Energy Penalty Analysis of Possible Cooling Water Intake Structure Requirements on Existing Coal-Fired Power Plants

U.S. Department of Energy

Office of Fossil Energy

National Energy Technology Laboratory

Argonne National Laboratory

October 2002

1.0 Executive Summary

Section 316(b) of the Clean Water Act requires that cooling water intake structures must reflect the best technology available for minimizing adverse environmental impact. Many existing power plants in the United States utilize once-through cooling systems to condense steam. Once-through systems withdraw large volumes (often hundreds of millions of gallons per day) of water from surface water bodies. As the water is withdrawn, fish and other aquatic organisms can be trapped against the screens or other parts of the intake structure (impingement) or if small enough, can pass through the intake structure and be transported through the cooling system to the condenser (entrainment). Both of these processes can injure or kill the organisms.

EPA adopted 316(b) regulations for new facilities (Phase I) on December 18, 2001. Under the final rule, most new facilities could be expected to install recirculating cooling systems, primarily wet cooling towers. The EPA Administrator signed proposed 316(b) regulations for existing facilities (Phase II) on February 28, 2002. The lead option in this proposal would allow most existing facilities to achieve compliance without requiring them to convert once-through cooling systems to recirculating systems. However, one of the alternate options being proposed would require recirculating cooling in selected plants.

EPA is considering various options to determine best technology available. Among the options under consideration are wet-cooling towers and dry-cooling towers. Both types of towers are considered to be part of recirculating cooling systems, in which the cooling water is continuously recycled from the condenser, where it absorbs heat by cooling and condensing steam, to the tower, where it rejects heat to the atmosphere before returning to the condenser. Some water is lost to evaporation (wet tower only) and other water is removed from the recirculating system as a blow down stream to control the building up of suspended and dissolved solids. Makeup water is withdrawn, usually from surface water bodies, to replace the lost water. The volume of makeup water is many times smaller than the volume needed to operate a once-through system.

Although neither the final new facility rule nor the proposed existing facility rule require dry cooling towers as the national best technology available, the environmental community and several States have supported the use of dry-cooling technology as the appropriate technology for addressing adverse environmental impacts. It is possible that the requirements included in the new facility rule and the ongoing push for dry cooling systems by some stakeholders may have a role in shaping the rule for existing facilities. The temperature of the cooling water entering the condenser affects the performance of the turbine -- the cooler the temperature, the better the performance. This is because the cooling water temperature affects the level of vacuum at the discharge of the steam turbine. As cooling water temperatures decrease, a higher vacuum can be produced and additional energy can be extracted. On an annual average, once-through cooling water has a lower temperature than recirculated water from a cooling tower. By switching a once-through cooling system to a cooling tower, less energy can be generated by the power plant from the same amount of fuel. This reduction in energy output is known as the energy penalty. If a switch away from once-through cooling is broadly implemented through a final 316(b) rule or other regulatory initiatives, the energy penalty could result in adverse effects on energy supplies.

Therefore, in accordance with the recommendations of the Report of the National Energy Policy Development Group (better known as the May 2001 National Energy Policy), the U.S. Department of Energy (DOE), through its Office of Fossil Energy, National Energy Technology Laboratory (NETL), and Argonne National Laboratory (ANL), has studied the energy penalty resulting from converting plants with once-through cooling to wet towers or indirect-dry towers. Five locations – Delaware River Basin (Philadelphia), Michigan/Great Lakes (Detroit), Ohio River Valley (Indianapolis), South (Atlanta), and Southwest (Yuma) – were modeled using an ASPEN simulator model. The model evaluated the performance and energy penalty for hypothetical 400-MW coal-fired plants that were retrofitted from using once-through cooling systems to wet- and dry-recirculating systems. The modeling was initially done to simulate the hottest time of the year using temperature input values that are exceeded only 1 percent of the time between June through September at each modeled location. These are the same temperature inputs commonly used by cooling tower designers to ensure that towers perform properly under most climatic conditions. The high temperature inputs correspond to the time of year when the highest power demands are observed and the needs for generating capacity are most critical due to the very high cost of buying replacement power on the spot market. Later, modeling was completed to estimate the monthly energy penalties, which were arithmetically averaged to generate an estimate of annual average energy penalty.

The results of the one-percent-high temperature modeling show that conversion to a wet tower could cause energy penalties ranging from 2.4 percent to 4.0 percent. This means that the plant will produce 2.4 percent to 4.0 percent less electricity with a wet tower than it did with a once-through system while burning the same amount of coal. That lost electricity could be made up at this plant or at some other existing or new plant by burning additional fuel. These peak-summer penalties are somewhat higher than those estimated by EPA in the technical documentation published with its April 9, 2002 proposal for existing facilities. DOE believes that EPA did not include all the relevant costs and made some inappropriate assumptions; these are described at the end of Chapter 4. When more appropriate costs and assumptions are considered, EPA estimates compare favorably with those in this report.

Conversion to an indirect-dry tower, where possible, could cause energy penalties ranging from about 8.9 percent to 12.14 percent using 20 degrees F for the approach (the difference between the inlet air dry-bulb temperature and the desired cold water temperature), and 12.7 percent to almost 16 percent using an approach of 40 degrees F. The industry norm for indirect dry towers – a 40-degree approach -- was evaluated initially, but the resulting pressures for the steam turbines were found to result in unacceptable operating conditions during the one-percent highest temperature times of the year. The mostly likely way that a company could operate a retrofitted indirect-dry tower at a 40-degree approach would be to reduce the power output from the plant (load shedding) during the hottest times of the year – just when the power demand is the greatest. This power output reduction imparts an immediate energy penalty. On completion of the analysis it was determined that even if load shedding was attempted on all the 40-degree approach cases it would still be technically infeasible to operate the turbines safely during the summer months. To provide more information on dry tower energy penalties, a more conservative approach of 20 degrees was subsequently modeled.

