an air-cooled steam plant employing an indirect Heller cycle in which the water flow rate between a direct-contact condenser and water-to-air heat exchangers is modulated to accommodate varying dry bulb temperatures.

Parallel air/water cooling systems have been used in a small number of newer fossil-fuel power plants where cooling water is limited. They have not been used in geothermal plants, which operate at lower heat input temperatures. The lower temperatures of geothermal plants mean that using hybrid cooling may provide an even greater relative performance benefit. Such a system could be used with both binary-cycle and steam-cycle geothermal power plants. Heller cycles have been used to provide air cooling in large coal-fired steam power plants, but there has been no use in geothermal plants. Our modeling showed how a Heller cycle with variable water flow rates will perform well for a steam-cycle geothermal power plant compared to a water-cooled plant. Much of the work carried out under this task has been reported in an interim report (Ashwood, 2010).

The objective for this study is to arrive at an efficient practical means for using water with ACCs to improve the performance of a binary geothermal plant with minimal use of water and minimal cost. Many schemes for using ACC systems in hybrid mode have been investigated and results reported in the literature. Our analysis aims to distill our results using a relevant set of goals to define a subset of most practical schemes. The results in this report are confined to the work carried out during the last year.

1.2 Background

Loss of power production capacity in geothermal power plants during hot summer days has been a subject of many studies reported in the literature. An interim report developed under the current ARRA-funded study covered the background literature (Ashwood, 2011).

The power output of air-cooled geothermal power plants can drop to less than half the design value on hot summer days when utilities need electricity the most. Maintaining capacity in summer weather is important for maximizing plant revenue. Analysis of hybrid cooling of a concentrating solar power plant has shown that, compared to a water-cooled plant, water consumption can be reduced by 80% with only a 2% drop in annual energy output. We believe that such systems, if properly optimized, can greatly boost summer output of a geothermal power plant with minimal use of cooling water. The cooling water can be obtained from wastewater treatment facilities, irrigation rights, or reverse osmosis of the geothermal brine.

No geothermal steam-cycle plants are air-cooled. Instead, these use the reservoir condensate for cooling. But this results in a drop in reservoir pressure, which has occurred at The Geysers geothermal complex in northern California. For EGS applications where the resource temperature is sufficiently high to run a steam cycle (~175^{o}C), returning water to the reservoir is preferable to using it for cooling because of potential reservoir depletion. An indirect Heller cycle provides a means by which a steam-cycle geothermal power plant can be cooled by air and water in sequence.

1.3 Approach

The work was initially divided into 3 phases of one-year duration each.

During the first year, the National Renewable Energy laboratory (NREL) performed simulations of 50-MW binary-cycle geothermal power plants at resource temperatures of 125^{o}C and 175^{o}C. These simulations were done for three different heat rejection systems: 1) water-cooled, 2) air-cooled, and 3) a hybrid cooling system that uses water over a limited number of hours during the year. For each system type, the overall plant design was optimized to produce the lowest levelized cost of electricity based upon typical economic assumptions used by utilities.

Our analysis included all costs such as parasitic power, water treatment, and operation and maintenance. The analysis was done for a range of water costs from $0.50 to $10 per thousand gallons. In analyzing the hybrid cooling system, the analysis considered a range of sizes for the water-cooled and air-cooled units ranging from zero to full-size.

The analysis was confined to one hypothetical U.S. location located in northern Nevada. The economic analysis included a determination of the profit from electricity sales for a range of flat electricity prices, as well as two time-of-day payment structures.

We also modeled a 50-MW geothermal power plant employing a direct contact condenser and a cooling liquid loop to an array of liquid-to-air heat exchangers. The results were compared to a directly air-cooled system and a water-cooled system using a surface condenser and a cooling tower.

As the test unit, during the second year, we selected a nominal 250-kWe geothermal power unit, originally built by Ormat Technologies, Inc. and operational at the Rocky Mountain Oilfield Test

Center (RMOTC), a U.S. Department of Energy (DOE) operated facility. Plans were made to test the air-cooled unit with and without hybrid water-cooling assist. Instrumentation and a data acquisition system were purchased and installed at the field site. However, just prior to the commencement of the experiments, the power plant broke down and has not been operational since. This caused changes in the scope of this study.

We modified the scope to conduct more detailed numerical studies of specific air-cooled systems and report the results here.

1.4 Case Studies

Description of a typical ACC System – A typical ACC system for a binary geothermal power plant consists of a large number of horizontal finned carbon-steel tubes arranged in stacks. A photograph of a typical tube is shown in Figure 1, showing a tube with an outer diameter (OD) of 1 inch and a wall thickness of 1/16 inch, covered by aluminum fins of 1/64-inch thickness at a spacing of 10 per inch, and has a 2.25-inch OD. The tubes are generally arranged in an isosceles triangle with a spacing of 2.375 inch. The tubes form a set of five rows, and the tubes may be 60 feet long.

Figure 1. A photograph of a typical ACC tube with fins. Photo by Desikan Bharathan, NREL 24524.

Usually, air is induced to flow over the tubes by large fans placed above the tube rows. A large condenser for a 5-MWe plant may cover a ground area of 14,400 square feet. Design air temperature for geothermal applications is usually 10°C, with a 15°C temperature rise at design at full load. The vapor and air go through the condensers in a single pass. The ACC system usually takes in superheated vapor exiting the turbine and condenses it, with some subcooling of the condensate. Typical overall ACC heating and cooling load curves are shown in Figure 2.

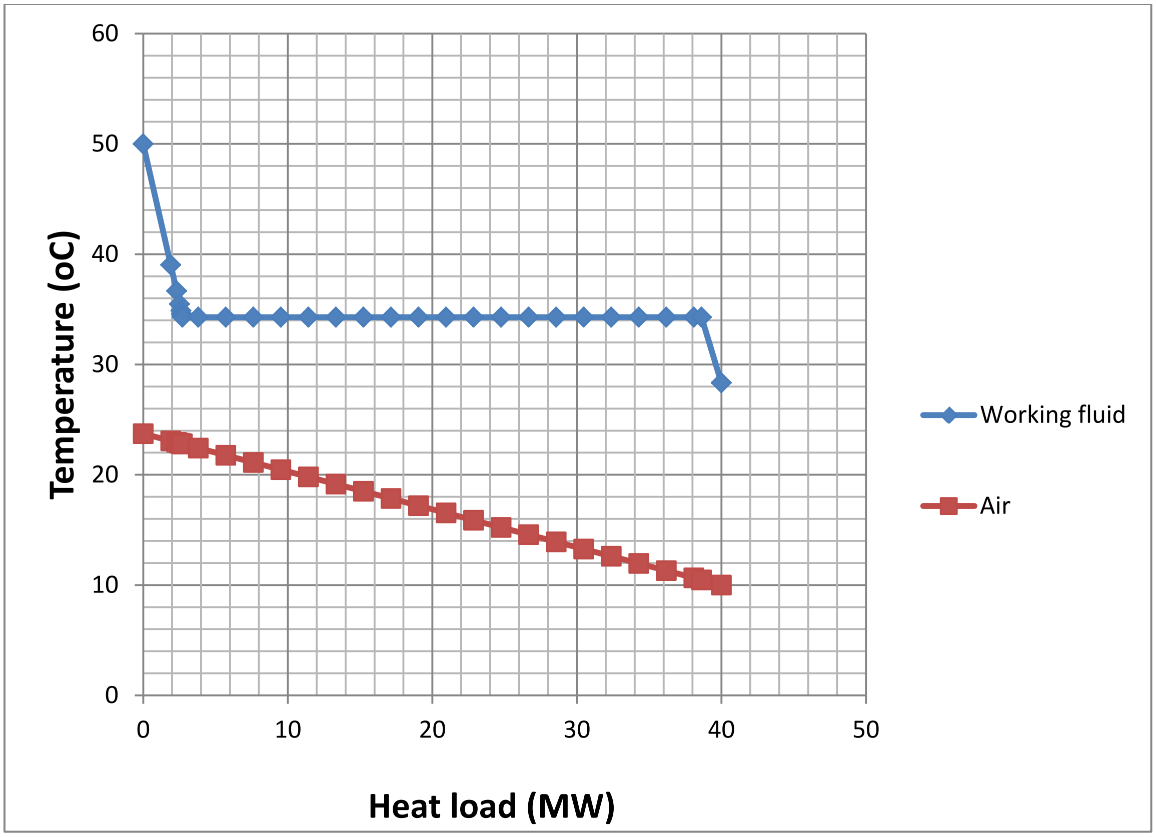


Figure 2. A typical condensing heat load curve, indicating desuperheating, condensing, and subcooling regions.

