Hybrid Cooling Systems for Low-Temperature Geothermal Power Production

Andrea Ashwood and Desikan Bharathan

NREL is a national laboratory of the U.S. Department of Energy, Office of Energy Efficiency & Renewable Energy, operated by the Alliance for Sustainable Energy, LLC.

Technical Report
NREL/TP-5500-48765
March 2011

Contract No. DE-AC36-08GO28308

NATIONAL RENEWABLE ENERGY LABORATORY

Hybrid Cooling Systems for Low-Temperature Geothermal Power Production

Andrea Ashwood and Desikan Bharathan

Prepared under Task No. ARGT.0910

NREL is a national laboratory of the U.S. Department of Energy, Office of Energy Efficiency & Renewable Energy, operated by the Alliance for Sustainable Energy, LLC.

National Renewable Energy Laboratory
1617 Cole Boulevard
Golden, Colorado 80401
303-275-3000 • www.nrel.gov

Technical Report
NREL/TP-5500-48765
March 2011

Contract No. DE-AC36-08GO28308

NOTICE

This report was prepared as an account of work sponsored by an agency of the United States government. Neither the United States government nor any agency thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States government or any agency thereof.

Available electronically at http://www.osti.gov/bridge

Available for a processing fee to U.S. Department of Energy
and its contractors, in paper, from:

U.S. Department of Energy
Office of Scientific and Technical Information
P.O. Box 62
Oak Ridge, TN 37831-0062
phone: 865.576.8401
fax: 865.576.5728
email: mailto:reports@adonis.osti.gov

Available for sale to the public, in paper, from:

U.S. Department of Commerce
National Technical Information Service
5285 Port Royal Road
Springfield, VA 22161
phone: 800.553.6847
fax: 703.605.6900
email: orders@ntis.fedworld.gov
online ordering: http://www.ntis.gov/help/ordermethods.aspx

Cover Photos: (left to right) PIX 16416, PIX 17423, PIX 16560, PIX 17613, PIX 17436, PIX 17721

Printed on paper containing at least 50% wastepaper, including 10% post consumer waste.

Acknowledgements

The authors would like to thank Dr. Chuck Kutscher, Mike Wagner, and Tom Williams of the National Renewable Energy Laboratory for their constructive comments and suggestions. This project was funded under the American Recovery and Reinvestment Act of 2009, with funding provided by Dr. Jay Nathwani, Program Manager, U. S. Department of Energy Geothermal Technologies Program.

List of Acronyms

ACC	air-cooled condenser
ACHX	air-cooled heat exchanger
C	Celsius
DCC	direct-contact condenser
EPRI	Electric Power Research Institute
Hg	Mercury
kg/s	kilograms per second
kW	kilowatt
kW/K	kilowatt per Kelvin
kWe	kilowatt electric
m^2	meter squared
MPR	market price reference
MW	megawatt
MWe	megawatt electric
MW/K	megawatt per Kelvin
NERC	North American Electric Reliability Corporation
RMOTC	Rocky Mountain Oilfield Testing Center
Pa	Pascal (metric unit for pressure)
psi	pounds per square inch
TOD	time-of-delivery
TMY	Typical Meteorological Year
UA	overall heat transfer coefficient times the heat transfer area
µm	micrometer
WCHX	water-cooled heat exchanger
W/m^2K	watt per meter squared Kelvin

Executive Summary

The overall objective of this investigation is to identify and evaluate methods by which the net power output of an air-cooled geothermal power plant can be enhanced during hot ambient conditions using minimal amounts of water.

Geothermal power plants that use air-cooled heat rejection systems experience a decrease in power production during hot periods of the day. This decrease in power output typically coincides with the time when utilities need power to address high air conditioning loads. Hybrid cooling options, which use both air and water, have been studied for this report to assess how they might mitigate the net power decrease.

Hybrid cooling options can be used in sites where some water is present for supplemental cooling, though not enough for a fully wet-cooled system. This report addresses binary power plants that use a hydrocarbon as the working fluid and utilize an air-cooled condenser (ACC) for heat load rejection. We considered two configurations to mitigate losses in power production: 1) evaporative pre-cooling of the ACC inlet air (without the use of any added heat exchanger) and 2) the use of a water-cooled condenser/heat exchanger in parallel or series with the ACC (or an air-cooled heat exchanger (ACHX)) to split the total condenser load.

Steam cycles, though not currently used in industry for low temperature geothermal resources, were also analyzed.

An indirect method of cooling, called the Heller system (which is currently not utilized in geothermal power production), was analyzed for both steam and binary plants. In the wet-cooling assisted Heller system, an ACHX is placed in series with a water-cooled heat exchanger. The Heller system can also be used with pre-cooled inlet air, though it was not explicitly studied in this analysis. This report contains analyses of the following:

1) ACC and Heller dry-cooled systems. These options were modeled for both binary and steam power plants as baseline cases. Water-assisted systems were then modeled for comparison to the baseline.

2) Systems that pre-cool the inlet air to the ACC, such as using wetted-media, fogging, and spray systems. The deluging of an ACC was also studied. These methods do not use an added heat exchanger. Since low temperature geothermal plants are typically binary cycle power plants, these analyses were only performed for the binary cycle power plants.

3) An ACC in parallel with a wet-cooled surface condenser (hybrid ACC system) was studied for both the binary and steam cycle power plants.

4) A wet-cooled heat exchanger in series with the ACHX used in the Heller system (hybrid Heller system) was analyzed for both the binary and steam cycle power plants.

In this study, we looked at using water to carry a nominal 30% of the heat rejection load from the power plant. By limiting the duration of operation with wet-assist to 1,000 hours during a year, the overall water consumption by the plant was capped at less than 3.5% of the water use in a fully wet-cooled power system.

A basic air-cooled plant requires added equipment to implement wet-assist schemes. For the various schemes, we evaluated the cost for the added equipment. We also evaluated the incremental power produced and the associated incremental revenue for these schemes. The overall benefit of the wet-assist is evaluated in terms of payback periods. The shorter the payback, the better the system is in an economic sense.

The payback periods for each system are detailed below.

Binary Systems

- Hybrid ACC System: The payback period for the hybrid ACC 125°C resource temperature plant varies from 4.5 to 4.7 years (as the water cost was varied from $0.3-$2.46 per thousand gallons). For the 158°C resource temperature hybrid ACC plant the payback periods are longer, varying from 5.7 to 6.1 years (as the water cost was varied from $0.3-$2.46 per thousand gallons).
- Hybrid Heller System: The payback for the 158°C resource temperature hybrid Heller plant varies from 3.8 to 4.0 years (as the water cost was varied from $0.3-$2.46 per thousand gallons). For the colder resource temperature plant, the payback periods are somewhat longer, ranging from 6.6 to 7.2 years (as the water cost was varied from $0.3-$2.46 per thousand gallons).
- Fogging System: The high cost of the system results in payback periods of 6.1 years (assuming a water cost of $1.38 per thousand gallons and that time-of-delivery (TOD) rates apply) for the 158°C resource temperature. The payback period for the 125°C resource temperature plant was 6.5 years (assuming a water cost of $1.38 per thousand gallons and that TOD rates apply).
- Spray System: The payback period for the 158°C resource temperature plant was 0.60 years (assuming a water cost of $1.38 per thousand gallons and that TOD rates apply). The payback for the 125°C resource temperature plant was 1 year (assuming a water cost of $1.38 per thousand gallons and that TOD rates apply).
- Deluge System: The payback period for the 158°C resource temperature deluge system plant was 0.13 years (assuming a water cost of $1.38 per thousand gallons and that TOD rates apply). The payback period for the 125°C resource temperature plant was 0.10 years (assuming a water cost of $1.38 per thousand gallons and that TOD rates apply).
- Wetted-Media System: Payback periods were 9.4 years for the 158°C resource temperature plant and 7.4 years for the 125°C resource temperature plant (assuming a water cost of $1.38 per thousand gallons and that TOD rates apply).

Steam Systems

- Hybrid ACC System: The payback period for the hybrid ACC system varies from 1.12-1.14 years (as the water cost was varied from $0.3-$2.46 per thousand gallons).
- Hybrid Heller System: The payback period from the hybrid Heller plant is 1.2-1.24 years (as the water cost was varied from $0.3-$2.46 per thousand gallons).

The payback period, however, does not tell the whole story. For each of the evaluated schemes, there are many advantages and disadvantages. One of the key considerations in our evaluation is that the wet-assist system should not interfere with the normal plant operation when the wet-assist is not operational (or needed).

With these criteria in mind, we find the following two systems as the most practical for use.

1) Pre-cooling the inlet air to the air-cooled heat rejection system using sprays. In this scheme, commercially available misting nozzles are placed in a grid in the path of the intake air. Mist eliminators are introduced downstream of the sprays to capture un-evaporated water droplets. The mist eliminators must be carefully selected to minimize air-side pressure loss. Pre-cooling of the inlet air has the potential to cool the air down close to its wet-bulb temperature with an effectiveness of about 75%. This scheme is effective in dry climates where there is a large difference between the air dry-bulb and wet-bulb temperatures. Payback for these systems was less than 2 years for both resource temperatures, assuming TOD rates are applicable.
2) Introduction of a wet-assist heat exchanger/surface condenser (hybrid ACC). In this scheme a conventional wet cooling tower is added to the system. Water from the tower takes heat away from either an added surface condenser or from the hot coolant. The tower and water streams are sized to handle about 30% of the overall heat rejection load from the plant. The other 70% of the load is carried by the air-cooled heat rejection system. This scheme uses conventional technology with readily available off-the-shelf commercial equipment. It is easy to implement. The payback period for this type of system was estimated to range from 4.5 to 6.1 years.

Considering the above two schemes, we find that the second approach requires little in terms of research and development. The first scheme, however, is suitable as a retrofit to existing air-cooled power plants. It requires evaluation of spray nozzles, manifolding, mist eliminators, and their effectiveness in actual plant operation. We propose to implement the pre-cooling inlet air approach at the air-cooled power plant currently operational at the Rocky Mountain Oilfield Testing Center (RMOTC).

Table of Contents

List of Figures

List of Tables

1 Introduction

Geothermal power plants use the heat from the earth to generate electricity. A recent summary of the technologies involved in establishing and operating geothermal power plants is provided by Kagel [Kagel 2008]. Power plant systems, working fluids, and configurations vary with the type and temperature of the available geothermal resource.

Of specific concern in this study are power plants that operate with hydrothermal resources in the low temperature range, nominally bracketed by 125° and 175°C. In this range of temperatures, the industry practice is to use a binary cycle, where heat is exchanged to a working fluid, such as isobutane or isopentane. The working fluid cycles through an evaporator, turbine, condenser, and pump to generate electrical power. This analysis, however, will also look at steam cycle power plants where the brine exchanges heat to closed-loop water/steam, which cycles through an evaporator, turbine, condenser, and pump to generate electrical power, using a Rankine cycle.

Geothermal power plants often use air-cooled heat rejection systems due to a lack of water in the arid sites where the plants are typically located. Heat rejection occurs in air-cooled condensers (ACC) with either forced-draft or induced-draft airflow. Air-cooled plants that operate using low temperature resources suffer a large loss in production capacity (as much as 50%) during times of the year when the air temperature (dry-bulb) is high. This loss is a result of increased condenser pressure and typically occurs during the afternoon hours in the months of June through September (during summer in the Northern Hemisphere) and at times when utilities need power to fulfill high air conditioning demands. High-demand times for electricity are also times when electricity sale prices attain their peak in the open market. Thus, any increase in power production using water-assisted cooling during hot ambient temperatures offers a potential increase in revenue for the power plant.

1.1 Background

The loss of production capacity with increasing ambient air temperature is well documented for geothermal power plants that use air as the heat load rejection medium. A comprehensive review of the basics of ACC design and performance is addressed in a 2005 report issued by the Electric Power Research Institute (EPRI) [EPRI 2005].

Early studies by Bamberger and Allemann [Bamberger 1982] assessed dry/wet cooling options for a 5-MWe geothermal plant at the Raft River power plant in Cassia County, Idaho. They summarized all of the prior works related to hybrid cooling investigations and concluded that evaporative condensers offered the least cost among systems studied for the replacement of an existing ACC.

More recently, Kanoglu and Çengel [Kanoglu 1999] analyzed a nominal 27 MW binary cycle air-cooled power plant, which could use water to evaporatively pre-cool the inlet air to the ACC to improve the power plant performance during hot days. Their analysis showed that if the inlet air temperature could be reduced close to the wet-bulb temperature, the result would be a 29% increase in net power output.

Bharathan and Nix [Bharathan 2001] studied the use of ammonia absorption pumps to improve power production in an air-cooled geothermal power plant. They found that increasing the brine flow rate in an absorption heat pump by 30% resulted in a 21% increase in net plant power. Such a scheme does not use additional cooling water, only the brine flow is increased. Kutscher and Gawlik [Kutscher 2000] evaluated alternative configurations for the fins of an ACC to improve condenser performance. Their results showed an increase in condenser performance, but also an increase in the pressure drop in the condenser.

Kutscher and Costenaro [Kutscher 2002] assessed the potential of using evaporative cooling enhancements to an ACC, and Kozubal and Kutscher [Kozubal 2003] looked at using a wet-cooled condenser in series with an ACC to increase the net power output during hot periods of the day in binary geothermal power plants.

Kutscher and Costenaro found that a deluging condenser system offered the lowest payback period and Kozubal and Kutscher found that the series arrangement of the ACC and wet-cooled condenser resulted in an increase in net power output, but the system had low economic viability. EPRI examined the use of water spray to cool the inlet air to the ACC in an effort to enhance the condenser performance [EPRI 2003]. They found that, depending upon the assumed price for electricity during high demand times, the payback period for the enhancement ranged from 1 to 3 years.

Because of the increasing demand and decreasing availability of water, air cooling is rapidly becoming the choice for heat load rejection for many thermal power plants. Geothermal power plants, which operate at a lower resource temperature and system efficiency than fossil fuel plants, suffer relatively greater losses in power production when using air cooling.

1.2 Objective

The goal of this project is to identify and analyze alternative heat rejection strategies that allow air-cooled geothermal power plants to both maintain a high electrical power output during periods of high ambient air dry-bulb temperatures and minimize plant water consumption. Cost and economic viability of the added systems are addressed in terms of simple estimated payback periods.

1.3 Scope of Study

We limited the scope of this study to a few air-cooled geothermal power plants. One of the plants investigated is a nominal 20 MW power plant, such as the existing binary cycle power plant in northern Nevada [Kanoglu 1999]. This plant was analyzed at its design resource temperature of 158°C.

A hypothetical 10 MW binary cycle plant was also analyzed with a lower resource temperature of 125°C, and a nominal 20 MW steam cycle power plant was analyzed with a resource temperature of 175°C.

Finally, a much smaller binary cycle plant, which has a capacity of 250 kW and a resource temperature of 92°C, located at the Rocky Mountain Oilfield Testing Center (RMOTC) was analyzed with the intent of using it for field testing and verification of the water-assisted heat load rejection systems.

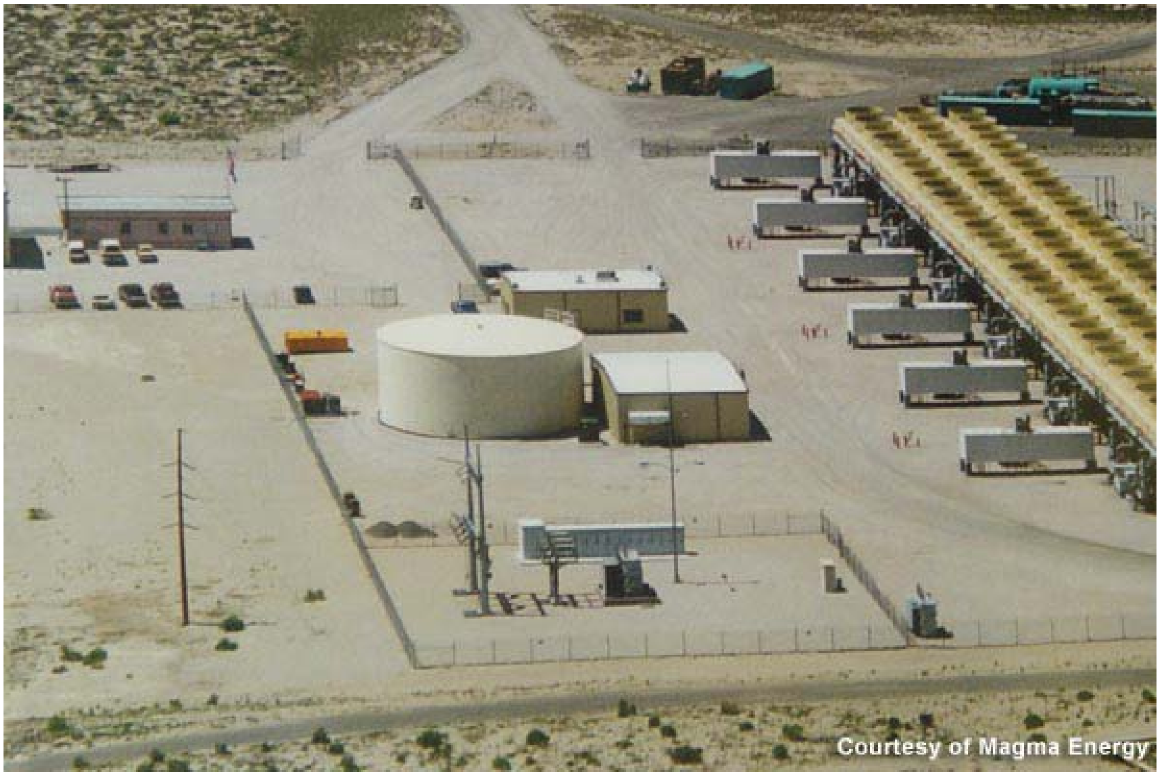

Figure 1. Photograph of the northern Nevada air-cooled binary cycle geothermal plant

2 Baseline Power Plants

2.1 Binary Cycle

2.1.1 20 MW (158°C resource temperature) ACC Power Plant

For this study, an existing air-cooled power plant with a nominal 20 MW net electric production capacity (depicted in Figure 1) was modeled. Many plant details are provided by Kanoglu and Çengel [Kanoglu 1999]. The working fluid used in the plant is isobutane. This plant suffers an approximate 50% loss of production capacity from winter to summer conditions. A smaller, hypothetical plant, with a nominal 10 MWe production output was also modeled for comparison.

The power plant was modeled using the commercially available process modeling software ASPEN Plus. A consistent set of assumptions regarding component efficiencies and heat-exchanger approaches were utilized in the models. Table 1 summarizes the nominal design conditions for various parameters. Except for the brine resource temperature (158^{o}C) and the ambient temperature at design (3°C), all of which were taken from the Kanoglu work [Kanoglu 1999], all of the values in Table 1 are approximated based on industry standards.

Table 1. Summary of Assumed Parameter Values for Analyses

Design conditions for ambient air	Value	Units
Dry-bulb temperature	3/10	^{o}C
Ambient pressure	101,322	Pa
Design conditions for cooling water		
Temperature	25	^{o}C
Pressure	101,322	Pa
Component efficiencies		
Turbine/generator	0.87	(---)
Brine/water/other pumps	0.69	(---)
Cooling air fan	0.52	(---)
Brine availability		
Temperature(s)	125/158/175	^{o}C
Component Operation		
Heat exchanger pinch point Delta-T	5	^{o}C
Air cooler/condenser air-side pressure drop	175	Pa

Figure 2 shows a schematic of a basic binary cycle power plant. The brine heat exchanger acts both as a preheater and boiler. An ACC is shown at the exhaust of the turbine. For organic working fluids, on account of their skewed thermodynamic dome characteristics, a superheater is not necessary to avoid moisture related problems (such as in the case of steam).

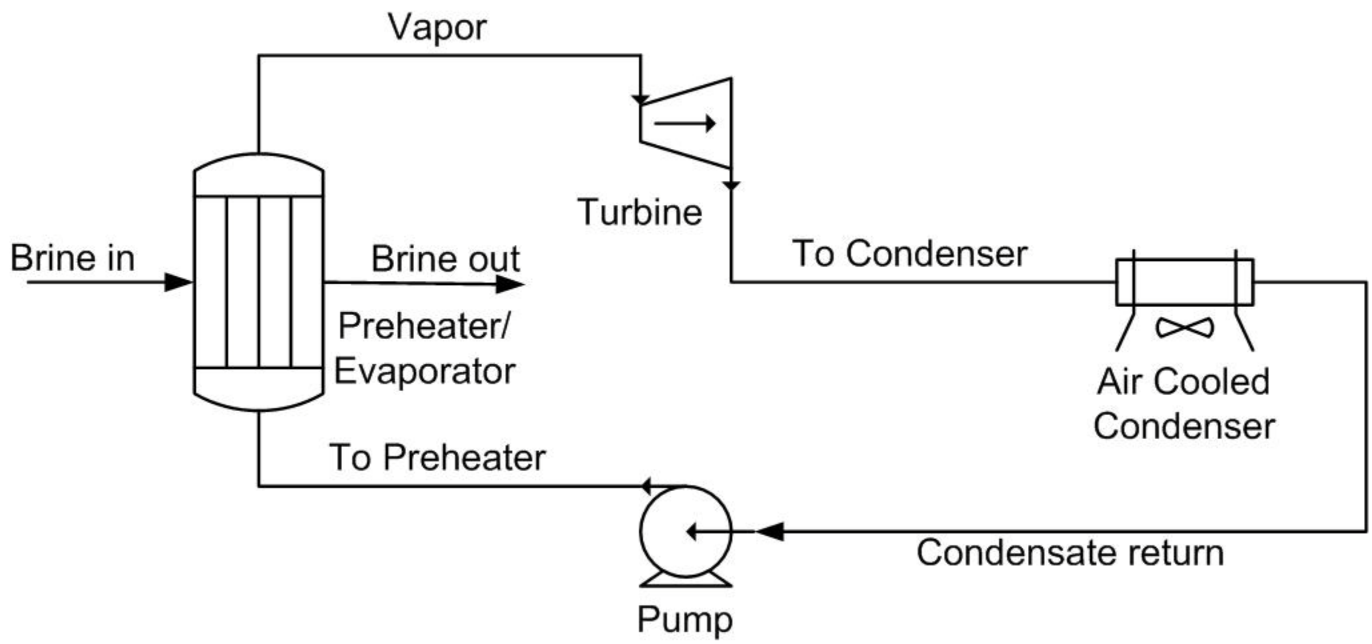

Figure 2. Basic schematic diagram of a binary cycle geothermal power plant

Figure 3 illustrates the plant process conditions as modeled in the ASPEN Plus simulation. The working fluid goes through a preheater and superheater. The superheater operates at pressures close to the critical pressure for isobutane.

In practice, the working fluid is not superheated prior to entering the turbine. However, in order to match the operating conditions quoted by Kanoglu, a small amount of superheat was necessary for the model. This may be because of the use of the thermodynamic property tables used in the model since explicit fluid property information was not indicated in the work by Kanoglu.

The vapor is expanded through the turbine and the turbine exhaust leaves at a superheated condition with an approximate 46°C superheat. The ACC desuperheats and condenses the vapor. Though the desuperheater and the ACC are shown as two separate entities in the simulation, in practice their functions occur in a single ACC.