The results of the annual energy penalty modeling show that conversion to a wet tower could cause energy penalties ranging from 0.8 percent to 1.5 percent. Conversion to an indirect-dry tower could cause energy penalties ranging from about 4.2 percent to 5.2 percent using 20 degrees F for the approach, and 7.9 percent to almost 8.8 percent using an approach of 40 degrees F.

A review of the "Environmental Directory of US Powerplants" (EEI 1996) indicated that in 1996, there were 258,906 MW of electric generating capacity in the United States that consisted of steam electric power plants employing once-through cooling. The one-percent highest temperature analysis modeled plants in just five locations and under very warm temperature conditions, but the modeled facilities are believed to be representative of the climatic conditions found throughout those portions of the country where once-through cooling is prevalent. It is quite possible that much of the Nation could experience very high temperatures at the same time (e.g., week of August 6, 2001), leading to results even more extreme than those calculated here.

Tables ES-1 and ES-2 demonstrate the effects on electric generating capacity during the one-percent highest temperature conditions if 10, 25, 50, or 100 percent of the existing once-through cooled power plants in the United States were required to convert to either wet or indirect-dry cooling towers. The example of a requirement for 100 percent of the plants to retrofit to either wet or dry towers is hypothetical since it would be technically infeasible to do either. The energy, time, and expense required to make up for these losses is significant and would not necessarily require building new plants. But for example in the "average" case, 19 additional 400-MW plants might have to be built to replace the generating capacity lost by replacing once-through cooling with wet cooling towers in 100 percent of existing steam plants. If some of those affected plants were required to retrofit an indirect-dry tower, the energy penalty impacts would be over three times higher. For example, the "average" case might require 66 new 400-MW plants to be built to replace the generating capacity lost by replacing once-through cooling with indirect dry cooling towers with a 20-degree air-side approach in 100 percent of existing steam plants. This example of new plants needed if 100 percent of existing plants were required to retrofit to dry towers is far too low since after thoroughly completing this analysis it has been determined that it would be impossible for most existing plants to be retrofitted to dry towers at many locations and therefore there would be a need for closures and far more new power plants than provided in the simple example above.

These new power plants may be needed to replace the energy lost as a result of the conversion from once-through to recirculating cooling, and do not reflect the need to build additional new generating capacity to meet the nation's growing demands for electricity. The U.S. Department of Energy's Annual Energy Outlook states that anticipated growth in electricity sales between

2000 and 2020 is about 1.8 percent per year (EIA 2001a). Alternatively, some of the existing plants that might have to retrofit to either wet or indirect-dry cooling systems may be able to just burn more fuel to replace the electricity lost due to the cooling system conversion. Either way, additional fuel will be burned and other adverse environmental impacts will be created such as increased emissions, land use, and noise pollution.

To more closely evaluate the impact of increased air emissions from burning additional fuel, several additional analyses were performed. Estimates of incremental air emissions were made using the average annual energy penalty results at the Delaware River Basin site and the South site. The results show that when once-through cooled plants are converted to wet cooling towers, the incremental air emissions are not large on a percentage basis (generally less than one percent), but the absolute increases in pounds or tons of key air pollutants (SO_2, NO_x, PM, mercury, and CO_2) are large nonetheless. If once-through cooled plants are converted to indirect-dry towers, however, the incremental air emissions can be significant. For dry towers with a 20-degree approach, the percentage increase in air emissions can exceed 4 percent depending on how the power company makes up the lost energy. For dry towers with a 40-degree approach, the percentage increase in air emissions can approach 8 percent and the number of additional pounds or tons is quite large.

Incremental air emissions are of greatest concern in nonattainment areas. Nonattainment areas are identified for "criteria pollutants" established under the 1970 Amendments to the Clean Air Act that do not meet standards set by EPA. The term "criteria pollutants" derives from the requirement that EPA must describe the characteristics and potential health and welfare effects of these pollutants. It is on the basis of these criteria that standards are set or revised. Although a national impact analysis is not performed in the present study, a general conclusion is that incremental air emissions are counterproductive to achieving standards set by EPA for air quality. There are a number of nonattainment locations throughout the United States where incremental air emissions could occur from an energy penalty associated with a requirement to add a cooling tower to existing power plants.

One important finding of this report is that neither indirect-dry nor direct-dry towers are viable as a retrofit technology at most U.S. locations under the one-percent-highest temperature conditions. As previously noted, many of the model runs evaluating conversion to indirect-dry towers resulted in calculated turbine pressures that exceeded the upper limit for safe turbine operation. This was true of all of the model runs made using the 40-degree approach assumption and for one quarter of the runs made at 20 degrees. The point should be made that the practice of load shedding, a method of reducing the steam load through the turbine, thereby reducing the condenser heat duty by a proportional amount, would not effectively lower the turbine backpressure enough for safe operation under the runs modeled with a 40-degree approach assumption. Even for those 20-degree approach cases in which the turbine pressures were below the upper safe limit, an indirect-dry tower would occupy huge amounts of space, which may not be available in an existing plant originally built with once through cooling. The results of sizing calculations to determine the required footprint area for a representative case of retrofitting to indirect dry towers at a 20-degree approach are discussed in section 10.2. Direct-dry towers are not practical either. In an existing plant, there simply is no room for the large-diameter ductwork required to conduct -atmospheric steam from the turbine exhaust hood to a direct-dry cooling tower.

Dry towers have been used as part of newly constructed cooling systems. If the entire power generating system (boiler, turbine, condenser, and cooling) is designed with dry cooling in mind, dry cooling does have applications. For retrofitted dry towers, the issues of large footprint and high energy penalty are important.