The vapor enters the condenser on the left side and its temperature is indicated as the working fluid temperature. The air enters from the bottom, heating up as it flows through the condenser. The overall heating load for a condenser of about a 40-MW consists of three regions; 1) desuperheating, where the temperature decreases rapidly to the vapor's saturation pressure, 2) a condensing region where the temperature remains constant with decreasing vapor quality, 3) and then a final subcooling region, where the condensate is cooled below the saturation temperature. In Figure 2, we note that the condensing load is about 90% of the overall heat duty of the condenser.

Tube and fin performance – The overall performance of the condenser is governed by three items; 1) the external air-side heat transfer, 2) the conduction through the tube and fins, and 3) the internal heat transfer by the condensing vapor. The major resistance to heat transfer arises from the air side and to reduce this resistance, fins are normally introduced on the air side. We looked into each aspect of these components in detail below.

Air side external heat transfer – The flow of air around a single tube was modeled using a computational fluid dynamics (CFD) program with the help of commercially available software, FLUENT. We discuss the details of the model below. Figure 3 illustrates the geometry of the tube along with its fin as modeled.

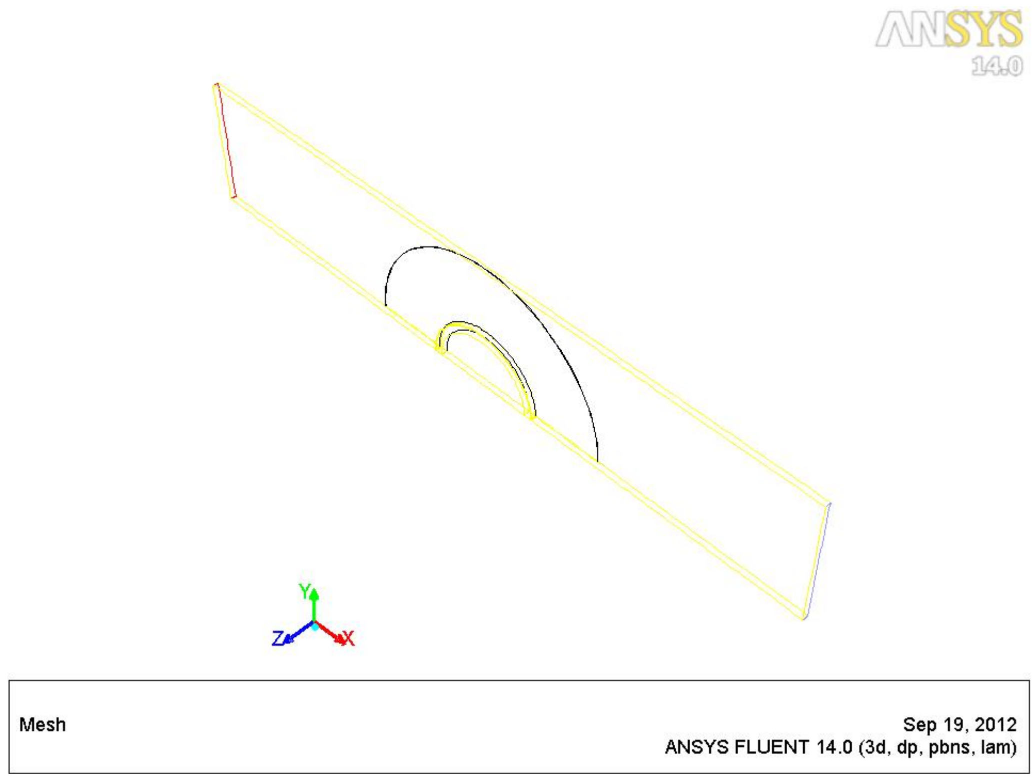


Figure 3. CFD model geometry for air flow over a single tube and fin (only one quarter of the tube and a single fin is used for modeling accounting for symmetry in the flow field).

When accounting for symmetry over two planes – namely XY and XZ planes – in the flow field, only one quarter of the fin and its associated tube need be modeled.

The air flow field indicating the velocity vectors is shown in Figure 4. Air flows from the right side of the figure toward the tube at a superficial velocity of 2 m/s, typical of ACCs. As air approaches the tube and fin, and because of the blockage due to the tube and fin, the air attains a maximum velocity of about 5.6 m/s adjacent to the tube.

Simulations were carried out at varied air approach velocities by fixing the tube internal temperature at 87°C and an air temperature of 27°C. Figure 5 shows the flow-field temperatures in Celsius. Air-side heat transfer coefficients were calculated based on the external heat transfer area and its mean temperature. Pressure losses for air flow were also calculated.

Figure 6 shows the simulation results. The heat transfer coefficient increases gradually with increasing air velocity. The pressure loss indicated increases much more rapidly. For applications in the field, a compromise must be made between the increased heat transfer coefficient and the associated pressure losses. In typical applications, superficial air velocity of about 2 m/s is commonly used. At this nominal velocity, the air-side heat transfer coefficient of about 40 W/m^2K is obtained, with a pressure loss of about 15 Pa. Note that this result is for a single tube. When multiple tubes are present, the air flow around one is affected by the others. An exact geometric arrangement of the tubes is necessary for proper modeling; flow over multiple tubes is addressed in the next section.

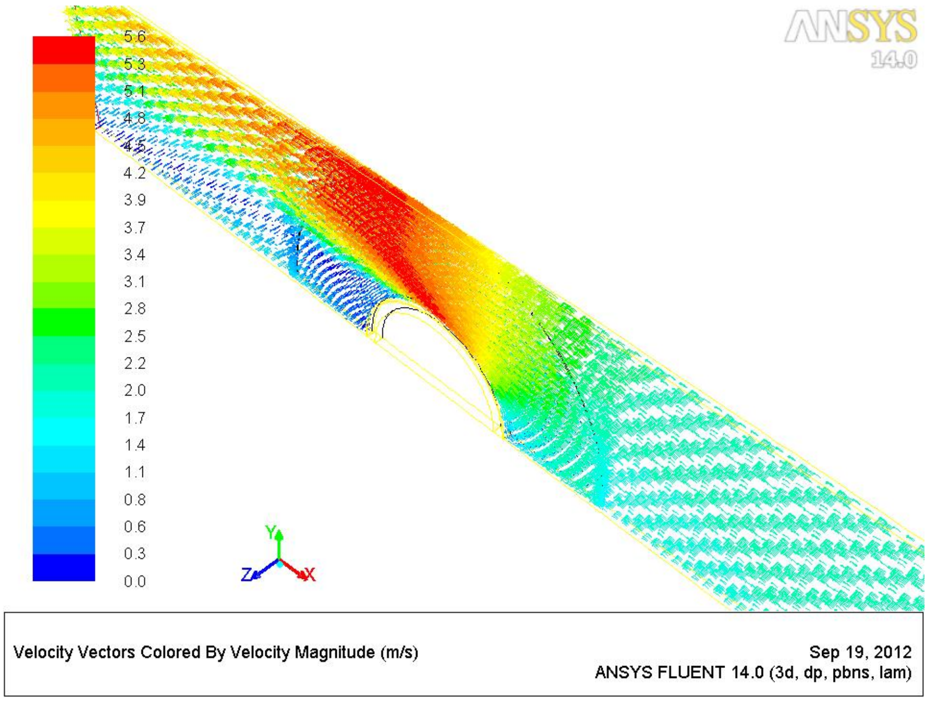


Figure 4. Velocity field around a single tube with fins.