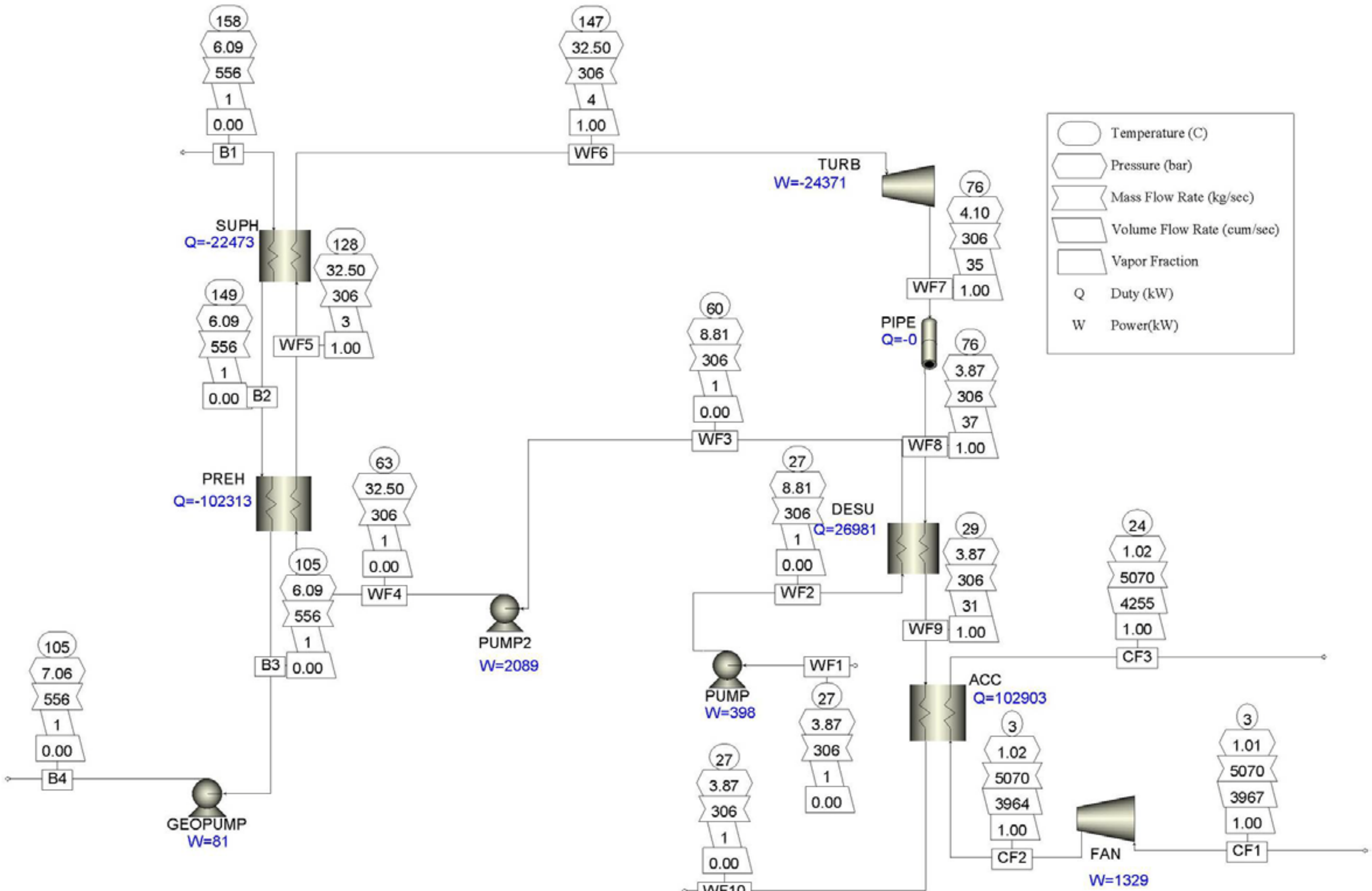

Figure 3. Binary cycle power plant with an ACC (158°C resource temperature) at design conditions as modeled in ASPEN Plus

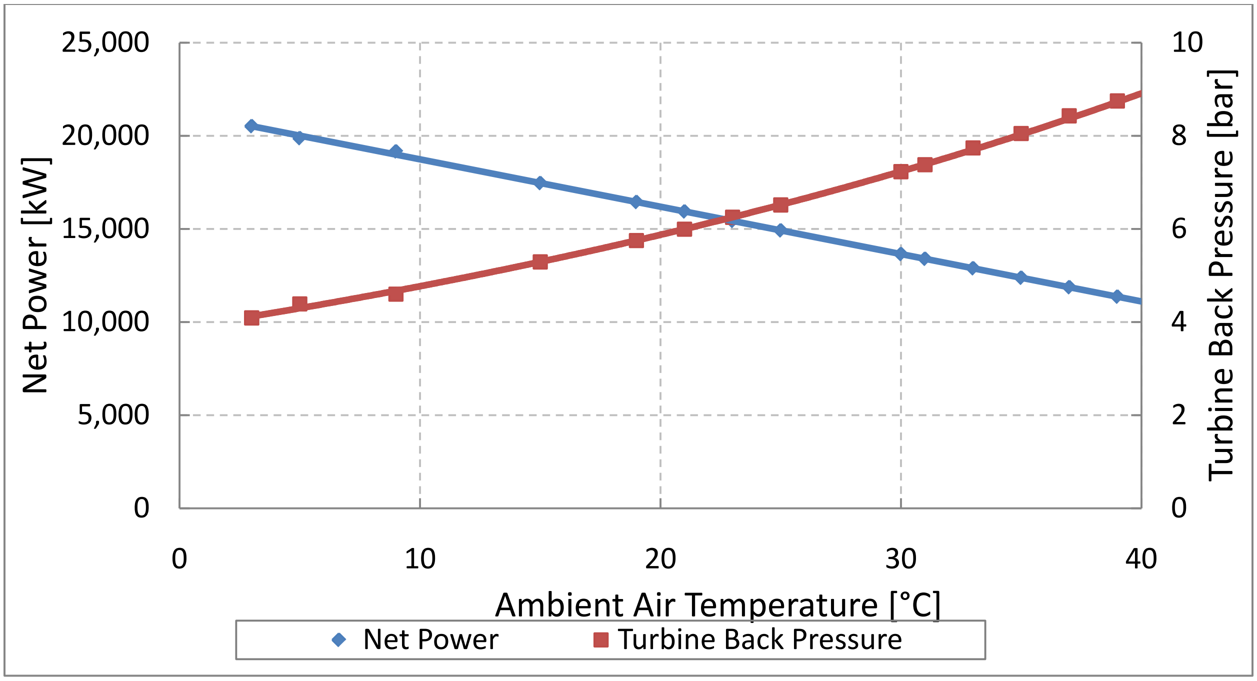

Figure 4. Variation of the power production capacity and turbine back pressure with ambient air temperature for the ACC binary (158°C resource temperature) cycle power plant

Figure 4 shows the predicted net power output and the turbine back pressure as functions of the ambient air temperature. Note that the condenser suction pressure would be lower than the turbine backpressure due to losses in the distribution pipes required for vapor. The back pressure on the turbine was modeled as the minimum pressure necessary for the vapor to be condensed at the exit of the ACC. The total power and turbine back pressure curves illustrated in Figure 4 were then idealized with best fit curves.

The net power from the plant decreases linearly with increasing temperature. For every 1°C that the ambient dry-bulb temperature is increased there is an approximately 2% decrease in net power output. The turbine back pressure, and thus the condenser pressure, increases exponentially with increasing air dry-bulb temperature.

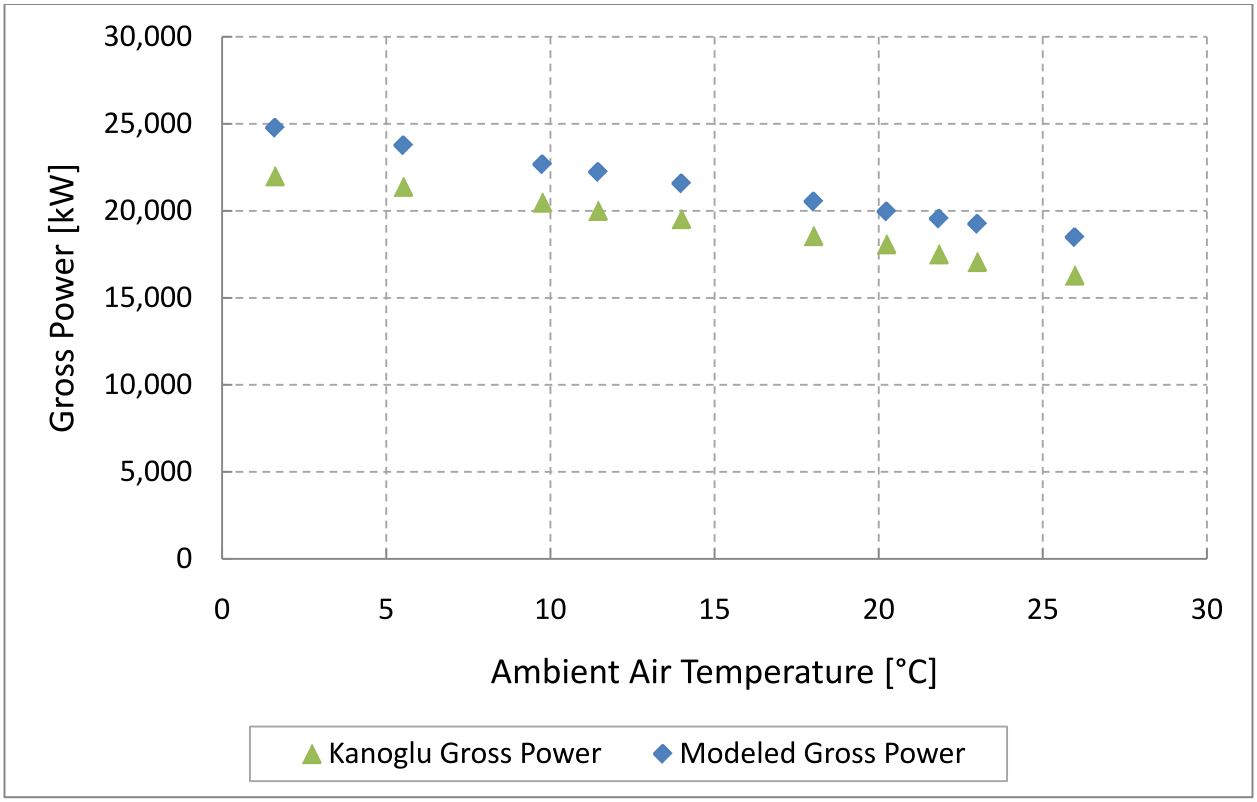

Figure 5. Comparison of the variation of the actual and modeled gross power with ambient temperature for the ACC binary (158°C resource temperature) cycle power plant

For model verification, we compared the gross power as provided by Kanoglu and Çengel [Kanoglu 1999] and the gross power obtained from our analysis as shown in Figure 5. The plant gross power is plotted with varying ambient air temperature. The gross power decreases with increasing ambient temperature. The modeled gross powers are approximately 2-3 MW larger than those provided by Kanoglu and Çengel at all temperatures resulting in a mean absolute error of approximately 10%.

This difference can be attributed to differing assumptions in equipment efficiencies. The turbine back pressure and the condenser pressure are not explicitly stated by Kanoglu, neither is the air flow through the ACC. The difference in gross power could also be due to differences in assumptions related to the condenser behavior. Our analysis assumes that the condenser operates in a counter-current manner, since heat exchangers can only be modeled in ASPEN Plus in a counter-current or co-current configuration, as opposed to the cross-flow pattern that would actually occur in the operation of an ACC.

2.1.2 10 MW (125°C resource temperature) ACC Power Plant

A similar set of analyses were carried out for a hypothetical power plant operating at a brine resource temperature of 125°C. We chose the plant design output at 10 MWe net for a modular plant for the following analysis. The air dry-bulb temperature at design was 3°C. Figure 6 shows the variation of the plant net power with ambient dry-bulb temperature. The net power decreases with increasing temperature at a rate of approximately 198 kW/K. This represents a 26% decrease compared to the 158°C resource temperature ACC plant.

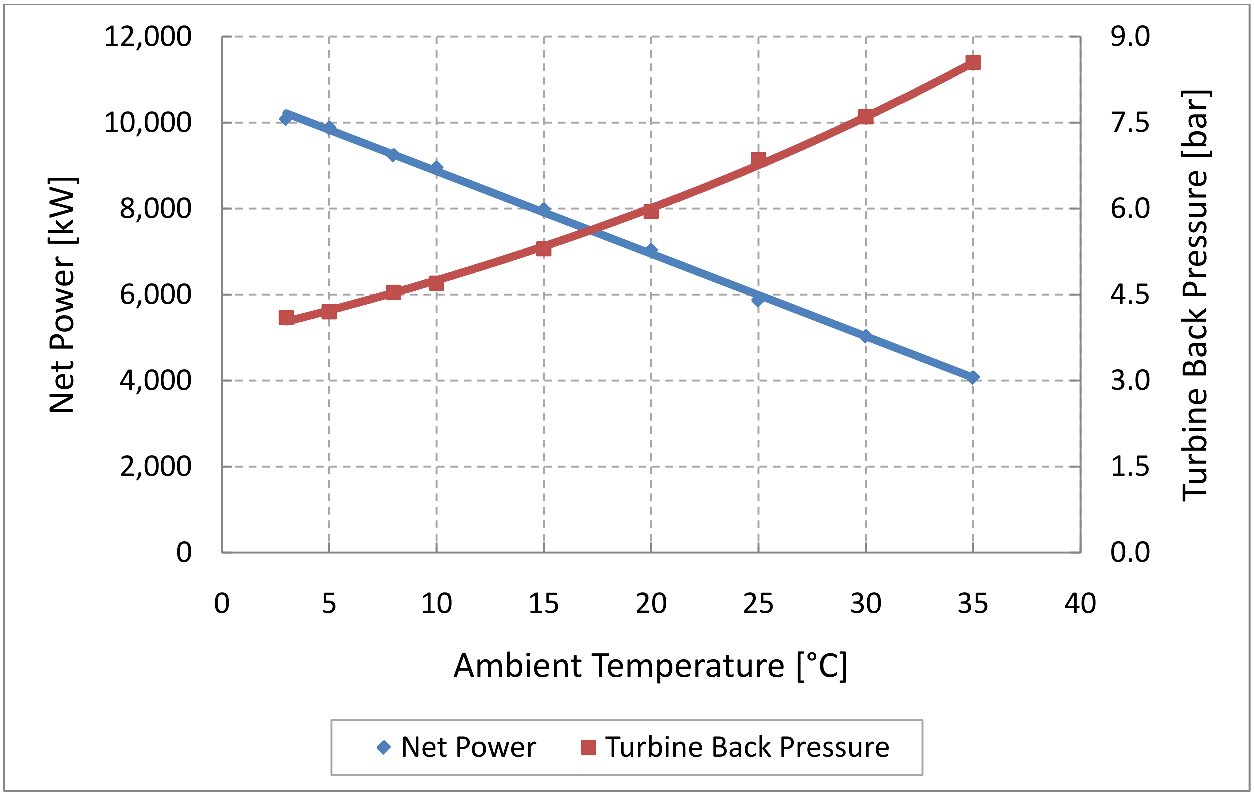

Figure 6. Net power variation with ambient temperature for the ACC binary (125°C resource temperature) cycle power plant

2.1.3 20 MW (158°C resource temperature) Heller System Power Plant

An indirect cooling method, also called a Heller system, is a system that cools the condensate generated in a direct-contact condenser (DCC) using air. The Heller system is usually associated with steam power plants; however, it can be used with other working fluids as well. We note that no indirect cooling systems as described in this section are operational in the United States. Those Heller systems that do exist in Europe are used in fossil power plants with a much larger power output than the systems described here.

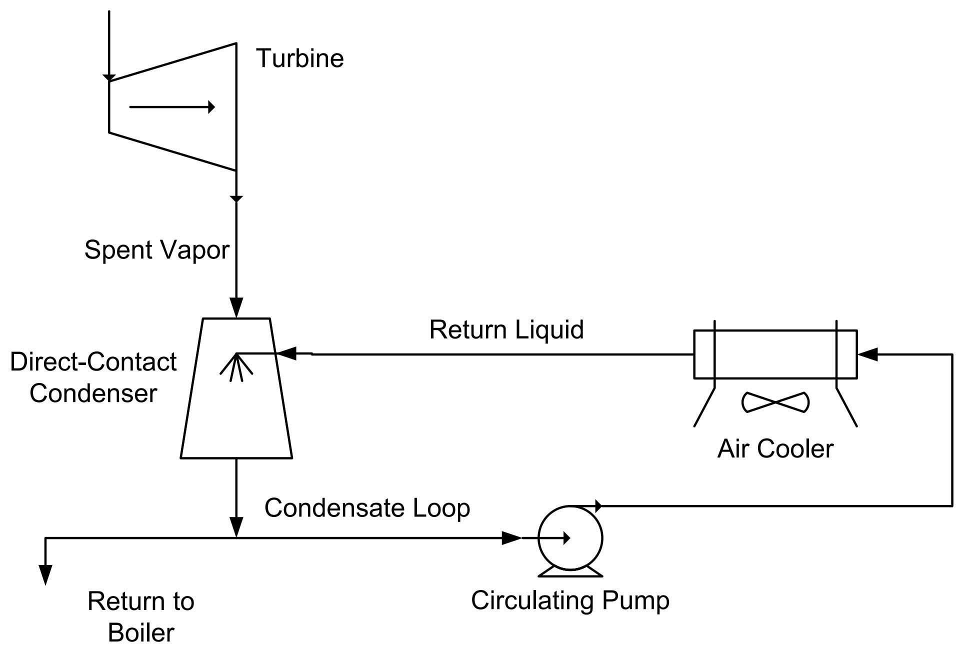

Figure 7. Schematic of the Heller system

The Heller system, shown in Figure 7, uses an indirect cooling configuration, where the vapor is condensed in a DCC and the condensate (working fluid) is sent to an air-cooled heat exchanger (ACHX) for rejecting the heat. The working fluid forms a closed loop sending the liquid back and forth between the DCC and ACHX, with the required condensate returned to the boiler feed.

The Heller system has several inherent advantages and disadvantages compared to the ACC system.

The advantages include:

1) The design offers a more simplified distribution arrangement in the ACHX for the liquid, with potentially “true” countercurrent flow of air and liquid.
2) The design also offers a new control variable, namely, the flow rate of the liquid, to allow for further flexibility in plant operation. The flow rate can be increased (or decreased) to allow for higher (or lower) operating temperature driving potential at the ACHX. The Heller system thus offers a means to increase the ACHX performance at high ambient temperatures without the use of any additional water.
3) The ACHX can be located away from the power plant. The heat exchanger modules can be arranged in a circular manner to take advantage of the natural tendency of the hot air to rise, and thus lower required fan power. The circular arrangement can also be less prone to local changes in prevailing wind flow and direction.

The disadvantages include:

1) Added complexity with the introduction of a secondary loop to carry the rejected heat, and its necessary piping.
2) A significant loss of temperature driving potential for the ACHX, which increases the heat exchanger surface area needed.
3) An additional pumping power requirement for the fluid loop.

Figure 8 shows the design operating conditions for the power plant using the Heller system for the 158°C resource temperature plant. In this model, the DCC is a pressure vessel with a height requirement of about 15 feet and is modeled simply as a mixer and splitter for present purposes.

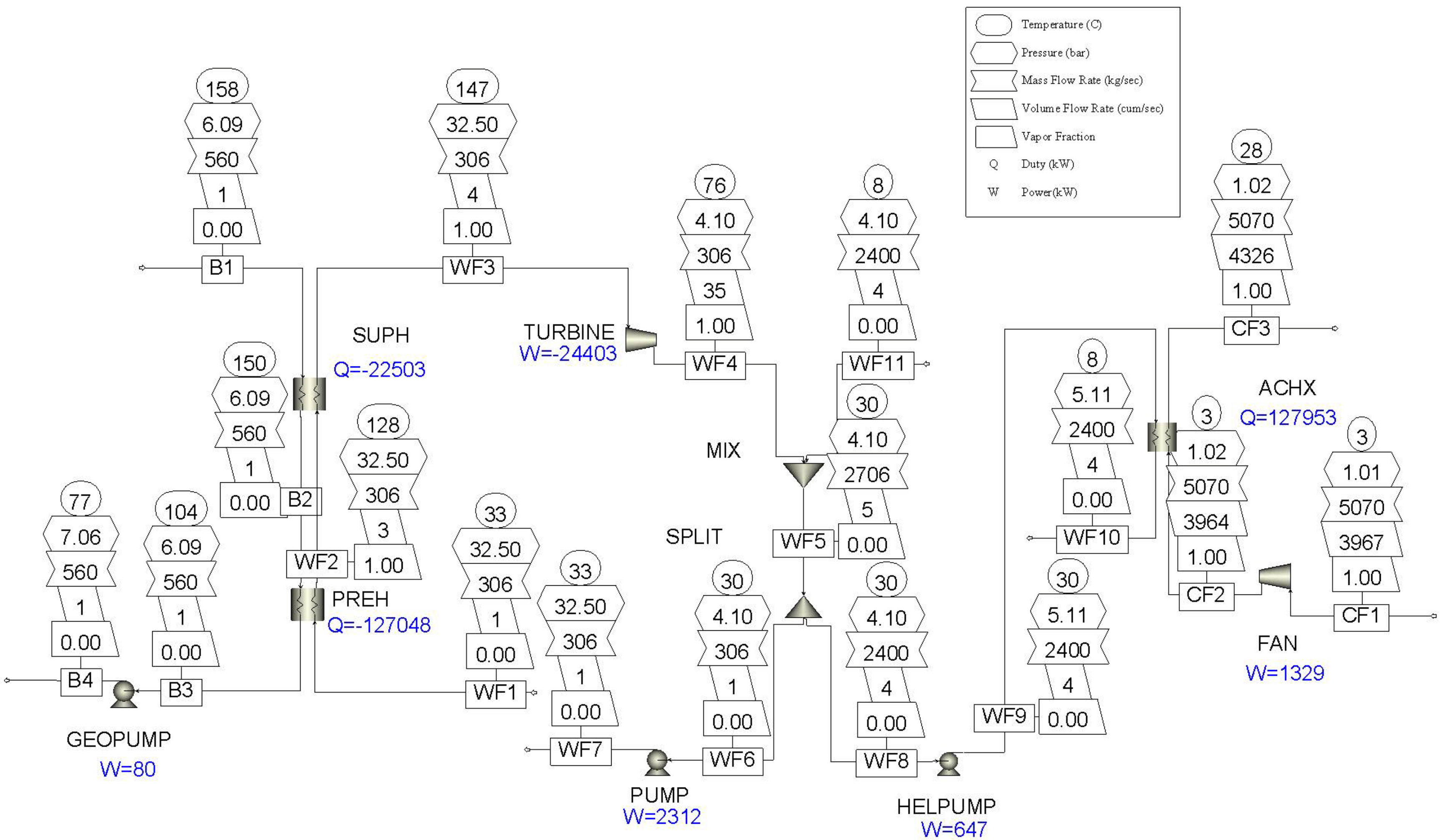

Figure 8. Design case for the baseline binary (158°C resource temperature) cycle Heller system

The performance of the system at varied ambient dry-bulb temperatures was simulated with the components fixed at design. The fan air flow and the secondary loop circulating flow rates were fixed at design conditions.

Figure 9 shows the net power production and condenser pressure as functions of the ambient dry-bulb temperature. The rate of net power reduction with ambient temperature is about 280 kW/K.

In comparison, the ACC system with the same resource temperature (158°C) resulted in a rate of 254 kW/K. Thus, the Heller system exhibits a 9.2% higher rate than the ACC at the same resource temperature. This is because the Heller system avoids vapor pressure losses in piping and is able to operate at lower condenser pressures. The large vapor distribution ducts required for an ACC are eliminated. The Heller system also performs better than the ACC system at lower temperatures.