Table ES-1 - Wet Cooling Tower Energy Penalties and Impact at One Percent Highest Temperature Conditions

Once-through Cooling Systems Required to Retrofit (%)	Wet, Recirculating Cooling Tower Retrofit Penalty (%)					
	Low Value* 2.4		Average Value* 3.0		High Value* 4.0	
	Energy Penalty (MW)	% of Total Steam Electric Capacity	Energy Penalty (MW)	% of Total Steam Electric Capacity	Energy Penalty (MW)	% of Total Steam Electric Capacity
10	621	0.24	777	0.30	1,036	0.40
25	1,553	0.60	1,942	0.75	2,589	1.00
50	3,106	1.20	3,883	1.50	5,178	2.00
100	6,212	2.40	7,766	3.00	10,356	4.00

* The energy penalties calculated for the Southwest site are not used here because once-through cooled plants are not likely to be found in that region.

Table ES-2 - Indirect-Dry Cooling Tower Energy Penalties and Impact at One Percent Highest Temperature Conditions

Once-through Cooling Systems Required to Retrofit (%)	Indirect-Dry (20° F Approach) Cooling Tower Retrofit Penalty (%)					
	Low Value* 8.8		Average Value* 10.2		High Value* 13.1	
	Energy Penalty (MW)	% of Total Steam Electric Capacity	Energy Penalty (MW)	% of Total Steam Electric Capacity	Energy Penalty (MW)	% of Total Steam Electric Capacity
10	2,278	0.88	2,641	1.02	3,392	1.31
25	5,696	2.20	6,602	2.55	8,479	3.28
50	11,392	4.4	13,204	5.10	16,958	6.55
100	22,784	8.80	26,408	10.20	33,917	13.10

* The energy penalties calculated for the Southwest site are not used here because once-through cooled plants are not likely to be found in that region.

2.0 Glossary

Many of the technical terms used in the report are defined here. Some of the definitions are taken from or adapted from Burns and Micheletti (2000).

Acid Rain Program - In 1990, Congress established the Acid Rain Program under Title IV of the Clean Air Act Amendments. The principal goal of the program is to achieve reductions of 10 million tons of sulfur dioxide (SO2) and 2 million tons of nitrogen oxides (NOx), the primary components of acid rain.

Approach - The minimum difference between fluid stream temperatures in a heat exchanger. The approach for a given heat exchanger is typically chosen as a design parameter that reflects how close the operation of that heat exchanger comes to the thermodynamic limits on the amount of heat that can be transferred between the two fluid streams. In designing any given heat exchanger, as the surface area is increased, a lower approach can be achieved, and the unit comes closer to transferring the highest amount of heat theoretically possible. In the limiting case of a zero approach, the maximum theoretical amount of heat possible is transferred, but it would require an infinitely large heat exchanger surface area to do this. Consequently, the selection of the approach temperature for designing a heat exchanger represents an engineering tradeoff of thermodynamic efficiency versus capital cost, size, and weight of the exchanger in question.

In the case of an evaporative wet cooling tower, the approach is the difference between the anticipated inlet air wet-bulb temperature and the discharge cold water temperature. In the case of an indirect dry cooling tower, it is the difference between the anticipated inlet air dry-bulb temperature and the discharge cold water temperature. In a condenser it is the difference between the condensing steam temperature and the temperature of the cooling water exiting the condenser.

Combined-Cycle Plant - A power plant that utilizes a highly efficient (50 to 60 percent, lower heating value basis) two-step process for production of electricity involving one or more gas turbine generator sets and a steam bottoming cycle. The hot exhaust gas from the turbine generator set(s), which would otherwise be exhausted to the atmosphere, is passed through a heat recovery steam generator to make superheated steam. The steam is then used to drive a separate steam turbine and its generator, which produces additional electricity.

Condenser - A device that cools and condenses steam discharging from a steam turbine. The most commonly used type of condenser in power plants is a shell-and-tube heat exchanger in which cooling water flows in the tubes and the turbine discharge steam enters the shell.

Deaerator - A process unit used to remove dissolved gases from a liquid stream. For a steam cycle the removal of dissolved gases (e.g. oxygen, carbon dioxide, ammonia, or hydrogen sulfide) from the boiler feed water is desirable to avoid problems associated with corrosion in the plant equipment. The steam plant uses a type of deaerator that is based on the dissolved gases becoming less soluble as the temperature of the water is increased by direct contact with a bleed steam stream.

Direct Dry Cooling Tower - A finned tube heat exchanger is used as a direct air-cooled condenser. The steam is condensed inside finned tubes and the heat of condensation is transferred directly to the surrounding atmosphere by using large diameter fans to blow ambient air over the tubes.

Dry Bulb Temperature - The temperature of ambient air as measured by a standard thermometer or other similar device.

Energy Penalty - The loss of electricity generating capacity incurred when a cooling system is unable to perform at design efficiency. The energy penalty is associated with insufficient cooling of the turbine exhaust steam and usually is manifested by an increase in steam turbine back pressure. This study expresses the penalty as "the percentage of plant output," or phrased differently, "the percentage of additional energy that would have to be used to generate the same amount of electricity." In this study, the energy penalty also includes additional power needed for pumps and fans in cooling tower systems.

Entrainment - The incorporation of eggs, larval stages, small fish and other aquatic organisms into the surface water intake stream that is used to supply a cooling water system. These organisms are small enough to pass through the screens and other barriers used in the intake structure.

Evaporative Heat Transfer - A form of heat transfer in which the evaporation of a liquid (e.g. water) by releasing latent heat of evaporation lowers the temperature of the remaining liquid. In a wet cooling tower, this released latent heat is absorbed by the flow-through air. Sensible heat transfer occurs simultaneously. In a wet cooling tower, evaporative heat transfer accounts for approximately 65 to 85 percent of the water cooling, with the remaining portion due to sensible heat transfer.

Helper Tower - In power plants (or industrial processes) where discharge water, either from once-through condensers or in blowdown streams exceeds permitted thermal regulations, additional cooling is required. This is often accomplished in a cooling tower denoted as a “helper”. This tower would be substantially smaller and less expensive then the cooling towers used in closed-loop cooling systems. Thus, the design is site specific.

Impingement - Entrapment of aquatic organisms on an intake structure during cooling water withdrawal from surface water bodies.