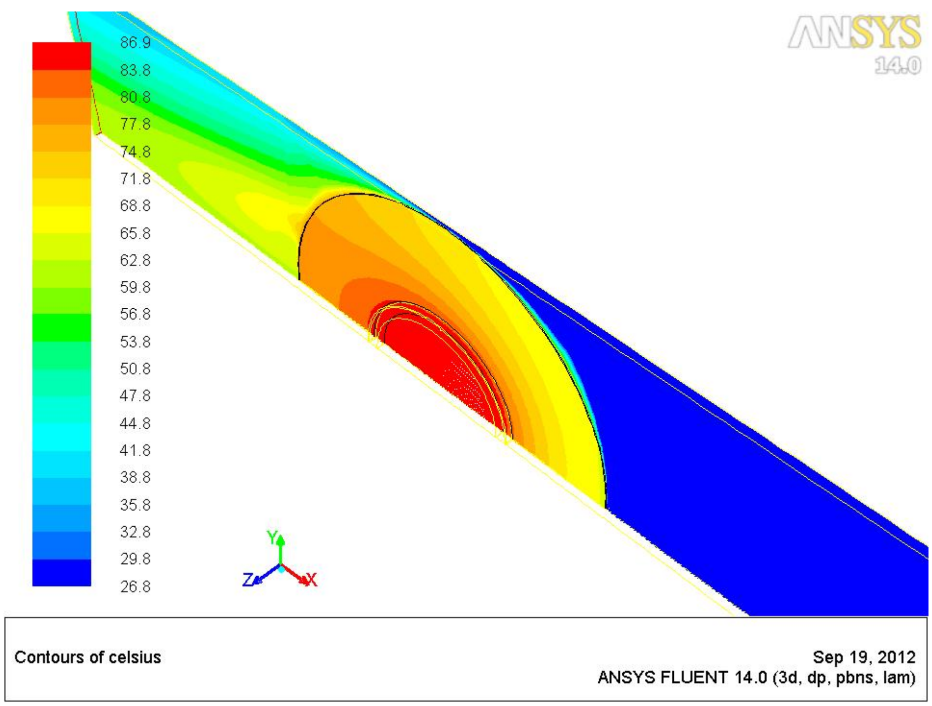


Figure 5. Temperature field around the single tube with fins.

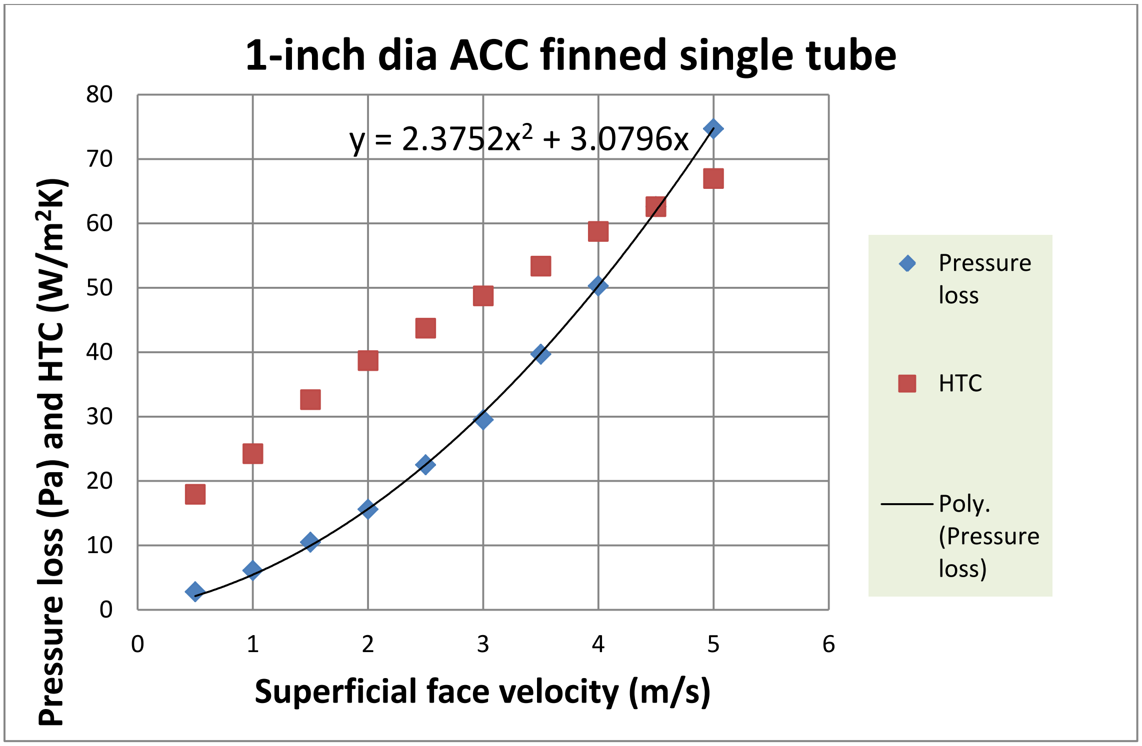


Figure 6. Plots of heat transfer coefficient and pressure loss as functions of air velocity.

1.4.1 Multiple Tubes

We also modeled air flow over five tubes arranged in a staggered manner to arrive at air-side heat transfer coefficients. Figure 7 shows the geometry along with air velocity vectors. Because of symmetry, only half the tubes are shown. In Figure 7 airflow is from left to right. The air tends to accelerate between the tubes due to restricted passages.

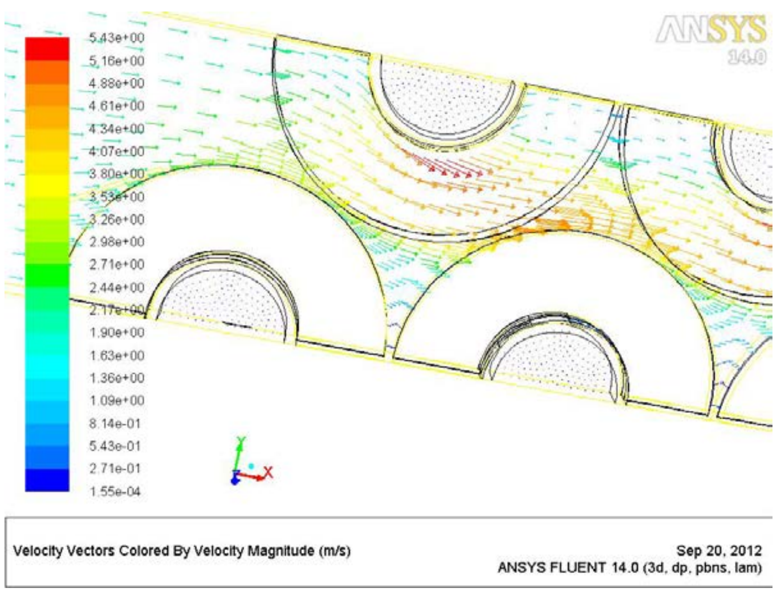


Figure 7. Velocity field around multiple tubes with fins.