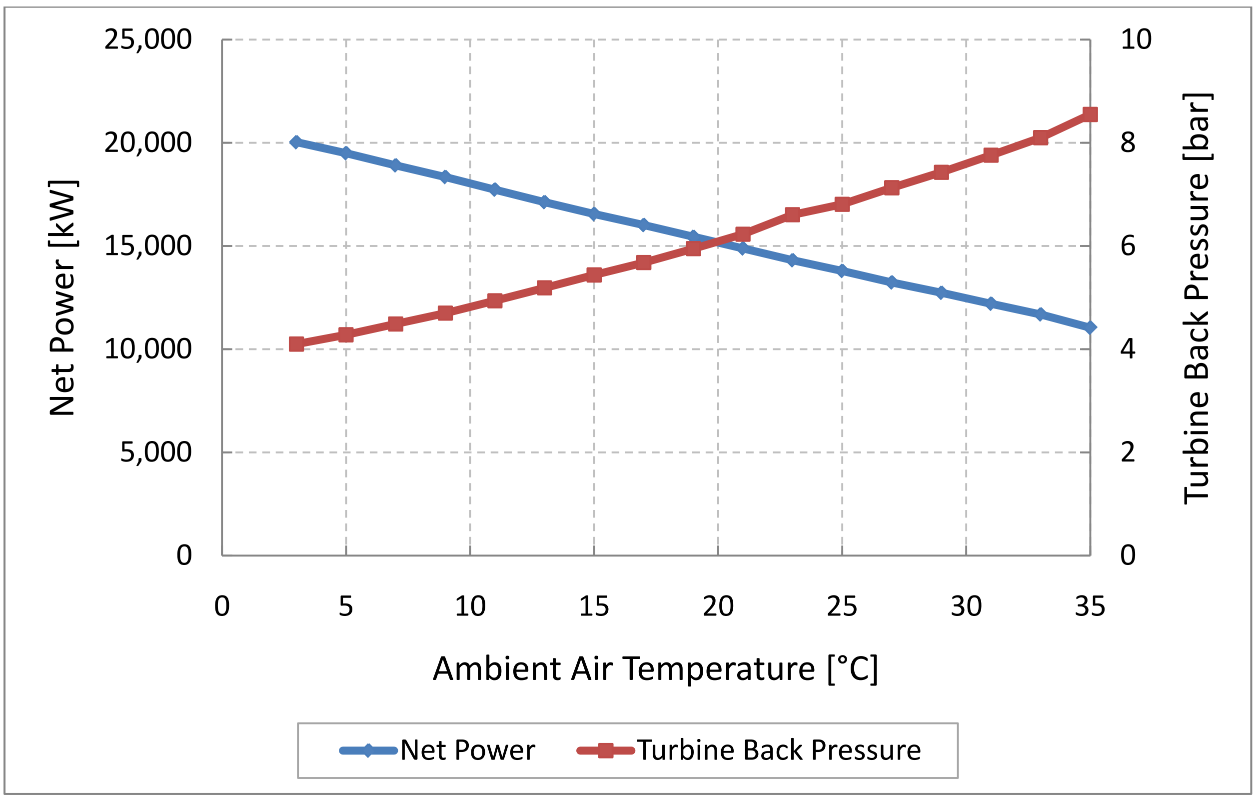

Figure 9. Variation of the net power and condenser pressure with ambient air temperature for the binary (158°C brine resource temperature) cycle Heller system

2.1.4 10 MW (125°C resource temperature) Heller System Power Plant

A similar set of analyses were carried out for a hypothetical Heller system power plant operating at a brine resource temperature of 125°C. Figure 10 shows the variation of the plant net power with ambient dry-bulb temperature. Again, the plant was modeled with the lowest condenser pressure necessary to condense all the vapor; the power and turbine curves are shown with idealized curve fits.

The net power decreases with increasing temperature at a rate of approximately 196 kW/K. This rate is 30% less than the rate of the 158°C resource temperature Heller system plant.

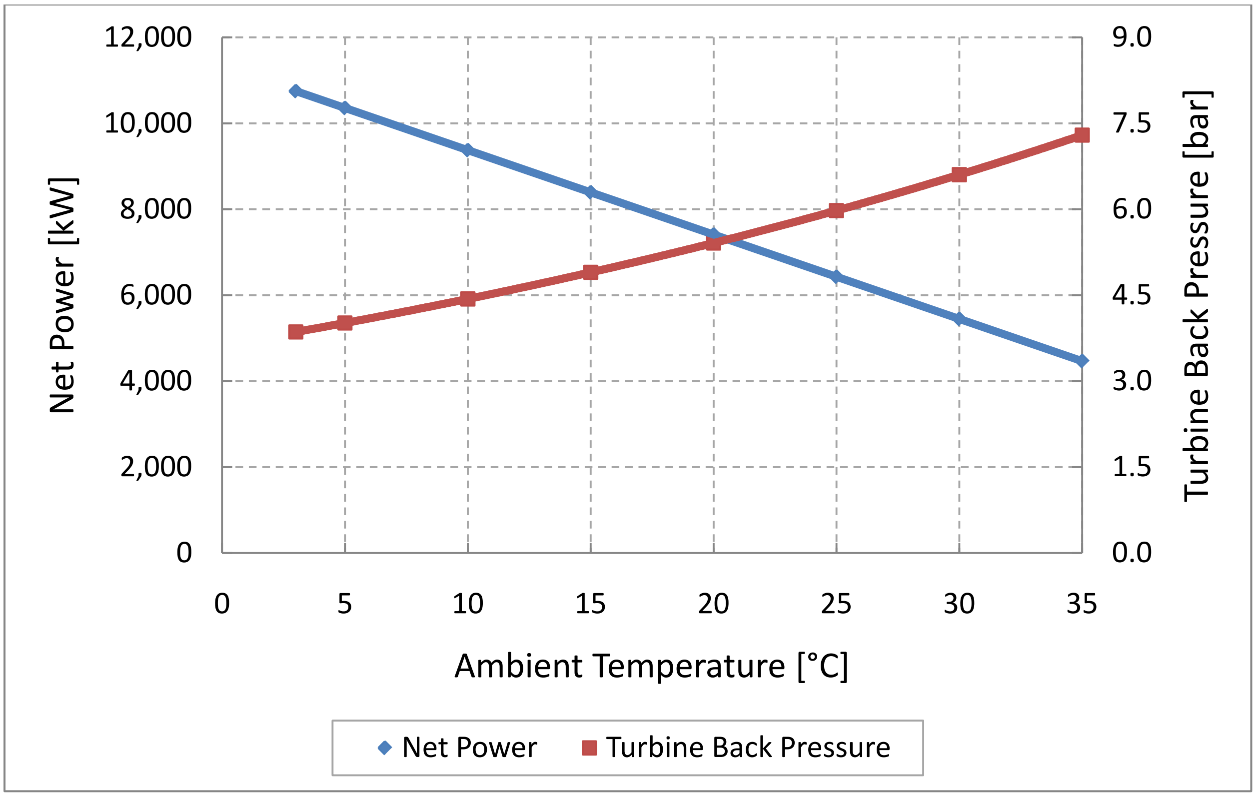

Figure 10. Variation of the net power and condenser pressure with ambient air temperature for the binary (125°C brine resource temperature) cycle Heller system

2.1.5 Baseline Binary Cycle Power Plant Result Summary

The Heller system power plants have a higher power output than the ACC systems for both resource temperatures. The rate at which the power decreases with increasing ambient temperature for the baseline Heller system for the 158°C resource temperature plant is 9% greater than the ACC plant with the same resource temperature. For the 125°C resource temperature, the rate is 4% greater with the Heller system plant than with the ACC plant.

The Heller system power plants have a higher power output than the ACC power plants at lower ambient air temperatures and the systems are able to operate at lower condenser pressures when compared to the ACC plants.

2.2 Steam Cycle

2.2.1 ACC (175°C resource temperature) Power Plant

For the purpose of this report, a steam cycle refers to plants that use steam as the working fluid in a closed loop to enable cooling with air. A power plant with a nominal design net power of 20 MWe was modeled using a brine resource temperature of 175°C. For the purposes of these analyses, the plant evaporator is modeled as a flash/condense system to generate the cooling water and keep the working fluid (steam) separate from the brine, as illustrated in Figure 11. For improved brine use, the evaporator can also be arranged as a simple heat exchanger; where the hot brine effluent can be used downstream in a flash/condense operation to generate usable fresh water.

In either case, it is possible to generate adequate amounts of fresh water to meet the needs for intermittent water use for heat rejection during hot periods of the day.

The drawbacks of this system, however, are that the reservoir may suffer a loss of pressure due to the water draw and that non-condensable emissions from the brine would have to be addressed.

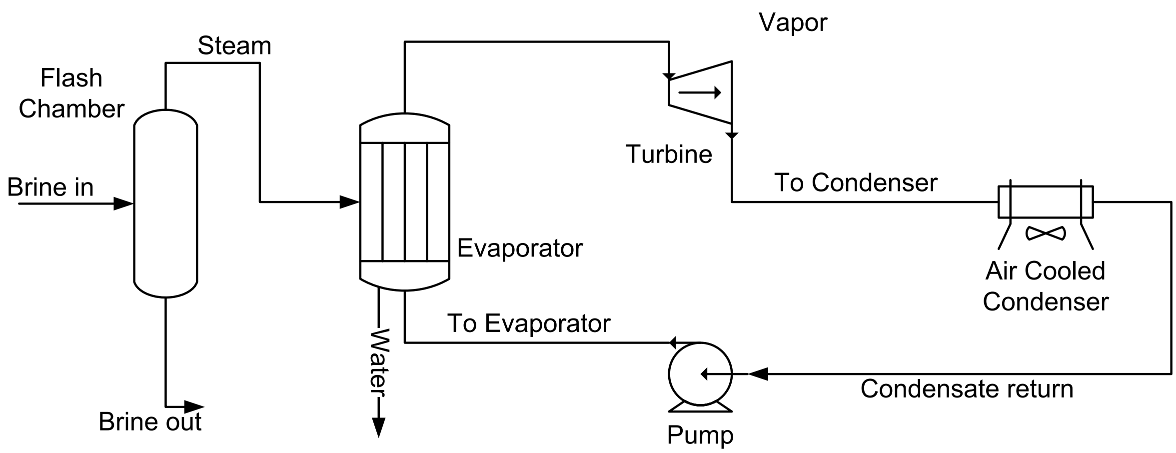

Figure 11. The ACC system for a steam cycle power plant

Figure 12 shows the steam cycle plant operating parameters for the design case with an ACC, as modeled in ASPEN Plus. The hot brine (at 175°C) enters a flash chamber and generates steam. The steam condenser/evaporator preheats and boils the working fluid, in this case, water. An approach temperature of 5°C is maintained in all heat exchangers. The steam in the binary loop expands through a turbine and is then condensed in an ACC. At design conditions, cooling air is available at 10°C.

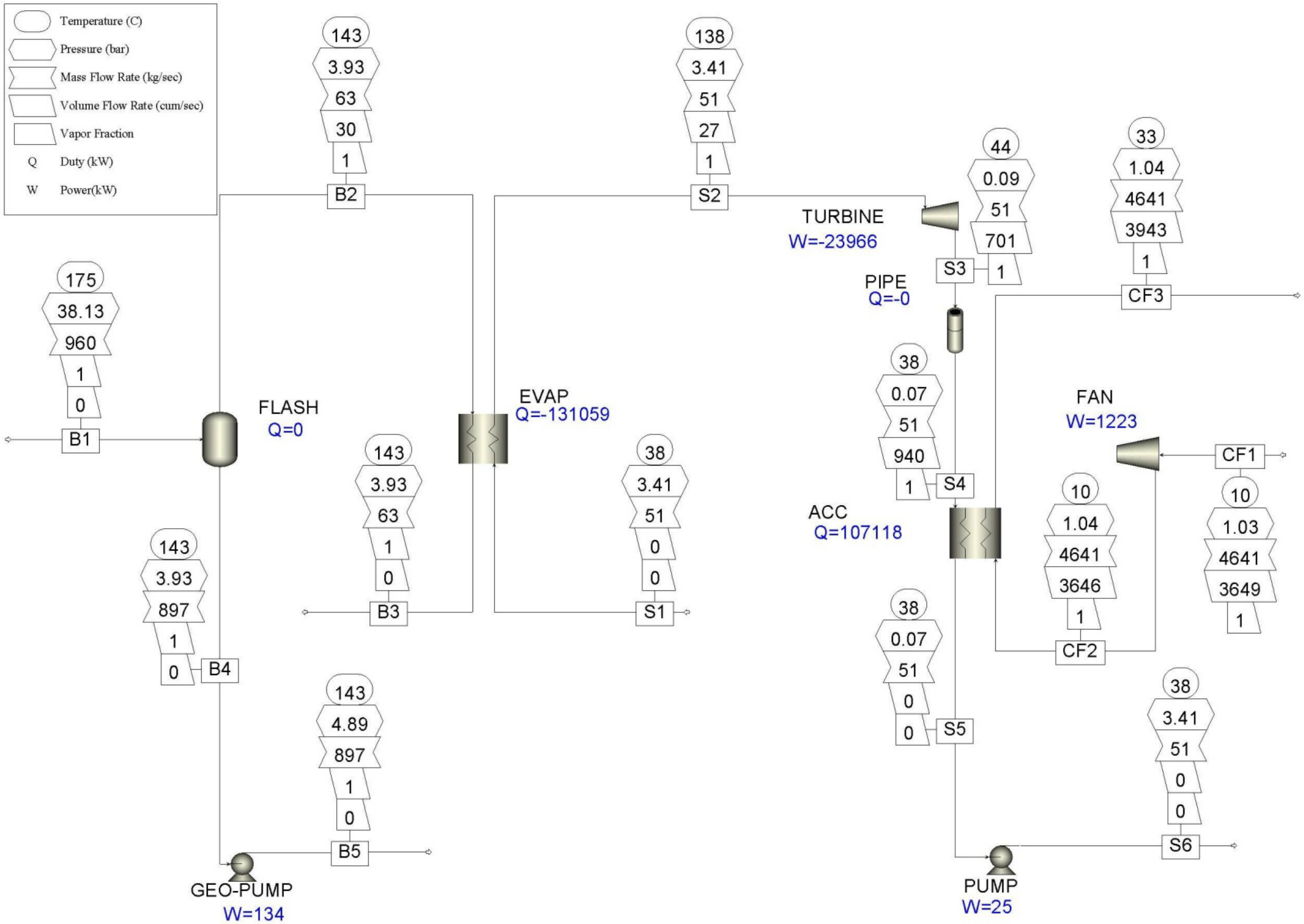

Figure 12. Plant design details for a geothermal ACC steam (175°C resource temperature) cycle power plant

The ACC system has a few advantages and disadvantages compared to the Heller system. To reiterate, it provides the highest temperature driving potential for the heat rejection. However, the condensing vapor must be piped to the ACC using large vapor distributor lines. The ACC also occupies a large land area. For a steam plant with an ACC, a minimum condenser pressure of 0.068 bar (2" of Hg) was used at design based on industry standards.

Due to losses in the distribution pipes for steam in the ACC system, this resulted in a turbine back pressure about 0.02 bar greater than the condenser suction pressure. Such pressure loss reduces the power that can be generated by the turbine. The ACC also limits the lowest sink temperature to the air dry-bulb temperature.

Figure 13 shows the performance of the steam cycle power plant as a function of the air dry-bulb temperature. The net power once again decreases with increasing air temperature at approximately 240 kW per °C temperature rise. This decrease is a direct result of the exponentially increasing condenser pressure, also illustrated in Figure 13.

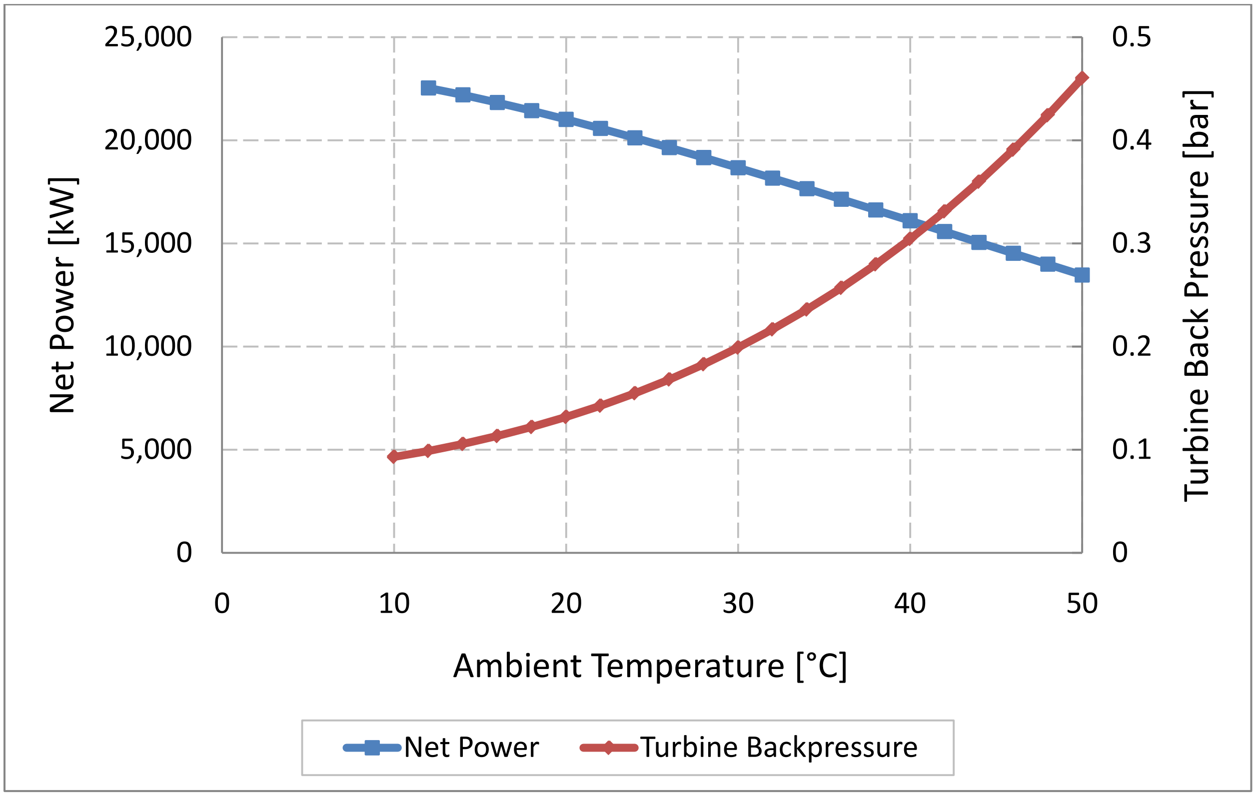

Figure 13. Variation of the power production capacity and turbine back pressure for the steam (175°C resource temperature) cycle ACC system

2.2.2 Heller System (175°C resource temperature) Power Plant

The Heller system is described in Section 2.1.2. The ASPEN Plus simulation of the steam Heller system is shown below in Figure 14.

The Heller system has many advantages and disadvantages, most of which were summarized in Section 2.1.2. An added advantage due to the use of steam instead of a binary working fluid is that this design dramatically reduces the volume of vapor that must be piped to the ACC. Figure 14 shows the design operating conditions for the power plant using the Heller system. Only the components downstream of the turbine are shown. The overall performance of the power plant is similar to the case using a binary working fluid, discussed in Section 2.1.2.

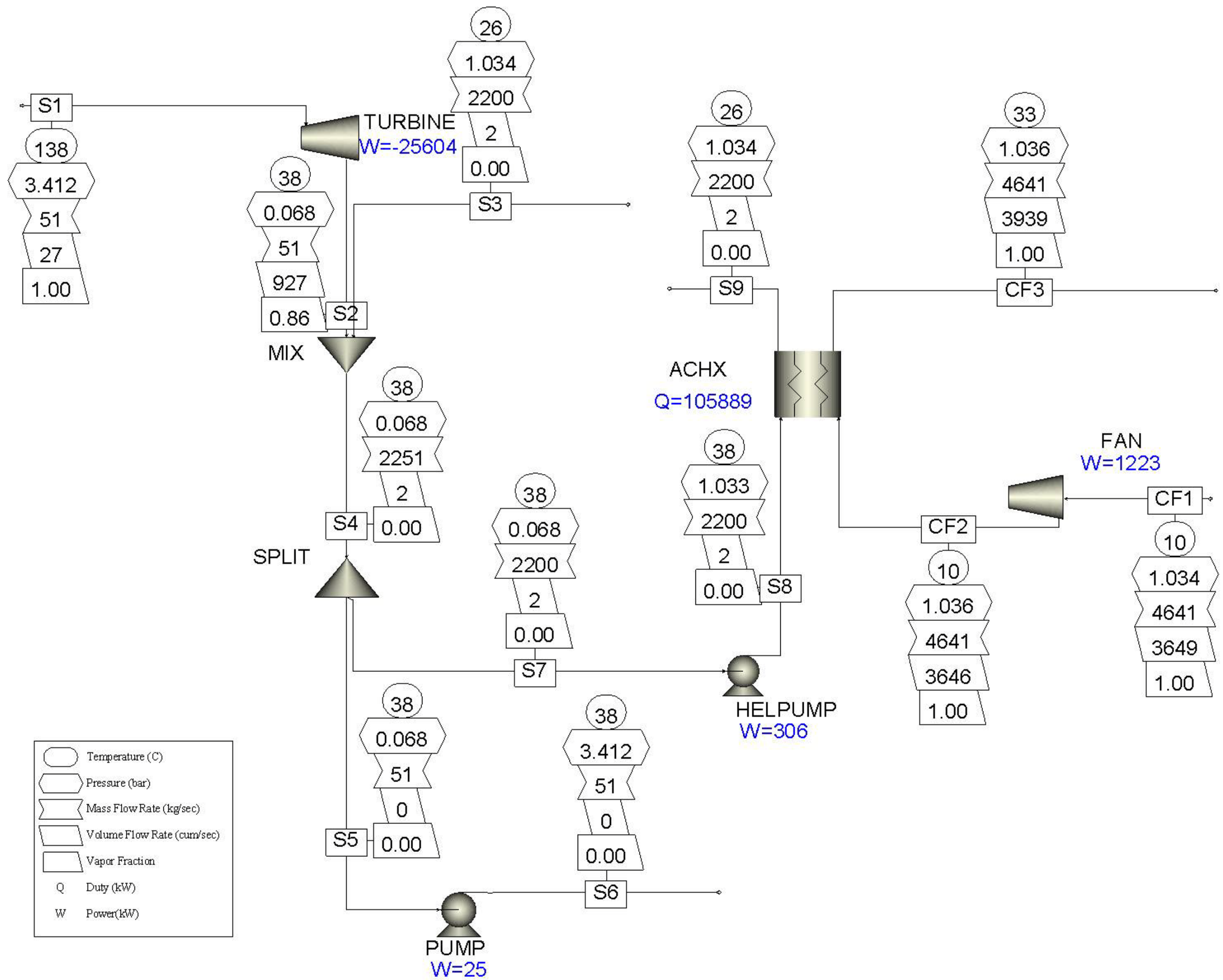

Figure 14. The Heller system design operating conditions for the steam (175°C resource temperature) cycle power plant

Figure 15 shows the net power production and condenser pressure as functions of the ambient dry-bulb temperature for the Heller system steam cycle plant. The rate of net power reduction with increasing ambient temperature is 274 kW/K, which is 12% higher than the steam cycle ACC power plant.

Figure 15 includes two lines, one with the condensate loop flow fixed at design conditions and another with this flow doubled. (In this illustration, neither an increased pressure drop penalty for the loop nor any improvement in heat transfer performance in the ACHX is accounted for). By doubling the loop flow, the temperature rise in the coolant is halved for the same duty. Therefore, the loop coolant temperature drifts higher, increasing the temperature driving potential for the ACHX. Doubling the loop flow increases the net power by about 260 kW (approximately 2%) at all air temperatures, without the use of any additional water.

