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September 30, 2013

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Dear AAD Document Control:

Subject: Final Report for Subtask 5.10 – Testing of an Advanced Dry Cooling Technology for Power Plants; Cooperative Agreement No. DE-FC26-08NT43291; EERC Fund 18202

Attached is the final report for the subject subtask. If you have any questions, please contact me by phone at (701) 777-5083, by fax at (701) 777-5181, or by e-mail at cmartin@undeerc.org.

Sincerely,

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SUBTASK 5.10 – TESTING OF AN ADVANCED DRY COOLING TECHNOLOGY FOR POWER PLANTS

Final Report

(for the period October 1, 2012, through September 30, 2013)

Prepared for:

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SUBTASK 5.10 – TESTING OF AN ADVANCED DRY COOLING TECHNOLOGY FOR POWER PLANTS

ABSTRACT

The University of North Dakota's Energy & Environmental Research Center (EERC) is developing a market-focused dry cooling technology that is intended to address the key shortcomings of conventional dry cooling technologies: high capital cost and degraded cooling performance during daytime temperature peaks. The unique aspect of desiccant dry cooling (DDC) is the use of a hygroscopic working fluid—a liquid desiccant—as a heat-transfer medium between a power plant's steam condenser and the atmosphere. This configuration enables a number of beneficial features for large-scale heat dissipation to the atmosphere, without the consumptive use of cooling water. The overall goal of this project was to accurately define the performance and cost characteristics of DDC to determine if further development of the concept is warranted. A balanced approach of modeling grounded in applied experimentation was pursued to substantiate DDC-modeling efforts and outline the potential for this technology to cool full-scale power plants. The resulting analysis shows that DDC can be a lower-cost dry cooling alternative to an air-cooled condenser (ACC) and can even be competitive with conventional wet recirculating cooling under certain circumstances. This project has also highlighted the key technological steps that must be taken in order to transfer DDC into the marketplace. To address these issues and to offer an extended demonstration of DDC technology, a next-stage project should include the opportunity for outdoor ambient testing of a small DDC cooling cell.

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SUBTASK 5.10 – TESTING OF AN ADVANCED DRY COOLING TECHNOLOGY FOR POWER PLANTS

EXECUTIVE SUMMARY

The Energy & Environmental Research Center (EERC) is developing a novel dry cooling technology to meet the cooling needs of power plants located in arid environments. This technology is intended to address the key shortcomings of conventional dry cooling technologies: high capital cost and degraded cooling performance during daytime temperature peaks. The key feature of desiccant dry cooling (DDC) technology is the use of a hygroscopic working fluid—a liquid desiccant—as a heat-transfer medium between a power plant's steam condenser and the atmosphere. This configuration affords several advantages for overall cooling system performance.

The overall goal of this project was to accurately define the performance and cost characteristics of DDC to determine if further development of the concept is warranted. In order to achieve this goal, necessary supporting project efforts were divided into several activities, including experimental performance measurement of a DDC system, extrapolation of the measured results to full-scale power plants, development of an economic model for DDC, and finally, case study calculations to compare the features of DDC to the conventional cooling options of wet recirculating cooling and an air-cooled condenser (ACC).

The resulting performance and economic models were used to evaluate case studies which were based on cooling a 300-MW_e net coal power plant for three different locations: Gillette, Wyoming; Atlanta, Georgia; and Phoenix, Arizona. The case study calculations indicate that DDC consistently maintained a lower annual cooling system cost compared to an ACC. Annual cooling costs for DDC averaged 60% of those of an ACC for the evaluated cases. Parasitic power requirements for DDC were also estimated to be lower, averaging 65% of those for a comparable ACC. Compared to wet recirculating cooling, the annual costs for DDC were within ±10% of the comparable wet system including the energy penalties associated with lost power production with the desiccant system. The breakeven cost of water for DDC ranged from a low of $1.72/kgal for Atlanta to a high of $3.35/kgal for Phoenix for the specific assumptions used.

Regarding the potential environmental impacts of DDC, experimental testing supports the hypothesis that DDC can be an environmentally benign cooling option. The key environmental concern, carryover of desiccant, appears to be manageable with proper design and operation of the cooling system. Measured drift rates were determined to be less than 0.00006% of the circulating working fluid rate. However, it is desirable to demonstrate an even lower limit for drift to avoid potential particulate emission limits that are currently imposed on conventional wet cooling towers.

This project has also highlighted the key technological steps that must be taken in order to transfer DDC into the marketplace. To address these issues and to offer an extended demonstration of DDC technology, a next-stage project should include the opportunity for outdoor ambient testing of a small DDC cooling cell.

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SUBTASK 5.10 – TESTING OF AN ADVANCED DRY COOLING TECHNOLOGY FOR POWER PLANTS

INTRODUCTION

Water is essential for thermoelectric power generation. It is the working fluid for the Rankine cycle, which is used to produce power from thermal energy, and it is used significantly as a conduit for numerous power plant operations such as environmental control and transport processes. But, by far, the largest and most critical water use at power plants has been to provide cooling. For example, at a new water-cooled coal power plant, cooling water represents 85%–90% of the plant's consumptive water use (1).

Water-based cooling is typically the most efficient and cost-effective solution where water is available. However, water demands for power plant cooling are increasingly seen to be in conflict with sustainable water use. This tension is the motivating force for pursuing improvements to utility-scale dry cooling of thermoelectric power plants.

Dry cooling options currently exist that can virtually eliminate a plant's need for cooling water, and there are a small number of dry cooling systems in the United States, with the most common configuration being the air-cooled condenser (ACC). In these systems, steam turbine exhaust is routed directly to an ACC where the latent heat of steam condensation is dissipated to the atmosphere through the sealed walls of the condenser. The key disadvantage of conventional dry cooling is a lower return on investment; ACCs are three to four times more expensive to construct than wet recirculating cooling systems (2), and the performance of an ACC degrades rapidly with hot weather—often limiting plant output during times of peak demand. The cost and performance gap associated with conventional dry cooling is a costly disadvantage for plants that are forced to use it because of resource limitations.

In response to these trends, the Energy & Environmental Research Center (EERC) is developing a novel dry cooling technology to meet the cooling needs of power plants located in arid environments (3, 4). This technology is intended to address the key shortcomings of conventional dry cooling technologies: high capital cost and degraded cooling performance during daytime temperature peaks.

BACKGROUND

The key feature of desiccant dry cooling (DDC) technology is the use of a hygroscopic working fluid—a liquid desiccant—as a heat-transfer medium between a power plant's steam condenser and the atmosphere. The process is schematically similar to a conventional wet evaporative cooling system, as shown in the simplified process diagram of Figure 1, but the underlying components are significantly different than those in conventional cooling systems because of the properties of the desiccant working fluid. In the EERC system, a circulating water loop cools the steam condenser and transfers thermal energy to a desiccant-based cooling circuit

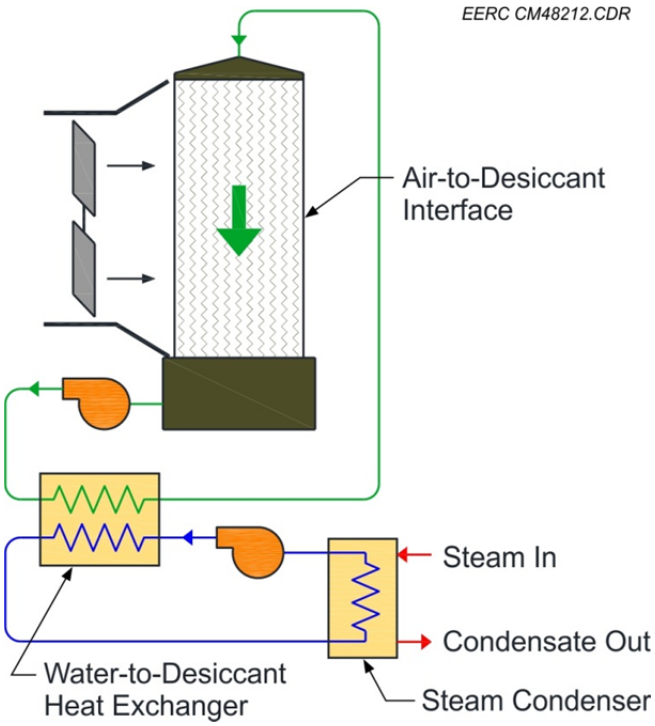

Figure 1. Simplified schematic of the DDC concept.

within a direct-contact, desiccant-to-air heat exchanger. The thermal energy absorbed by the desiccant is dissipated directly to a convective flow of ambient air. Contact between the desiccant and the air in the heat exchanger is similar to that in a conventional wet cooling tower. However, unlike evaporative cooling with water, the desiccant is stable under contact with the atmosphere (i.e., it will not completely evaporate, but remains as a liquid) and can be recycled continuously without requiring significant makeup fluid. It is assumed that the initial charge of working fluid should last for the life of the system.

The significant improvement of the EERC's DDC system over other state-of-the-art approaches is the result of the direct-contact heat exchange that occurs in the air heat exchanger. In this heat exchanger, heat transfer from the desiccant to the atmosphere is primarily through sensible heating of the air, but the process is augmented by transient absorption and desorption of atmospheric moisture. This configuration affords several advantages for overall cooling system performance, including the following:

- The heat-transfer area is defined by the interfacial area between the working fluid and the air. Therefore, it is possible to create very large heat-transfer surfaces with relatively inexpensive wetted packings or structured film fill material.

- Combined heat and mass transfer occur between the working fluid and the air, which improves heat-transfer efficiency at low temperature differentials. In a conventional ACC, heat transfer is limited by the outside surface convection rate; higher rates require faster, more turbulent flow, which necessitates high fan power consumption.

Combined energy and mass exchange in the EERC system adds an additional driving gradient (vapor pressure) to enhance heat transfer without the need to increase fan power.

- Daily fluctuations in ambient temperature cause cyclic absorption and evaporation of moisture in the working fluid. The cyclic periods of evaporation can be tuned to offset the daytime performance degradation that hinders conventional dry cooling systems. This feature is shown conceptually in Figure 2, where the fluctuating sensible heat transfer (representative of an ACC) is compared to the dampened, net heat transfer of the DDC system that can use both sensible and latent heat-transfer components.

The benefits associated with direct-contact operation rely heavily on the hygroscopic, or moisture-retaining, properties of the desiccant fluid. Several candidate fluids have been identified for this process, and the most promising are nonproprietary materials that are available in industrial quantities. While the fluids are, in general, corrosive and their emission to the atmosphere as drift aerosols is not permissible, they have established handling procedures and are commonly used in industry. The corrosive nature of these fluids will require some handling and safety precautions at a plant site. However, the candidate fluids are not included in the U.S. Department of Homeland Security's chemicals of interest listing.

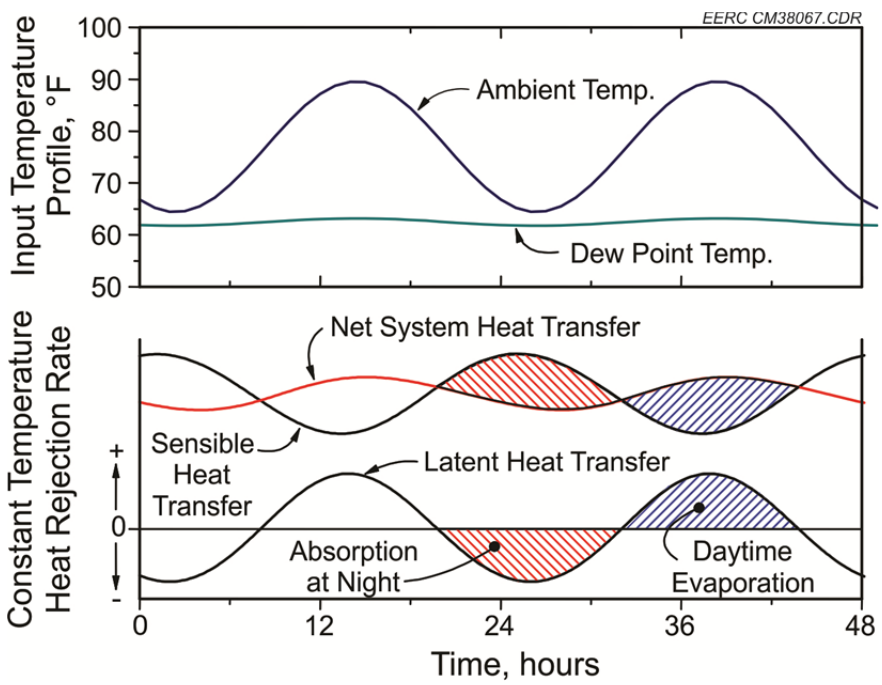

Figure 2. Conceptual diagram of the DDC system's net heat-transfer performance (lower graph) in response to a diurnal ambient air temperature profile (upper graph).

GOALS AND OBJECTIVES

The overall goal of this project was to accurately define the performance and cost characteristics of DDC to determine if further development of the concept is warranted. Specific objectives were identified based on the results from a previous stage of development (4) and are outlined below:

- The first objective was to modify the experimental system by adding a water-to-desiccant working fluid heat exchanger that was identified as a promising configuration during previous work on the concept.

- A second objective was to conduct performance testing with the modified system configuration over a range of climatic conditions representative of the continental United States.