Inch of Mercury – The pressure exerted by a 1-inch high column of liquid mercury at standard conditions, as read from a mercury manometer. One inch of mercury is equivalent to a pressure of 0.49 pounds per square inch.

Indirect-Dry Cooling Tower - A cooling tower in which a hot liquid such as condenser coolant rejects heat to the atmosphere without the evaporation of water. Heat from the water is transferred to the surrounding atmosphere in finned-tubes, which are cooled by large diameter fans blowing air over the finned surfaces. The cooled water is then returned to the condenser to repeat the cycle.

Gross Power - The total amount of electricity produced at the generator terminals.

Net Power - The gross power output of a plant minus the power used internally by the plant’s auxiliary systems (e.g., pumps, fans, lighting). This is the amount of power available to distribute to external users.

NO_x SIP Call - In October, 1998, EPA finalized the "Finding of Significant Contribution and Rulemaking for Certain States in the Ozone Transport Assessment Group Region for Purposes of Reducing Regional Transport of Ozone." (Commonly called the NO_x SIP Call.) The NO_x SIP call was designed to ensure that NO_x emissions reductions are achieved to mitigate the regional transport of ozone across State boundaries in the eastern half of the United States.

Once-Through Cooling System - A cooling system in which water (generally surface water) is used as the condenser coolant and is then discharged after just a single pass through the condenser.

Overfire Air – A method of staging the combustion air that reduces nitrous oxide from coal combustion. Overfire air is one of the lower cost ways to achieve significant nitrous oxide emissions reductions and is almost always implemented in combination with the installation of low-NOx burners. The combination is the most cost-effective NOx reduction modifications for existing units.

Range - The temperature difference realized in a particular flow stream of fluid in a heat exchanger. One example would be the temperature difference between the cold water entering and the hot water leaving a condenser.

Recirculating Cooling System - A cooling system in which the condenser coolant water is not directly discharged but is recirculated to a separate structure for cooling and then is returned to the condenser. Most recirculating cooling systems employ cooling towers, which can be either wet or dry towers. Operators of wet-tower recirculating systems must extract or "blow down" a portion of the recirculating water on a regular basis to avoid undesirable build up of suspended and dissolved solids. Makeup water is added to replace the water lost to blow down, evaporation, and entrained droplets of mist discharged from wet cooling towers, commonly called "drift loss."

Selective Catalytic Reduction - A chemical treatment process used to reduce the amount of NOx emissions in a fossil fuel fired power plant's flue gas exhaust. In the SCR process, ammonia or a compound of ammonia is injected into the flue gas stream, passing over a catalyst. The resultant chemical reaction between the flue gas and the ammonia yields free nitrogen and water vapor. Typically, NOx emission reductions of 80-90% are achieved.

Sensible Heat Transfer - A form of heat transfer in which a warm fluid is cooled by contact with a cooler fluid. In a dry cooling tower, this is the only method of heat transfer. In a wet cooling tower, the water is cooled not only by sensible heat transfer but also by evaporative heat transfer that occurs simultaneously.

Terminal Temperature Difference or TTD - The difference between the turbine exhaust steam temperature and the hot cooling water temperature.

Turbine Back Pressure - The pressure at the discharge of a turbo expander. In the case of a steam turbine this would be the operating pressure on the steam side of the condenser. Departures from design turbine back pressure have a major effect on electric generating efficiency. An operating back pressure greater than design means lower power from the steam turbine and thus lower generating efficiency.

Wet Bulb Temperature - The temperature of ambient air as measured by a thermometer in which the bulb is kept moistened and ventilated. The resulting measurement equates to the dynamic equilibrium temperature attained by a water surface when the rate of heat transfer to the surface by convection equals the rate of mass transfer away from the surface by evaporation.

The wet bulb temperature is the lowest temperature at which evaporation can occur for specific ambient conditions (dry bulb temperature and relative humidity).

Wet Cooling Tower - A cooling tower in which water rejects heat to the atmosphere through evaporation and sensible heat transfer to the ambient air flowing through the tower. The flow of ambient air through the tower is maintained by fans (mechanical draft) or through buoyancy effects (natural draft).

3.0 Introduction

3.1 Legal Background for Cooling Water Intake Structure Requirements

Section 316(b) of the Clean Water Act, enacted by Congress in 1972, addresses withdrawal of cooling water from surface water bodies, as follows:

> Any standard established pursuant to section 301 or section 306 of this Act and applicable to a point source shall require that the location, design, construction, and capacity of cooling water intake structures reflect the best technology available for minimizing adverse environmental impact.

In 1976, the U.S. Environmental Protection Agency (EPA) promulgated final §316(b) regulations (April 26, 1976; 41 FR 17387). However, those regulations were successfully challenged by a group of 58 utilities [*Appalachian Power Co. v. Train*, 10 ERC 1965 (4th Cir. 1977)]. In 1979, EPA formally withdrew its §316(b) regulations (June 1979; 44 FR 32956). As a consequence of the vacuum created by the absence of Federal regulations, many States adopted their own cooling water intake regulations to implement the §316(b) requirements. The broad statutory language facilitated widely differing interpretations by the States. Some adopted comprehensive programs, others imposed less rigorous requirements, and still others never developed formal regulations.

In the mid-1990s, a coalition of environmental groups, headed by the Hudson Riverkeeper, filed suit against EPA over failure to repromulgate §316(b) regulations [*Cronin, et al. v. Reilly*, 93 Civ. 0314 (AGS)]. On October 10, 1995, the U.S. District Court, Southern District of New York, entered a Consent Decree between the parties, directing EPA to regulate cooling water intake structures within 7 years. Under the Consent Decree, EPA agreed to propose regulations by June 1999 and promulgate a final rule by 2001. The Consent Decree was modified on November 21, 2000 to: a) finalize new facility regulations by November 9, 2001; b) propose existing source large utility and non-utility power producer regulations by February 28, 2002 and issue final regulations by August 28, 2003; and c) propose regulations by June 15, 2003 and issue final regulations by December 15, 2004 for other existing facilities not covered in b) above.