Once again, based on the temperature distribution, we are able to calculate the air-side heat transfer coefficients and pressure losses.

The overall variations of the heat transfer coefficient and the pressure loss for a set of five tubes arranged in a staggered geometry are indicated in Figure 8. As before, the heat transfer coefficient increases gradually with increasing air velocity. The pressure loss, however, increases much more rapidly with increasing air velocity.

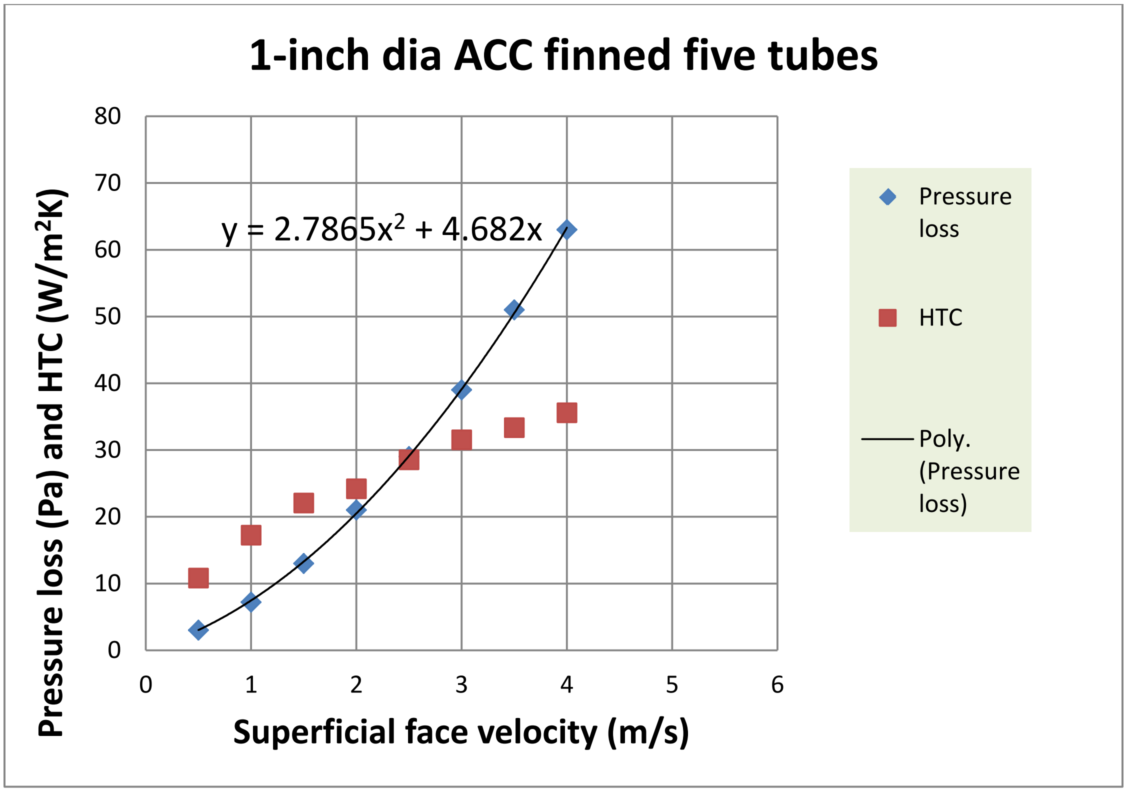


Figure 8. Variations of heat transfer coefficient and pressure loss for air flow around multiple tubes.

External heat transfer coefficient, h_o

Numerical values generated by the CFD studies of the air flow around the finned tubes can be used in the evaluation of the external heat transfer. For five tubes in a staggered arrangement, the CFD results indicate a heat transfer coefficient of about 24 W/m^2K, based on all available outer heat transfer area.

We arrive at an external heat transfer coefficient of about 24 W/m^2K at an air superficial velocity of 2 m/s for this set of five tubes.

Using this value, our estimated overall heat transfer coefficient, U, is quite consistent with the condenser specifications as noted later in the report.

Internal heat transfer coefficient on the working fluid side, h_i

Little data on condensation heat transfer for n-pentane are available in the literature. However, many correlations exist for hydrocarbon refrigerants used for internal flow in condensation tubes. A comprehensive summary of various correlations can be found in Chapter 8 of Engineering Handbook III by Wolverine Tube Inc. (Wolverine, 2001)

We find that the condensation heat transfer coefficients range from 2000 to 6000 W/m^2K for the majority of hydrocarbons. We'll use a nominal value of 2000 W/m^2K for our analysis of pentane condensation as the working fluid. Any error resulting from this assumption is expected to be minimal for our estimates, because the resistance to heat transfer is dominated by the air in ACCs.

Tube wall resistance

The wall resistance is simply calculated as conduction through the wall of a given thickness. Sometimes, the contact resistance between the fin wrap and the tube external wall can offer some significant resistance when the overall resistance is low. In the present case of air cooled condenser, this contact resistance is negligible and can be ignored in the analysis.

Fouling resistances (r_i and r_o)

Fouling occurs over time both on the internal side and external walls, caused by accumulating impurities during service. Typical values for the fouling resistances for air coolers are:

88e-6 (m^2K/W) [0.0005 ($Hr.ft^{2o}F/Btu$)] and 0.00035 (m^2K/W) [0.002 ($Hr.ft^{2o}F/Btu$)] on the tube side and external surfaces, respectively.

Overall heat transfer coefficient (U)

Using these values, we calculate an overall heat transfer coefficient as follows:

$$(1/U) = (A_o/A_i)*(1/h_i) + r_i * (A_o/A_i) + (t/k)*(A_o/A_{av}) + r_o + 1/h_o$$

Where:

A_i, A_o, A_{av} are the heat transfer areas; internal, outer and average, respectively,

and h_i, h_o are the internal and external heat transfer coefficients,

r_i and r_o are the internal and external fouling resistances, and

t and k are the thickness and thermal conductivity of the tube wall.

Typical steel tube used in the ACCs has an OD of 25.4 mm and a thickness of 1.65 mm. At a nominal superficial air velocity of 2 m/s, we get an overall heat-transfer coefficient of:

$$1/U = (1/2000)*22 + 88e\text{-}6*22 + (0.00204/40)(11) + 0.00035 + (1/23.4);$$

or an overall U value of 17.6 (W/m^2K) based on the outer area of the tube. This value is close to the air-side heat transfer coefficient, this being the major resistance to heat transfer.

This value is also close to the overall heat transfer coefficient for the ACC system under tested conditions. The above set of detailed calculations indicates that in the case of a wetted tube, the external heat transfer coefficient can be improved considerably and the internal heat transfer may become the governing resistance.

1.4.2 On Subcooling

As we noted earlier, subcooling of the condensate is generally unavoidable in the case of single-pass vapor flow. However, sometimes subcooling is preferred to avoid cavitation in the condensate feed liquid pumps. However, subcooling is not necessary if adequate liquid head can be made available at the suction side of the pump. This is the case for many geothermal plant installations where the condenser tubes are placed high above the ground and the feed pumps are generally located at ground level.

The following section looks at the condensation process along the length of the tubes for a set of five tube bundles. For this analysis, five 18.3 m (60 feet) long tubes are modeled. Each tube is divided into 183 segments. Mass and heat balance calculations are carried out for each of the segments.