- A third objective was to design a full-scale module of the cooling technology to serve as a basis for estimating the capital costs associated with the technology.

- The final objective of the project was to develop a performance and cost model of the cooling technology based on the information collected during this project.

METHODS

In order to achieve the outlined objectives, necessary supporting project efforts were divided into the following five activities:

- Activity 1 – Modify Experimental System
- Activity 2 – Performance Testing
- Activity 3 – Full-Scale Unit Design Study
- Activity 4 – Economic Model Development
- Activity 5 – Case Study Evaluations

Activity 1 – Modify Experimental System

Activity 1 consisted of installing a water-to-desiccant fluid heat exchanger in the test facility constructed under Stage 1 at the EERC (4). Stage 1 testing began with the evaluation of a desiccant-cooled steam condenser since this arrangement eliminates a heat-transfer step associated with the intermediate water heat-transfer loop, as shown in Figure 1. However, testing and feedback from industry regarding the Stage 1 results did not favor this configuration, and activities were focused on the current configuration with the water-to-desiccant heat exchanger.

The modified test facility with the water interface loop is diagrammed in Figure 3. The facility has provisions for supplying a temperature- and humidity-controlled airstream for exposing the air heat exchanger, and it also has an electrically heated boiler that can either

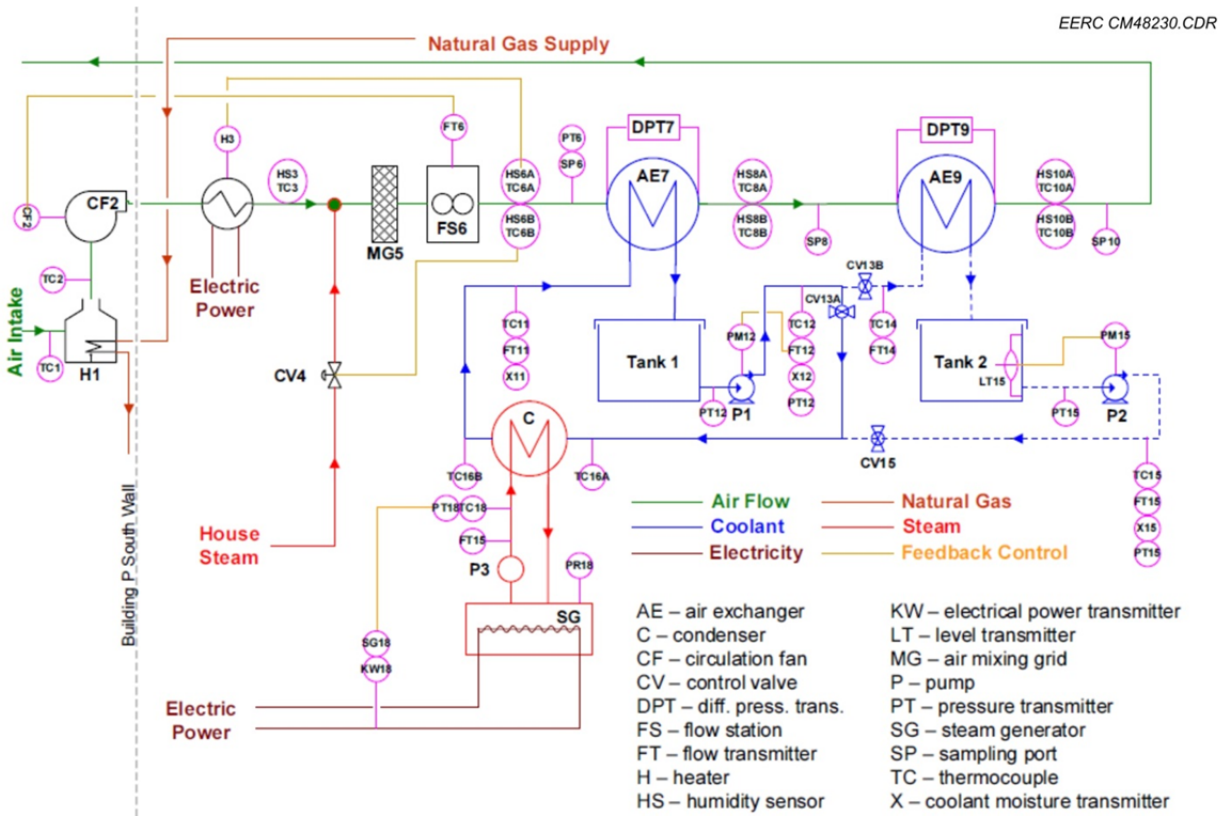

Figure 3. Process diagram for the dry cooling test facility.

produce low-pressure steam to mimic the thermal load of steam turbine exhaust or, as used in this work, generate a stream of hot water that is heated by circulation through the boiler.

As shown in Figure 3, air temperature regulation is divided into two stages: air is first drawn through a gas-fired preheater and is then blown across a bank of resistance heating elements for trim temperature control. Following the trim heater, steam is injected to maintain the desired humidity set point. An expanded metal grid is located further downstream to induce turbulence in the airflow and promote thorough mixing. After passing through the air exchanger test sections, the flow is turned 180° and exhausted back outdoors. Figure 4 contains photos of the system during construction that show the outdoor air-preheating system and the air circulation ductwork.

The system's cross-sectional flow path is square in shape and 0.76 m to a side. Volume flow rate is monitored by an averaging differential pressure flow station with an integral flow straightener section, FS6 in Figure 3. As shown in Figure 3, flow, temperature, and humidity data collected at Station 6 are used to provide feedback to the circulation fan, trim heater, and steam valve, respectively. At the inlet and outlet of the air exchanger test sections, dual measurements of temperature and humidity are made on opposite sides and at different elevations within the duct and averaged in order to minimize effects of stratification.

Figure 4. Exterior (left) and interior (right) views of the cooling system test facility.

The heat load used to challenge the cooling technology is generated by an electrically heated steam boiler that can be maintained under vacuum to simulate power plant turbine exhaust; however, for this work a stream of hot water was generated by simply circulating water through the boiler. A photograph of this system is provided in Figure 5; in this configuration, a polymer tube bundle was inserted into a shell made from 6-inch carbon steel piping. Hot-water flow was through the shell side of the heat exchanger; water was pumped into the bottom of the shell, and it flowed back into the boiler by overflowing and falling down through the vertical riser tube. Desiccant flow was through the tube side of the heat exchanger, and it entered and exited through the horizontally oriented flanges. Existing commercially available options that would be suitable for the water-to-desiccant heat exchanger were evaluated for this testing, and a fluropolymer tube bundle was ultimately selected.

The most appropriate working fluids for use in the EERC's dry cooling technology are necessarily hygroscopic, but they are also generally corrosive to common engineering materials and pose a concern for operating personnel safety. It is believed that a safe cooling system can be engineered, but additional safety precautions may be needed compared to conventional wet or dry cooling systems. To satisfy safety needs during testing, curtains were installed around the test system to contain unexpected splashes and keep all but authorized operators out of the equipment zone. This is highlighted in the photograph of the completed system (Figure 6). System operators were required to wear the appropriate personal protective equipment, consisting of protective overalls, gloves, eye protection, and a hard hat with face shield when working behind the protective curtains.

In addition to the physical components of the system, a LabView-based control and data acquisition system was customized for the test facility and implemented on a PC laptop. This control system processed the feedback loops that maintained the airflow-conditioning set points, liquid flow set point, and heat load input. A necessary feature of the control system was the ability to set either static or dynamic airflow-conditioning set points. As the name implies, static set points maintain constant air temperature and relative humidity values at the air exchanger inlet. In dynamic mode, the temperature and humidity set points were continually updated in order to mimic the diurnal weather cycle of outdoor air.

Figure 5. Photograph of the hot-water boiler and attached water-to-desiccant heat exchanger.

Figure 6. Photograph of the completed dry cooling test facility during a drift-sampling measurement.

The control system also displays and records data from the system's transducers, and it calculates a number of mass and energy balance parameters instantaneously to aid interpretation of the system's performance during operation. Additionally, a number of automatic alarms and emergency shutdown routines have been incorporated into the control system in case of unexpected events.

Activity 2 – Performance Testing

Once the modifications were made to the test facility, an additional series of performance tests were conducted under Activity 2 using a range of simulated climatic conditions in order to extract the operational data needed for accurate performance modeling. Testing consisted of exposing heated desiccant solution to both static and dynamically changing ambient air conditions in the air–desiccant contactor. Static testing presented a constant temperature and moisture content airstream to the contactor in order to provide experimental validation points for the three possibilities of combined latent and sensible heat transfer: 1) sensible heat dissipation only, with no net change to desiccant concentration; 2) combined sensible and latent heat dissipation, accompanied by an increase in desiccant concentration; and 3) sensible heat dissipation of both the hot-water thermal load and the heat of absorption as the desiccant absorbs atmospheric moisture and becomes less concentrated.

Experimentally derived measurements of the air-to-desiccant heat-transfer coefficient were made at several airflow velocities during static condition testing. In order to eliminate latent heat-transfer effects from these measurements, the desiccant solution was allowed to reach equilibrium so that there was no net mass transfer taking place. These data were then used to derive an effective heat-transfer coefficient correlation as a function of air velocity. It was difficult to achieve the analogous situation to identify a mass transfer coefficient since the vapor pressure changed with concentration change of the desiccant. Instead, it was found that a suitable estimate could be derived using the heat and mass transfer analogy and the aforementioned heat-transfer coefficient.

Dynamic condition testing was also conducted in order to evaluate the time-varying features of the cooling technology. These tests used the functionality of the test facility's controller to supply an inlet airstream with a sinusoidal temperature profile that was intended to be representative of day-to-night ambient temperature variations. The air moisture content for these tests was held constant since the moisture content of outdoor air is typically more stable than ambient temperature. This testing clearly demonstrated the predicted periods of moisture absorption during the simulated nights, with evaporation during the days. As with the static condition testing, these experimentally measured conditions were used to validate the modeling assumptions used for the performance modeling.

A further aspect of experimental testing was to make measurements regarding the possible environmental impact of DDC. Carryover and emission of desiccant solution in the exhaust airstream could lead to localized equipment corrosion and would possibly pose safety concerns. Furthermore, carryover of working fluid would create a situation comparable to the drift that leaves conventional wet cooling towers and would contribute to a plant's overall particulate air emissions.

Drift sampling was conducted downstream of the air exchanger test section, SP10 in Figure 3. An extractive sampling technique based on U.S. Environmental Protection Agency sampling methodology for stack and plume emissions was used to sample for water-soluble species of the specific desiccant used during testing. A diagram of the sampling setup is shown in Figure 7, with a photograph of the equipment in Figure 8. As indicated in the schematic, the nozzle was inserted through the test facility duct wall and used to isokinetically extract a sample of the exhaust airflow. The gas sample passed through a particulate filter (heated to prevent moisture condensation), through an umbilical tube, and into an impinger train to absorb any gas-phase vapors. The vapor collection train was probably not necessary since the thermodynamic properties of the working fluid suggest that it is very unlikely that the fluid would exist as a vapor. Instead, it is much more likely that constituents of the desiccant, if present, would be detected on the particulate filter or in the nozzle rinse.

Recovery of the sample consisted of emptying the impinger solutions into a collection flask and rinsing the nozzle, umbilical tube, filter holder, and all connecting glassware into the same flask. The particulate filter itself was then submerged in the same rinse liquid where any water-soluble aerosols would enter the liquid phase. The resulting solution was then sampled and analyzed for the working fluid's characteristic chemical species using ion chromatography.

Activity 3 – Full-Scale Unit Design Study

Activity 3 consisted of conducting a design study for a full-scale module of the cooling technology. This exercise provided the basis for estimating the capital costs associated with the DDC and led to identification of many characteristic features. The core of this effort was to create and validate a simulation of the cross-flow, air-to-desiccant interface since it is the performance of this component that most impacts the overall design of the cooling system. This simulation generates an estimate of the heat dissipation performance of the air–desiccant interface scaled for a full-size system. With this performance estimate, the size and, ultimately, the cost of the key system parameters could be estimated in a straightforward manner.

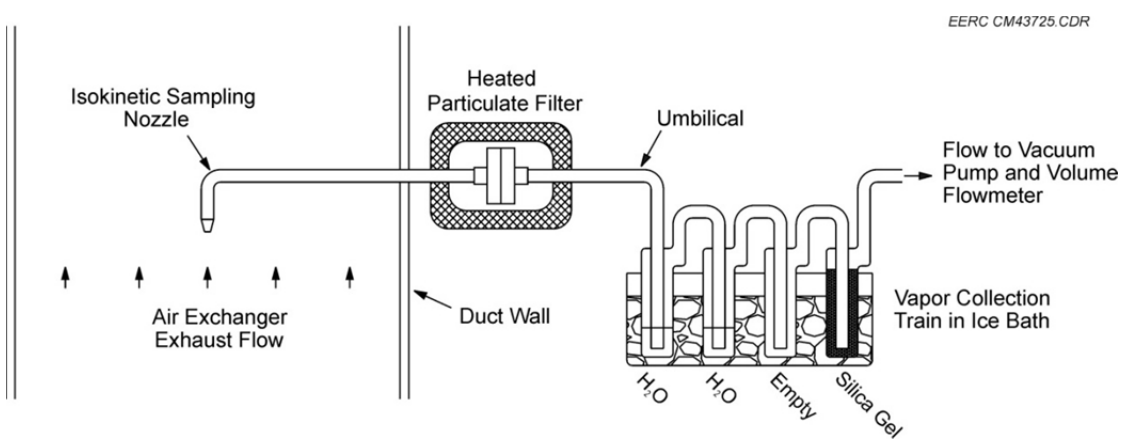

Figure 7. Schematic of the drift measurement apparatus.