3.2 Purpose of This Report

EPA adopted 316(b) regulations for new facilities on December 18, 2001 (66 FR 65256). Under the final rule, most new facilities could be expected to install recirculating cooling systems, primarily wet cooling towers. The EPA Adminstrator signed proposed 316(b) regulations for existing facilities on February 28, 2002. The lead option in this proposal would allow most existing facilities to achieve compliance without needing to convert once-through cooling systems to recirculating systems. However, one of the alternative options proposed requires recirculating cooling in selected plants. Until this rule is finalized, retrofitting to recirculating cooling remains a regulatory option.

Although neither the final new facility rule nor the proposed existing facility rule require dry cooling towers as the national best technology available, the environmental community and several States have supported the use of dry-cooling technology as the appropriate technology for addressing adverse environmental impacts. It is possible that the requirements included in the new facility rule and the ongoing push for dry cooling systems by some stakeholders may have a role in shaping the rule for existing facilities. Recognizing that over 50 percent of the existing coal-fired power plants employ once-through cooling systems, a decision to require many or all of these plants to install dry- or wet-cooling tower systems could have impacts on electricity costs and availability as well as secondary environmental impacts.

The purpose of this report is to quantify the loss of net electric output from an existing coal-fired power plant that would result from the replacement of its once-through cooling system to either a wet- or a dry-cooling tower. The reduction in net electric output is known as the energy penalty and is discussed below. Modeling was done for five locations to simulate the hottest time of the year using temperature values that are exceeded only 1 percent of the time between June through September at each modeled location. This corresponds to the time of year when the highest power demands are observed. To give an idea of the energy penalty at times other than the hottest period of the year, additional modeling was conducted on a monthly basis. This technique allowed for the calculation of an annual average energy penalty value at each site.

In order to compensate for the electricity lost as a result of the energy penalty, utilities would need to produce more electricity through burning additional fuel, thereby generating additional air emissions. A second purpose of this report is to quantify the additional amount of air emissions that would result at existing coal-fired plants using wet or dry cooling systems. Estimates of incremental air emissions were made at the Delaware River Basin and the South sites.

4.0 Overview of Cooling Systems at Steam Electric Power Plants

4.1 Cooling Water Use

Water is used in many industrial applications to cool machinery or to condense steam. The largest industrial user of cooling water is the steam electric power industry. Data from a recent U.S. Geological Survey (USGS) report indicate that steam electric power generation uses approximately 190 billion gallons of water per day (USGS 1998). In 1999, more than 60 percent of the utility power generating capacity in the United States (382,270 MW) utilized the steam-electric process (EIA 2000). At nuclear and fossil-fuel power plants, electricity is produced by heating purified water to create high-pressure steam. The steam is expanded in turbines, which drive the generators that produce electricity. After leaving the turbines, the steam passes through a condenser that has multiple tubes and a large surface area. A large volume of cool water circulates through the tubes, absorbing heat from the steam. As the steam cools and condenses, the temperature of the cooling water rises.

4.2 Types of Cooling Systems at Steam Electric Power Plants

Most power plants use either once-through cooling or recirculating cooling. Once-through cooling systems withdraw large volumes of water -- typically in the range of tens of millions to billions of gallons per day from a river, lake, estuary, or ocean. The water is pumped through the condenser and finally returned to the same or a nearby water body. Recirculating cooling systems receive their cooling water from and return it to a cooling tower and basin, cooling pond, or cooling lake. Because evaporation and planned cooling tower blowdown (periodic discharges of portions of the recirculating water to remove build up of solids and other undesirable constituents) removes cooling water from the evaporative system, regular additions of "makeup" cooling water are needed. Makeup volumes are much lower than daily once-through volumes, and may range from hundreds of thousands to millions of gallons per day. The USGS estimates that about 2 percent of the water withdrawn for steam electric power generation was consumed as a result of once-through cooling, cooling towers, or pond cooling (USGS 1998).

This report considers two types of recirculating cooling systems – wet towers and indirect-dry towers. These are defined in Section 2 and described in Section 5.3.

4.3 How Cooling Water Affects Steam Power Plant Performance

High-pressure steam is generated in a boiler whose heat source is a high temperature atmospheric pressure furnace fired by some type of fossil fuel or a nuclear reactor. The high-pressure steam is expanded through a multistage turbine that turns a generator to produce electricity. Spent exhaust steam exiting the turbine is condensed and recycled to the boiler for steam production. During the condensation process, a large quantity of low-grade heat is absorbed by the condenser coolant, which is typically water.

The steam side of the condenser operates under vacuum conditions (i.e., a pressure below normal atmospheric pressure). The magnitude of the condenser vacuum depends chiefly upon the condenser design and the incoming temperature of the condenser coolant. Lower coolant temperatures will produce a larger vacuum in the condenser that, to a certain extent, has a favorable effect on performance. Likewise, higher condenser coolant temperatures are associated with a smaller vacuum, resulting in reduced energy output. These relationships are based on the laws of thermodynamics and hold true regardless of the type of cooling system used (once-through or recirculating).

4.4 The Energy Penalty

Steam condensers are designed to produce a vacuum at the outlet end of the turbine, thereby increasing the efficiency of the system. The temperature of the cooling water exiting the condenser affects the performance of the turbine -- the cooler the temperature, the better the performance. As cooling water temperatures decrease, a higher level of vacuum can be produced and additional energy can be extracted. On an annual average, once-through cooling water has a lower temperature than recirculated water from a cooling tower. Because most of the heat rejection in a wet cooling tower is due to evaporation, the temperature of the recirculated cooling water is limited by the ambient air wet-bulb temperature. It can never be lower than the wet-bulb temperature and generally is about 5° to 10° F higher. As a result of switching from a once-through cooling system to a cooling tower, less energy can be generated by the power plant from the same amount of fuel.