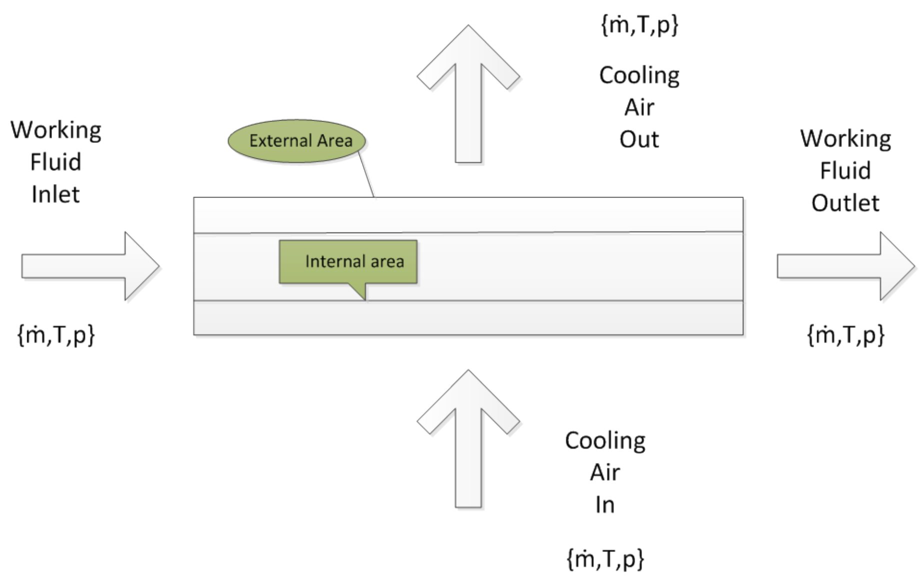


Figure 9. Elemental segment of condenser tube showing cross flow of air.

For each segment, heat transfer calculations are made as follows. Based on an assumed overall heat transfer coefficient (assumed constant for this analysis), the heat lost from the working fluid is calculated based on the available heat transfer area for the segment and the average driving temperature difference between the working fluid and air at each segment. In cross flow, the air flows upward along a y-axis, whereas the working fluid flows horizontally through the tubes along the x-axis. Given the entering temperatures of the working fluid and air, heat balances are calculated at each segment and iterated to arrive at the exit temperature of the corresponding fluids.

Certain simplifying assumptions were made for the following results. The vapor flow through all tubes was assumed to be the same, and pressure loss calculations were not made. Note that varying pressure losses through the tubes will cause uneven flow through the tubes. The overall heat transfer coefficient was assumed to be constant at a value of 24 W/m^2K. In real conditions, the heat transfer coefficient will also vary.

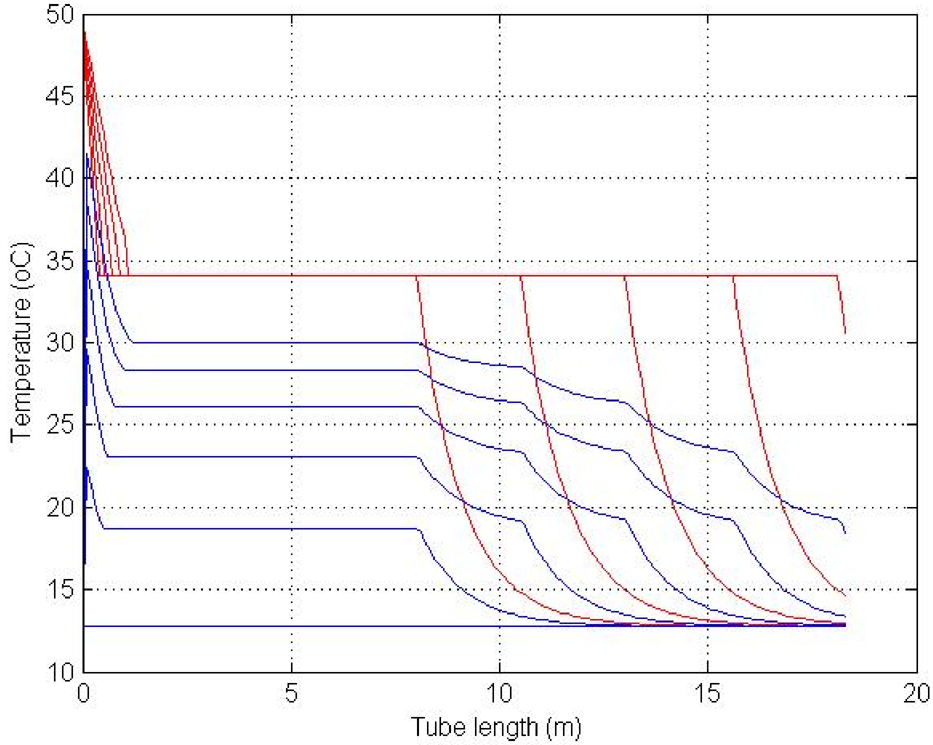
If the working fluid is superheated or subcooled at the given condenser pressure, sensible heat transfer occurs; otherwise, condensation happens.

Since the first row of the five-tube rows encounters the coldest air, condensation happens quickly in it, condensing all the working fluid within about half the tube length. However, as the air gets heated from the first row to the next row and so on, the condensation occurs more and more gradually, until all of the condensation is completed in the last row using the tube's full length.

Figure 10 shows the temperature distribution in the working fluid and the air over the entire field of five tubes. Once again, the air enters the field from below and the working fluid enters from left. Initial air temperature is set at 12.7^{o}C and the condenser pressure is set at saturation value for n-pentane at a temperature of about 34^{o}C.

The lower six blue lines in Figure 10 indicate the air temperature in between the tubes. The top five red lines represent the temperature of the working fluid. At the entry section of each tube length, desuperheating is seen as a rapid decrease in working fluid temperature from 48^{o}C to 34^{o}C. During condensation, the working fluid temperature lines merge at the 34^{o}C saturation temperature. Upon completion of condensation, the condensate begins to cool within the lowest tube in the stack first. The subcooling region on this tube occupies over half of the total tube length. This subcooled region length decreases for each subsequent tube as the air gets hotter while it flows upward. The condensation region occupies almost the entire length of the fifth (uppermost) tube. We note that the air does not reach the maximum temperature possible once the subcooling begins in the first tube. The qualitative trends depicted in Figure 10 are typical of any cross-flow single-pass condenser or heat exchanger; a more rigorous analysis that includes calculations of pressure losses and varying heat-transfer coefficients would refine the analysis.

<u>Significantly, these results suggest there may be methods to improve the utilization of the tubes.</u> As we note, almost half the total length of the tubes are in the subcooling region when subcooling is not necessary. If one can reduce the region of subcooling, a smaller, more efficient condenser can be realized.



***Red** line is for the working fluid temperature.*

***Blue** line is for the air temperature*

Figure 10. Temperature variations of the working fluid and air along the tube.

2 Hybrid Cooling Options

Basic Requirements – Typically, most binary power plants use ACCs. Their power output ranges from fractions of a MW up to about 20 MW. While an evaporative condenser would be favored from a cycle efficiency perspective, lack of available water is the dominant factor in the selection of air cooling. Plants with ACCs are also simpler to set up and operate. Their typical operating power-conversion efficiency runs about 12% to 15%. Cooler ambient weather allows for more efficient operation of an ACC.

However, during hot ambient weather, air-cooled plants suffer higher condenser pressures and lower conversion efficiencies. As a rule, reduced power conversion efficiency of 0.2% to 0.25% occurs with each degree Kelvin increase in the ambient temperature.

If water is available, a hybrid cooling scheme can help mitigate the reduction in performance during hot weather. However, since these plants are located in arid regions, water is often scarce. Water that is available is often of low quality. Further, water use may be restricted to certain times of the day. An EPRI study (see References) suggests that to minimize water consumption, its use in hybrid condensers should be restricted to 1000 operating hours during summer months.

Many schemes for implementing hybrid cooling have been explored both in the form of model analyses and experiments in the literature. Yet, to our knowledge, not a single geothermal power plant operates using hybrid cooling on a routine basis.