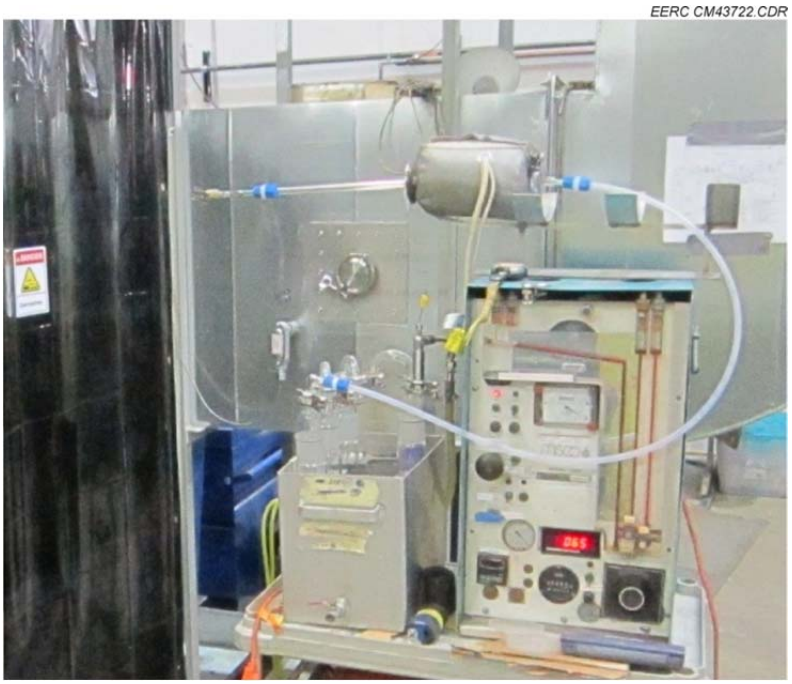

Figure 8. Photograph of drift-sampling equipment.

Cross-flow configuration was selected for the full-scale design study for the primary reason of not needing a liquid disengagement zone such as that required by a counterflow contactor. Impingement of liquid droplets on the liquid pool or other surfaces is believed to contribute to drift formation, and this mechanism can be eliminated in a cross-flow design by extending the fill surface down into the liquid pool. A secondary reason to pursue a cross-flow design compared to a countercurrent one is that the cross-flow footprint has the potential to be more compact for typical DDC operating conditions.

Modeling efforts began with an extraction of the needed modeling parameters from the experimental test data such as estimates for the heat- and mass transfer coefficients. These data were used in a finite element model that tracked the heat and mass exchange of individual fluid elements as they flowed down the fill in the presence of a cross-flow of air. Results from the model were then corroborated with experimental static test results and the trends from dynamic testing. This basic cross-flow interface model was then extended to predict performance for larger cross-flow sections that would be typical of a full-scale power plant cooling system. While the total heat load dissipated by a utility-scale cooling system is approximately 4 orders of magnitude larger than the experimental system, the scale-up for an individual cross-flow surface was limited to a factor of 100.

Full-scale system sizing was then performed given input design conditions of ambient dry-bulb temperature, thermal load, and the desired steam condensation temperature. First, the necessary hot desiccant temperature was derived using knowledge of the expected heat exchanger temperature differentials. The cross-flow interface simulation was then used to estimate heat transfer to the air in the air contactor for a single film fill surface. The total

required volume of film fill was subsequently estimated by scaling the performance of a single surface to meet the desired thermal load; the remaining physical sizes, capacities, etc., were then derived from the fill volume estimate. These parameters were used as inputs to the economic model for calculating DDC system cost estimates.

Activity 4 – Economic Model Development

The primary task under this activity was to develop a capital cost-estimating model that assigned costs to the full-scale designs generated under Activity 3. In parallel to DDC economic model development, models were also created for conventional cooling systems including a wet recirculating system and an ACC.

DDC-specific costs were based on a ground-up approach of costing major system components, as outlined in Table 1.

For comparative purposes, economic models were also derived for the conventional cooling options of wet recirculating cooling and dry cooling with an ACC. Similar to the DDC economic model, capital costs for the conventional systems began with sizing estimates based on the thermal load and the heat-transfer performance of each system. The performance of these systems was calculated based on established heat-transfer assumptions for each system and results from the developed models were compared to available data and models from the literature.

The remaining economic modeling parameters included estimates for various direct and indirect costs that are not specific to cooling system type. These costs and the justifying assumptions used for case study modeling are summarized in Table 2.

An economic model evaluation meeting with a cooling system manufacturer was also associated with the completion of Activity 4. This meeting provided the opportunity to gather constructive feedback about the model and recommendations for the case study evaluations. Engineers from a manufacturer of wet and dry cooling systems, SPX Cooling Technologies, were consulted to review the economic modeling methodology and provide feedback on the case study trends.

Activity 5 – Case Study Evaluations

In order to make the results of this project more accessible to potential end user utilities, the economic model was used under Activity 5 to evaluate specific case studies comparing DDC to conventional cooling options. These studies considered cooling of a 300-MW_e (net) output power plant, with an efficiency assumption typical of a modern coal-fired facility.

Three U.S. locations were selected for the case studies based on the variety of climatic conditions represented and their potential for future energy development. The chosen locations included Gillette, Wyoming; Atlanta, Georgia; and Phoenix, Arizona. Five years of historical weather data were collected for each location, and data for 2008 were selected for use in the case

Table 1. Summary of Major DDC Cost Components and the Associated Costing Methodology

DDC System Component	Costing Basis
Air Contactor	
Cooling Tower Structure	Sizing based on enclosed volume consisting of film fill and air plenum. Cost per unit volume metric based on conventional fiberglass tower construction.
Film Fill	Cost per unit volume metric based on the conventional film fill used during testing.
Desiccant Solution	Delivered price normalized for desiccant component concentration.
Liquid Basin	Cost per unit liquid volume metric established from conventional cooling tower correlations and scaled to account for added costs of corrosive liner.
Liquid Heat Exchangers	
Steam Condenser	Sized according to steam load using the same correlation as for conventional wet recirculating systems.
Water-to-Desiccant Heat Exchanger	Cost per unit surface area derived from an estimate of the potential performance of an advanced heat exchanger. Actual cost and performance targets are currently being evaluated under a separate project.
Major Mechanical	
Air Circulation Fans	Cost per unit fan power for conventional cooling tower fans used. Fan power determined by air travel depth and exhaust air velocity.
Desiccant Circulation Pumps	Cost per unit pump power correlation developed for corrosion-duty pumps.
Water Circulation Pumps	Used same cost per unit pump power correlation as for conventional wet recirculating cooling systems.

studies since they were generally representative based on historical averages for the given locations. Figure 9 highlights the differing climates these locations present in a plot of the correlated wet- and dry-bulb temperatures for 2008. Gillette presents an arid but comparatively cooler climate and is already the site of several air-cooled power plants. In comparison, Phoenix is also arid and has a demonstrated need for water conservation, but has a noticeably hotter climate. Atlanta represents another end of the climate spectrum; being at a latitude similar to Phoenix, it also has a hot climate, but it experiences significant moderating effects of more frequent rainfall and higher humidity. However, recent droughts in the southeastern United States have highlighted the vulnerability of power production to water consumption, even in this region.

Table 2. Economic Modeling Assumptions for Items Common to All Cooling Technology Options (i.e., Wet Recirculating, ACC, and DDC)

Total Capital Cost	
Direct Costs	
Purchased Equipment	Derived from cost correlations developed for the three cooling technologies.
Taxes	3% of purchased equipment.
Freight	1% of purchased equipment.
Total Purchased Equipment (TPE)	Equipment including taxes and freight.
Installation Costs (site preparation, electrical, painting, and fire and lightning protection)	14% of TPE.
Indirect Costs	
General Facilities Capital	10% of direct costs.
Engineering and Supervision	10% of direct costs.
Start-Up/Performance Test	7% of direct costs.
Construction Interest	3-year construction period at 8% interest.
Contingencies	15% of direct costs.
Annual Cooling Cost	
Maintenance Materials	1% of total capital cost.
Labor	
Maintenance	50% of materials.
Operations	4 hours/shift for wet; 8 hours/shift for ACC and DDC at $24.00 per hour (no overhead).
Supervisor	30% of maintenance and operating labor.
Utilities	
Parasitic Power	Billed according to energy charge schedule.
Water	$2 per 1000 gallons.
Annualized Capital Recovery	30-year economic life, with a 12% discount rate.
Annual Indirect Costs	
Overhead	60% of materials and labor.
Property Tax	1.5% of total capital.
Insurance	1.5% of total capital.
Administration	2% of total capital.

While it is unlikely that a moderate-sized coal-fired facility would be considered viable for all of the case study locations, this application was kept consistent among the three locations to allow comparisons as a function of climate. Pertinent plant application data, including condenser thermal load, are summarized in Table 3. Given the different efficiencies for a coal-fired power plant versus the bottoming cycle of a combustion turbine, the 423-MW_{th} heat load shown in Table 3, is roughly 1.3 times the thermal load that would need to be dissipated by the bottoming cycle of a 500-MW_e net natural gas combined cycle plant (which is estimated to be 316 MW_{th}).

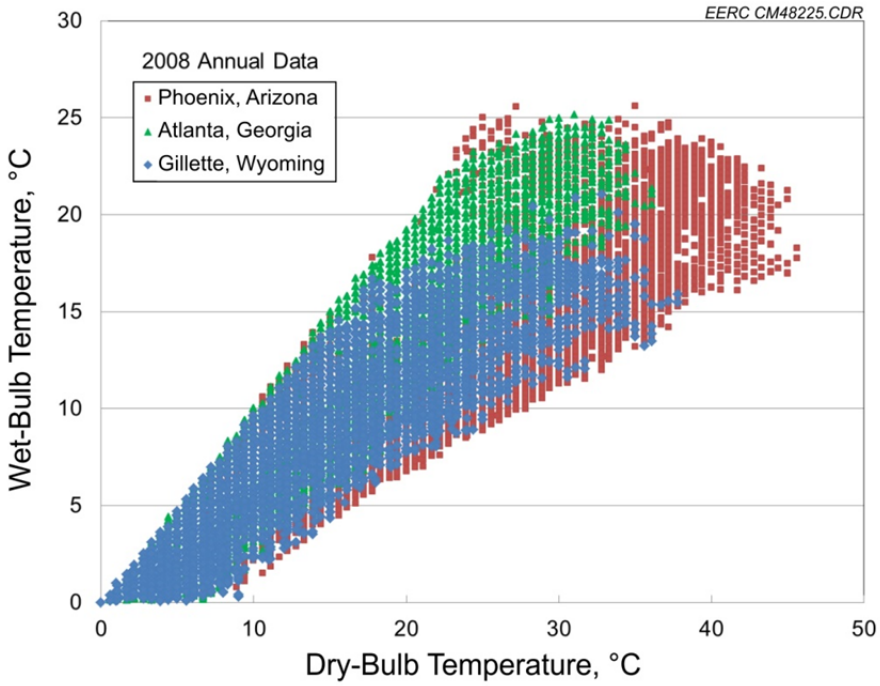

Figure 9. Plot of the 2008 wet-bulb-versus-dry-bulb data for the case study locations.

Table 3. Power Plant Conditions for the Case Studies

Plant Size	300 MW_e net (325 MW_e gross)
Fuel Type	Coal
Net Unit Efficiency	34.5%
Heat Rejected to Condenser	423 MW_{th}
Annual Capacity Factor	0.8
Elevation	1830 m for Gillette; 335 m for Atlanta and Phoenix
Weather Simulation Year	2008

For each case study, the cooling performance for each system was computed on an hourly basis in response to the constant thermal load from the plant and the ambient air conditions recorded for that time; steam condensation pressure and parasitic power consumption were the return values. For cases where the condenser could be cooled to its design pressure or lower, the plant was credited to have produced its design power capacity less the parasitic power required by the cooling system itself. For cases when the cooling system could not cool the condenser to its design pressure, the plant's power output was reduced by an amount based on the difference from its design condenser pressure according to the penalty schedule in Table 4. Annual generation totals were prepared from the sum of these hourly power production estimates.

Revenue from power production was calculated using the hourly estimates of the power produced, scaled by an energy charge rate that changed hourly on a daily cycle. The daily energy charge values are summarized in Table 5 and were intended to incorporate an element of time

Table 4. Lost Generation Values Used for the Case Study Calculations Due to Elevated Condenser Pressure (linear interpolation was used for intermediate-pressure values)

Condenser Pressure, inches of Hg	Plant Output Reduction, MW_e
Less than 2.5	0
Design Pressure: 2.5	0
4	5.6
6	13.1
Maximum Pressure: 8	20.6
Greater than 8	Not permitted; simulations limited to a maximum condenser pressure of 8 inches Hg.