In a related manner, the performance of a dry cooling system is limited by the ambient air dry-bulb temperature because all of the heat rejection in a dry cooling system is attributable to sensible heating of the surrounding air. Since dry-bulb temperatures are higher than corresponding wet-bulb temperatures, the performance of dry cooling systems will be less than wet systems (either once-through or recirculating). In fact, a recent analysis of cooling system options for combined-cycle power plants found that at nearly all locations and under nearly all climatic conditions in the United States, the performance of a properly designed and operated recirculating cooling system would be superior to a comparable direct-dry cooling system (Burns and Micheletti 2000).2001). Therefore, switching a once-through cooling system to a dry cooling system would mean that the decline in power generation for a given amount of fuel would be even greater than for a once-through to a recirculating wet cooling system retrofit.

Veil et al. (1992) summarized literature values for the energy penalty associated with retrofitting once-through cooled plants with wet-cooling towers. The majority of the data points for the energy penalty for fossil-fueled plants were clustered in a band between 1.5 percent to 2.5 percent. Results for nuclear power plants show greater variability, ranging between 1 percent and 5.8 percent. The data points were not as clearly clustered in a narrow range as were the data points for the fossil plants. Veil et al. (1992) selected a range of 2 percent to 3 percent for the decrease in net electrical power that could be experienced if existing nuclear power plants retrofit from once-through to wet cooling.

In a more recent study, Burns and Micheletti (2000) estimate the maximum energy penalty values for a new generic 750-MW combined-cycle power plant using either a wet recirculating cooling system or a direct dry cooling system at sites in five different parts of the country. In this study, the energy penalty is defined as the loss of electricity generating capacity incurred when a cooling system is unable to perform at design efficiency. Then for both types of cooling systems, the maximum energy penalty occurs during the hottest times of the year when ambient wet-bulb and dry-bulb temperatures are greatest. This period normally is represented by 1 percent of the time during the four warmest months, which also happen to coincide with the times of national peak electricity demand. For recirculated wet cooling, the estimated maximum energy penalty was less than 1 percent for any of the five sites. For direct dry cooling, the estimated maximum penalty ranged from 11.6 percent to 18.1 percent, depending on site climatic conditions. Although the estimates prepared by Burns and Micheletti indicate a dramatic difference in the maximum energy penalties expected from using wet and dry cooling systems, the results are not directly comparable to this study for two reasons. First, the Burns and Micheletti estimates were based exclusively on new cooling systems for new plants and did not consider any of the retrofit complexities associated with an existing once-through cooling system at an existing plant. Second, the Burns and Micheletti estimates were based on a direct-dry cooling system, while an indirect dry cooling system would be a more suitable retrofit option for an existing once-through cooling system (see subsequent discussion in Section 5.3).

For its 316(b) regulation development, EPA researched and derived energy penalty estimates based on empirical data and proven theoretical concepts for a variety of conditions (EPA 2002). To estimate nationally representative energy penalties, EPA sought data to estimate representative regions. These four regions include Northeast (Boston, MA), Southeast (Jacksonville, FL), Midwest (Chicago, IL) and Northwest (Seattle, WA). The Agency calculated the turbine component of the energy penalty by examining the empirical effect on net plant heat rates resulting from changing turbine exhaust pressures for fossil-fueled, combined-cycle, and nuclear plants. The Agency related the turbine exhaust pressure to ambient conditions for the selected locations. Because the source water temperature for once-through cooling systems and the ambient wet bulb and dry bulb temperatures for cooling towers varies with location and time of year the Agency used empirical coastal water temperatures at the four selected locations.

For calculation of monthly average wet and dry bulb temperatures, EPA calculated time-weighted averages during the daytime period between 8 AM and 4 PM. Since the energy penalty will vary over time as ambient climatic and source water temperatures vary, the calculation of the total annual energy penalty for a chosen location integrated the results of individual calculations performed on a periodic, monthly basis. EPA used design temperatures to calculate peak-summer penalties for the selected locations based on the temperature that ambient conditions equaled or exceeded one percent of the time.

EPA derived the turbine exhaust pressure values for alternative cooling system scenarios in conjunction with the empirical temperature values. EPA used these turbine exhaust pressure values to estimate the associated change in turbine efficiency. EPA then calculated either the peak-summer (design) or the monthly energy penalty. Annual values were calculated by averaging the 12 monthly values. The annual average energy penalty values for fossil-fueled plants at the four regional sites ranged from 1.5 to 1.8 percent and the peak-summer energy penalties ranged from 1.4 to 2.0 percent.

It should be noted that EPA's annual average energy penalties were based on assuming that plants operate at just 67 percent of maximum load. The DOE does not agree with this assumption for base load plants and has brought this to EPA's attention. Subsequently, EPA estimated a 1.1% annual energy penalty based on assuming that plants operated at 100 percent of the maximum load.

The annual average penalties presented in Chapter 8 of this report are similar to those calculated by EPA, but the EPA estimates of peak summer energy penalties are considerably lower than those presented in Chapter 7 of this report. The reasons for this discrepancy are:

- EPA does not include all the pumping costs associated with a wet tower retrofit. The additional pumping costs could add approximately 0.2 to 0.7 percent to EPA's energy penalty estimates.
    - EPA uses a range assumption at or near 20 degrees, which is higher than that used in most of the DOE model runs. Those runs were based on actual temperature data provided by EPA for each of the model locations (see descriptions in Chapter 7). If a 15-degree range were used, the energy penalty would increase by about 0.5 percent.
    - EPA's analysis assumes that the condenser duty is the same when converting from once-through cooling to wet cooling towers. DOE estimates that this could result in a maximum additional penalty of 0.5 percent.

Adding the contributions from these three items yield a possible increase in the EPA peak energy penalty of 1.0 to 1.5 percent and a revised EPA peak energy penalty of 2.7 to 3.2 percent for conversion to wet towers. These revisions to the EPA analysis to adjust to similar basis with this study shows approximate agreement to the DOE results.