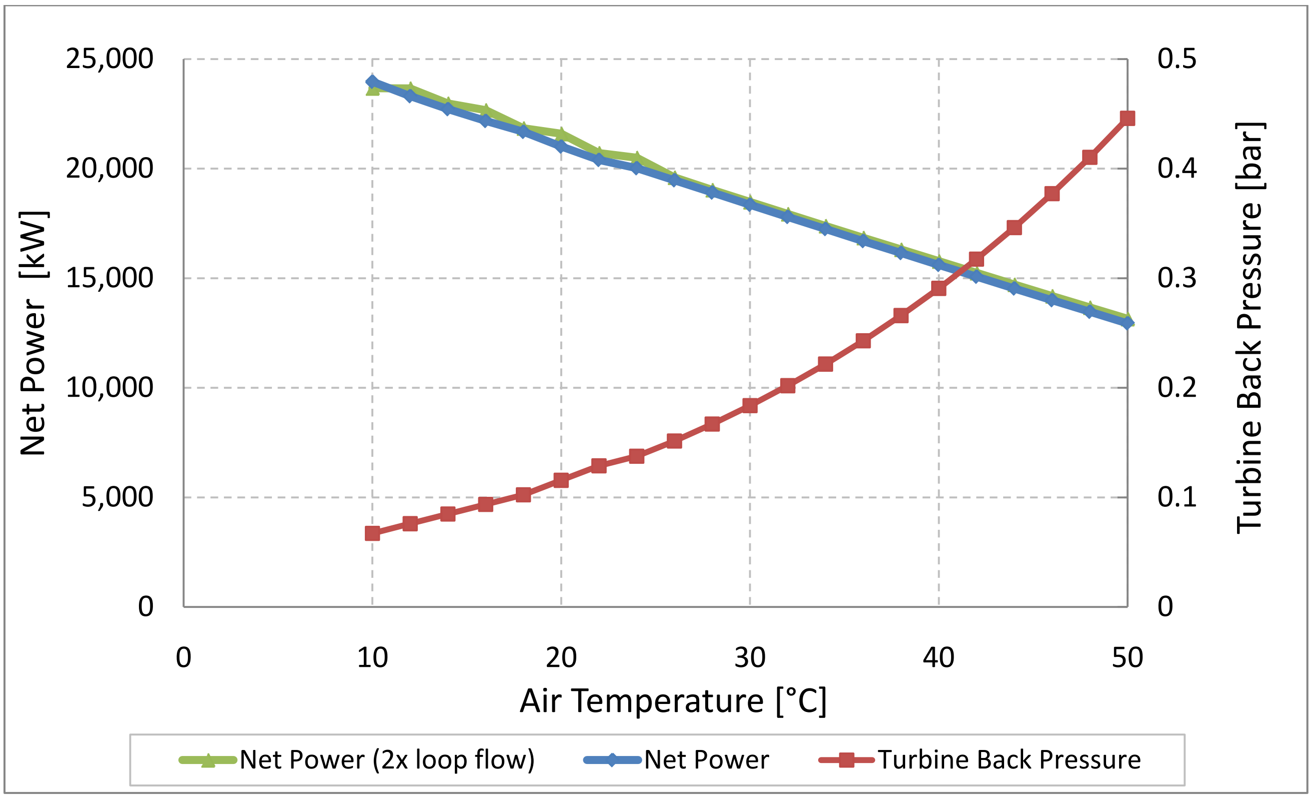

Figure 15. Variation of net power and condenser pressure for the steam (175 °C resource temperature) cycle Heller system; the influence of doubling the loop flow is also indicated

2.2.3 Baseline Steam Cycle Power Plant Result Summary

The doubling of the flow through the secondary loop of the Heller system results in a 2% increase in net power without the addition of any water cooling. The Heller system plant has a 12% higher rate of power reduction per °C increase in ambient air temperature than the ACC system power plant. This results in a large potential to increase the power output with a wet-assist system.

3 Hybrid Cooling Options

Since the baseline air-cooled configurations experience a sharp decline in the net power production during hot periods of the day, hybrid cooling options were also analyzed. For the sake of this report, hybrid cooling refers to use of water in addition to air for plant heat load rejection.

Hybrid cooling can be achieved with several designs. This investigation, however, is confined to the following two methods that introduce wet cooling without added heat exchangers: 1) evaporative cooling of the inlet air and 2) the use of water deluge to cool the air cooler/condenser. Two configurations that require an additional heat exchanger are 1) the use of a parallel wet-cooled surface condenser with an ACC and 2) the use of a wet-cooled heat exchanger in series with the ACHX in the Heller system.

3.1 Pre-Cooling the Inlet Air to the ACC (Evaporative Cooling)

3.1.1 Wetted Media, Spray, or Fog

One method to improve the performance of the ACC (or the ACHX, though not analyzed in this study) is to cool the intake air by evaporating water in the air flow. This method works well in climates where the air is predominantly dry. The following three methods are commonly used to reduce the air temperature, namely, flow through an evaporative water/air contact media or direct water spray or water fog in the intake air. Figure 16 shows schematic diagrams of these processes.

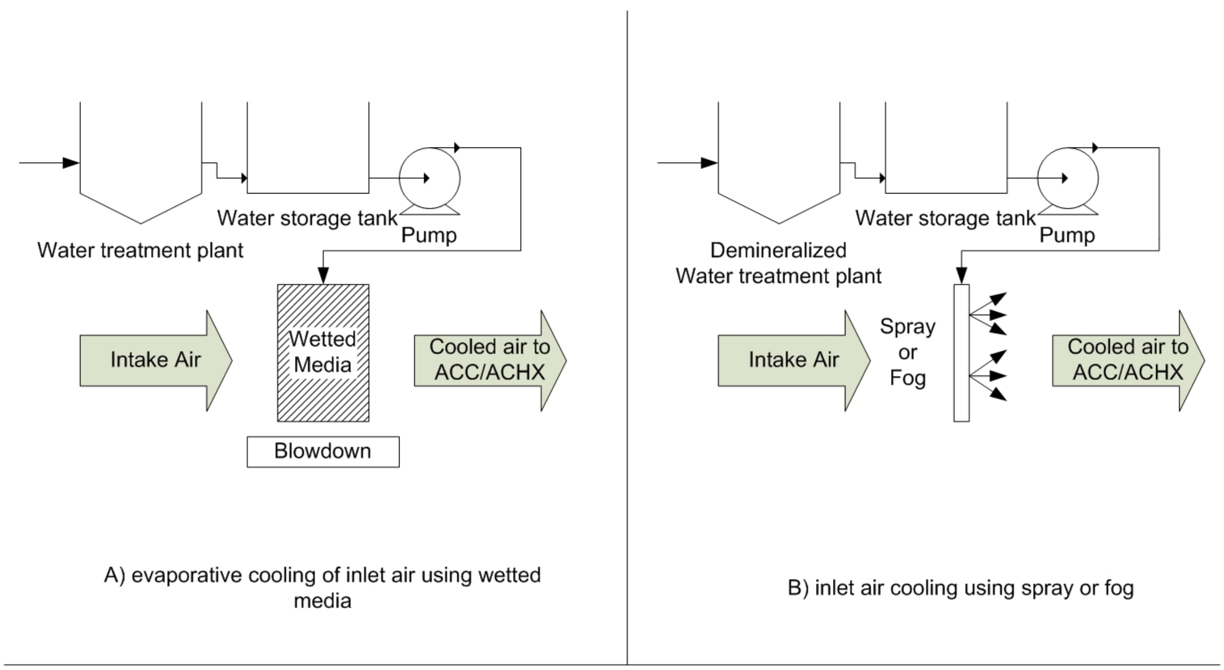

Figure 16. Alternative arrangements for inlet air evaporative cooling
A) Uses wetted gas-liquid contact media; B) Uses water spray or high pressure fog

The intake air enters the contact area to be humidified and cooled. In the case of wetted media, the air can approach the wet-bulb temperature with an effectiveness of 80%. A drain pool below the packing collects the overflow and redistributes it over the contact media. Brackish water may be used in this arrangement, but care has to be taken to minimize biofouling and clogging of the packing. A certain amount of blowdown is necessary to maintain the water within acceptable levels of dissolved solid content.

Work done by Kutscher and Gawlik [Kutscher 2003] at the Mammoth Pacific power plant showed that the wetted media can be placed on hinged doors at air intake areas allowing the operator to remove the packing when it is not in use. Thus, one can avoid excess pressure losses in the air when cooling is not needed. In the particular system tested, they also observed an increase in the pressure drop across the fan, but a decrease in the mass flow rate through the fan. Thus, there was no net increase in the fan parasitic load.

When using high-pressure sprays (1,000-2,000 psi) to cool the inlet air (such as those used for gas-turbine inlet air cooling), a compressed air stream helps generate water sprays of droplet diameters less than 10 µm. This is called a fogging system. Droplet dispersion and overspray is avoided with this type of system because the small droplets can more easily be evaporated into the inlet air stream.

A spray system uses nozzles at a much lower pressure than a fogging system (typically less than 300 psi) and produces larger droplets on the order of 50 µm. Overspray and dispersion of the droplets should be avoided, if possible. Local wind speed and direction may also carry the cooled air away from the needed areas.

Both the spray and fogging systems can saturate the inlet air, i.e., the air can be fully cooled down to its wet-bulb temperature. The purity of the water must be maintained for these systems so that the nozzles do not get clogged and deposits do not accumulate over the condenser fin surfaces. Proper water treatment will add to the system cost. Key advantages of these systems are that they are easily installed with no severe obstructions to the air flow and that they can be turned on and off as needed with a flip of a switch. The systems must be drained during cold seasons to avoid freezing.

3.1.2 Deluge of the ACC

In a deluge system, the finned condenser tubes are deluged using water sprays. Air flows over the wetted condenser tubes. With this method, the air is evaporatively cooled at the condenser tubes. This system requires a large volume of water flow, which must be evenly distributed to wet all of the heat exchanger surfaces. The air stream is able to carry the evaporated water away, fully saturated at the air outlet temperature. A schematic diagram of the wetted surface air cooler is shown in Figure 17.

Once again, brackish water can be used in this arrangement. Only about 3% to 5% of the circulating water stream can be expected to evaporate, requiring large water reflux. The water stream accumulates dissolved matter making a certain amount of blowdown necessary. An advantage to this system from a cost perspective is that the same piece of equipment can be used when air-cooling or water-cooling. However, dissolved solids can leave a residue on the condenser hardware, which can result in corrosion of the condenser tubes and fins. It would be essential to thoroughly wash the deluged tubes often throughout the day to remove all traces of low-quality water. Lack of attention to the washing can cause corrosion of the heat exchanger fins and surfaces and result in potentially costly replacements.

An ideal application of this system is to use a specially fabricated evaporative condenser/cooler that uses no fins and has galvanic corrosion protection. Such hardware is generally available in the industry but at an added cost to the power plant.

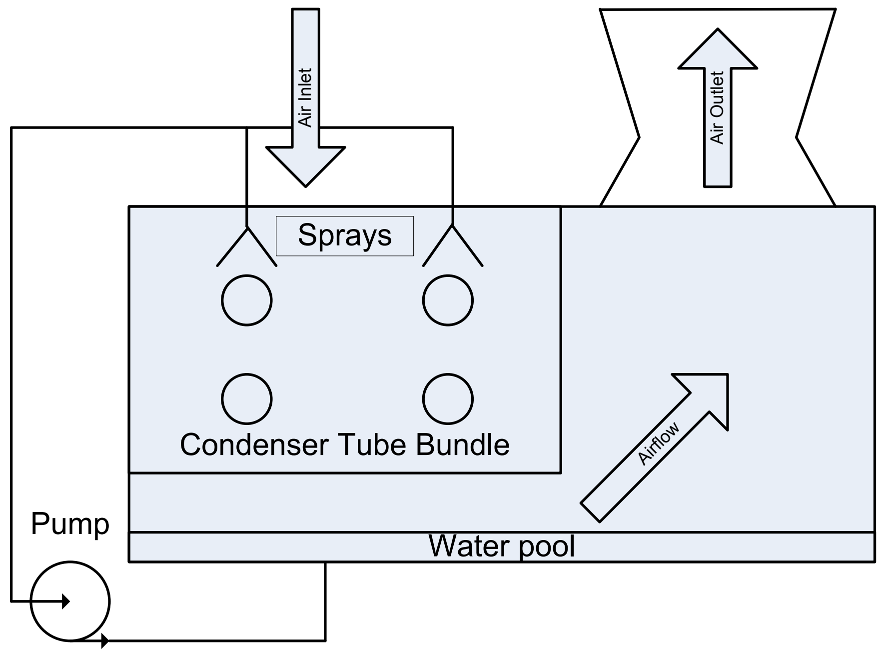

Figure 17. Schematic of an ACC with deluged heat-exchanger surfaces

For the sake of this analysis, it was assumed that the water used in these systems was at a constant temperature of 25°C (see Table 1).

3.2 Hybrid ACC

Another method to improve the ACC performance is to use a wet-cooled surface condenser in parallel with the ACC. This arrangement (hereafter called the hybrid ACC system) allows for the ACC to reject a smaller load and operate at a lower internal temperature difference (ITD), which increases the system performance. Figure 18 shows a schematic diagram of the arrangement.

Depending on the availability of water, a fraction of the turbine exhaust vapor is diverted to a surface condenser. In this analysis, a range of sizes for the water-cooled portion, ranging from no water use to the water use necessary to reject 30% of the total condenser load, were looked into. This surface condenser is supplied with circulating water from a wet cooling tower. While it can be turned on and off at will, freeze protection during cold nighttime conditions is required.

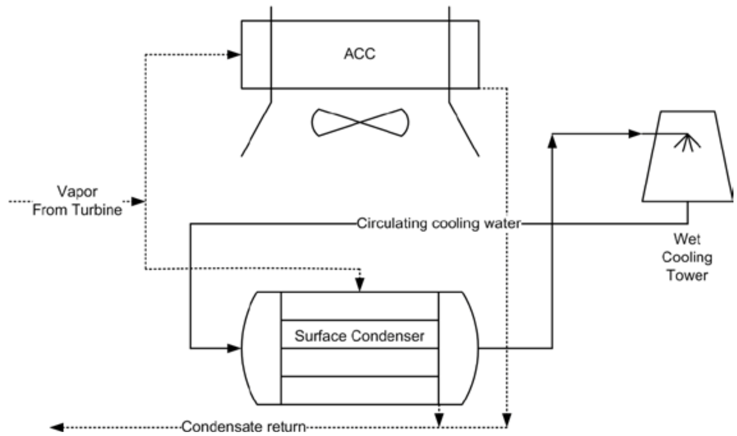

Figure 18. An ACC with supplemental wet-cooled surface condenser in parallel (hybrid ACC)

3.3 Hybrid Heller

The hybrid Heller system uses an auxiliary wet-cooled heat exchanger in series with the ACHX and a wet cooling tower for the added cooling, shown in Figure 19. The series arrangement makes it possible to cool the entire coolant with water for this system, if necessary.

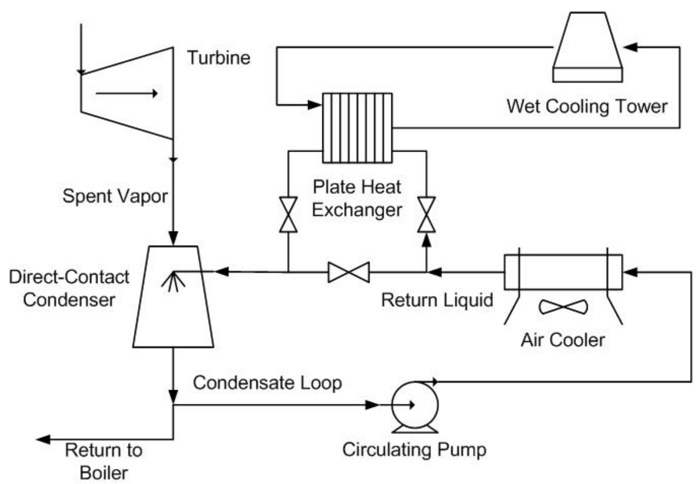

Figure 19. Schematic of the hybrid Heller system (wet-cooled heat exchanger in series with ACHX)

A range of sizes for the wet-cooled portion (ranging from no water use to the water use necessary to reject 30% of the total condenser load) were studied. While the system can be turned on and off at will (via a bypass valve), freeze protection during cold nighttime conditions would also be required.

4 Hybrid Cooling Results

The hybrid ACC and hybrid Heller systems are discussed in this section. As discussed in sections 3.2 and 3.3, the hybrid ACC system is a dry-cooled ACC system with a wet-cooled surface condenser in a parallel configuration and the hybrid Heller system is a dry-cooled Heller system with a wet-cooled heat exchanger in a series configuration.

The pre-cooled inlet air systems (i.e., fogging, spray, wetted media) and deluge systems are also discussed. These systems were analyzed for the binary cycle power plants only and were not modeled directly. Given a particular dry air and water mass flow rates, the extent of temperature reduction in the inlet air is calculated first. Then, the power output from these systems was calculated from the baseline system power curves using the inlet air temperature for the given water consumption rate. Varied approaches to the wet-bulb temperature (saturation efficiency) were used depending on the kind of evaporative cooler used in the system.

4.1 Binary Cycle Power Plants

The primary aim of this effort is to assess the benefits of water use to improve the power yield of a dry-cooled power plant during hot periods of the day. Since the purpose of this analysis is to minimize water use, heat rejection to air will remain the primary and major load carrier for heat rejection. In the following discussions, water assisted heat rejection load is limited to a maximum of 30% of the overall condenser load.

Given a particular rate for the evaporative water used, the sensible load on the ACC is reduced by the amount of heat that can be carried away by the water evaporative flux. A load reduction on the ACC results in lower condenser operating temperature and suction pressure and increased turbine output.

Variations in the plant net power output as functions of the load percentage that the wet-cooled assist carries at two different ambient temperatures are shown in Figure 20. The dry air mass flow was maintained at the design condition to generate these results. At each temperature, as expected, the net power increases with increasing water consumption. The net power increases at a rate of approximately 133 kW per (kg/s) of water consumed, with 11 kg/s representing the maximum consumption at 30% of the total heat rejection load.

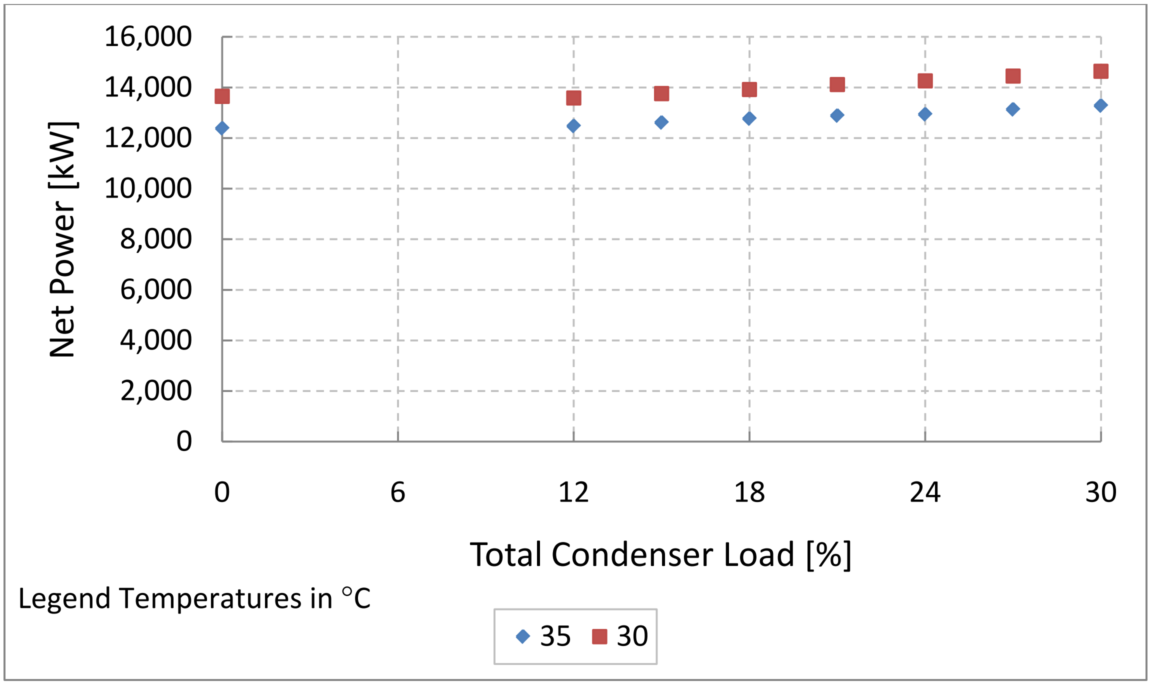

Figure 20. The net power variation as a function the wet-assist load fraction for the binary (158°C resource temperature) cycle hybrid ACC system

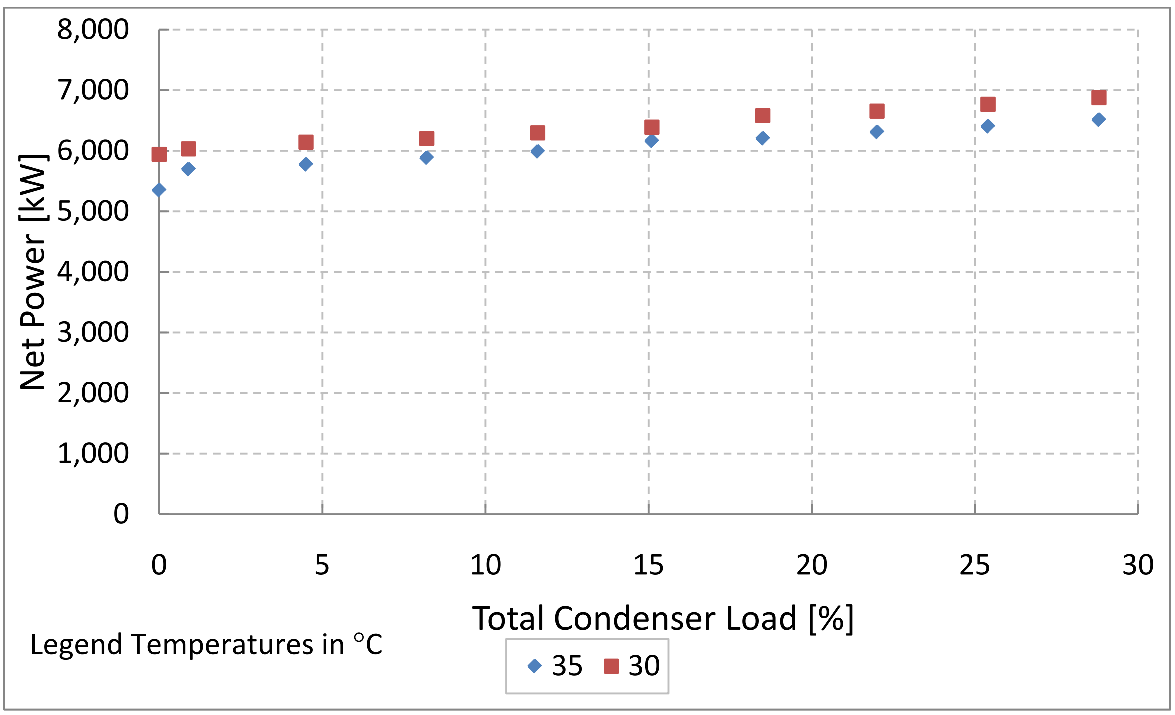

Figure 21. The net power variation as a function the wet-assist load fraction for the binary (125°C resource temperature) cycle hybrid ACC system

The net power output increases by approximately 122 kW per (kg/s) of evaporative water use for the 125°C resource temperature hybrid ACC system (see Figure 21), approximately 8% less than the 158°C resource temperature hybrid ACC system shown in Figure 20.

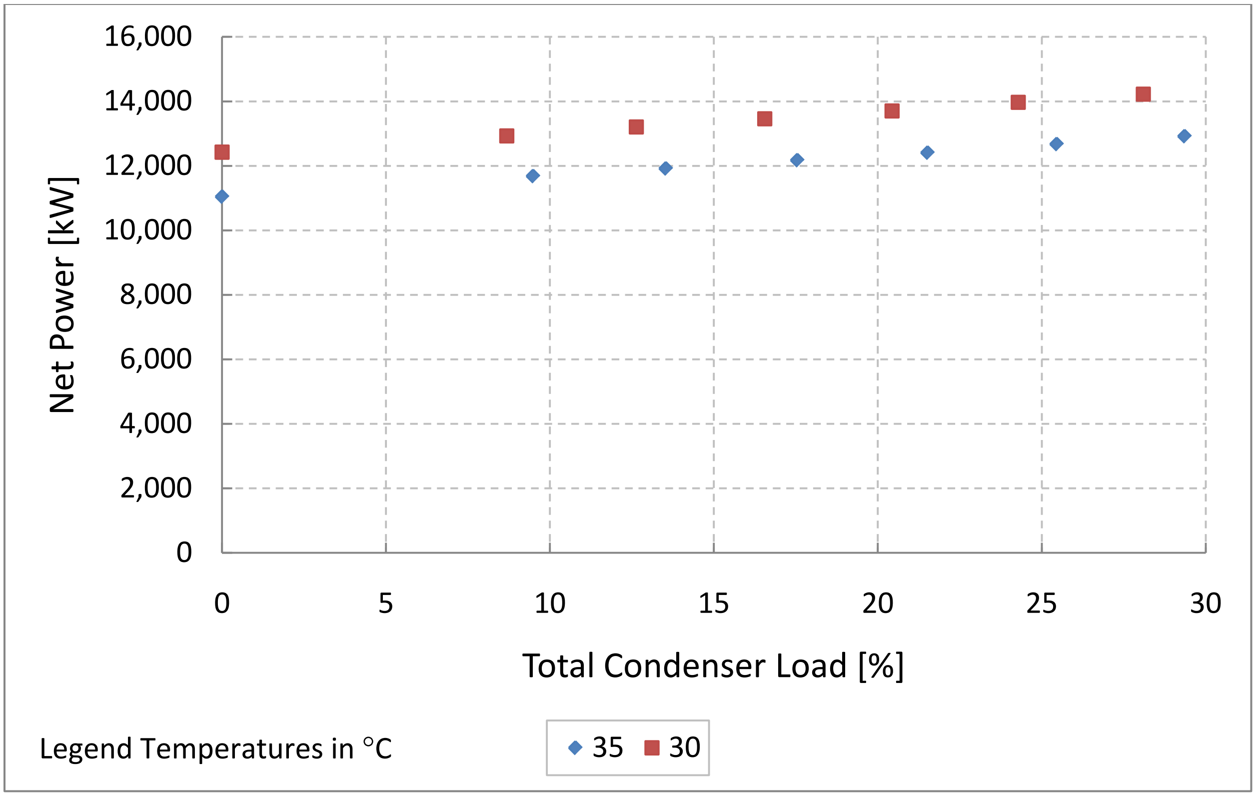

Figure 22. The net power variation as a function the wet-assist load fraction for the binary (158°C resource temperature) cycle hybrid Heller system

The net power output increases by approximately 132 kW per (kg/s) for the hybrid Heller system with a resource temperature of 158°C (Figure 22) and 103 kW per (kg/s) when using the hybrid Heller system with a resource temperature of 125°C (see Figure 23). The water consumption represented at 30% of the total condenser load is 16 kg/s and 10 kg/s for the hotter and colder resource temperatures, respectively.