Table 5. Energy Charge Schedule Used for the Case Study Modeling

Local Time	Electricity Value, $/MWh
0:00	21.79
1:00	18.54
2:00	17.02
3:00	16.79
4:00	17.95
5:00	21.66
6:00	29.89
7:00	39.41
8:00	39.46
9:00	39.92
10:00	38.76
11:00	36.85
12:00	34.09
13:00	34.76
14:00	34.21
15:00	33.71
16:00	30.61
17:00	32.28
18:00	38.28
19:00	36.08
20:00	32.77
21:00	28.71
22:00	23.22
23:00	20.68

sensitivity into the model. Specifically, the cooling systems were penalized more for not satisfying the daytime peak, which is precisely when they are most challenged to provide cooling. The energy charge schedule in Table 5 was applied year-round and, therefore, it does not include seasonal extremes that may be observed for certain locations. The values were based on examination of recent (2012) retail power values for several independent system operators across the United States.

RESULTS AND DISCUSSION

Activities 1 through 5 have generated the information needed to compile a reasonably complete technical description of DDC. System characteristics as derived from these sources are presented, followed by the location-specific case studies.

DDC System Characteristics

Thermal Interface

The proposed implementation schematic for DDC, shown in Figure 1, includes an intermediate water circulation loop to transfer heat from the steam condenser to the desiccant in the air contactor. An alternate approach using a desiccant-cooled condenser was evaluated under Stage 1, and while this configuration had the advantage of eliminating the intermediate water loop, the associated heat exchanger and the water circulation pump, testing showed that these advantages were outweighed by other concerns. Feedback from industry also expressed the desire for a water-interfacing cooling system to minimize the chance for power plant contamination, enable retrofits, or provide supplemental cooling.

The effect of including the intermediate thermal loop, as shown in Figure 1, is to place practical limits on the initial temperature difference (ITD) of the cooling system, which is the difference between the steam condensation temperature and the ambient dry-bulb temperature. With the water-cooled condenser shown in Figure 1, thermal energy from the condensing steam must be transferred across several temperature differentials including the steam condenser terminal temperature difference (TTD), the sensible temperature range of the water-cooling loop, the water-to-desiccant heat exchanger TTD, and the final approach temperature between the cool desiccant and the ambient air's dry-bulb temperature. High and low ranges for these temperature approaches are presented in Table 6, and when summed together, they represent a minimum feasible ITD range for DDC. While ACCs can be designed for even lower ITD values, the DDC concept will have a practical minimum range of 14°–23°C.

Table 6. Expected Temperature Ranges for a DDC System (values from industry and data gathered from experimental testing and modeling)

	Ranges, °C	
	Low	High
Steam Condenser TTD	4	5
Sensible Heating Range (water and desiccant)	7	10
Water-to-Desiccant Heat Exchanger TTD	3	3
Desiccant Approach to Ambient Dry-Bulb Temperature	0	5
Overall ITD	14	23

Observed terminal temperature differentials with the experimental system's water-to-desiccant heat exchanger were in the range of 7° to 10°C, but it is believed that the smaller value of 3°C in Table 6 can be achieve by increasing the surface area of contact while still meeting the cost requirements. A separate project is under way to verify the feasibility of such a heat exchanger.

Performance

A sensitivity study was performed to identify the preferred range of operating conditions for DDC technology. The key parameter adjustments that can be made include the following:

- Air velocity through the fill
- Desiccant loading on the fill surface
- Air travel depth
- Film fill height
- Quantity of reserve desiccant
- Thermal loading of the desiccant

Choosing the values of these parameters presents a multivariable optimization problem with the dual metrics of minimizing capital cost and parasitic power consumption as objective functions. Experimental testing has been used to identify feasible ranges for many of the parameters, and subsequent performance simulation has enabled the rational selection of an optimized DDC configuration.

Airflow velocity was selected to be at the low end of its feasible range since it contributes substantially to parasitic power consumption but it does not increase heat transfer to the same degree. This is shown in Figure 10, which compares the linear increase in the experimentally determined heat-transfer coefficient to the parabolic increase in the power needed to force airflow through the fill. With an airflow range established, desiccant loading was then predetermined to some extent based on the achievable desiccant temperature range and the air available for energy exchange. Too much desiccant flow down the fill results in an incomplete temperature drop and excessive pumping loads, while too little results in excessively large heat exchangers.

The overall film fill dimensions were selected to maintain the established liquid-to-gas flow ratio while further minimizing parasitic power. For maximum heat transfer per sheet of film fill, the air travel depth and fill height were not completely independent. As shown in the example fill surface heat-transfer results of Figure 11, the optimum zone of heat transfer is outlined by tightly clustered isotherms that trace a diagonal path from upper left to lower right. Minimum heat-transfer activity is shown as the near-isothermal zones in the upper right and lower left corners. Optimal fill sheet size was selected to include as much of the active heat-transfer zone as possible. The key limitation was total fill height, which was capped by the desiccant pump's parasitic power requirement; taller fill can be more compact but necessitates a higher lift to circulate the desiccant solution.

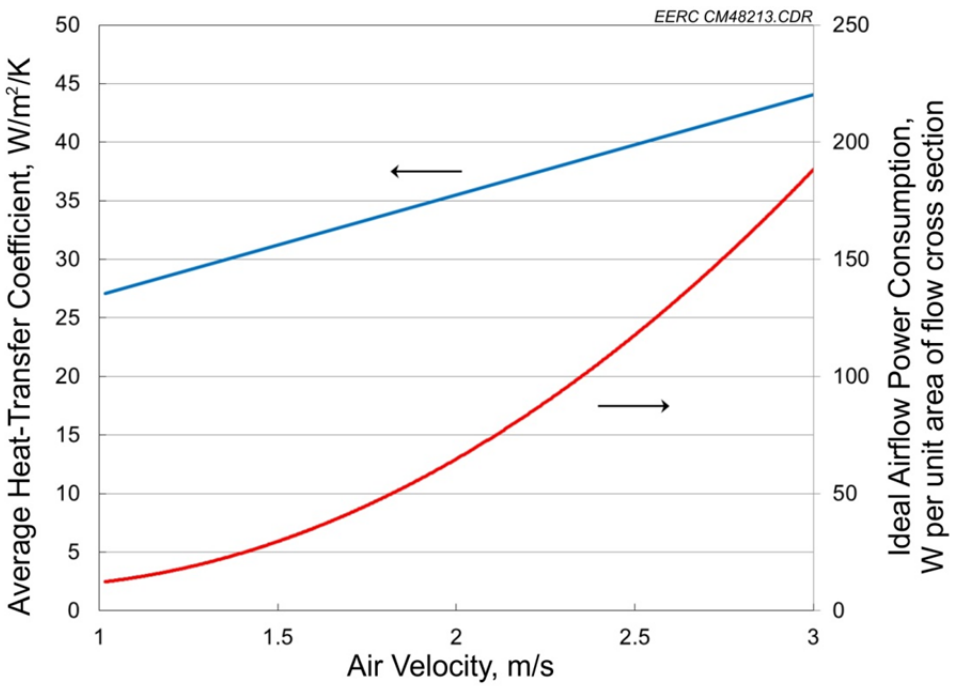

Figure 10. Linear increase in the experimentally determined heat-transfer coefficient compared to the parabolic increase in the power needed to drive airflow.

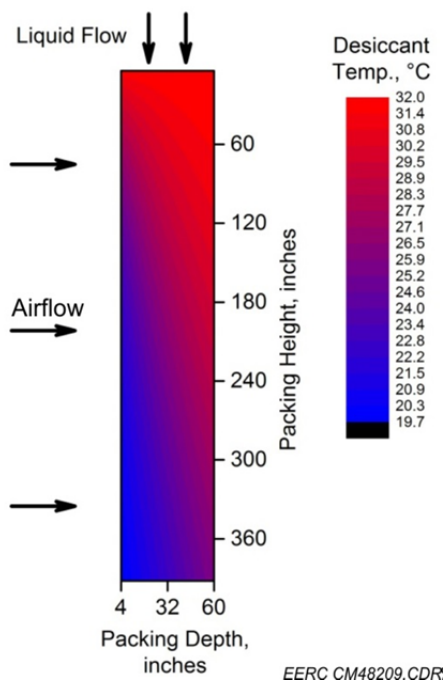

Figure 11. Example simulation results for desiccant temperature on the surface of the cross-flow film fill.

The quantity of reserve desiccant is unique in this discussion in that it does not directly impact the characteristics of the desiccant–air interface. Instead, it is a key parameter that determines the time constant of the cooling system and the consequent degree of day-to-night thermal storage, or more accurately, the amount and timing of the evaporative cooling boost shown in Figure 2. The experimental system was able to vary the diurnal cycle in order to examine the effect of reserve desiccant on time constant. Testing was conducted with simulated diurnal periods ranging from 1 to 12 hours.

The summary effect of the cooling system's time constant is to decrease the significance of the latent component of heat transfer with decreasing system time constant. That is, the faster the system cycles through desiccant solution and reaches equilibrium, the more it can quickly adapt to changing ambient conditions, and the delay time for evaporative cooling is minimized. The quantitative effect of desiccant reserve solution on the dampening effects of the system's time constant is show in Figure 12, which is a plot of the condenser temperature amplitude ratio as a function of the average desiccant residence time in storage. The amplitude ratio is a comparison of the range of steam condenser temperature variations for damped and undamped systems in response to a sinusoidal temperature-forcing function having a period of 24 hours. At a desiccant residence time of zero, the ratio approaches 1, which indicates no thermal damping in the system. However, even with a small amount of reserve solution (less than 1 hour), the amplitude ratio drops rapidly to 0.6–0.7, meaning that the peak-to-trough diurnal temperature variation experienced by the steam condenser in an undamped system would be reduced to 60%–70% of that value in the damped system.

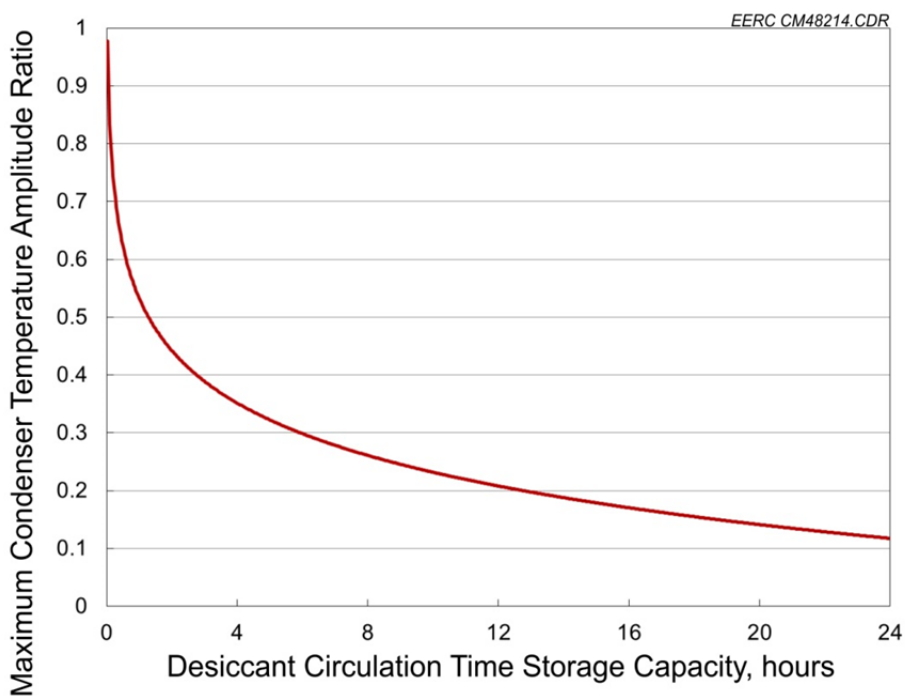

Figure 12. Diurnal temperature swing dampening of DDC as a function of desiccant reserve storage.

As shown by Figure 12, with enough reserve solution, the daily temperature fluctuations can be nearly eliminated; however, a review of the associated economic costs shows that such excessive quantities of desiccant are not cost-effective despite the temperature-stabilizing effect. Instead, simulation results have shown that the best trade-off between cost and benefit is by taking advantage of the 30% to 40% of thermal damping that even small quantities of desiccant can provide. A final item to note is that while the presented time constant analysis was based on simple recirculation of a fixed quantity of reserve desiccant in the tank, alternative strategies have been identified to modify the system time constant beyond simply increasing the desiccant mass.

Thermal loading of the desiccant is the final key system parameter to be mentioned. It also affects system time constant by dictating the hot-desiccant return temperature. Since this factor strongly affects the cost of DDC, its effect was evaluated as the primary sensitivity parameter during the case studies.

Cost

The characteristics of DDC technology point toward specific system features, including low airflow velocity, liquid flow to match the heat capacity of the air, shorter and more shallow fill profiles to reduce parasitic power, and relatively small reserve volumes of desiccant in order to maximize the thermal storage benefit of the desiccant on hand. The dimension of system cost has been explored to further narrow down these general characteristics into specific dimensions and capacities of a DDC system.