Difficulties arise not only in the form of cost, but also in general ease of operation and plant maintenance. Ideally, a plant operator must be able to simply turn the hybrid cooling option on and off as needed with minimal preparation. The hybrid option must also operate reliably and consistently. Despite the fact that hybrid cooling is conceptually simple, these difficulties have prevented implementation in many cases.

Hybrid cooling can typically be implemented 1) either by pre-cooling the air entering the ACCs, or 2) by using a dedicated condenser that uses water as the heat reject media, supplementing the ACCs. Our prior report, Ashwood (2011) addresses the varied schemes in detail. In 2003, Kutscher (2003) completed a study of hybrid cooling options for geothermal power plants. We shall look into some of the details of the more popular ideas incorporated by those studies in the following sections.

Using Packings – Packings or cooling tower “fills” are commonly used to precool incoming dry ambient air. Generally, water flows down the fill materials while air flows across; the fill ensures good contact between the cooling water with the incoming air. Water evaporates using the sensible heat from the air, thus cooling and simultaneously increasing the relative humidity of the air. Figure 11 shows the installation of packings placed around the condenser module to precool the intake air.

Figure 11. Setup to use packings to precool intake air at a geothermal power plant. Photo by Keith Gawlik, NREL 24495.

Kutscher (2002) found that the packing provided an evaporative effectiveness of about 80%, which helped increase the plant output by 62% during hot weather. They found that evaporative cooling can reduce the cost of production of electricity by about 0.3¢/kWh and increase capacity revenues as well. However, it was not practical to use the packing on a continued basis because of the required packing maintenance and restricted access to the condenser.

Using Sprays and/or Mist – The revised scope of the present work was addressed at looking into the use of sprays and mist in more detail. Our assessment and early experience with tests conducted at a field site suggested that sprays were not highly practical. During tests, we found that the sprays and spray-cooled air get diverted from entry into the ACC tube bundle by local winds. Spray droplets invariably get deposited on the fins of the condenser tubes. These wet surfaces are prone to corrosion and attract deposition of hard water substances and air-borne dust and particles. The deposits tend to accumulate quickly to fill the passages and block the air flow in those areas.

The purpose of sprays and mists is to evaporatively cool the incoming air. Large droplets do not evaporate readily and cause wetting of the heat transfer surfaces. As noted above, such deposition has been known to cause severe corrosion. Methods to generate smaller droplets have been attempted. Droplets of less than 50 μm have been known to evaporate quickly and completely. However, wetting of the surfaces cannot be avoided because of the variability of local air flow and gusts. Smaller droplets are more susceptible to local wind gusts and tend to be carried away from the heat transfer surfaces and become ineffective in strong winds. Our best efforts using sprays indicate that an evaporative effectiveness of 60% to 70% to the wet bulb is often the outcome. <u>Achieving zero deposition of droplets is impossible and the potential corrosion of the tubes in the ACC system, which is perhaps the major cost item in the power plant, is unavoidable.</u>

To generate droplets of less than 50 μm diameter takes substantial work input. The work comes in the form of compressed air flowing with the water through fine nozzles at a pressure of over

500 psi. New spray nozzles introduced to the industry use a different approach by impinging two separate streams of liquid against each other to generate small diameter sprays. They can operate at lower pressures of 250 psi. However, sophisticated nozzles generate another stringent requirement because the water has to be treated to contain a minimal amount of solid and dissolved matter. This additional equipment adds to the cost of the overall system.

In general, it is difficult to handle the full flow volume of air for the ACC system with any kind of treatment. Systems that deal with a fraction of the total air flow are likely to be more practical.

We contacted three different vendors for incorporating a spray/mist type evaporative cooler. We sized the systems to fill the needs for the 250-kW Ormat Technologies, Inc. test unit located at RMOTC. The quotations received were exorbitantly expensive. The pressure loss through the spray system was also extremely high. Most of the systems required a compressed air supply as well. Not all the data required for evaluation of the spray system such as droplet spectral data were available.

Table 1. Relative costs of spray/mist cooling hardware for use with a 250-kW binary power system.

Company	Spray system cost	Pressure required	Remarks
A	$143,000	Not specified	Claimed 8μm SMD
B	$373,000	Not specified	Need compressor at 110 psi
C	$120,000	870 psi	

Despite the initial focus on spray systems, upon reexamining further details and assessing their practicality in the use at power plants, we decided to drop further investigation of the spray/fog/mist systems.

Deluge Cooling (or evaporative condenser) – A deluge of water over the condenser tubes was found to be the most effective means for wetting the tubes and enhancing the heat transfer. Kutscher (2002) show a schematic of the deluged cooling system in the Figure 12.

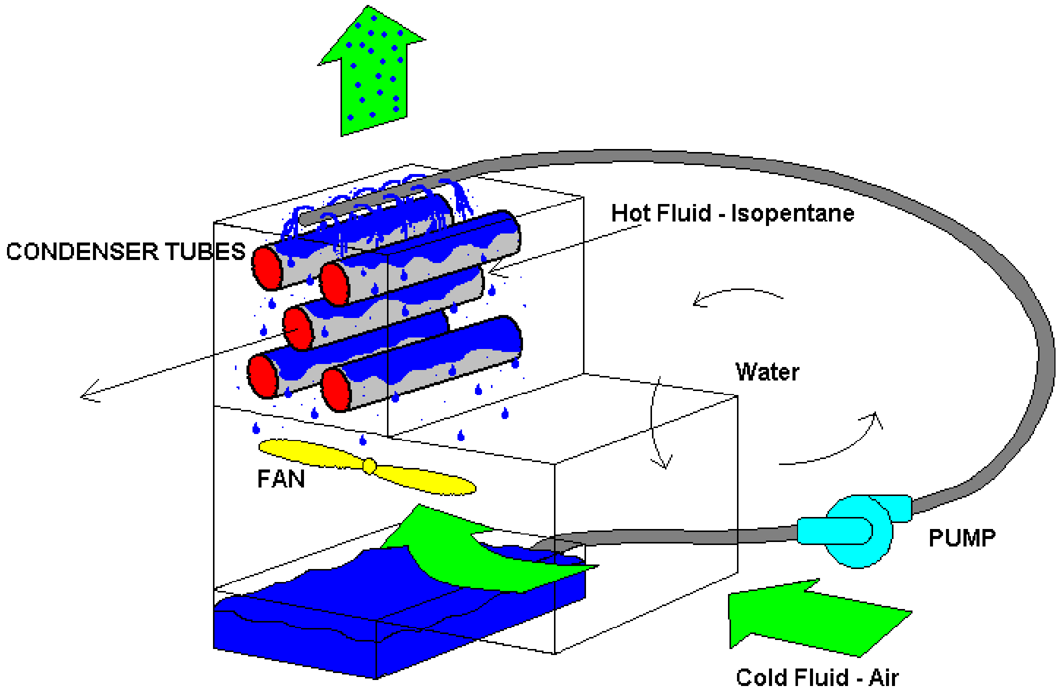


Figure 12. Schematic diagram for deluge cooling.

In this scheme, the exiting air gets warmed and fully saturated. Thus, it is able to carry more moisture than the previous cases discussed. They found that this system yielded excellent performance. This system nearly restored the design power yield for the plant during hot ambient conditions, and yielded the least cost increment for use of the hybrid system.

Using a Surface Condenser in Parallel or Series – The use of a conventional surface condenser in parallel or series with an ACC was also examined. Ashwood (2011) concludes that the simple payback for a surface condenser operating in series runs about 4 to 6 years.

2.1 A Relative Comparison of Options

The different cooling schemes offer varied levels of advantages and disadvantages. Full details of the comparisons are provided in Ashwood (2011). Table 2 summarizes the findings of this study in a succinct manner.