For the 125°C resource temperature hybrid Heller system, there is a 22% decrease in the net power production per (kg/s) of evaporative water use compared to the 158°C resource temperature hybrid Heller system. The difference in wet-assist improvements between resource temperatures is more pronounced in the Heller system than the ACC system.

The hybrid ACC system for the 125°C resource temperature also has more potential for power increase with a wet-assist compared to the hybrid Heller with the same resource temperature. There is a 19% difference between the two systems. For the 158°C resource temperature, the systems perform similarly, with an approximate 1% difference between the two.

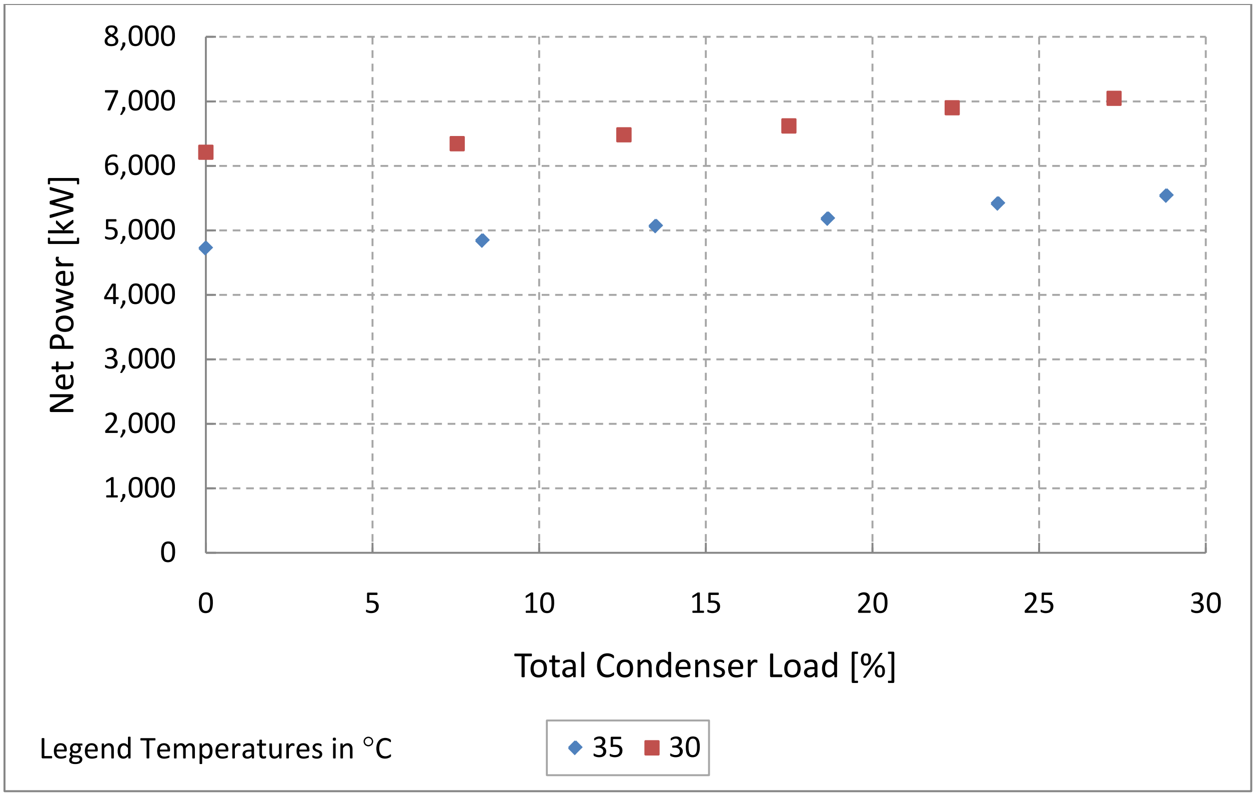

Figure 23. The net power variation as a function the wet-assist load fraction for the binary (125°C resource temperature) cycle hybrid Heller system

4.2 Steam Cycle Power Plants

Vapor from the turbine is split at its highest pressure (right after the turbine exhaust) to a wet-cooled condenser and an ACC (recall Figure 16) for the steam cycle hybrid ACC system. The wet-cooled condenser is shown as a deluged condenser; it can also be a surface condenser cooled using a wet-cooling tower.

Figure 24 shows the variations of the net power of the hybrid ACC system with a wet-cooled condenser in parallel at two different ambient dry-bulb temperatures. The net power increases linearly with increased water use and decreasing ambient temperature at a rate of approximately 174 kW for each additional (kg/s) of water used.

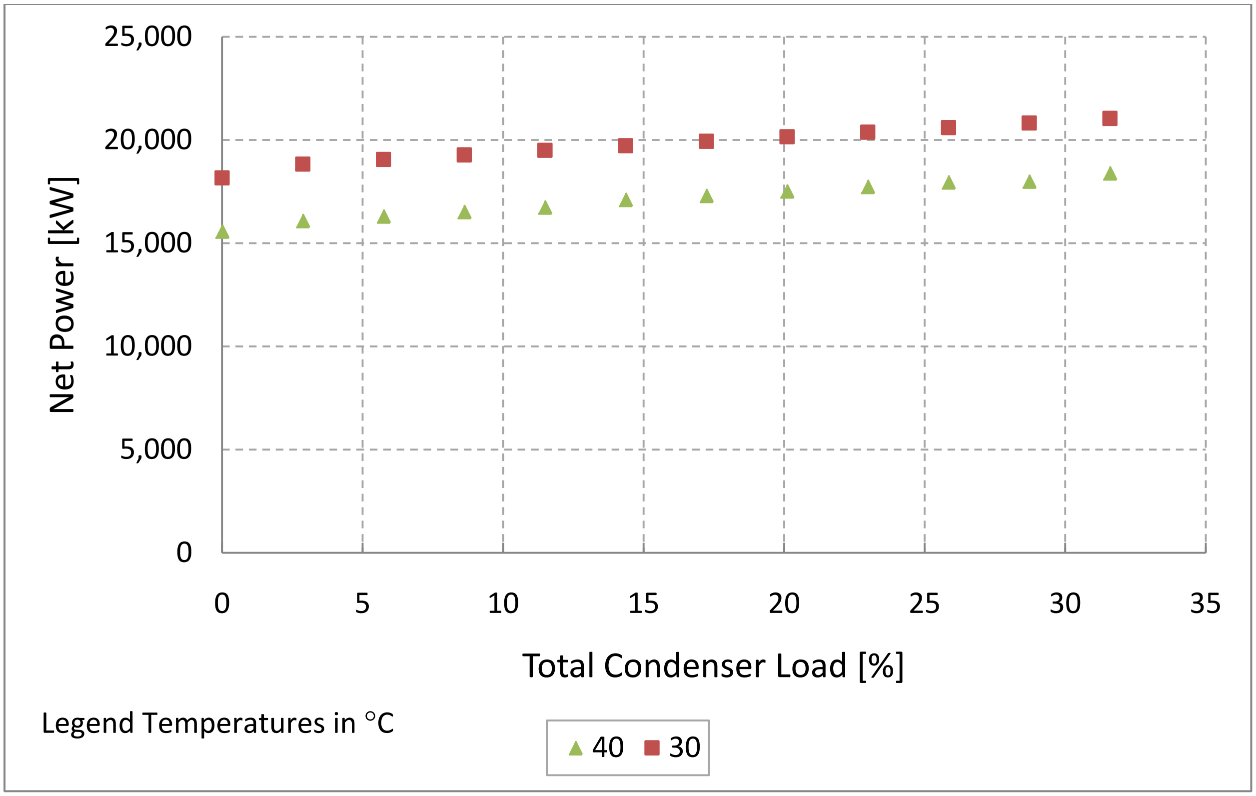

Figure 24. Net power production as a function of the wet-assist condenser load for the steam (175°C resource temperature) cycle hybrid ACC system

The performance of the hybrid Heller system was modeled and the results are summarized in Figure 25. The net power produced by the plant is plotted as a function of evaporative water use at three different ambient dry-bulb temperatures. The net power increases with increased water use at a rate of approximately 170 kW per (kg/s) of evaporative water used.

Both hybrid steam cycle systems exhibit a similar increase of power with the addition of evaporative water use with the ACC hybrid system performing 2% higher than the hybrid Heller.

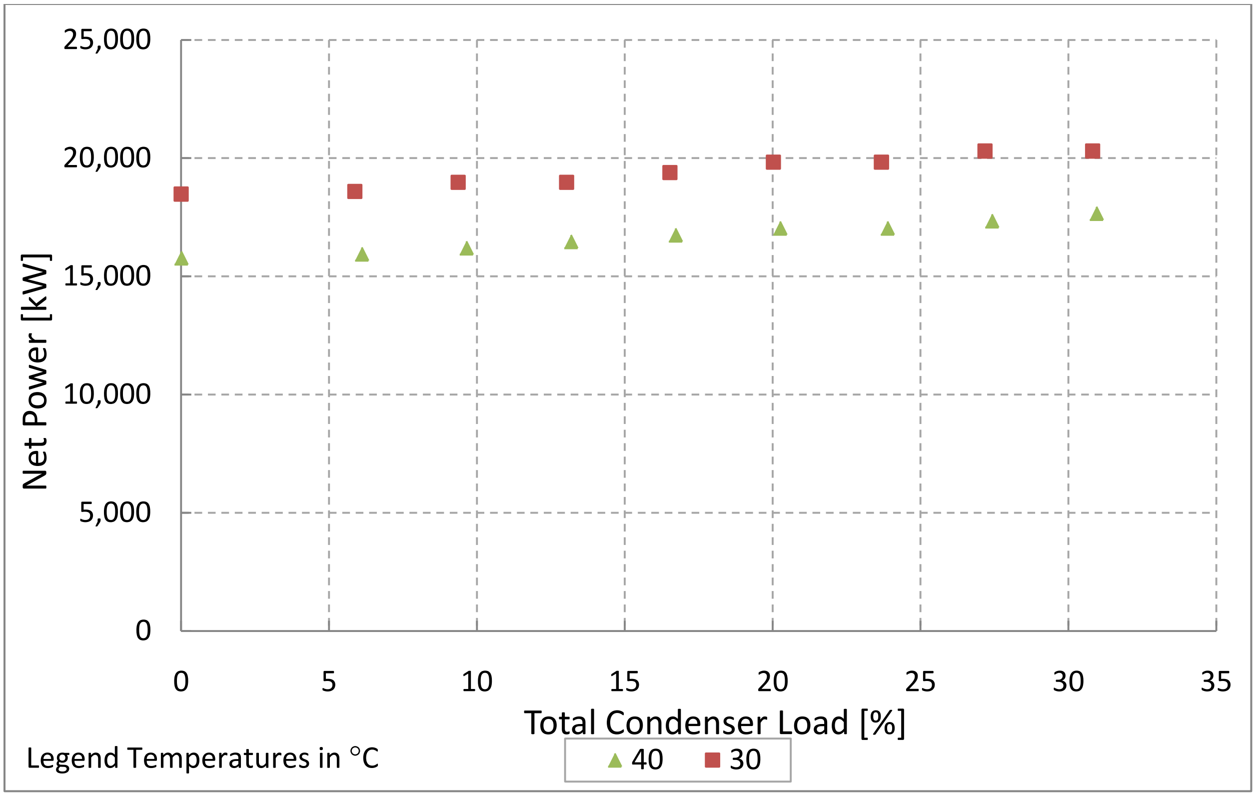

Figure 25. Net power as a function of the wet-assist condenser load for the steam (175°C resource temperature) cycle hybrid Heller system

5 Economic Evaluations

Economic evaluations for these systems were conducted assuming time-of-delivery (TOD) factors are available for the plant. This factor represents the cost multiplier for the base electricity price (i.e., what is paid to the electricity seller); it increases during high demand times.

It is available for the state of California and increases by a factor of 3 during times of high demand (see Table 2) [Public Utilities Commission 2009]. The average market price reference (MPR) value (i.e., the average flat rate price of electricity) used for these analyses was 0.1 $/kWh [Public Utilities Commission 2009]. TOD factors and the market price reference (MPR) value work together to determine the plant revenue throughout the day. These factors are structured such that a plant running 24 hours a day, 7 days a week, with a constant output would see no change in annual revenue. These factors incentivize production improvements during peak hours.

Table 2. 2009 Time of Delivery Values for California

Season	Period	Definition	Factor
June-September	On Peak	Weekdays 12 p.m. – 6 p.m. except NERC holidays	3.13
	Mid-Peak	Weekdays 8 a.m. – 12 p.m., 6-11 p.m. except NERC holidays	1.35
	Off Peak	All other hours	0.75

Consider an average July day temperature variation at the plant location, obtained using typical meteorological year (TMY) weather data at Reno, Nevada [Wilcox 2008], as shown in Figure 26. The ambient dry-bulb temperature peaks at around 1 p.m. at a value of 33°C. The wet-bulb temperature is lower and reaches a maximum of about 15°C during mid day. This difference between the dry-bulb and wet-bulb temperature allows for evaporatively cooling the intake air. With an air flow rate of about 6,600 kg/s, the amount of water that can potentially be evaporated at intake reaches a maximum of 45 kg/s.

However, to maintain the goal of using a minimal amount of water, we limited the maximum amount of water use to carry only 30% of the overall condenser load for the wet-assist analyses. The maximum water evaporated for each system is shown in Table 3. The pre-cooled inlet air systems were limited to the hybrid ACC consumption values.

Table 3. Water Consumption for all of the Systems at 30% of the Heat Rejection Load

System	Resource Temperature (°C)	Evaporative Water Consumption (kg/s)
Water	-	45
Binary hybrid ACC	158	11
	125	8
Steam hybrid ACC	175	14
Binary hybrid Heller	158	16
	125	10
Steam hybrid Heller	175	20

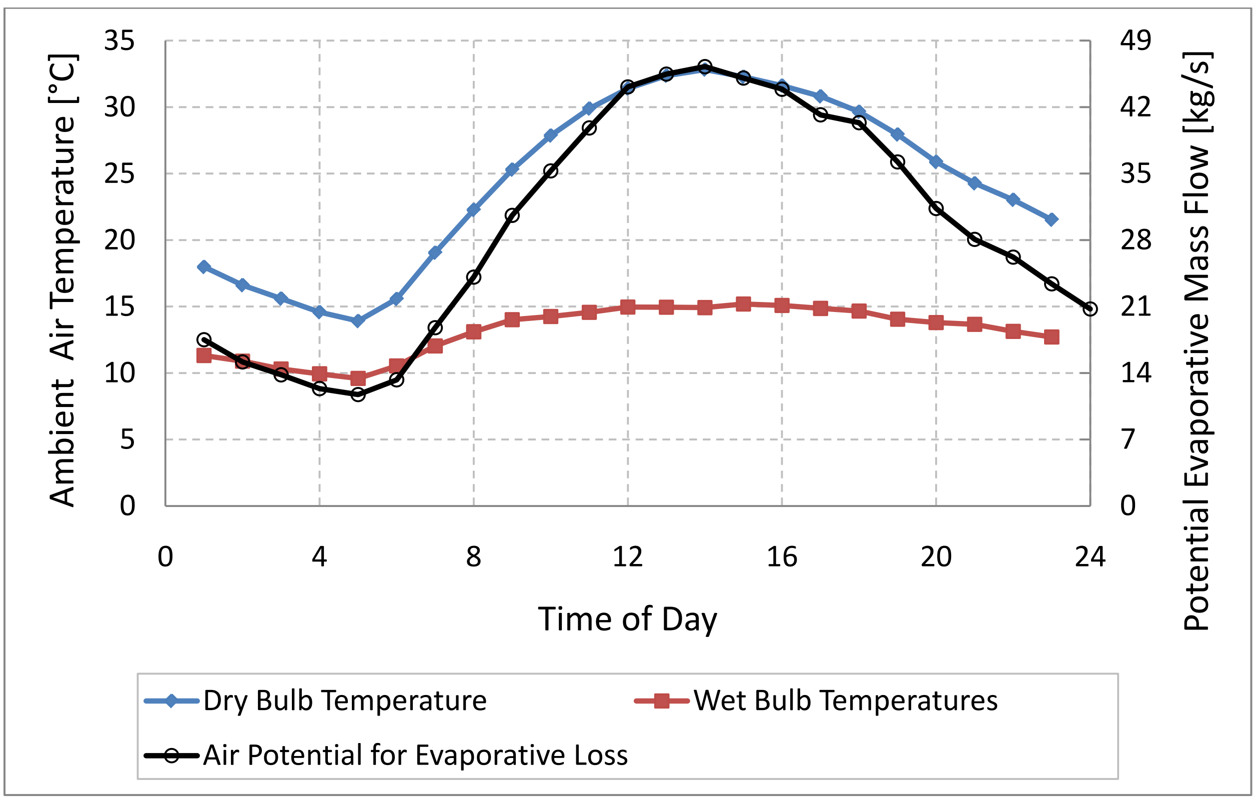

Figure 26. Average July day temperature profile and air evaporation potential in Reno, Nevada

5.1 Binary Cycle Power Plant Results

Figure 27 shows the variation of net power from both the 125°C and 158°C resource temperature power plants as a function of the time of the day (without any water use) for the weather data in Figure 26. The power curves take on a sinusoidal variation, similar to the ambient dry-bulb temperature, with a decrease in the net power during hottest periods of the day.

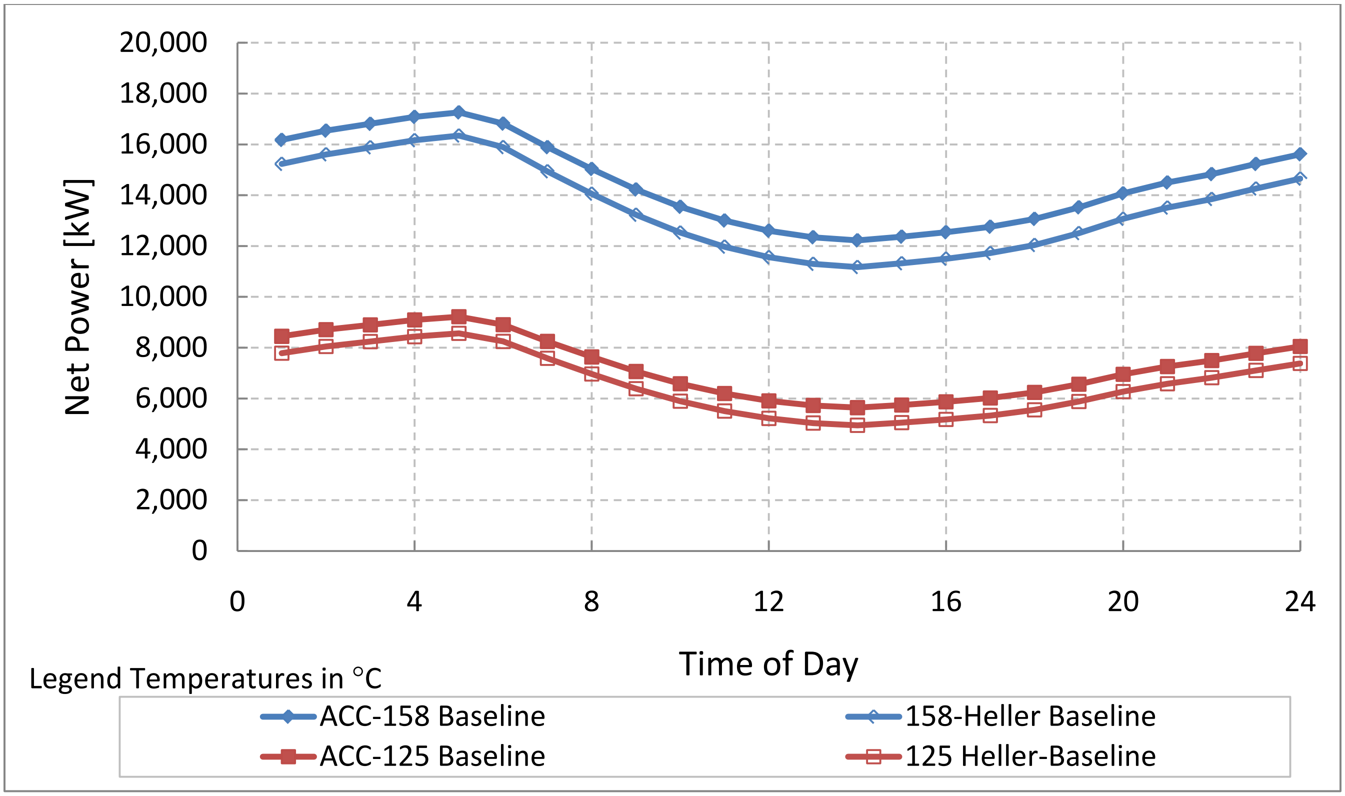

Figure 27. Net power variation for an average day in July in Reno, Nevada, for the binary cycle plants with time of the day for the dry-cooled systems

The net power results shown in Figure 28 were acquired using the addition of a wet-cooling system and limiting the amount of water used to a maximum of 30% of the condenser load. The wet assist system is turned on at 10 a.m. and off at 6 p.m. and, as Figure 28 illustrates, limiting the water consumption to 11 kg/s and maintaining a saturation efficiency of 80% for the hybrid ACC, fogger, and wetted-media systems, and 75% for the spray system.

This yielded a maximum increase of 1.4 MW in power production. The deluge system, which was modeled assuming that the outlet air of the ACC was completely saturated, exhibited a maximum of a 2 MW increase in power production.