System capital cost estimates build on the experimentally derived performance modeling and the individual cost correlations that were the outcome of Activity 4. Figure 13 shows a parametric study of capital costs as a function of cross-flow interface surface dimensions. Each parametric curve corresponds to a specific airflow travel depth; the locus of minimum cost for each depth produces a similar trend as indicated by the diagonal high-intensity heat-transfer zone of Figure 11.

Capital cost only represents one of the minimization objectives; the other is parasitic power consumption. The three primary consumers of parasitic power in DDC are the fan to create airflow through the fill, the desiccant circulation pump and the water interface loop circulation pump. Sensitivity results for parasitic power (relative to the thermal power being dissipated) are shown in Figure 14, which correlate to the same conditions depicted in Figure 13. Each fill depth has a corresponding height that minimizes parasitic power ratio; however, these sizes are only poorly correlated with the cost minimums of Figure 13.

A representative breakdown of major system costs is plotted in Figure 15 for the minimum cost condition of 2 ft of air travel depth in Figure 13. Over two-thirds of the estimated capital costs are associated with the air exchanger; these include the tower structure, fill material, desiccant solution, and desiccant storage basin. The remaining costs are distributed among the liquid heat exchangers and the major mechanical equipment, including liquid circulation pumps and airflow fans.

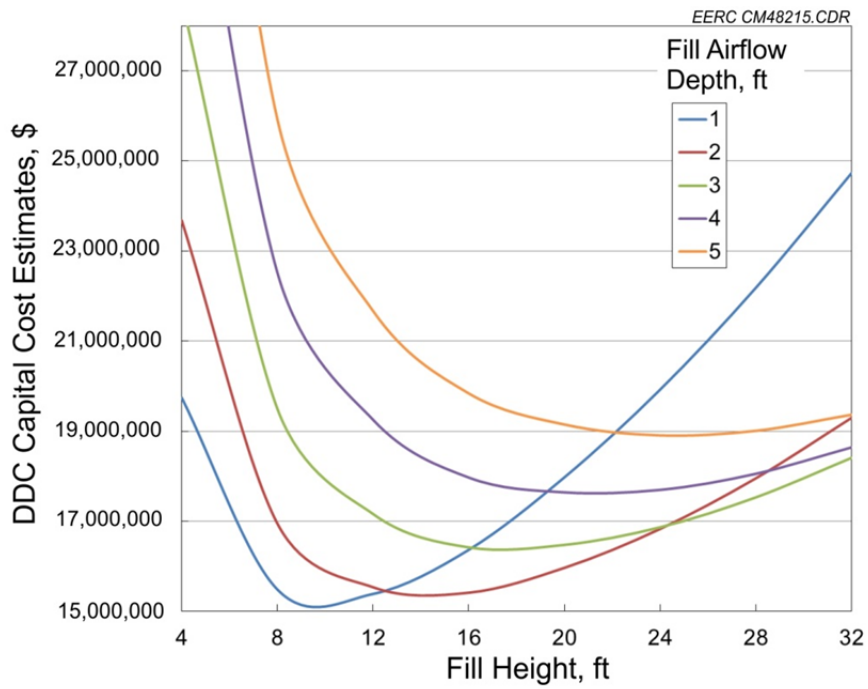

Figure 13. Parametric results for capital cost-versus-cross-flow interface dimensions. Conditions: 316 MW_{th} heat load, 40°C condenser temperature, and 19°C ambient dry-bulb temperature.

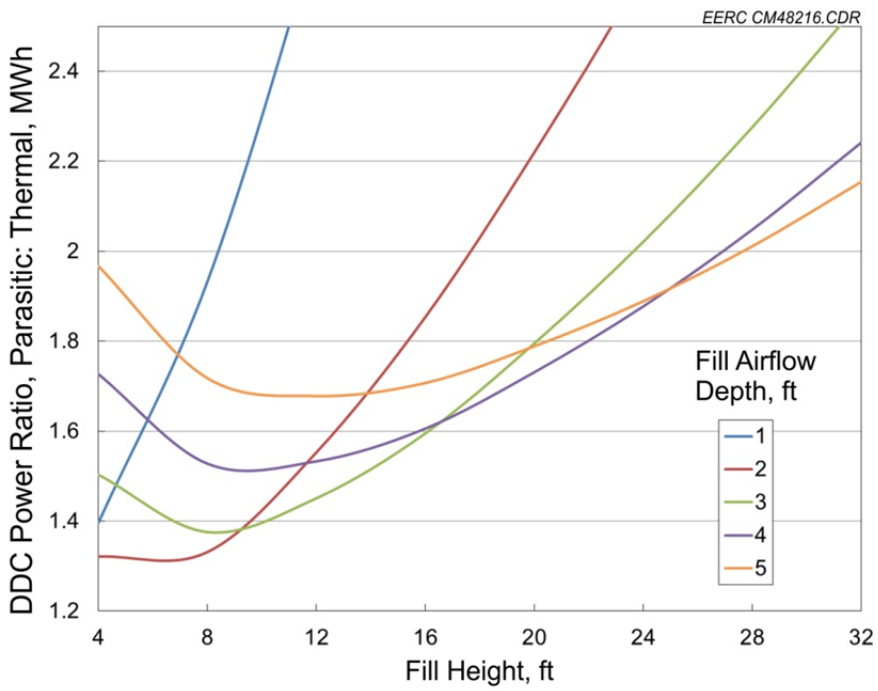

Figure 14. Parametric results for DDC parasitic power ratio-versus-cross-flow interface dimensions. Conditions: 316 MW_{th} heat load, 40°C condenser temperature, and 19°C ambient dry-bulb temperature.

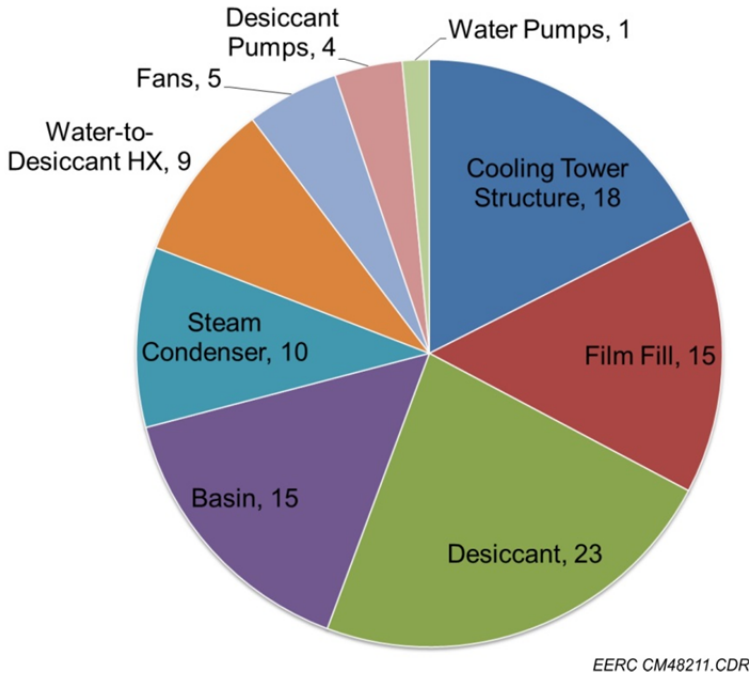

Figure 15. Representative breakdown of major DDC system capital costs. Values are percent of total.

The estimated overall costs for the water-to-desiccant heat exchanger are roughly equivalent to the estimated steam condenser cost for the same heat load. Therefore, on a heat-transfer basis the proposed water-to-desiccant heat exchanger is targeted to be comparable to conventional noncorrosive service heat exchangers.

Environmental Impact

As suggested in Figure 15, the key environmental impact that DDC is envisioned to potentially have is carryover of the desiccant by the airflow within the direct-contact heat exchanger. This issue is well known from conventional wet recirculating cooling tower operation and is termed drift. Compared to conventional wet cooling towers, optimal DDC operation generally dictates that less liquid loading and lower airflow velocities are used. This helps to minimize the potential for carryover, but it also appears necessary to minimize or eliminate sources of entrainable droplets. These sources include splashing, where droplets fall into a pool of liquid, and foaming, where bubbles are allowed to burst, which can release fine aerosols.

With precautions taken to prevent unwanted droplets, experimental measurements have shown drift rates to be lower than 0.00006% of the circulating liquid rate. This compares to the current wet cooling tower industry standard of 0.0005%, but given the higher concentration of dissolved desiccant solids with DDC a 0.00006% drift rate might result in a particulate loading to the environment similar to a conventional wet cooling tower. From an emission standpoint, it is preferred to demonstrate an even lower limit of drift from a DDC system; however, to achieve

this accuracy will require longer-duration sampling exposures averaged over the entire air exhaust flow.

Beyond the emission aspect, another reason to virtually eliminate drift is to minimize desiccant consumption for purely economic reasons. As indicated in the sample cost breakdown of Figure 15, the desiccant solution itself is not an insignificant cost component compared to the other major items. Therefore, maximizing the lifetime of the solution will be important to maintaining the economic competitiveness of DDC since introducing an ongoing consumptive charge for desiccant would adversely impact the technology's projected economics.

An additional environmental impact may involve the treatment and disposal of accumulated dusts and other foreign matter within the direct-contact heat exchanger itself. It seems clear that some amount of contamination from extended contact with the ambient air will take place, but the degree to which it occurs and how it is manifested was not studied in detail at this stage. A follow-on study that includes extended-duration testing of a pilot field demonstration unit would be the ideal opportunity to evaluate these effects.

Case Studies

Since the performance of DDC incorporates some constituency of latent and sensible cooling, the timing and sequence of weather patterns in addition to the current conditions play a role in determining how effective its cooling performance will be. Because of the dynamic behavior of DDC, comparisons to conventional cooling technologies have been based on the simulated performance of the cooling systems in response to actual historically recorded weather data. Three different case study locations were evaluated for an application at a medium-scale coal-fired power plant, 300 MW_e net electrical output, and are discussed further below.

Gillette, Wyoming

The first case study is for a location near Gillette, Wyoming. Gillette is centrally located in the Powder River Basin and was selected as a region of interest for dry cooling since a number of air-cooled plants are already located there. The case study calculations examined the sensitivity of cooling system cost and performance as a function of each system's design cooling capacity. Characteristic performance curves for each system were evaluated to identify trends among the systems and to identify designs suitable for direct comparison.

The first set of characterization data for Gillette is shown in Figure 16, which is a plot of the estimated lost revenue due to insufficient cooling as a function of the total annual cooling cost, which, as shown in Table 2, includes the annualized capital costs and other annual costs for each cooling system. Intuitively, this figure agrees with general expectations that within certain ranges, annual cooling costs can be modified (by changing the modeled cooling system configuration), with a resulting inverse effect on cooling performance. A smaller, less expensive cooling system can be chosen over more expensive systems with higher cooling capacities, but cooling performance and power production are expected to degrade.

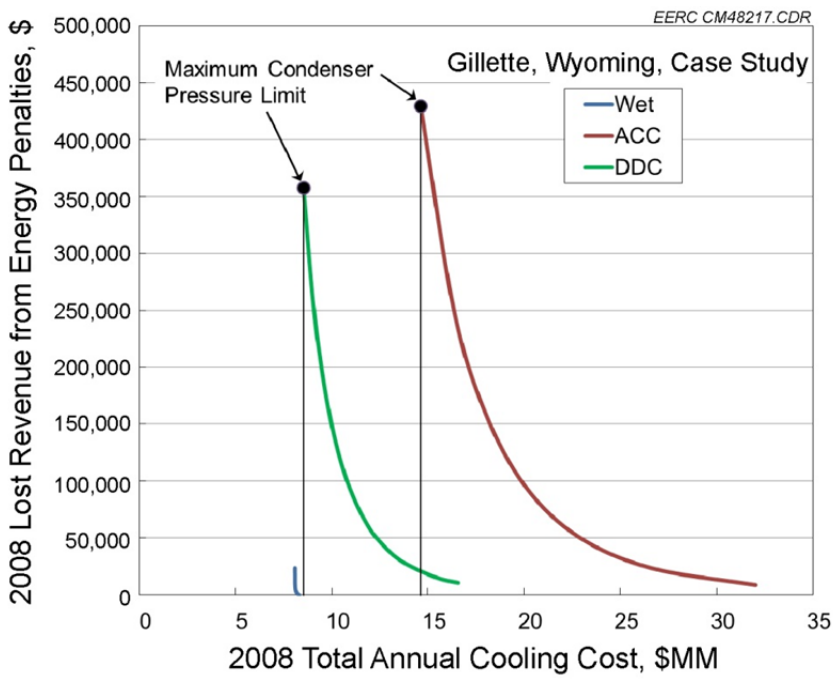

Figure 16. Calculated energy penalties as a function of annualized cooling costs for a 300-MW_e (net) coal plant located in Gillette, Wyoming.

Figure 16 indicates that without including other considerations like water availability, wet cooling is the clear best option; its use of highly effective evaporative heat dissipation allows a system to be specified that could provide sufficient cooling for all of the recorded weather variations without significant effect on annual cost. For dry technologies, either an ACC or DDC, a choice must be made regarding annual costs versus penalties to power generation. At the extreme high end of annual cooling cost, each technology could be designed to adequately cool the plant under the hottest weather of the simulation year. The annual costs for these scenarios are estimated to be approximately $17MM for DDC and $33MM for an ACC. However, the reduced sensitivity of penalties in this high-capital-cost region shows that these excessively large systems suffer from diminishing returns on the investment cost.