Table 2. Summary of hybrid cooling options and their payback periods.

Evaporative Cooling System	Incremental Equipment Costs (M$)	Incremental Annual Net Revenue (M$)	Simple Payback Period (Years)
Wetted-Media	3.3	0.36	9.4
Fogging	2.2	0.36	6.1
Spray	0.21	0.36	0.60
Deluge	0.06	0.49	0.13

Table 2 was generated in our previous work for a nominal 20-MWe air-cooled power plant. Economics play a major role in the selection and use of any particular scheme. Relative cost comparisons suggest that spray or deluge systems are the most economical. It is possible to incorporate the deluge system in series (in vapor flow) with the dry cooling sections of the ACC system.

2.2 Additional Findings

For a hybrid system to be incorporated in a power plant, it must be functional, adaptable, and practical. Key considerations include:

1. It is difficult to contain and control the entire volume flow of air that is required by the ACC. Ideally a hybrid system should handle only a fraction of the air, perhaps around 10%. Thus, evaporative precooling of the entire air stream to the condenser is considered impractical.

2. Use of water spray or fog must be confined to a portion of the ACC such that water does not pool on the ground, making operation and maintenance difficult. Use of water in contact with air must be confined in devices; for example, within a wet cooling tower or within an evaporative cooler.

3. Use of a conventional wet cooling tower with a surface condenser was found to be prohibitively expensive with very long payback periods. However, evaporative condensers offer good performance and should be explored further to achieve lower costs. Evaporative systems work best in low humidity climates.

4. Deluge evaporative condensers were found to offer the best combination of simplicity, effectiveness, and cost.

5. For evaporative coolers/condensers, we should avoid using fins on tubes to prevent potential deposition and corrosion in contact areas. It is suggested that tubes be

galvanized to prevent corrosion on the outside, similar to commercially available evaporative condensers.

6. Considering that the ACC system normally consists of many bays, each being served by a set of fans, it is possible to isolate one of the bays as an evaporative cooler.

7. The evaporative cooler should be sized to handle about 30% to 50% of the condenser load. This condenser must be placed last in series with the remaining bays of the ACC system which are purely air cooled. This arrangement avoids the use of flow splitting valves and controls.

8. Each bay of the ACC system already has all the components needed to make it an evaporative cooler. The fans and the tubes for the vapor flow exist. It is just a matter of providing the water flow to deluge the tubes and recirculate the water. Water quality is not a major concern, because adequate blowdown rates can be incorporated at the offset.

9. To convert a bay to an evaporatively cooled condenser, fans, tube selection, fins, and water spray devices and their distribution must be carefully reevaluated.

10. Once a deluge section is created, the operator simply regulates water flow to the deluge section as needed. The operator can run no water if the ambient air is cool enough, or regulate the flow of air and water such that maximum benefit is obtained without using too much water.

2.3 Our Proposed Solution

With these practical considerations in mind, we came up with the following scheme to implement hybrid cooling.

In an ACC system, one of a total of eight to ten bays is "converted" to a deluged evaporatively-cooled condenser. This volume of the condenser is isolated or enclosed such that water sprays do not get carried into the other ACC sections. Air is directed to flow from bottom to top. Water sprays deluge the entire set of tubes within this bay. The entire tube-side flow from the air-cooled sections is channeled through this section, where the final condensation and subcooling occurs. During times when this section is used with water deluge, we expect about 30% to 50% of the condensation to occur here. Therefore, the air alone provides a remaining load of 70% to 50%. This level of condensation and corresponding heat load would be typically equivalent to an "effective" reduction in the intake air temperature of about 4^{o}C to 8^{o}C. That is, the condenser saturation temperature decreases by that amount. The tubes and manifolds must be properly designed such that they can accommodate this level of the condenser duty.

We believe that this type of arrangement for hybrid cooling is practical for power plant adaptation. For this reason, we are seeking intellectual property protection for this concept for implementation with geothermal and other power plants or ACC systems used in other industrial applications.

2.4 Summary Remarks

A variety of schemes for enhancing the performance of a geothermal power plant during times of high ambient temperatures using hybrid cooling was investigated. Deluge cooling was found to be the most economical option. However, potential scaling and corrosion of the condenser tubes remain obstacles. Limiting the use of water to one thousand hours of operation during a year will help reduce water consumption for hybrid cooling options.

Acknowledgements

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Appendix A – Method to Eliminate Subcooling in ACC Systems

Heat transfer from the condenser tubes was modeled as follows. A set of five horizontal tubes arranged in a staggered manner is modeled. The working fluid vapor enters the tube on one end and leaves as condensate on the other. All tubes are identical, and air flows from below the tubes upward as shown in Figure A-13.

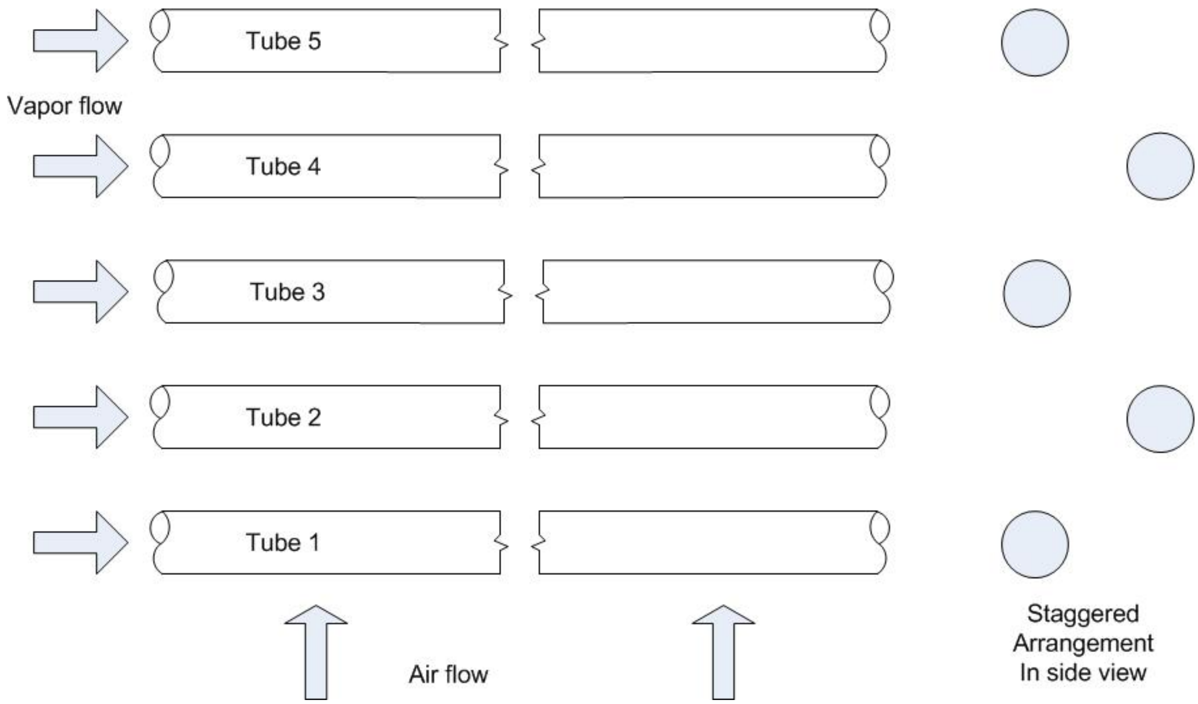
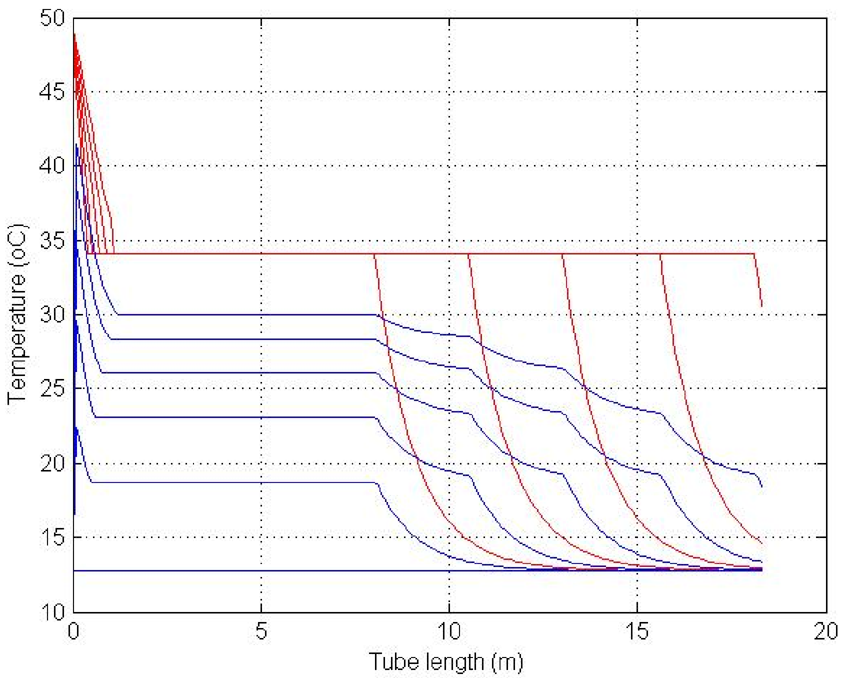