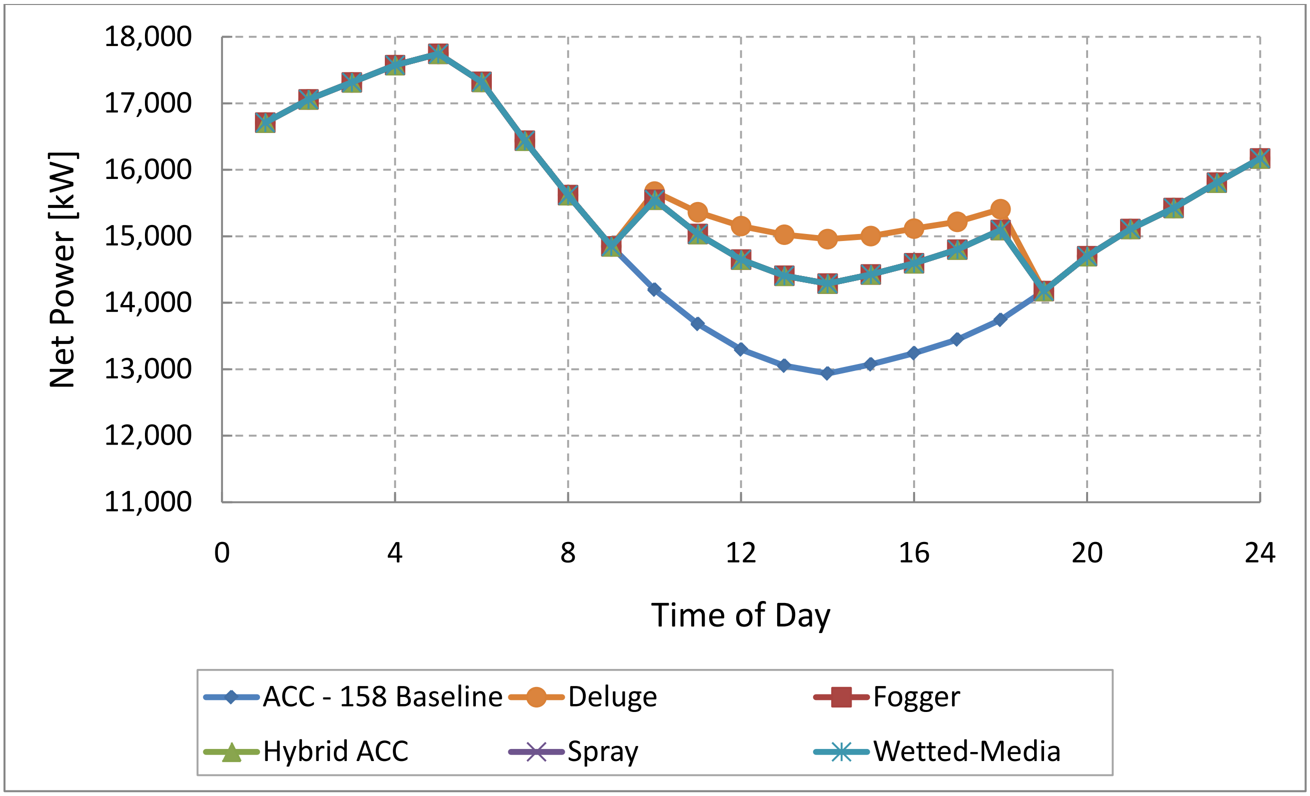

Figure 28. Net power over the course of the day for the (158°C resource temperature) wet-assisted ACC systems

5.2 Steam Cycle Power Plant Results

Figure 29 shows the variation of net power from both the ACC and Heller baseline power plants as a function of the time of the day (without any wet-cooling). The power curves take on a sinusoidal variation, similar to the ambient dry-bulb temperature, with a decrease in the net power during hottest periods of the day. Note also that the Heller system performs slightly better than the ACC system at lower ambient temperature, but similarly during hot periods of the day.

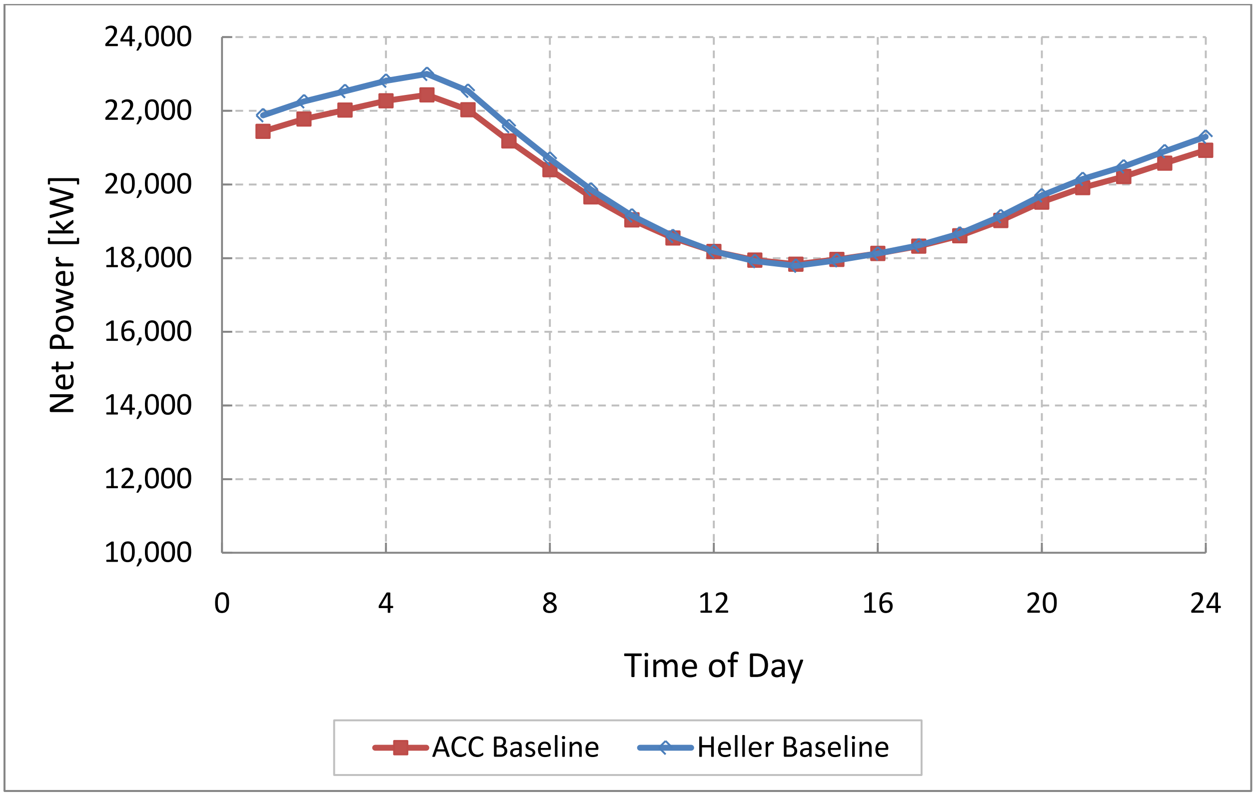

Figure 29. Net power variation for the dry-cooled steam cycle power plants over the course an average day in July in Reno, Nevada

The wet-assist system is turned on at 10 a.m. and off at 6 p.m. and, as Figure 30 illustrates, the hybrid Heller system exhibits a larger increase in power during this time (compared to the Heller baseline) than the hybrid ACC does compared to the ACC baseline. Lower condenser suction pressure in the ACC does not directly translate to a lower turbine back pressure on account of the steam distribution manifolds.

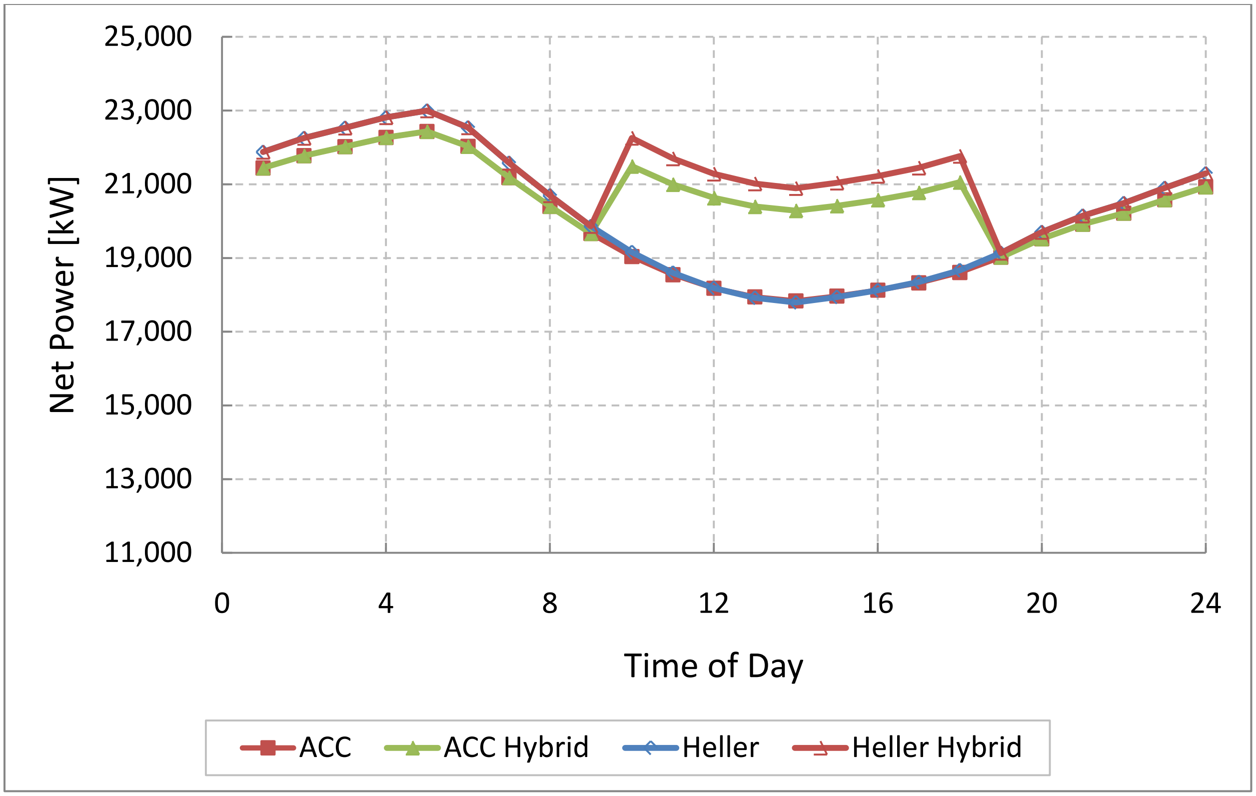

Figure 30. Net power for the steam cycle plants (175°C resource temperature) over the course of an average day in July in Reno, Nevada, using hybrid cooling systems

6 Relative Performance and Cost

This section discusses the benefit of adding the hybrid cooling options to the dry-cooled systems for the hottest periods of the day in Reno, Nevada. The benefit is presented as both an incremental net power (the difference between the net power from the dry-cooled system and the hybrid system) and incremental revenue. The revenue of the dry-cooled systems was calculated using the MPR value and TOD factors presented in Section 5.

Equipment costs were evaluated for surface equipment only (e.g., heat exchangers, pumps, fans, cooling towers). The revenue of the hybrid systems was calculated in the same manner as the dry-cooled systems with the additional cost of water subtracted out. It was assumed that the wet-assist system operates at a maximum of 30% of the total condenser load for 1,000 hours per year during weekday, non-holiday hours from 10 a.m. to 6 p.m..

6.1 Binary Cycle Power Plants

To assess the economics of water use, we dealt with incremental power yield as a function of the incremental water use and the capital cost associated with the incremental needed equipment. The additional revenue obtained (because of the water use) was isolated by subtracting the baseline design case from that for the water-assisted case.

Assuming that water is purchased from outside sources, the cost of water (assumed to be an average of $1.38/1,000 gallons [Turchi 2010]) was subtracted from that revenue to obtain a net return from the auxiliary wet cooling system.

Figures 31 and 32 show the incremental power (i.e., the power output with the hybrid cooling assist minus the power from the baseline dry-cooled system) for a typical hot day in July for a plant located in Reno, Nevada, for the hybrid ACC system. The wet-cooling system can be expected to operate for about 1,000 hours (or about 125 days of 8-hour operation/day) during the course of a year.

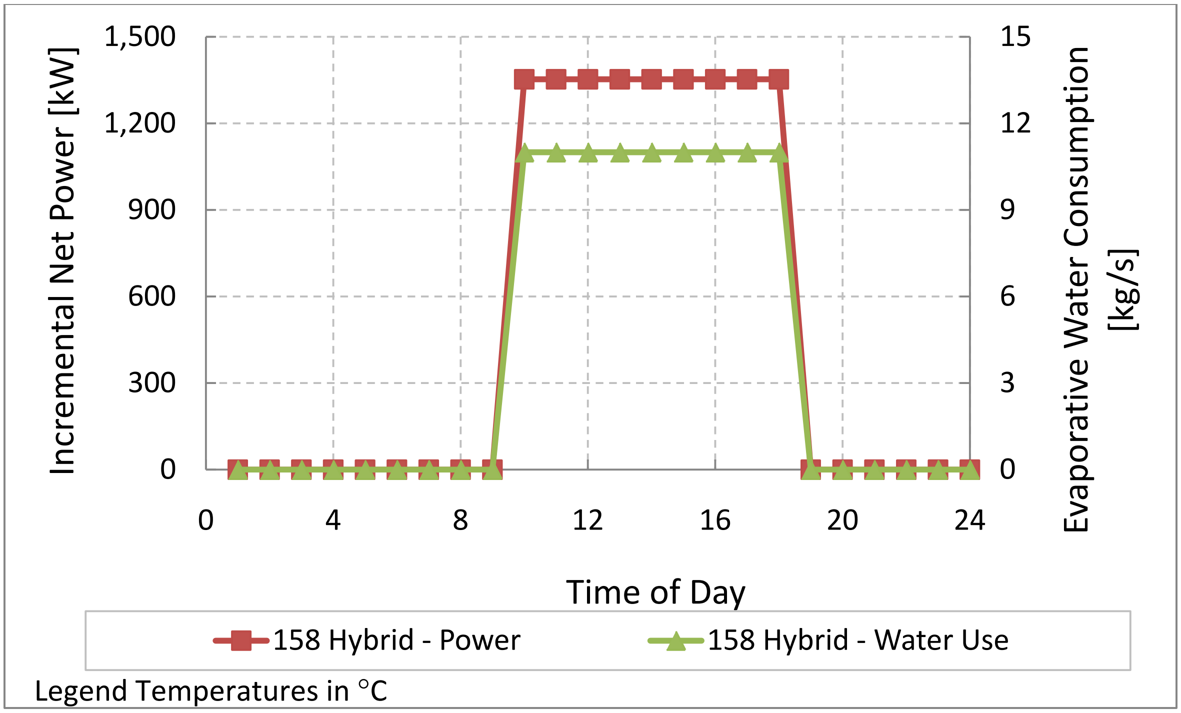

Figure 31. Incremental net power and water consumption for the 158°C resource temperature binary cycle hybrid ACC system

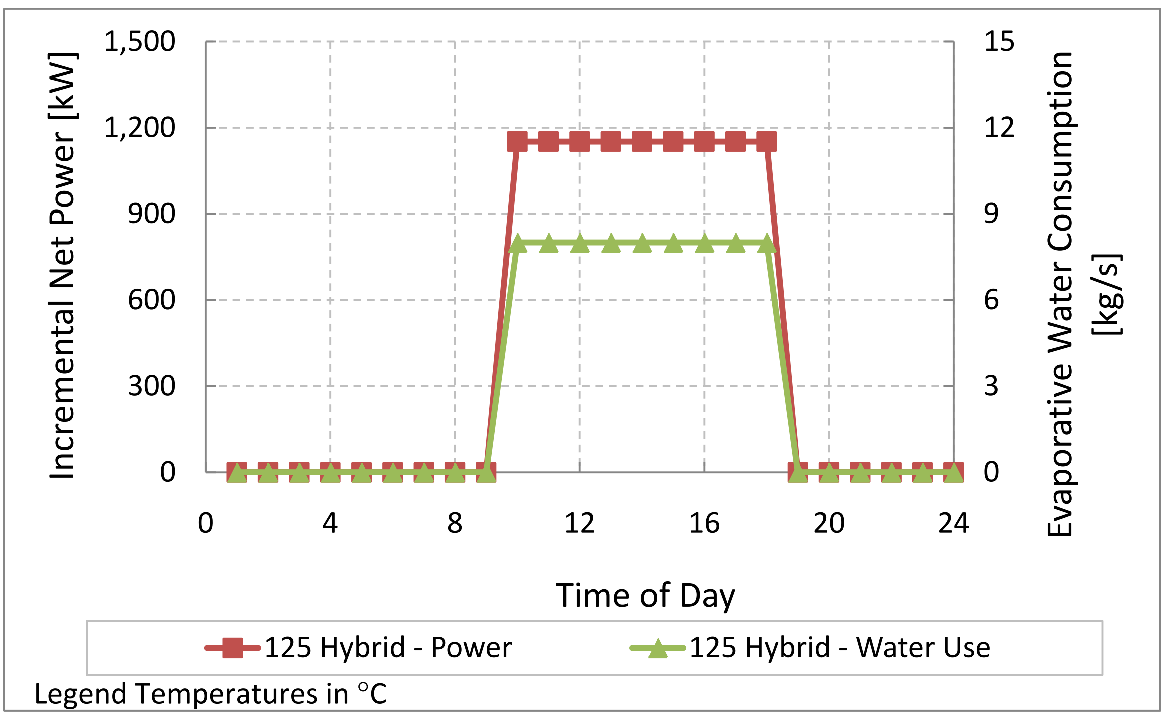

Figure 32. Incremental net power and water consumption for the 125°C resource temperature binary cycle hybrid ACC system

The water consumption is limited during the hottest periods of the day to the water consumption necessary to handle 30% of the total condenser load. The incremental power during the times when the wet-assist is used is constant for both resource temperature plants.

Figures 33 and 34 illustrate the incremental net power for the hybrid Heller systems. Sharp changes in the slope are periods of time where the TOD factor changes. The hybrid Heller system is sensitive to these changes for both resource temperatures. The incremental net power for the 158°C resource temperature plant is approximately 40% higher than that seen in the hybrid ACC system. The lower resource temperature plants are more similar in incremental power, with the hybrid ACC system yielding a 17% higher incremental power output than the hybrid Heller system.

These Figures (30-34) also show the water consumed for all of the cases. It should be noted that though the 30% load case is shown, the plant operator can reduce the water use at any time by adjusting the cooling load distribution.

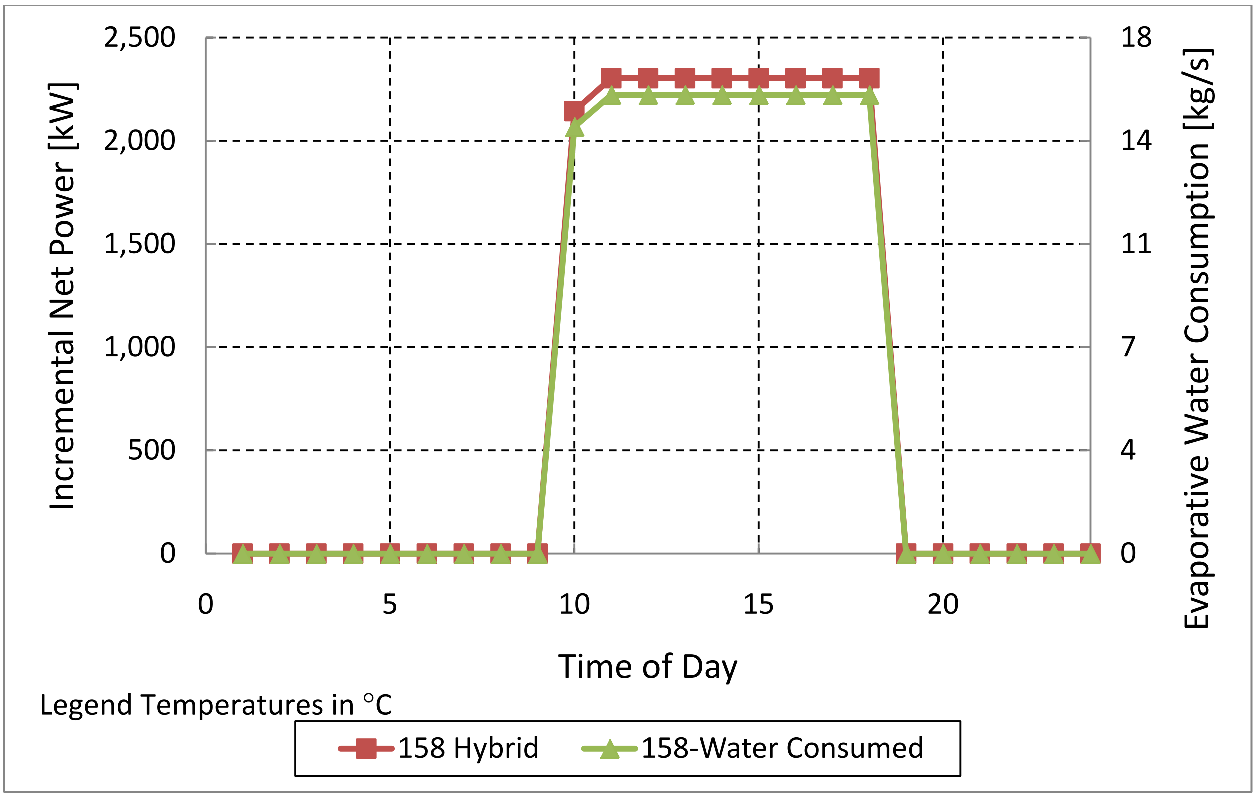

Figure 33. Incremental net power and water consumption for the 158°C resource temperature binary cycle hybrid Heller system

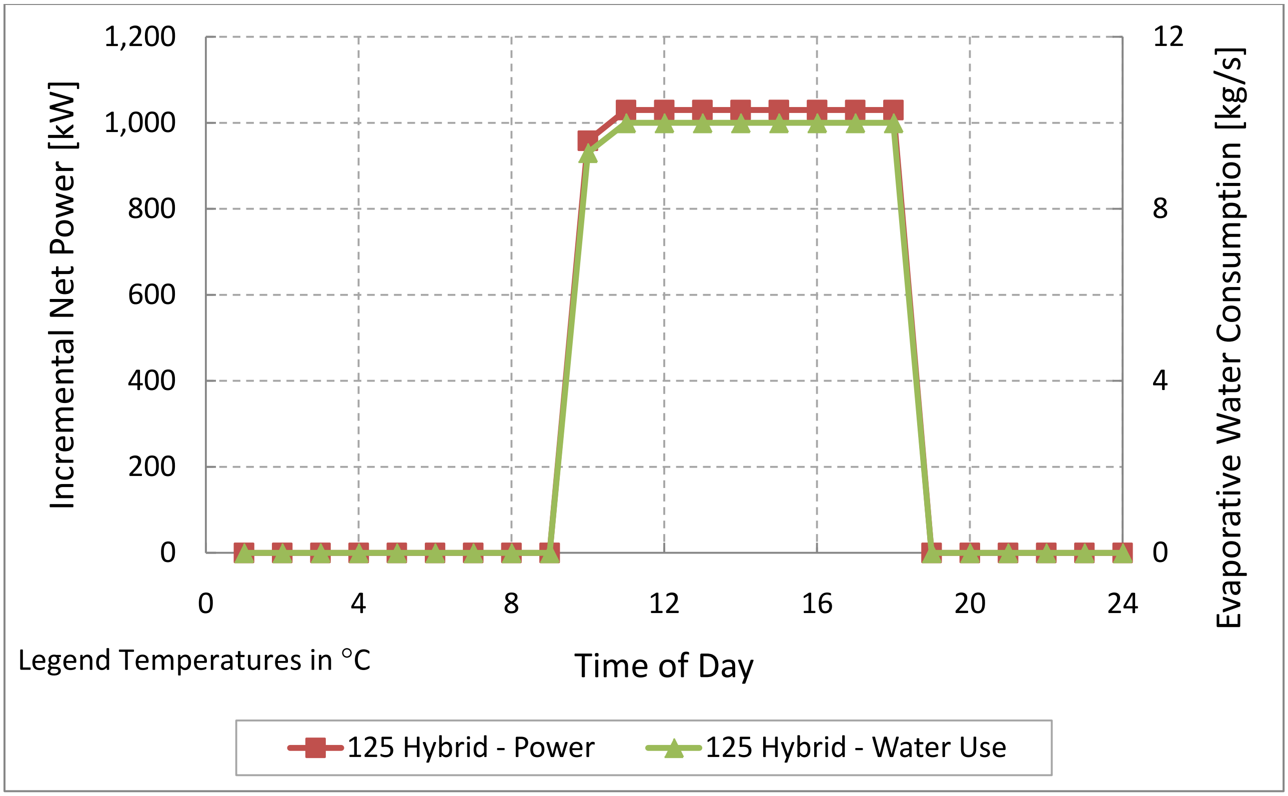

Figure 34. Incremental net power and water consumption for the 125°C resource temperature binary cycle hybrid Heller system

In order to assess the economic viability of the supplemental wet cooling option, a simple payback for each system is calculated. The revenues are assessed below and the costs associated with the water use are assessed in Section 6.1.1. These calculations lead to an estimation of the potential payback period and are presented later.

Figures 35 and 36 translate these results into the added revenue realized based on the incremental water use. The added revenue takes on a similar shape to the incremental power production curves. Any sharp increase in the curves is associated with the changes in TOD factor. The incremental return resulting from the hybrid ACC systems are estimated to be \$0.37 million per year for the 158°C resource temperature plant and \$0.27 million per year for the 125°C resource plant. The incremental return resulting from the hybrid Heller systems are estimated to be \$0.56 million per year for the 158°C resource temperature plant and \$0.27 million per year for the 125°C resource temperature plant.