The opposite end of each dry system characteristic curve in Figure 16 is capped by the maximum condenser pressure limit, which was chosen to be 8 inches of Hg for these simulations. To reduce the total annual cooling cost below the limiting points in Figure 16 would mean that the condenser maximum pressure would be exceeded at some point during the simulation year.

The most simple criterion to evaluate design points along the dry cooling characteristic curves in Figure 16 is to weigh the reduction in annual lost power revenue against the corresponding increase in the projected annual cooling cost. Given the rough order of magnitude difference between the range of annual costs and the lost power penalties, no purely economic benefit is gained from oversizing the cooling system, even for the most sensitive regions near the limiting pressure design point. This conclusion implies that the most economical DDC and ACC

systems are those that fall just short of reaching the limiting operating pressure for the simulated weather conditions. In reality, there are other factors that guide cooling system selection since exact weather patterns will not be known in advance. Furthermore, a specific power plant application may be subject to overriding priorities (e.g., reliability to provide grid services) beyond minimizing cost, but those types of considerations are beyond the scope of this comparison.

In addition to increasing cooling costs, the parasitic power requirements for the cooling system's fans and pumps also generally increase with an increase in cooling capacity. Figure 17 highlights this trade-off specifically; it is a plot of the calculated megawatt-hours of lost generation versus the maximum rated power required to operate the cooling system. As the trends for the dry cooling systems show, there is an increase in the cooling system's full-load parasitic power rating as the cooling capacity increases. If the cooling system consumed the full-load quantity of power shown in Figure 17 over the entire year, the small number of megawatt-hours to be saved suggests that there is, again, little incentive to increase capacity of the dry cooling systems beyond the specified minimum.

For this comparative study, the lowest-annual-cost options for DDC and an ACC (i.e., the condenser pressure-limited designs in Figure 16) were selected to compare with a wet recirculating system that was designed to handle the cooling duty over the entire simulation year. Figure 18 is a plot of the calculated steam condenser temperatures for these cooling systems along with the ambient temperatures for Gillette in 2008. Also identified in the figure are the temperatures corresponding to the design condenser pressure (42°C) and the maximum allowable pressure (67°C); condenser temperatures that fell between these limits were subject to the lost energy penalties discussed in relation to Table 4. Summary economic parameters for each cooling system are presented in Table 7.

Observations from Table 7 show that the significant difference between the dry cooling options stems from the higher capital cost for the ACC relative to DDC. The estimated capital costs for an ACC are nearly double those for a DDC system having a similar energy production penalty. As the economic modeling assumptions in Table 2 show, many of the included annual operating and indirect costs are derived from the total capital estimate, which further increases the cost difference between the ACC and a DDC system.

Comparing wet cooling to the dry options in Table 7 indicates that a higher fraction of the total annual wet cooling costs are associated with the annual operating costs instead of those derived from capital or indirect charges. The majority of wet cooling's annual operating expense is related to water usage, i.e., $2.15MM out of $3.52MM. In an attempt to directly compare wet and dry cooling options, a breakeven cost of water has been calculated for the ACC and DDC systems. This value represents the cost of water that would be required to increase the annual wet cooling cost to equal the respective dry cooling system. The higher annual cost for an ACC results in a breakeven water cost that is almost four times the $2.00/kgal water cost assumed for the wet system, while the DDC breakeven cost appears more like a near-term possibility at $2.21/kgal.

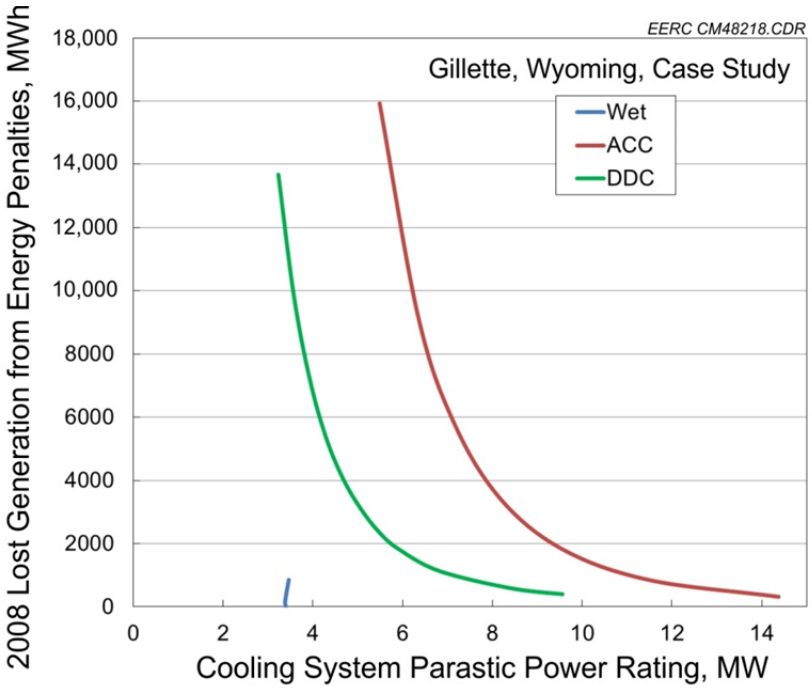

Figure 17. Calculated lost energy production as a function of the cooling system's parasitic power rating for a 300-MW_e (net) coal plant located in Gillette, Wyoming.

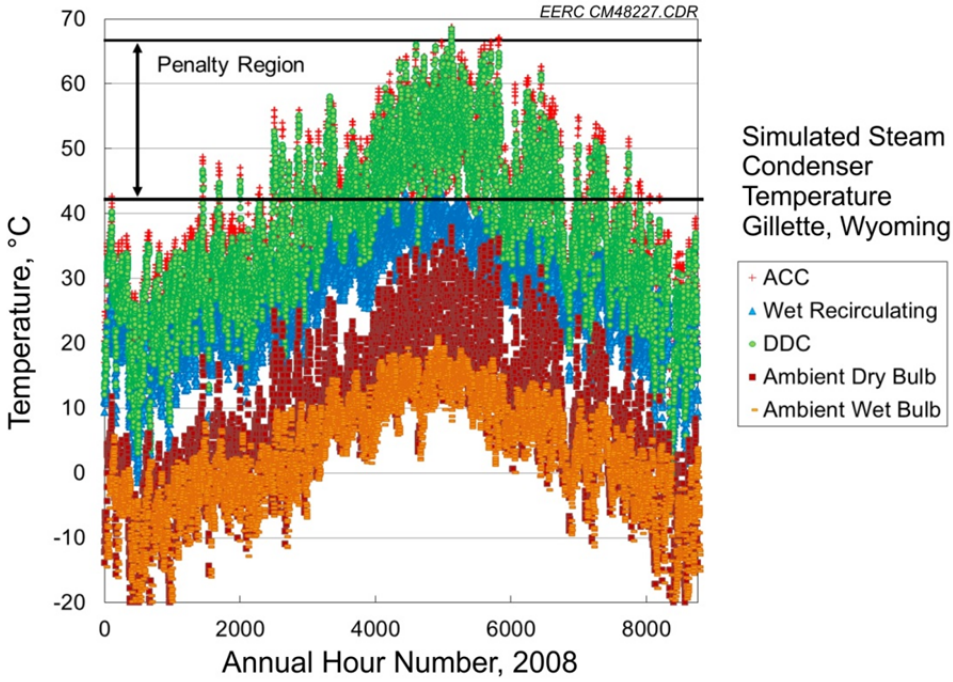

Figure 18. Simulated steam condensation temperatures for Gillette, Wyoming, in 2008.

Table 7. Summary of Economic Parameters for the Three Cooling Systems Considered for Gillette, Wyoming

	Wet Recirculating	ACC	DDC
System Total Capital Cost, $MM	25.1	61.3	33.0
Annualized Capital Cost, $MM	3.12	7.61	4.10
Annual Operating Cost, $MM	3.52	3.02	2.13
Annual Indirect Cost, $MM	1.67	4.00	2.31
Total Annual Cooling Cost, $MM	8.31	14.6	8.54
Energy Penalty for 2008, $	0	429,000	357,000
2008 Lost Generation, MWh	0	15,900	13,700
Full-Load Parasitic Power Rating, MW	3.4	5.5	3.2
Annual Water Cost at $2/kgal, $MM	2.15	NA[1]	NA
Breakeven Water Cost, $/kgal	NA	7.86	2.21

[1] Not applicable.

The energy penalties associated with the dry cooling systems in Table 7 are relatively small components of the total annual cooling cost, representing only 2% to 3% of the total annual cost. Significant increases to the assumed energy charge schedule would result in assigning more importance to minimizing these production penalties. However, the assumed charge schedule is believed to be representative of the current postrecession economy, with substantial electricity production from low-cost natural gas. Furthermore, the low penalty values suggest that optimization is possible by further reducing the cooling capacity and cost of the dry cooling systems if the concomitant penalty to energy production can be tolerated.

The annual plot of temperatures in Figure 18 is useful to convey the trade-off between limiting condenser temperature and the number of hours per year that would be affected. However, to gain insight into performance differences with DDC, a more detailed view like Figure 19 is needed where individual diurnal cycles can be distinguished. By comparing the steam condenser temperatures for the ACC and for DDC, the thermal inertia provided by DDC is clearly observed. While the ACC temperatures accurately track the ambient dry-bulb temperature, DDC temperatures rise more slowly and typically do not reach the same maximum values as the ACC. This incremental reduction in maximum steam condenser temperatures is the primary underlying reason why the energy penalty values in Table 7 are not identical for the two dry technologies.

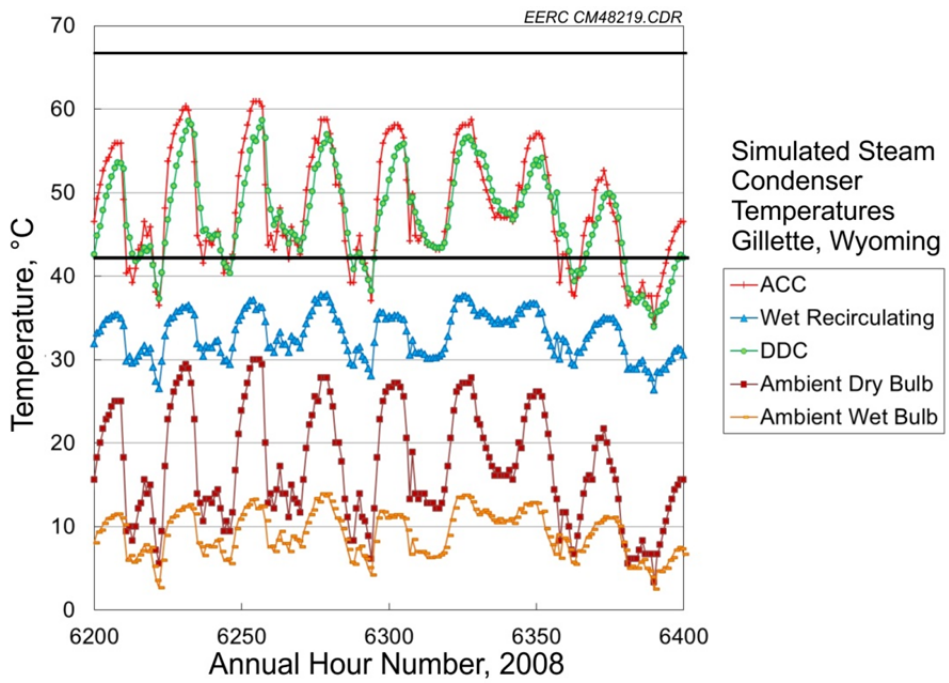

Figure 19. Detail of Gillette simulation results highlighting the thermal inertia provided by DDC.

Atlanta, Georgia

In contrast to Gillette, Atlanta presents a hotter and more consistently humid climate. Dry cooling is not regularly employed in this region, but recent droughts and pending U.S. Environmental Protection Agency action regarding once-through cooling have highlighted water availability as a vulnerability for power generation in the southeastern United States.

Energy penalty-versus-annual cooling cost curves are shown for each cooling option in Figure 20. The annual cooling cost ranges for Atlanta are similar to those estimated for Gillette, but the energy penalty charges are roughly three times as large. This is suggestive of the fact that the peak dry-bulb temperatures for Atlanta and Gillette are similar, but Atlanta experienced the higher temperatures more frequently. Despite the higher penalties, the most economically viable dry cooling design points remain those that just satisfy the maximum condenser pressure requirement. Also as with Gillette, wet cooling would be the preferred design choice given its good performance with low annual expense.

Figure 21 shows the relationship between lost energy generation and the cooling system's parasitic power ratings. Trends are similar to Gillette aside from the greater quantity of lost generation due to more frequent hot weather.