Figure A-13. Single pass arrangement of the condenser tubes as modeled.

The tubes are divided into many segments along the length of the tubes. At each segment, heat and mass balance calculations were performed. In the superheated and subcooled regions, sensible heat transfer occurs. In between, latent heat transfer occurs with decreasing quality with increasing tube length. Heat transfer was calculated based on a constant heat transfer coefficient and the temperature difference between the average vapor temperature and the average air temperature.

As described in the main text, the temperature variations for single-pass cross-flow air cooled condenser are shown in Figure A-14. Distance along the tube length is indicated on the x-axis. The vapor temperatures are indicated in red and the air temperatures in blue in the y-axis. The vapor flows from left to right in Figure A-14, and the tube length is indicated on the x-axis. Full condensation occurs first in tube 1, midway through the tube. The condensate subcools the rest of the way. In the last row of tube, full condensation occurs at the very end of tube 5. The corresponding air temperatures above and below each tube are also shown in Figure A-14. We note that the exit air temperatures are lower at the downstream end of all tubes.



Red *line is for the working fluid temperature.*

Blue *line is for the air temperature*

Figure A-14. Variations of temperature of the working fluid and air along the length of the tube.

From Figure A-14, it is clear that the downstream ends of the tubes are not utilized to their full extent and that subcooling occurs when it is not needed.

To avoid subcooling, a countercurrent flow arrangement between the two fluids is needed. To approximate a countercurrent flow, we arrived at a configuration where the vapor flows from the top tube to the bottom tube in steps. For convenience, we chose a set of five tubes for illustration. Figure A-15 shows the tube vapor flow and air flow arrangement.

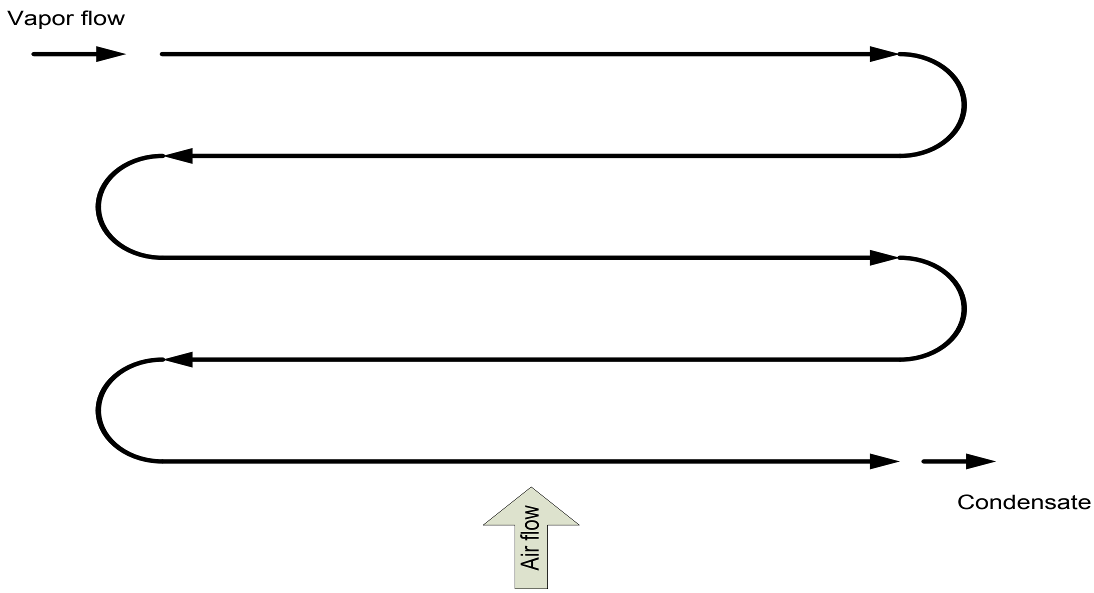


Figure A-15. Countercurrent arrangement for the working fluid and air using folded tubes.

With an odd number of tubes, vapor enters on one side and the condensate can be collected on the other. The overall tube length is adjusted in the analysis to achieve full condensation at the end of the tube.

The vapor is assumed to enter the tube at saturation temperature with a quality of 1. It remains at this temperature throughout the tube as its quality decreases to 0. The air temperature increases as it passes over each tube as shown in Figure A-16. Since subcooling is eliminated in this arrangement, the required overall tube length turns out to be about 25% less than the single pass case. Correspondingly, the air flow requirement is also reduced by the same amount.

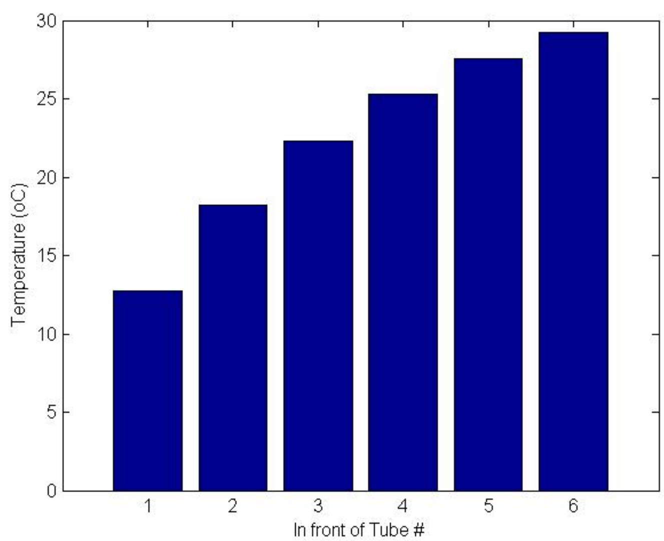


Figure A-16. Air temperatures in between tube rows.

Since the tubes are now shorter, the overall height of the condenser tube bundle can be lowered while still providing adequate air intake area. This height reduction may result in lower overall ACC system cost.

The current analysis does not take into account the vapor-side pressure losses or header modifications. Introduction of pressure losses will reduce the tube length reduction somewhat.