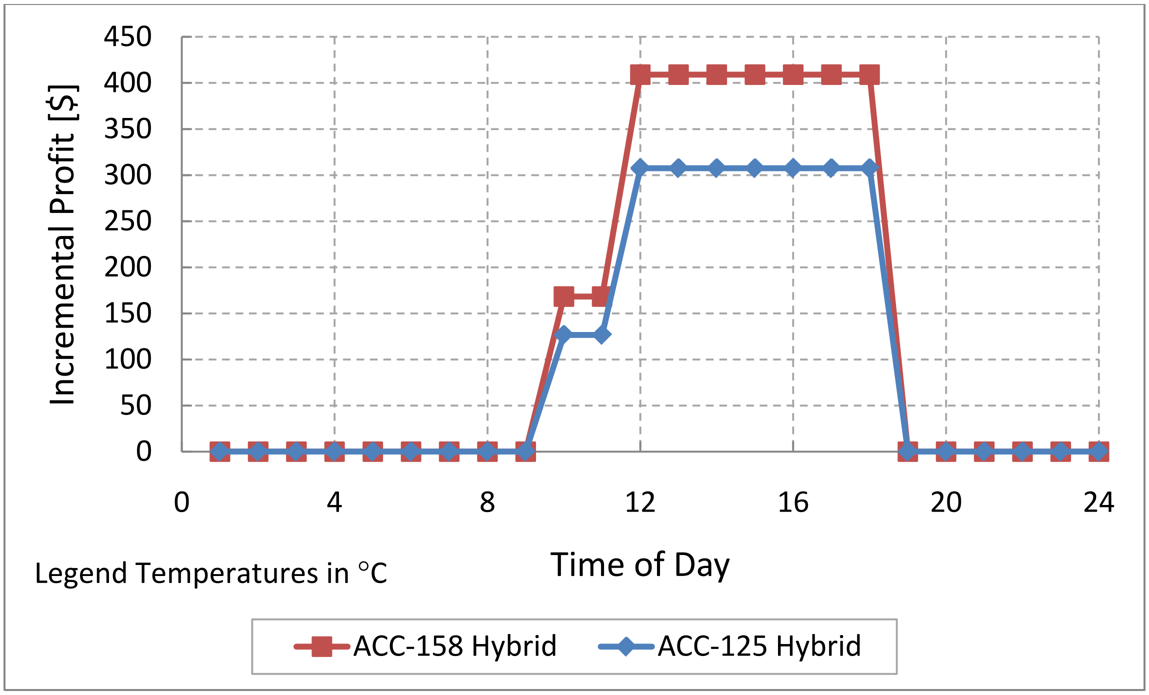

Figure 35. Incremental profit for the binary cycle hybrid ACC systems

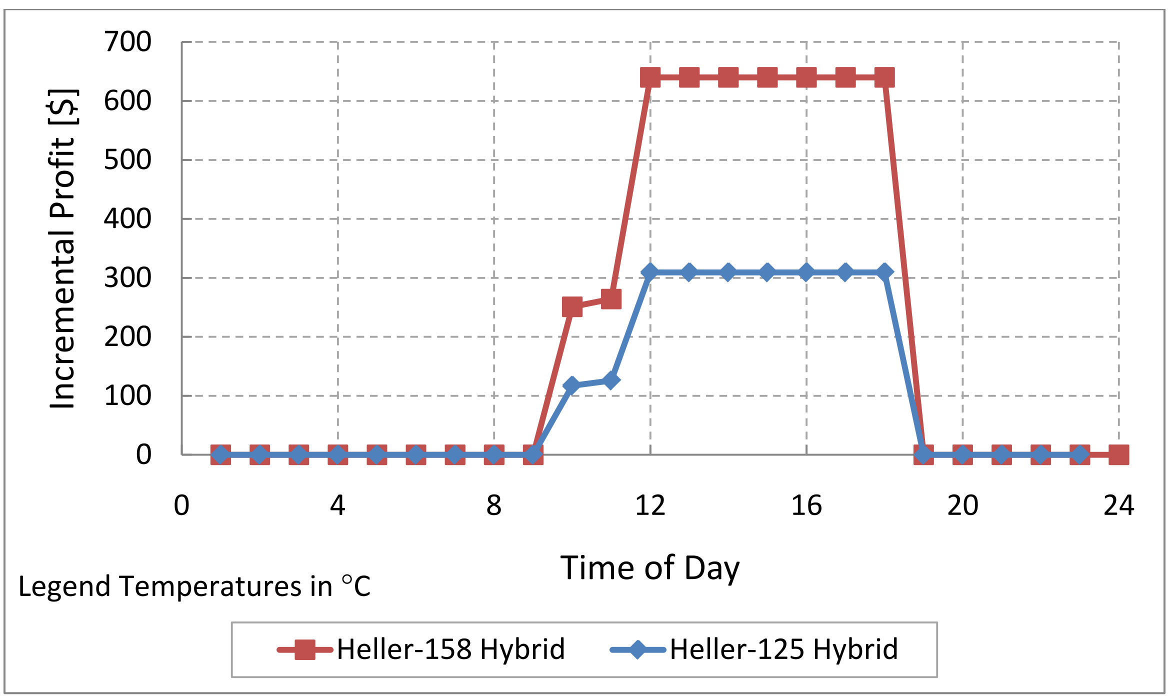

Figure 36. Incremental revenue for the binary cycle hybrid Heller systems

6.1.1 Payback Period for the Binary Cycle Power Plants

The payback period was calculated based on the added system costs associated with the wet-cooling equipment. Table 4 summarizes the incremental costs for the hybrid plants. The wet-cooling assist is limited to handling a maximum of 30% of the overall condenser load.

Correspondingly, the wet cooling tower is sized to handle a maximum of 30% of the heat rejection load.

Costs for the cooling tower, surface condenser, and water pumps are included in the table. These costs were derived using cost equations from Matches.com for the cooling tower [Matches 2003]. The pump and turbine costs were calculated using cost equations from Product and Process Design Principles [Seider 2009] and the heat exchanger costs were calculated using a quote from GEA Heat Exchangers [Smith 2010].

Table 4. Results Summary for the Binary Cycle Hybrid Systems

System	Resource Temperature (°C)	Cooling Tower (M$)	Pump (M$)	WCHX (M$)	Water-Cooled Condenser (M$)	Total Cost (M$)	Simple Payback Period (Years)
Binary hybrid ACC	158	1.8	0.06	N/A	0.22	2.1	5.9
	125	1.2	0.04	N/A	0.13	1.4	4.6
Binary hybrid Heller	158	1.8	0.09	0.39	N/A	2.2	3.9
	125	1.3	0.08	0.42	N/A	1.8	6.9

*Calculated assuming an average cost of water of $1.38 per thousand gallons

The payback period was calculated using the incremental cost of equipment between the baseline power plants and the hybrid plants. Since water costs are relatively low compared to equipment cost, increased water costs lengthen the payback times only slightly for both when water costs were varied from $0.30 - $2.50 per 1,000 gallons.

The payback period for the hybrid ACC 125°C resource temperature plant varies from 4.5 to 4.7 years. For the 158°C resource temperature hybrid ACC plant, the payback periods are longer, varying from 5.7 to 6.1 years.

The payback for the 158°C resource temperature hybrid Heller plant varies from 3.8 to 4.0 years. For the colder resource temperature plant, the payback periods are somewhat longer, ranging from 6.6 to 7.2 years.

Recall that the hybrid Heller system for the larger plant exhibited a much higher incremental return compared to the hybrid ACC system. As Table 4 illustrates, the incremental equipment costs are relatively similar for both hybrid systems. So, the increase in the incremental revenue available for the hybrid Heller systems represents a much smaller payback period for the 158°C resource temperature power plant.

For the 125°C resource temperature power plant the incremental revenue is higher for the hybrid Heller system. The equipment costs for the hybrid Heller system are also higher than the hybrid ACC system. This results in a longer payback period.

Table 5 shows the estimated incremental equipment costs, incremental revenue, and payback period calculations for the fogger systems, spray systems, wetted-media systems, and the deluge systems. Since detailed information regarding the ACCs modeled in this report is not known, system quotes were obtained from vendors using the unit at RMOTC. A power law relationship was then used to scale costs to the systems analyzed in this report.

The incremental equipment costs are for surface equipment only and do not include additional piping or installation costs. The incremental equipment cost for the fogging systems were estimated using an average cost from three vendors: Atomizing Systems, Inc. [Elkas 2010], MEE Industries [Peterson 2010], and Caldwell Energy, Inc. [Shepherd 2010] and an exponent of 0.7. The spray system equipment cost estimate is based on costs from an EPRI study using sprays to cool the inlet air to an ACC in a fossil plant [EPRI 2002] and an exponent of 0.7. The wetted-media cost was estimated based on an equipment quote from Munters, Inc. [Delman 2011] and an exponent of 0.75. It was assumed that the only incremental equipment necessary for the deluge system is a pump to re-circulate the water over the condenser tubes.

The same amount of water consumption as the hybrid power plants shown in previous sections was used for the pre-cooled inlet air systems. The fogger and wetted-media systems were modeled with a wet-bulb approach effectiveness of 80%. The spray systems were modeled using a 75% wet-bulb approach effectiveness. For the deluge system, the air at the outlet of the ACC was assumed to be fully saturated.

The deluge system provides the shortest payback period since the only necessary incremental equipment is the pump needed to deluge the ACC. The costs associated with fin degradation, which is an issue for a deluge system such as a water treatment system, were not included in the costs listed in Table 5. The water used to pre-cool the inlet air (and for the deluge system) was assumed to be at a constant temperature, and cooling water equipment costs were not included in the incremental equipment costs for any of these systems.

Table 5. Results Summary for the Binary Cycle Pre-cooled Inlet Air Systems

Evaporative Cooling System	Resource Temperature (°C)	Incremental Equipment Costs (M$)	Incremental Annual Net Revenue (M$)	Simple Payback Period* (Years)
Wetted-Media	158	3.3	0.36	9.4
	125	2.0	0.27	7.4
Fogging	158	2.2	0.36	6.1
	125	1.7	0.27	6.5
Spray	158	0.21	0.36	0.60
	125	0.27	0.27	1.0
Deluge	158	0.06	0.49	0.13
	125	0.04	0.38	0.10

*Calculated assuming an average cost of water of $1.38 per thousand gallons and applying TOD factors

Incremental revenue based on time-of-delivery rates will depend upon the local market conditions as well. A flat rate for electricity will result in smaller incremental revenue, which would increase the payback period for these systems.

6.2 Steam Cycle Power Plants

Figure 37 shows the incremental revenue for the hybrid systems on a typical hot day in July for a plant located in Reno, Nevada. TOD rates and the MPR value discussed previously (Table 2) were used in calculating the revenue. The incremental revenue resulting from the hybrid systems (calculated assuming the wet-assist is used for 1,000 hours of the year, during weekday, non-holiday hours from 10 a.m. to 6 p.m.) are estimated to be $1.4 million per year for the ACC hybrid and $1.3 million per year for the Heller hybrid system assuming 100% plant availability.

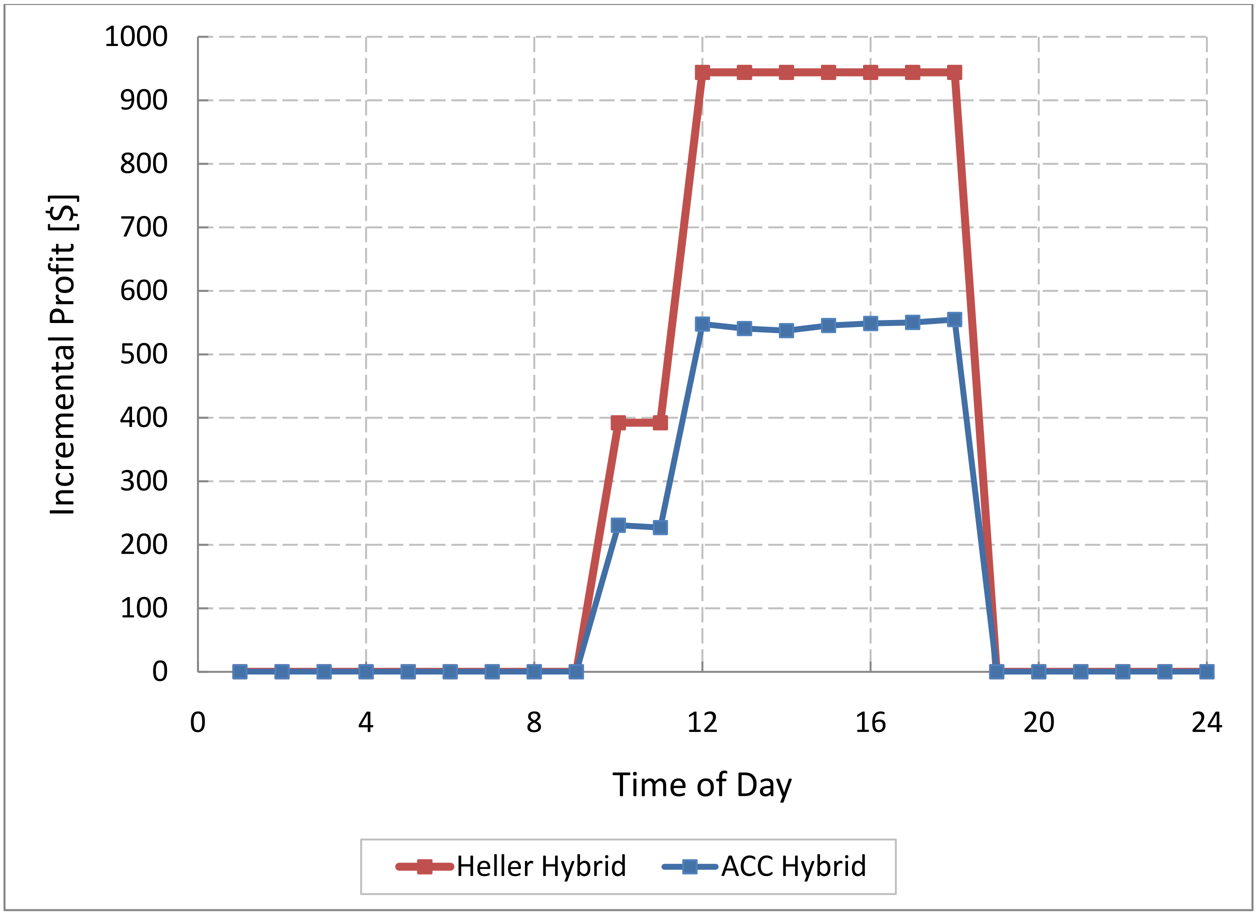

Figure 37. Incremental profit for the steam cycle hybrid ACC and hybrid Heller systems

If the water used in the hybrid systems is obtained from the flash/condense operation at the evaporator, adequate water should be available for six to eight hours of operation a day for the wet-cooling assisted systems. The water must be cooled and stored for use at appropriate times, the cost of which is not included in this study.

6.2.1 Payback Period for the Steam Cycle Power Plants

Estimates for the cost of the auxiliary wet-cooled system vary with installed capacity. Costs associated with the conversion of the baseline plant to a hybrid (for the ACC this is the parallel cooling system; for the Heller system this represents the wet-cooling assist in series) system are summarized in Table 6.

Table 6. Results Summary for the Steam Cycle Hybrid Systems

System	Resource Temperature (°C)	Cooling Tower (M$)	Pump (M$)	WCHX (M$)	Water-cooled Condenser (M$)	Total Cost (M$)	Simple Payback Period* (Years)
Steam hybrid ACC	175	1.53	0.039	N/A	0.012	1.58	1.1
Steam hybrid Heller	175	1.53	0.157	0.026	N/A	1.71	1.2

*Calculated assuming an average cost of water of $1.38 per thousand gallons and applying TOD factors

The total incremental cost of the hybrid Heller system is higher than the incremental costs for the hybrid ACC system largely due to the pumping requirement. The hybrid Heller system also requires a larger wet-cooled heat exchanger because of the introduction of the secondary loop, which results in a reduction of the driving potential for heat exchange in the ACHX.

Similarly to the results of the binary cycle power plants, the payback period for the steam cycle power plants increases with increasing water costs. The effect of the cost of water on the payback for the steam cycles, however, is negligible. The payback period for the hybrid ACC system varies from 1.12-1.14 years and the payback period for the hybrid Heller plant is 1.2-1.24 years with water cost varying from $0.3-$2.46 per thousand gallons.

7 Summary

An investigation of wet-cooling additions to heat rejection systems of geothermal power plants rejecting heat to air was carried out in this report. Two binary power plants (using brine resource temperatures of 158°C and 125°C) were analyzed as well as a steam power plant (using a brine resource temperature of 175°C).

Binary plants were analyzed with the following hybrid heat rejection configurations: cooling the inlet air to an ACC, deluge of the ACC, a wet-cooled surface condenser in parallel with the ACC, and a wet-cooled heat exchanger in series with an ACHX.

Steam power plants were also analyzed with analogous configurations. The next phase of the project is to implement a hybrid cooling strategy to an existing power plant. Since most low temperature geothermal power plants are binary power plants, steam plants with the pre-cooled inlet air or deluge systems were not analyzed.

To address the limited availability of water, we restricted the water use to handle a maximum of 30% of the total condenser load. The time of operation with the wet assist was also limited to a cumulative time period of 1,000 hours of operation (during the hottest times of the day) in a year. As a result of these restrictions, water use was reduced to about 3.5% of the evaporative water use for a fully wet-cooled plant.

Table 7 provides a summary of results for the baseline plants with the cost of water at $1.38/1,000 gallons. Plant costs and water consumption amounts for the wet-cooled heat rejection system are included for comparison. Plants with air-cooled heat rejection systems cost more and yield less power.

Table 7. Results Summary for the Baseline Power Plant Cooling Systems

Cooling Configuration	System	Resource Temperature	Energy Output*	Surface Equipment Cost#	Net Revenue w/ Flat Rate+	Consumptive Evaporative Water Use
Units		(°C)	(MWh/yr)	(M$)	(M$/year)	(1,000 gallons/year) (acre-ft/year)
Air-Cooled	Binary ACC	158	157,830	16.2	15.8	0
		125	85,450	9.7	8.6	0
	Steam ACC	175	202,550	16.4	20.3	0
Indirect-air cooling	Binary Heller	158	149,970	14.8	15.0	0
		125	79,770	9.7	8.0	0
	Steam Heller	175	206,200	17.0	20.6	0
Water-Cooled	Binary Water	158	183,960	14.4	18.4	374,910 (1151)
		125	96,360	8.7	9.6	374,910 (1151)
	Steam Water	175	219,000	12.8	21.9	374,910 (1151)

*Averaged over the year based on weather conditions in Reno, Nevada. Plant availability is assumed to be 100%.
#Quoted values are restricted to surface equipment only (e.g., heat exchangers, pumps, fans, turbine, cooling tower). For full plant costs including wells, pipe lines, brine pumps, installation and operations and maintenance costs and more rigorous financial controls, refer to the System Advisor Model [Gilman 2010] or the Geothermal Electricity Technology Evaluation Model [GETEM 2009].
+An average market price reference of $0.1/kWh was used to calculate the flat rate net revenue.

Table 8 shows the impact of pre-cooling the inlet air to the ACC and the deluge system. Incremental costs for the added equipment based on vendor system quotes are included. However, these costs do not include water treatment costs.

The incremental power output and the revenue from its sale are included. The incremental revenues for the plants are estimated for both a flat rate and TOD rates. Simple payback periods for these systems are calculated based on the incremental equipment cost. The payback periods are shorter when TOD rates are applied.

The deluge system resulted in the shortest payback period. This is because the only equipment necessary to deluge an existing ACC is ground water and a simple pump. However, the deluge system comes with many drawbacks. One of the major drawbacks is the potential for corrosion of the fins and condenser tubes. This would have to be addressed using water rinses or a protective coating on the fins and therefore is considered an R&D item. Further advantages and disadvantages of all of the hybrid systems are discussed in Table 10.

Table 8. Results Summary for the Pre-cooling Inlet Air and Deluge Systems

Binary Wet-assist configuration	System	Resource Temperature	Incremental Energy Output	Incremental Equipment Cost	Incremental Net Revenue with TOD**	Incremental Net Revenue with a Flat Rate+	Payback Period with TOD	Payback Period with a Flat Rate	Consumptive Evaporative Water Use++
		(°C)	(MWh/yr) In (%)	(M$) In (%)	(M$/yr)	(M$/yr) In (%)	(yr)	(yr)	(1,000 gallons/yr) (acre-ft/yr) (%)
	Wetted-Media§	158	1,353 (0.8)	3.3 (20.4)	0.36	0.12 (0.8)	9.4	27.7	10,464 (32) (2.8)
		125	911 (1.1)	2.0 (20.6)	0.27	0.09 (1.1)	7.4	21.8	7,620 (23) (2.0)
	Fogging§§	158	1,353 (0.8)	2.2 (13.6)	0.36	0.12 (0.8)	6.1	17.8	10,464 (32) (2.8)
		125	963 (1.1)	1.7 (17.5)	0.27	0.09 (1.1)	6.5	19.0	7,620 (23) (2.0)
	Spray§§	158	1,353 (0.8)	0.21 (1.3)	0.36	0.12 (0.8)	0.60	1.8	10,464 (32) (2.8)
		125	963 (1.1)	0.27 (2.8)	0.27	0.09 (1.1)	1.0	3.0	7,620 (23) (2.0)
	Deluge‡	158	1,892 (1.2)	0.06 (0.4)	0.49	0.17 (1.1)	0.13	0.37	10,464 (32) (2.8)
		125	1,391 (1.6)	0.04 (0.4)	0.38	0.13 (1.4)	0.10	0.28	7,620 (23) (2.0)

**Assuming the wet-assist is turned on for 1,000 hours/year, during weekday non-holiday hours from 10 a.m.- 6 p.m. and applying TOD factors shown in Table 2.
++Assuming the wet-assist system operates at maximum of 30% of the total condenser load for 1,000 hours/year.
§Wetted-media such as Munter's packing with a hinged door type system for removal when system not in use.
‡A water wash is required after deluging the ACC to prevent fouling and corrosion. These costs are not included in the incremental equipment costs.

Table 9 presents the results for systems that use an auxiliary wet-cooled condenser/heat exchanger either in parallel with the ACC or in series with the Heller arrangement. Payback periods based on the incremental costs and revenues range from a minimum of less than 2 years to a maximum of about 7 years.

Table 9. Results Summary for the Hybrid Cooling Systems Using an Auxiliary Wet Surface Condenser or Heat Exchanger

Wet-assist configuration	System	Resource Temperature	Incremental Energy Output	Incremental Equipment Cost	Incremental Net Revenue with TOD[**]	Incremental Net Revenue with a Flat Rate[+]	Payback Period with TOD	Payback Period with a Flat Rate	Consumptive Evaporative Water Use[++]
		(°C)	(MWh/yr) In (%)	(M$) In (%)	(M$/yr)	(M$/yr) In (%)	(yr)	(yr)	(1,000 gallons/yr) (acre-ft/yr) (%)
Systems with a wet-assist heat exchanger	Binary hybrid ACC	158	1,335 (0.8)	2.1 (13)	0.36	0.12 (0.8)	5.9	17.4	10,464 (32) (2.8)
		125	1,152 (1.3)	1.4 (14)	0.30	0.10 (1)	4.6	15.3	7,620 (23) (2.0)
	Steam hybrid ACC	175	3,100 (1.5)	1.6 (10)	1.4	0.23 (1)	1.1	7.0	13,320 (41) (3.6)
	Binary hybrid Heller	158	2,286 (1.5)	2.2 (15)	0.57	0.21 (1)	3.9	10.6	15,216 (47) (4.1)
		125	994 (1.2)	1.8 (19)	0.26	0.09 (1)	6.9	20.9	9,540 (29) (2.5)
	Steam hybrid Heller	175	3,100 (1.5)	1.6 (9)	1.31	0.22 (1)	1.2	7.2	19,020 (58) (5.0)

Table 10 lists the major advantages and disadvantages of each of the hybrid cooling systems. The deluge, spray, and fogging systems require a higher quality of water than the other systems.

While the deluge system allows a single piece of equipment to operate both as a dry and a wet cooler and exhibits the shortest payback period, a large volume of re-circulating water is necessary to fully wet all of the heat exchange surfaces. Furthermore, an ACC operating in deluge mode is vulnerable to corrosion of the thin aluminum fins attached to the tubes. Commercial evaporative condensers are not subject to fin corrosion because they use bare (i.e., no fins) galvanized condenser tubes, but they will not work as ACCs because of their lack of extended surface area.