Comparative data for the zero-penalty wet cooling system and the condenser pressure-limited dry cooling systems are presented in Table 8. In comparison to the rankings calculated for Gillette, the different climate presented by Atlanta resulted in DDC being ranked as the least

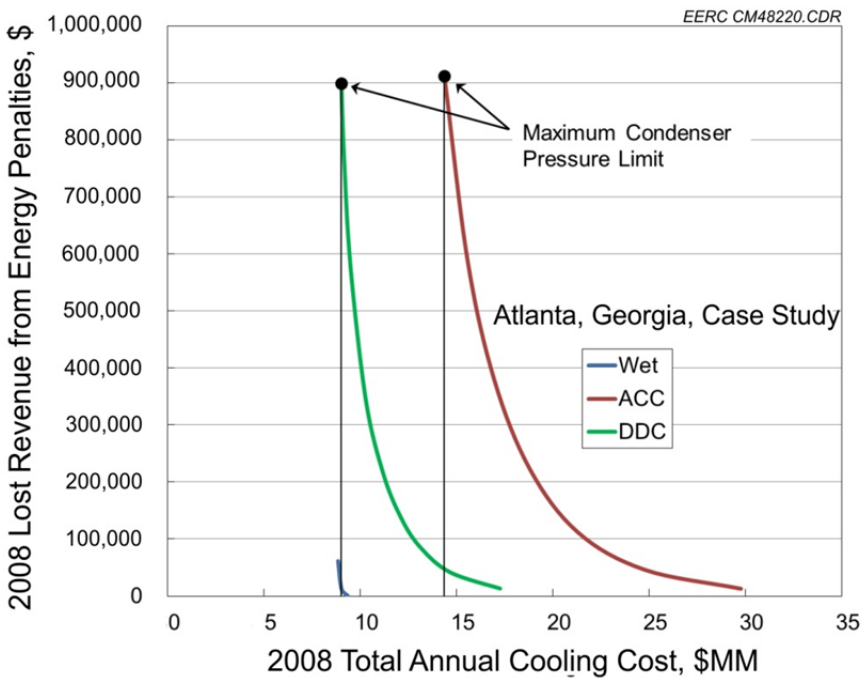

Figure 20. Calculated energy penalties as a function of annualized cooling costs for a 300-MW_e (net) coal plant located in Atlanta, Georgia.

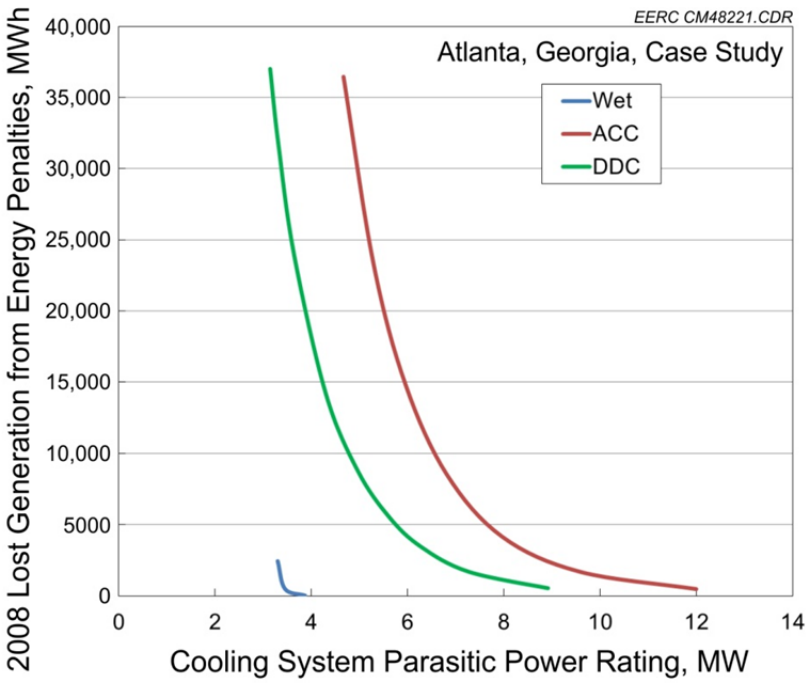

Figure 21. Calculated lost energy production as a function of the cooling system's parasitic power rating for a 300-MW_e (net) coal plant located in Atlanta, Georgia.

Table 8. Summary of Economic Parameters for the Three Cooling Systems Considered for Atlanta, Georgia

	Wet Recirculating	ACC	DDC
System Total Capital Cost, $MM	28.6	58.0	32.8
Annualized Capital Cost, $MM	3.55	7.20	4.07
Annual Operating Cost, $MM	3.92	3.37	2.65
Annual Indirect Cost, $MM	1.88	3.80	2.29
Total Annual Cooling Cost, $MM	9.34	14.4	9.01
Energy Penalty for 2008, $	0	912,000	899,000
2008 Lost Generation, MWh	0	36,500	37,000
Full Load Parasitic Power Rating, MW	3.9	4.7	3.1
Annual Water Cost at $2/kgal, $MM	2.36	NA	NA
Breakeven Water Cost, $/kgal	NA	6.25	1.72

expensive cooling option on an annual cost basis, which included the performance penalties incurred with DDC. Key differences in Atlanta's 2008 weather compared to that of Gillette were a lower maximum dry-bulb temperature with a higher average wet-bulb temperature. The lower maximum dry-bulb temperature in 2008 allowed for smaller dry cooling systems to be specified, but the higher wet bulb required a slightly larger wet cooling system. The estimated advantage of DDC is further shown by the breakeven cost of water for DDC, which was calculated to be less than the $2.00/kgal value assumed for the wet system.

Figure 22 is a plot of the calculated steam condensation temperatures for Atlanta, and compared to Gillette, most of the dry cooling temperatures do indeed fall within the penalty zone. Figure 23 is a detail plot of the Atlanta results, again showing the thermal dampening associated with DDC.

Phoenix, Arizona

The final case study location was selected to be Phoenix, Arizona, in order to represent an area of the United States that is expected to continue with population growth but one that also has a demonstrated lack of natural cooling water options. Key differences in Phoenix 2008 weather included higher summer temperature extremes and an overall lower relative humidity level during hot weather. The energy penalty-versus-annual cooling cost data are plotted in Figure 24. Overall maximum penalties are similar to those of Atlanta, which is indicative of a higher frequency of hot days; however, the key differences in Figure 24 are the higher values of

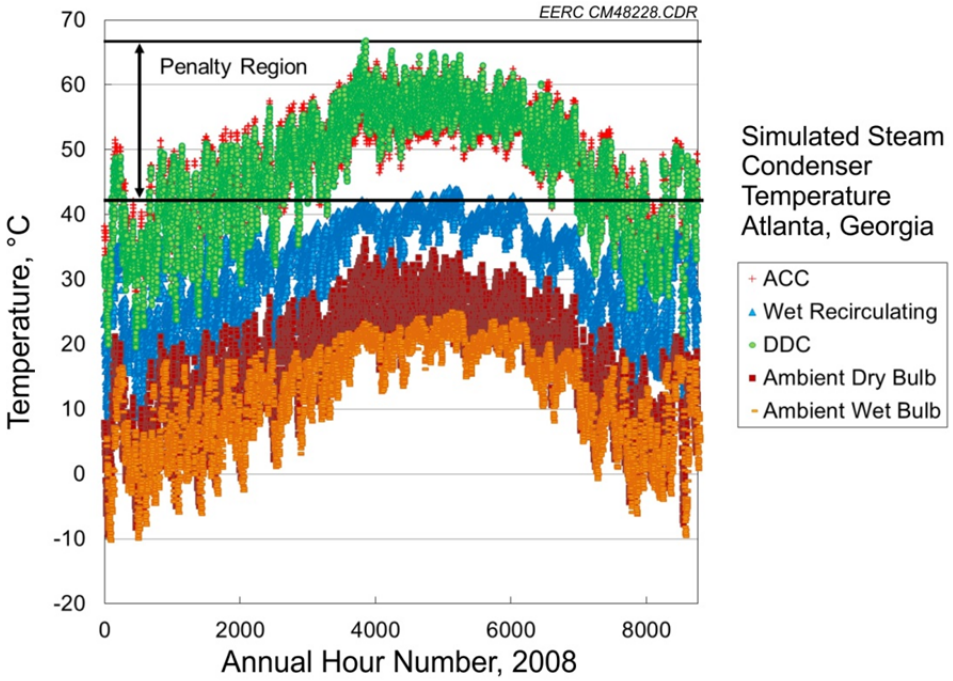

Figure 22. Simulated steam condensation temperatures for Atlanta, Georgia, in 2008.

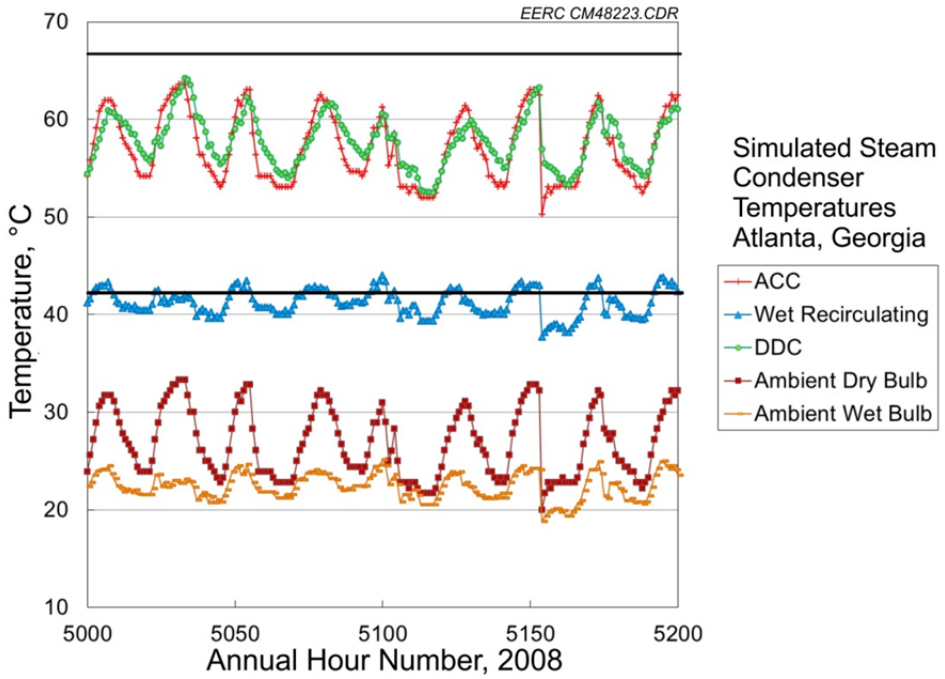

Figure 23. Detail of Atlanta simulation results highlighting the thermal inertia provided by DDC.

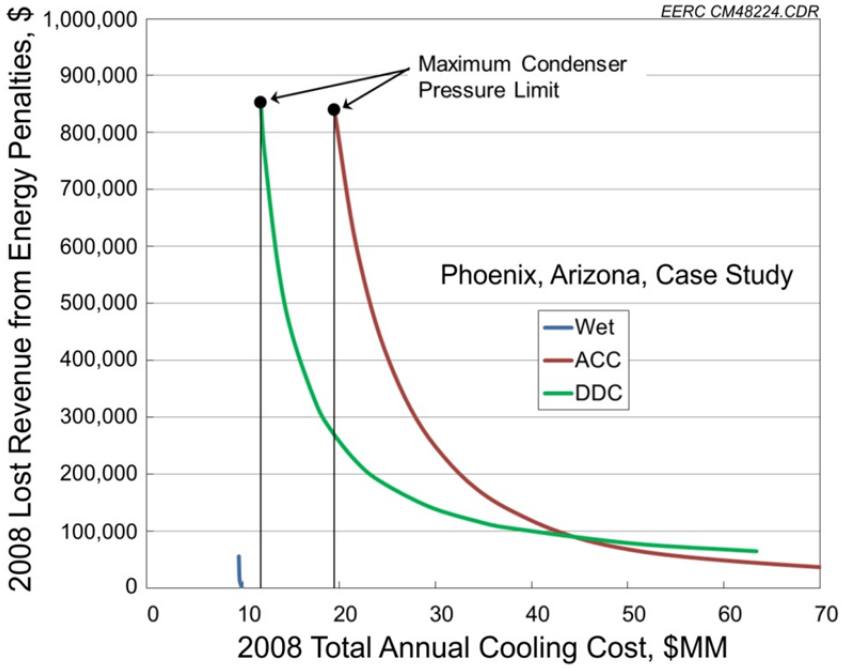

Figure 24. Calculated energy penalties as a function of annualized cooling costs for a 300-MW_e (net) coal plant located in Phoenix, Arizona.

total annual dry cooling system costs. The increased costs for dry cooling are primarily due to needing larger-capacity systems to cope with the high dry-bulb temperatures in Phoenix. The need for larger-capacity dry cooling systems is also reflected in the parasitic power rating data of Figure 25.

The crossover of the DDC characteristic curves with those of the ACC in Figures 24 and 25 is a consequence of the extra heat-transfer steps required for DDC compared to an ACC. As the 2008 dry-bulb data for Phoenix show in Figure 26, the extreme ambient temperature that the cooling systems are responding to is approximately 46°C, which necessitates an ITD of roughly 20°C in order to avoid the condenser pressure limit. According to the temperature differential data compiled in Table 6 for DDC, an ITD of 20°C is possible, but the system's performance sensitivity to capacity changes will begin to flatten as the limiting temperature differentials are approached.