5.0 Description of Models and Modeling Efforts

5.1 - Background on ASPEN Model

ASPEN (Advanced Simulator for Process Engineering) PLUS is a simulator software package commercially available from Aspen Technology (the original development was co-sponsored by the DOE) that is used worldwide by companies and universities to examine both commercially available and conceptual processes. Examples of technologies for which ASPEN has been used as part of the development process include integrated gasification combined cycle (IGCC) power plants, pulverized-coal power plants, fuel cells, advanced gas turbine systems and Vision 21 systems (see the NETL web site at http://www.netl.doe.gov). The simulator includes a suite of built in physical property packages and engineering process models and an additional flexibility for adding user-generated models. The ASPEN model provides a steady state representation of the overall process units (or process sections) that includes sufficient detail to accurately predict the energy and mass balances.

5.2 Specific Model for Pulverized Coal Power Plant

An ASPEN PLUS 10.2 model developed for a pulverized coal power plant in an earlier study by NETL (Shah et al. 2001) was used as a starting point for this analysis. For this study, cooling-tower systems (i.e. “Wet” and “Indirect Dry”) were added as options to the original model’s “Once-Through” steam condenser cooling.

The ASPEN model used for this study was based on a detailed design by Buchanan et al. (1998) for a power plant feeding pulverized coal to a conventional steam boiler and steam turbine. The process design uses a single reheat steam power cycle to generate nominally 400 MW of power. The steam boiler can be viewed as containing two major heat-transfer sections, a radiant section and a convective section. The radiant section consists of a natural circulation, wall-fired, subcritical unit arranged with a water-cooled dry-bottom furnace. The convective section consists of a superheater, reheater, and economizer heat exchangers. An additional air heater is external to the steam boiler. The furnace burners were a low-NOx type. The flue gas was desulfurized by treating it with lime slurry.

In the design, air is preheated in the air heater by exchanging heat with the flue gas. Coal and hot air are fed to the boiler from the bottom. High-pressure steam is generated in the radiant section. Flue gas from the radiant section enters the convective section at 2,200 ^oF. In the convective section, thermal energy from the flue gas is transferred to high-pressure steam (in the superheater heat exchanger), intermediate-pressure steam (in the reheat heat exchanger), and feed water (in the economizer heat exchanger). Flue gas leaves the convective section at 600 ^oF and passes through the air heater to preheat combustion air. An ESP is used to remove particulates and the flue gas is then sent to a sulfur dioxide (SO_2) scrubber with the aid of an induced draft fan. Lime slurry is employed to scrub SO_2 from the flue gas. The treated flue gas leaves through stacks.

High-pressure steam is superheated in the convective section. Superheated steam at 2,415 psi and 1,000 ^oF is expanded in the high-pressure turbine to an intermediate pressure of 604 psi. The intermediate-pressure steam is reheated in the convective section to 1,000 ^oF and is then expanded in the intermediate-pressure steam turbine. Finally, the exhaust from the intermediate-pressure steam turbine is expanded in the low-pressure turbine to approximately 1 psia and is then sent to a condenser. The condensate water is sent to a series of low-pressure feed heaters. The heated water is sent to the deaerator to remove dissolved gases. De-aerated water is passed through high-pressure water heaters and is then fed to the economizer portion of the convective section. Water is further heated to close to its saturation temperature in the economizer and then sent to radiant section for boiling.

5.3 - Model Adaptation for Cooling Systems

The ASPEN model described above provides the heat duty (heat of condensation for the exhaust steam) for the steam-cycle condenser. For the purposes of this study the following options were added to the model described above:

– Once-Through Cooling - this modification considers that cooling water is used in a single open-loop pass in a shell-and-tube heat exchanger. The simulator estimates the cooling water requirements and associated circulating water-pump power.

– Wet Cooling Tower - a detailed model for a wet cooling tower (Enick et al. 1994) was added to the simulator. The cooling tower operates in a closed-loop with the steam condenser. The tower cools the hot cooling water from the steam condenser by both evaporation of some of the entering water and sensible heating of the ambient air entering the tower. Estimates for blowdown and drift losses were assumed. Makeup water is provided for these losses and for evaporative losses. The cooling tower air fans' power requirements were predicted based on induced-draft fan design.

– Indirect Dry Cooling Tower - a cooling tower in which a hot liquid such as condenser coolant rejects heat to the atmosphere without the evaporation of water. Heat from the water is transferred to the surrounding atmosphere in finned-tubes, which are cooled by large diameter fans blowing air over the finned surfaces. The cooling tower air fans' power requirements were predicted based on induced-draft fan design.

The hot cooling water from the steam condenser enters countercurrent to the entering ambient air. Since the dry tower uses only sensible heat transfer to cool the water, the required air-flow rate and fan power is considerably higher than for the wet-cooling tower.

For each of the above options, the steam turbine exhaust pressure to the steam condenser is dependent on the assumptions (such as cooling water range and approach temperatures) for a particular case.

A direct-dry cooling system was not considered for use with the ASPEN model because the focus is to provide a cooling system that can be retrofitted to existing plants. In an existing plant, there simply is no room for the large-diameter ductwork required to conduct sub-atmospheric steam from the turbine exhaust hood to a direct-dry cooling tower. Additionally, existing plants have steam turbine designs that result in only allowable maximum backpressures of approximately 5.5 inches of mercury. This limit would probably be exceeded with the choice of a direct-dry cooling system when ambient temperatures are above 90° F.

5.4 Air Emissions Calculations

The process for estimating increased air emissions as a result of an energy penalty associated with conversion of a once-through cooling system to a cooling tower is focused on existing coal power plants. Calculation of air emission increases will depend on the extent and type (e.g., wet or dry cooling tower) of cooling system conversions. For illustrative purposes, this analysis assumes that all once-through cooling systems at existing coal power plants are converted to a recirculating cooling tower. The procedure to conduct this analysis is described in the following discussion.