Fogging systems are expensive because of the high pressure necessary to produce very small droplets (~10 microns). These systems also require a large number of nozzles each operating at low flow rates. The advantage of a fogging system is that it is modular and easy to install and operate as needed. The small droplets evaporate more readily into the air stream resulting in a low potential for mineral deposition on the condenser tubes.

Spray systems that operate at pressures less than approximately 300 psi result in larger sized droplets, which will require a mist eliminator in the air intake stream. This results in a small additional pressure drop that must be overcome. These systems, however, are much less expensive than the fogging systems for the same capacity. As with the fogging systems, the spray system is easy to install and operate as needed.

The wetted-media system is another high cost option. The wetted-media will create a high pressure drop in the air stream. As a result, it is beneficial to mount the media on hinged doors so that the wetted-media can be removed when not in use. This adds maintenance costs as well as equipment costs.

Of the pre-cooled inlet air systems, the spray system represents the most viable option in dry climates since it does not require a large volume of water and it allows the inlet air to potentially be fully saturated.

The hybrid (hybrid ACC and hybrid Heller) systems with auxiliary wet-assist have similar incremental equipment costs when compared to the fogger systems. Hybrid cooling systems are a known technology. The equipment necessary is readily available, modular, and can be easily installed.

Table 10. Summary of the Advantages and Disadvantages of Each Wet-assist Cooling System

Wet-assist scheme	System	Advantages	Disadvantages
Systems with a wet-assist heat exchanger	Hybrid ACC	• Lowers ACC load • Lowers condenser suction pressure • Off-the shelf equipment • Readily designed and installed • Easily bypassed when not needed	• Relatively high equipment cost • Relatively long payback period
	Hybrid Heller	• Series arrangement for loop flow • Potential to cool full load with auxiliary heat exchanger • Off-the shelf equipment • Readily designed and installed • Easily bypassed when not needed	• Relatively high equipment cost • Relatively long payback period
ACC pre-cooled inlet air systems	Binary Wetted Media	• Off-the shelf equipment • Readily designed and installed • Lower quality water may be used	• Increased air flow pressure drop • Higher fan power required to maintain air flow rate • Packing must be replaced often • Chemicals necessary to minimize deposits and fouling • Media must be removed when not in use • Longest payback period
	Binary Fogging	• Inlet air can be cooled down to the wet bulb temperature • Modular technology • Negligible parasitic load and pressure drop • Small droplets evaporate easily in inlet air stream	• High quality water necessary to minimize clogging of nozzles • High pressure(~1,000-2,000 psi) is needed to generate fog • Relatively long payback period
	Binary Spray	• Inlet air can be cooled down to the wet-bulb temperature • Negligible parasitic load and pressure drop • Relatively short payback period	• Local wind may carry spray away • High quality water necessary to minimize clogging of nozzles • Potential for un-evaporated droplets to impact heat exchanger fins
	Binary Deluge	• Lower quality water may be used • Outlet air can be saturated • Shortest payback period	• High potential for corrosion of heat exchanger tubes and fins • High quality water needed for washing to minimize corrosion and fouling of fins • Requires lower air flow to limit water use • Heat exchanger coatings are not yet available in the market • Large volume water reflux flow is needed to fully wet all heat exchanger surfaces

Key findings are presented below.

Binary Systems

- Baseline Plants: The Heller systems have a higher power output than the ACC systems for both resource temperatures. The rate at which the power decreases with increasing ambient temperature for the baseline Heller system for the 158°C resource temperature is 9% greater than the ACC with the same resource temperature. For the 125°C resource temperature, the rate is 4% greater with the Heller system than with the ACC.
- Hybrid ACC System: The payback period for the hybrid ACC 125°C resource temperature plant varies from 4.5 to 4.7 years (as the water cost was varied from \$0.3-\$2.46 per thousand gallons). For the 158°C resource temperature resource temperature hybrid ACC plant, the payback periods are longer, varying from 5.7 to 6.1 years (with water cost varying from \$0.3-\$2.46 per thousand gallons).
- Hybrid Heller System: The payback for the 158°C resource temperature hybrid Heller plant varies from 3.8 to 4.0 years (with water cost varying from \$0.3-\$2.46 per thousand gallons). For the colder resource temperature plant, the payback periods are somewhat longer, ranging from 6.6 to 7.2 years (with water cost varying from \$0.3-\$2.46 per thousand gallons).
- Fogging System: The fogging system is an effective means of increasing the power output during hot periods of the day by pre-cooling the inlet air to the ACC. Implementation of the system, however, is hindered by high system costs. The high cost of the system results in payback periods of 6.1 years (assuming a water cost of \$1.38 per thousand gallons and that TOD rates apply) for the 158°C resource temperature. The payback period for the 125°C resource temperature plant was 6.5 years (assuming a water cost of \$1.38 per thousand gallons and that TOD rates apply).
- Spray System: The spray system resulted in the same incremental power output as the fogging system. However, the spray system is much more economical. This resulted in a shorter payback period. The payback period for the 158°C resource temperature plant was 0.60 years (assuming a water cost of \$1.38 per thousand gallons and that TOD rates apply). The payback for the 125°C resource temperature plant was 1.0 year (assuming a water cost of \$1.38 per thousand gallons and that TOD rates apply).
- Deluge System: The deluge system is found to be economical because it requires very little additional equipment and can yield a large power increase over a dry-cooled configuration. The payback period for the deluge system was 0.13 years (assuming a water cost of \$1.38 per thousand gallons and that TOD rates apply) for the 158°C resource temperature plant. The payback period for the 125°C resource temperature plant was 0.10 years (assuming a water cost of \$1.38 per thousand gallons and that TOD rates apply). However for the deluge system, the potential for corrosion of the heat exchanger components must be dealt with effectively.
- Wetted-Media System: The wetted-media systems have a similar incremental power output as the fogging and spray systems. Economically, the incremental equipment costs are more than the costs for the deluge and spray systems. Payback periods were 9.4 years for the 158°C resource temperature plant and 7.4 years for the 125°C resource temperature plant (assuming a water cost of \$1.38 per thousand gallons and that TOD rates apply).

Steam Systems

- Baseline Plants: The doubling of the flow through the secondary loop of the Heller system results in a 2% increase in net power without the addition of any cooling water. The Heller system has a 12% higher rate of power reduction per °C increase in ambient air temperature than the ACC system. This results in a large potential to increase the power output with a wet-assist system.
- Hybrid ACC System: The payback period for the hybrid ACC system varies from 1.12-1.14 years with water cost varying from $0.3-$2.46 per thousand gallons.
- Hybrid Heller System: The payback period from the hybrid Heller plant is 1.2-1.24 years with water cost varying from $0.3-$2.46 per thousand gallons.

8 Conclusions

Geothermal power plants that use air as the heat rejection media suffer large losses in production capacity (>50%) during hot summer days. Such hot periods often occur about 1,000 hours during the course of a year. One method for reducing this loss is to use the available water to assist in heat rejection and decrease the condenser pressure during those hot periods of the day.

In this study, we looked at using water to carry a nominal 30% of the heat rejection load from the power plant. By limiting the duration of operation with wet-assist to 1,000 hours during a year, the overall water consumption by the plant was capped at less than 3.5% of the water use in a fully wet-cooled power system.

We evaluated pre-cooling the intake air (using fog, sprays, and evaporative cooling media) and using wet-cooled heat exchangers (in parallel or in series) with the air-cooled heat rejection system. A basic air-cooled plant requires added equipment to implement wet-assist schemes. For the various schemes, we evaluated the cost for the added equipment. We also evaluated the incremental power produced and the associated incremental revenue for these schemes. The overall benefit of the wet-assist is evaluated in terms of payback periods. The shorter the payback period, the better the system is in an economic sense. However, the payback period does not tell the whole story.

For each of the evaluated schemes, there are many advantages and disadvantages. One of the key considerations in their evaluation is that the wet-assist system should not interfere with the normal plant operation when the wet-assist is not operational (or needed).

With these criteria in mind, we find the following two systems as the most practical for use.

1. Pre-cooling the inlet air to the air-cooled heat rejection system using sprays. In this scheme, commercially available misting nozzles are placed in a grid in the path of the intake air. Mist eliminators are introduced downstream of the sprays to capture un-evaporated water droplets. The mist eliminators must be carefully selected to minimize air-side pressure loss. Pre-cooling of the inlet air has the potential to cool the air down to close to its wet-bulb temperature with an effectiveness of about 75%. This scheme is effective in dry climates where there is a large difference between the air dry-bulb and wet-bulb temperatures. The payback period for such systems was less than 2 years for both resource temperatures, assuming TOD rates were applicable.

2. Introduction of a wet-assist heat exchanger/surface condenser (hybrid ACC). In this scheme a conventional wet cooling tower is added to the system. Water from the tower takes heat away from either an added surface condenser or from the hot coolant. The tower and water streams are sized to handle about 30% of the overall heat rejection load from the plant. The other 70% of the load is carried by the air-cooled heat rejection system. This scheme uses conventional technology with readily available off-the-shelf commercial equipment. It is easy to implement. The payback period for this type of system was estimated to range from 4.5 to 6.1 years.

Considering the above two schemes, we find that the second approach requires little in terms of research and development. The first scheme, however, is suitable as a retrofit to existing air-cooled power plants. It requires evaluation of spray nozzles, manifolding, mist eliminators, and their effectiveness in actual plant operation. We propose to implement the pre-cooling inlet air approach at the air-cooled power plant currently operational at RMOTC as described in Section 9.

9 Recommendations for Future Work

Provided funding, we plan to verify the performance of a retrofitted evaporative cooling system at a field site with an operational geothermal power plant during the summer of calendar year 2011 and to publish the results of the field work at the end of the year.

9.1 Field Test Unit

An Organic Rankine Cycle (binary cycle) unit that uses isopentane as a working fluid is located at RMOTC in Casper, Wyoming, and is a potential candidate for field verification tests (see Figure 38).

Figure 38. Photograph of a 250 kW Ormat binary cycle geothermal plant with an ACC installed at RMOTC
(NREL PIX 17507)

The unit uses isopentane as the working fluid. The oil field has an excess of low temperature water (at a resource temperature of 92°C) of approximately 40,000 bpd (4.42 m^3/s). Currently, the power plant is not monitored for hourly measurements of flow rates, temperatures, pressures, and net power output. However, knowledge of the daily averages of the power output and ambient temperature are available.

Figure 39 illustrates the average power output in August and November of 2009 with increasing ambient dry-bulb temperature. In practice, the resource flow rate is varied (when excess water is available) to maintain the net power output at approximately 215 kW. However, Figure 39 illustrates that there is an approximate 60% decrease in the power output from November to August.

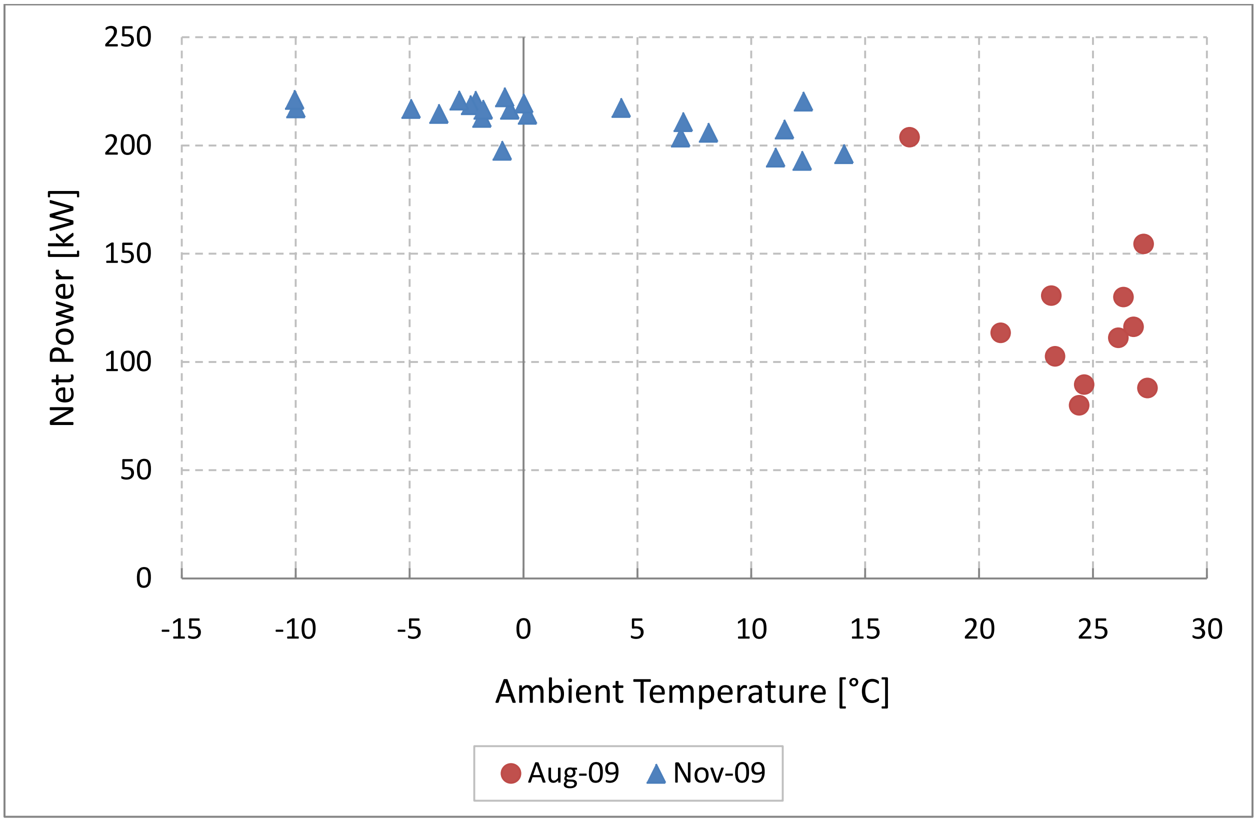

Figure 39. Actual net power output from the air-cooled Ormat unit at RMOTC

The performance of the system was also estimated by modeling the system similarly to those described earlier in this report. Figure 40 shows the variation in plant net power with increasing ambient temperature for this unit. The plant yields about 250 kW at design (ambient air temperature of 14°C). The available power decreases by nearly 75% when the ambient temperature approaches 30°C.

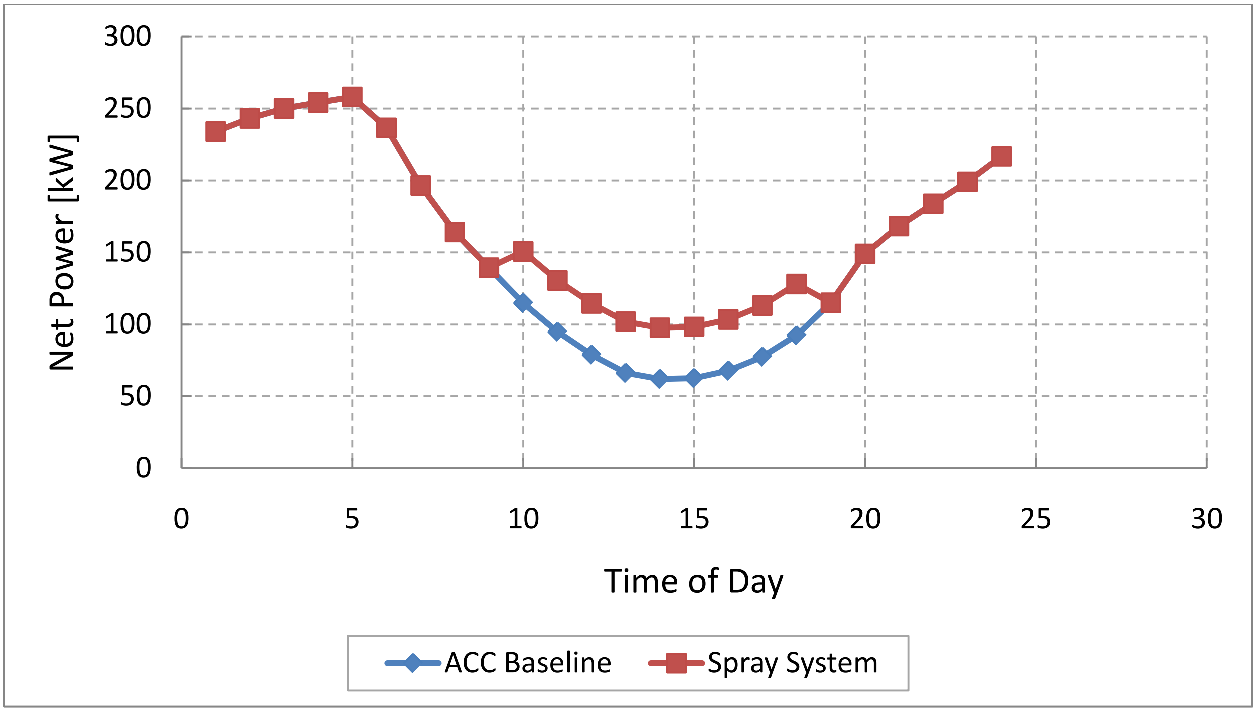

Figure 40. Potential net power increase using a wet-cooling assist (spray system) from 10 a.m. to 6 p.m. on the air-cooled Ormat unit at RMOTC

Pre-cooling the inlet air to the ACC would be well suited for tests with this unit since humidity levels at the site are generally low, averaging a maximum of 29% during the hottest times of the summer days. These conditions are ideal for evaluating the performance of a spray system for the air-cooled plant. The addition of a spray during the hot periods of the day could mitigate the decrease in power production by nearly 36%, as illustrated in Figure 40.

References

Backer, L.D.; Wurtz, W.M. (2003). “Why every air cooled steam condenser needs a cooling tower.” Cooling Tower Institute, Paper TP03-01.

Bamberger, J.A.; Allemann, R.T. (1982). “Economic evaluation of four types of dry/wet cooling applied to the 5-MWe Raft River geothermal power plant.” PNL-4053. Richland, WA: Pacific Northwest Laboratory.,

Bharathan, D.; Nix, G.. (2001). “Evaluation of an Absorption Heat Pump to Mitigate Plant Capacity Reduction Due to Ambient Temperature Rise for an Air-Cooled Ammonia and Water System. Geothermal Resources Council Transactions. Vol. 25, 2001; pp. 563-567.

Delman, Donald. (17 January 2011). Equipment quote. Munters, Inc. Dallas, TX.

Electric Power Research Institute (EPRI). (2002). *Spray cooling enhancement of air-cooled condensers*. Report 1005360.

Electric Power Research Institute (EPRI). (2003). *Water and sustainability (volume 3) U.S. water consumption for power production- the next half century*. Report 1006786.

Electric Power Research Institute (EPRI). (2005). *Air-cooled condenser design, specification, and operation guidelines*. Report 1007688

Elkas, Michael. (14 December 2010). Equipment quote. Atomizing Systems, Inc. New Jersey.

Kagel, A. (2008). “The State of Geothermal Technology Part II: Surface Technology.” Geothermal Energy Association.

Konaglu, M.; Çengel, Y.A. (1999). “Improving the performance of an existing air-cooled binary geothermal power plant: a case study.” J. Energy Resources Technology, v.121, pp 196-202.

Kozubal, E.; Kutscher, C. (2003). “Analysis of a Water-Cooled Condenser in Series with an Air-Cooled Condenser for a Proposed 1-MW Geothermal Power Plant,” GRC Transactions, Vol. 27. Davis, CA: Geothermal Resources Council pp. 587-591.

Kutscher, C.; Gawlik, K. (2000). “Development of a Porous Fin Air-Cooled Condenser,” GRC Transactions, Vol. 24. Davis, CA: Geothermal Resources Council pp. 485-489

Kutscher, C.; Costenaro, D. (2002). “Assessment of Evaporative Cooling Enhancement Methods for Air-Cooled Geothermal Power Plants,” GRC Transactions, Vol. 26. Davis, CA: Geothermal Resources Council pp. 775-779.

Kutscher, C.; Gawlik, K. (2003). “Report on Measurements of the Mammoth Pacific Power Plant Evaporative Pre-Cooling Systems taken on September 26-27 and November 13,2002,” Golden, CO: National Renewable Energy Laboratory.

Matches. (2003). www.matches.com. Accessed June 6, 2010.

Peterson, Ross. (12 December 2010). Equipment quote. MEE Industries, Inc. California.

Public Utilities Commission of the State of California. (2009). "Energy Division Resolution E-4298." http://docs.cpuc.ca.gov/PUBLISHED/FINAL_RESOLUTION/111386.htm. Accessed June 6, 2010.

Seider, W.; Seader, J.; Lewin, D.; Widagdo, S. <u>Product and Process Design Principles</u>. John Wiley and Sons, Inc. Hoboken, New Jersey. 2009.

Shepherd, Don. (22 December 2010). Equipment quote. Caldwell Energy, Inc. Kentucky.

Smith, Andrew. (9 June 2010). Equipment quote. GEA Heat Exchangers / GEA Power Cooling, Inc. Lakewood, CO.

Turchi, C. (July 2010). "Parabolic Trough Reference Plant for Cost Modeling with the Solar Advisor Model (SAM)." NREL/TP-550-47605. Golden, CO: National Renewable Energy Laboratory.

Wilcox, S.; Marion, W. (2008). *User's Manual for TMY3 Data Sets*. NREL/TP-581-43156. Golden, CO: National Renewable Energy Laboratory.

REPORT DOCUMENTATION PAGE			*Form Approved* *OMB No. 0704-0188*

The public reporting burden for this collection of information is estimated to average 1 hour per response, including the time for reviewing instructions, searching existing data sources, gathering and maintaining the data needed, and completing and reviewing the collection of information. Send comments regarding this burden estimate or any other aspect of this collection of information, including suggestions for reducing the burden, to Department of Defense, Executive Services and Communications Directorate (0704-0188). Respondents should be aware that notwithstanding any other provision of law, no person shall be subject to any penalty for failing to comply with a collection of information if it does not display a currently valid OMB control number.

PLEASE DO NOT RETURN YOUR FORM TO THE ABOVE ORGANIZATION.

1. **REPORT DATE** *(DD-MM-YYYY)*	2. **REPORT TYPE**	3. **DATES COVERED** *(From - To)*
March 2011	Technical Report	

4. **TITLE AND SUBTITLE**	5a. **CONTRACT NUMBER**
Hybrid Cooling Systems for Low-Temperature Geothermal Power Production	DE-AC36-08GO28308
	5b. **GRANT NUMBER**
	5c. **PROGRAM ELEMENT NUMBER**
6. **AUTHOR(S)**	5d. **PROJECT NUMBER**
Andrea Ashwood and Desikan Bharathan	NREL/TP-5500-48765
	5e. **TASK NUMBER**
	ARGT.0910
	5f. **WORK UNIT NUMBER**

7. **PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)**	8. **PERFORMING ORGANIZATION REPORT NUMBER**
National Renewable Energy Laboratory 1617 Cole Blvd. Golden, CO 80401-3393	NREL/TP-5500-48765
9. **SPONSORING/MONITORING AGENCY NAME(S) AND ADDRESS(ES)**	10. **SPONSOR/MONITOR'S ACRONYM(S)**
	NREL
	11. **SPONSORING/MONITORING AGENCY REPORT NUMBER**

12. **DISTRIBUTION AVAILABILITY STATEMENT**
National Technical Information Service
U.S. Department of Commerce
5285 Port Royal Road
Springfield, VA 22161

13. **SUPPLEMENTARY NOTES**

14. **ABSTRACT** *(Maximum 200 Words)*
This paper describes the identification and evaluation of methods by which the net power output of an air-cooled geothermal power plant can be enhanced during hot ambient conditions with a minimal amount of water use.

15. **SUBJECT TERMS**
geothermal; electricity generation; power plant; air-cooling; air-cooled condenser; hybrid air-cooling

16. **SECURITY CLASSIFICATION OF:**			17. **LIMITATION OF ABSTRACT**	18. **NUMBER OF PAGES**	19a. **NAME OF RESPONSIBLE PERSON**
a. **REPORT**	b. **ABSTRACT**	c. **THIS PAGE**			
Unclassified	Unclassified	Unclassified	UL		19b. **TELEPHONE NUMBER** (*Include area code*)

Standard Form 298 (Rev. 8/98)
Prescribed by ANSI Std. Z39.18