Comparative data for the zero-penalty wet cooling system and the condenser pressure-limited dry cooling systems are presented in Table 9 for Phoenix. As noted for Figures 24 and 25, Table 9 also indicates higher capital costs and parasitic power requirements for the dry cooling options; annual dry cooling costs are estimated to be 30%–35% higher than for Atlanta. Wet cooling costs are almost identical to those in Atlanta, the key difference being the higher water consumption in Phoenix. The high cost for dry cooling in Phoenix results in the highest breakeven water cost for DDC, $3.35/kgal, which is not absurdly high given the arid nature of the southwestern United States. Conversely, the assumption of $2.00/kgal used for the wet system may actually be too low for this region.

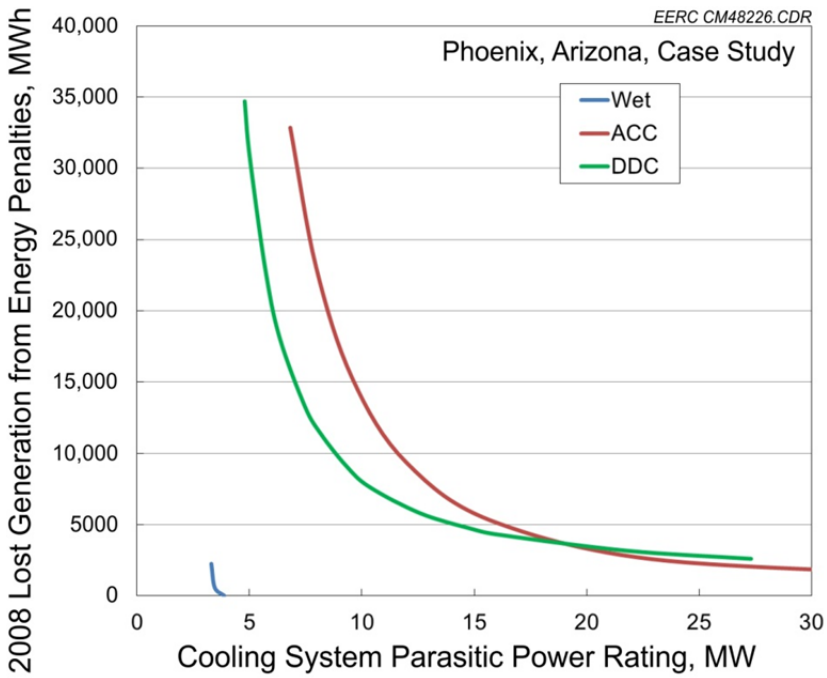

Figure 25. Calculated lost energy production as a function of the cooling system's parasitic power rating for a 300-MW_e (net) coal plant located in Phoenix, Arizona.

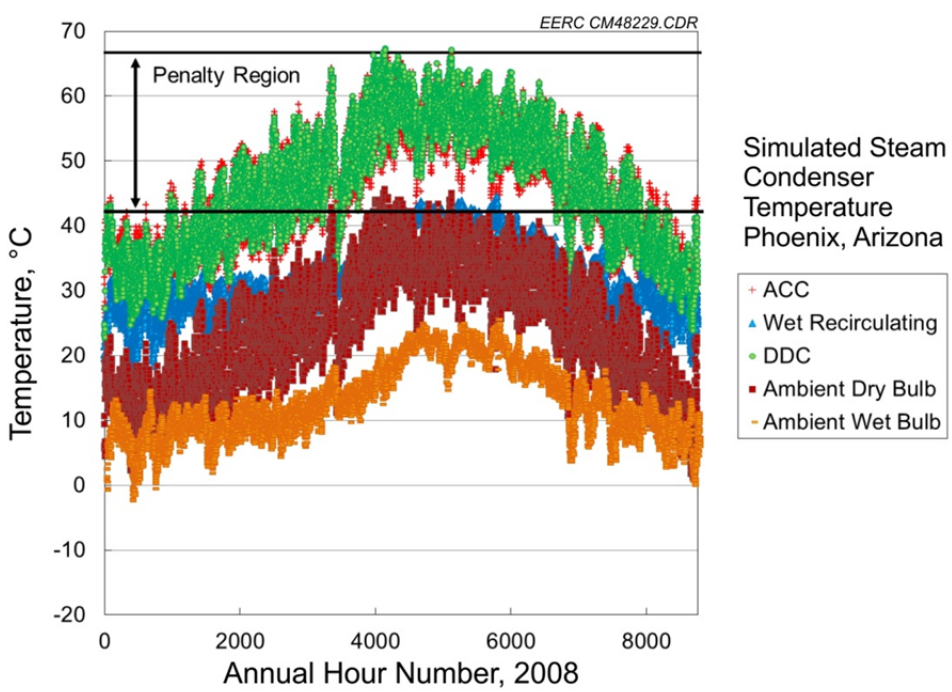

Figure 26. Simulated steam condensation temperatures for Phoenix, Arizona, in 2008.

Table 9. Summary of Economic Parameters for the Three Cooling Systems Considered for Phoenix, Arizona

	Wet Recirculating	ACC	DDC
System Total Capital Cost, $MM	28.6	81.3	45.2
Annualized Capital Cost, $MM	3.55	10.1	5.61
Annual Operating Cost, $MM	4.45	4.18	3.20
Annual Indirect Cost, $MM	1.88	5.20	3.04
Total Annual Cooling Cost, $MM	9.88	19.5	11.8
Energy Penalty for 2008, $	$0	841,000	853,000
2008 Lost Generation, MWh	0	32,800	34,700
Full Load Parasitic Power Rating, MW	3.9	6.8	4.8
Annual Water Cost at $2/kgal, $MM	2.90	NA	NA
Breakeven Water Cost, $/kgal	NA	8.63	3.35

Figure 27 shows a sample of the diurnal details for the Phoenix case study during the hottest weather of 2008. The interesting part of Figure 27 is the fact that even under this extreme of high temperature and low humidity, some amount of daily moisture exchange still occurs with the DDC system. While not large in magnitude, the effect reduces the day-to-night temperature swing by a finite amount and allows the DDC system to be sized for an average temperature roughly 1°C higher than the ACC. Figures 19 and 23 show this value to be fairly consistent with the other locations considered, despite the different ambient conditions encountered.

Hybrid DDC Operation

This investigation into DDC was to evaluate the technology as a dry cooling alternative; however, a discussion of DDC hybrid cooling potential is included since hybrid augmentation is under active consideration as a strategy to reduce the costs of dry cooling.

Hybrid operation in the context of DDC means to supplement its normally dry operation (i.e., no net water consumption) with evaporative cooling using water supplied from an external source. ACCs take advantage of water supplementation using a multistep process where water is evaporated to cool the incoming airstream which increases the ACC's ITD and, ultimately, enhances its sensible heat transfer into the airstream. With DDC, the implementation of hybrid augmentation is more direct and effective: liquid water is introduced into the desiccant solution where it absorbs heat directly in the water-to-desiccant heat exchanger and then evaporates readily in the air contactor to augment overall heat transfer.

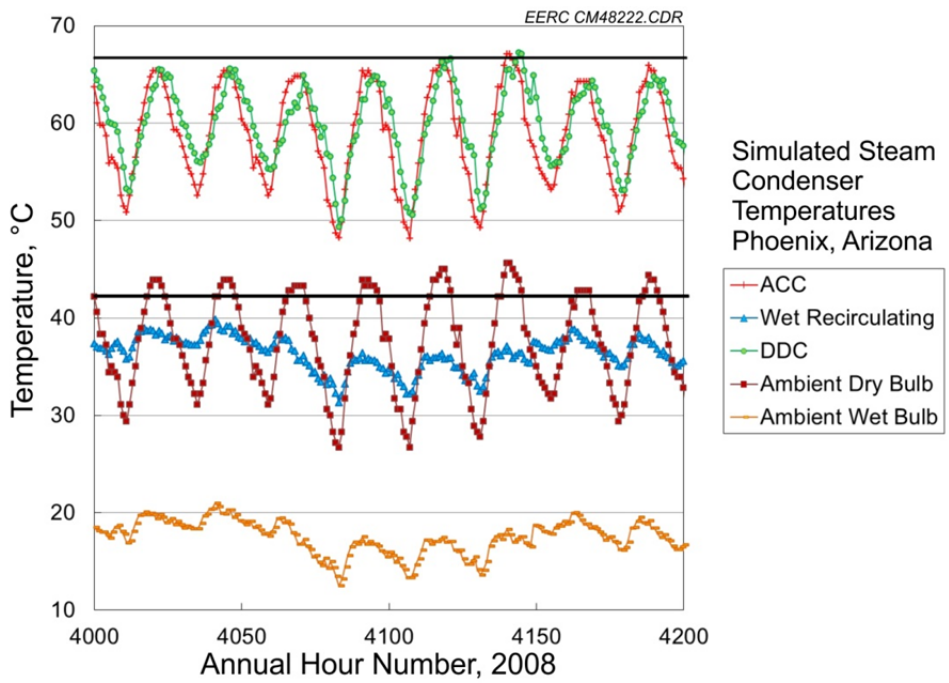

Figure 27. Detail of Phoenix simulation results highlighting the thermal inertia provided by DDC.

Hybrid augmentation is limited in its capacity to significantly lower the energy penalties encountered by dry cooling systems since steam condensation is limited by the hottest temperature in the cooling circuit, and for dry systems, this is governed by the ambient dry-bulb temperature. To reduce energy production penalties by lowering the steam condensation temperature would require a significant shift of the plant's cooling load to the hybrid cooling mechanism. The limit is reached when it is desired to lower the steam condensation pressure down to the ambient dry-bulb temperature, at which point the hybrid cooling component would have to handle virtually all of the plant's cooling load.

Hybrid augmentation is, however, a way to optimize the required size of a dry cooling system by providing a low-cost method (if sufficient water is available) to absorb periodic ambient temperature extremes. For locations like Gillette that have an overall cool climate punctuated with high-temperature excursions, this approach could have a significant impact. It is estimated that redesigning the Gillette DDC system to handle all but the hottest 1.5% of hours in 2008 would result in a 16% reduction in the estimated capital costs and would lower the annual cooling cost for DDC to approximately $7.8MM. Such a change would make DDC's annual cost more competitive than wet recirculating cooling and would lower its breakeven cost of water to approximately $1.53/kgal.

On the other hand, consistently warmer climates like Atlanta and Phoenix would benefit less.

The key challenge in adapting DDC for hybrid augmentation is to mitigate the impacts to long-term desiccant solution chemistry. DDC is intended to operate as a closed-loop system that does not require conventional makeup and blowdown streams. Hybrid water augmentation would introduce a potentially uncontrolled source of dissolved solids that may lead to adverse effects on cooling system performance over time. Techniques to manage dissolved solids content for hybrid DDC augmentation are currently under evaluation.

CONCLUSIONS AND RECOMMENDATIONS

The activities completed under this project have supported a balanced approach of modeling grounded in applied experimentation; together, they have generated the necessary information to substantiate DDC modeling efforts and outline the potential for this technology to cool full-scale power plants. The analysis shows that DDC can be a lower-cost dry cooling alternative to an ACC and can even be competitive with conventional wet recirculating cooling under certain circumstances. Specific conclusions regarding DDC include the following:

- Case study calculations indicate that DDC consistently maintained a lower annual cooling system cost compared to an ACC. Annual cooling costs for DDC averaged 60% of those of an ACC for the evaluated cases. Parasitic power requirements for DDC were also estimated to be lower, averaging 65% of those for a comparable ACC.

- Compared to wet recirculating cooling, the annual costs for DDC were within ±10% of the comparable wet system, including the energy penalties associated with lost power production. The breakeven cost of water for DDC ranged from a low of $1.72/kgal for Atlanta to a high of $3.35/kgal for Phoenix using the stated assumptions.

- Experimental testing supports the hypothesis that DDC can be an environmentally benign cooling option. The key environmental concern, carryover of desiccant, appears to be manageable with proper design and operation of the cooling system. Measured drift rates under these conditions were determined to be less than 0.00006% of the circulating working fluid rate. However, it is desirable to demonstrate an even lower limit for drift in future testing to avoid potential particulate emission limits that are currently imposed on conventional wet cooling towers.

This project has also highlighted the key technological steps that must be taken in order to transfer DDC into the marketplace. In terms of performance, the remaining unproven component is the water-to-desiccant heat exchanger. While the commercially available version tested under this project worked, it was determined to be too expensive on a per unit heat-transfer basis to be feasible for full-scale implementation. Development of a lower-cost liquid heat exchanger that has a cost per unit of heat transfer comparable to existing utility-scale heat exchangers is needed. Fortunately, this need has materialized into a separately funded project that is currently under way.

Another aspect that will need to be addressed for DDC is the ongoing and long-term impacts of direct-contact air exposure. This project has shown that the fundamental aspects of

heat and mass transfer associated with DDC are relatively well understood, but these process are still somewhat unique in industry given the proposed scale and duration of direct air–desiccant contact that is needed to provide cooling for a power plant. Since the desiccant portion of the solution is nonvolatile and drift losses appear negligible, it has been assumed that the initial charge of working fluid should last for the life of the cooling system, with minimal makeup requirements. However, the accumulated effects of long-term exposure to atmospheric air and its contaminants are currently unknown. To address these issues and to offer an extended demonstration of DDC technology, a next-stage project should include the opportunity for outdoor ambient testing of a pilot DDC cooling cell.

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