The ASPEN Model was used to determine the peak and annual energy penalty estimates associated with replacing once-through cooling with cooling towers. The incremental air emissions resulting from combustion of additional fuel to make up for these energy penalties are estimated using the following process. First, the regional power system that is associated with the location of the model plant is defined (see Figure 1). The Delaware River Basin model plant is located in the Mid-Atlantic Area Council (MAAC) regional power pool. The MAAC Region, geographically the same as the PJM Interconnection (a company responsible for the operation and control of the bulk electric power system) control area, encompasses nearly 50,000 square miles. MAAC encompasses approximately 58,000 MW of installed generating capacity of which 20,000 MW is coal-fired capacity.

The South (Atlanta) model plant is located in the Southeastern Electric Reliability Council (SERC) regional power pool. The SERC Region covers an area of about 464,000 square miles and includes parts or all of 13 southeastern and south central States. The Region is divided geographically into four diverse Sub regions - Entergy (the geographical area of the Entergy Operating Companies and Associated Electric Cooperative, Inc.), Southern (the geographical area of the Southern electric system), TVA (the Tennessee Valley Authority area), and VACAR (the Virginia-Carolinas area).

The MAAC and SERC Regions have 332 coal-fired boiler generator sets connected to various cooling systems. The MAAC Region has 73 coal-fired boiler generator sets with about 46 percent of its capacity using cooling towers. The dominant type of cooling tower is natural draft (36 percent), followed by forced mechanical draft (8 percent). The natural draft towers will cost more but emit less pollutants. The least common type of cooling tower installed in the MAAC Region is the induced draft (2 percent) design. Similarly, the SERC Region has about 36 percent of its coal-fired capacity cooling system operating with cooling towers. The SERC Region has 259 coal-fired boiler generator sets and its coal-fired capacity is approximately four times larger than the MAAC Region capacity. The cooling tower type is dominated by the natural draft design (20 percent) followed by forced mechanical draft (10 percent) and then induced draft (5 percent).

Next, the total of all atmospheric emissions of concern associated with coal-fired power plants in this region are estimated for a baseline time period (1998). The basis for these estimates was taken from the NETL database for coal-fired power plant operations in 1998. The database contains power plant equipment details as well as an accounting of existing air emissions from these plants. The database was linked to results of ASPEN model simulations of energy penalties for model plants located in the power pool regions. The air emission model was comprised of the NETL database and logical code for translating energy penalty into increased air emissions. Also, the regional plant capacity and electricity generation are defined for the baseline time period in order to determine the amount of lost generation and lost plant capacity.

In the third step, the annual energy penalty estimates from the ASPEN simulations are used to develop estimates of plant-level emissions increase of sulfur dioxide (SO_2), nitrogen oxides (NO_x), particulate matter (PM), mercury (Hg), and carbon dioxide (CO_2), under the assumption that all coal-fired plants currently employing once-through cooling systems will need to retrofit to either wet or dry cooling towers. The model considers three different scenarios for making up the energy lost to the energy penalty. Scenario 1 assumes that new coal-fired power plants will be built to replace lost power generation capacity. Scenario 2 assumes that the replacement capacity is provided by a new gas-fired combined-cycle unit. Scenario 3 assumes that no new plant construction is needed to replace the loss of generating capacity, but that existing units are able to supply the needed power through increased dispatching of these units. This scenario implies the availability of power outside each regional power pool or sufficient marginal capacity to achieve reliable operations while reducing reserve margins within that power pool. Power generation availability is a function of supply and demand. Forecasts for power generation availability in these regions indicate that supply will be able to meet demand over the next decade. However, these forecasts do not account for energy penalties associated with installations of cooling towers. As the energy penalty increases, the availability of power generation will likely diminish. For low energy penalties (e.g., wet cooling towers), there is less risk to realizing insufficient availability than for high energy penalties (e.g., dry cooling towers). An independent availability analysis was not performed for the present study.

The replacement options in scenarios 1 and 2 consider either an advanced design coal-fired power plant or an advanced design gas-fired combined cycle power plant that meets or exceeds the Clean Air Act's New Source Performance Standards (NSPS). These scenarios will result in less emissions since the newer plants have greater efficiency and, on average, have better pollution controls. Reference plant designs for each of these options are taken from DOE's Market-Based Advanced Coal Power Systems report (DOE 1999).

Pollution control equipment at existing coal-fired power plants is accounted for in the NETL database for coal-fired power generation. Pollution control equipment is comprised of particulate control devices, SO_2 control equipment and NOx control devices.

For the MAAC region, particulate controls are predominantly cold-side electrostatic precipitators (ESPs) and represents about 94 percent of the capacity for power generation. Baghouses are used at about 5 percent of the power generation capacity with the remaining capacity equipped with mechanical devices. Most of the capacity (86 percent) in the MAAC region does not have flue gas desulfurization controls installed to reduce SO_2 emissions. The balance of coal-fired power plant capacity (14 percent) has wet scrubbers to control SO_2 emissions. NOx control equipment includes combustion controls (low NOx burners and/or overfire air) and post combustion controls (selective catalytic reduction [SCR] and selective noncatalytic combustion controls). Only 13 percent of the coal power plant capacity in the MAAC region do not have some form of combustion controls –this means that NOx emissions are higher. On the other hand, only 18 percent of the capacity are equipped with post combustion NOx controls (SCR). Therefore 82 percent, or a majority of systems in the MAAC region, are emitting high levels of NOx. This amounts to 2 to 4 times the emissions rate of the Advanced Coal-Fired replacement plants.

For the SERC region, particulate controls are predominantly cold-side ESPs (78 percent of capacity) with a lesser amount of hot-side ESPs (17 percent) and baghouses (4 percent). Mechanical devices to control particulates are installed at less than 1 percent of the capacity of coal power plants. Approximately 85 percent of the capacity in the SERC region does not have controls to reduce SO_2 emissions. SO_2 control is predominantly wet scrubbers (14 percent) with only 1 percent using spray dryer absorbers. Combustion controls to reduce NOx emissions are installed in nearly all coal-fired power plants located in the SERC region (94 percent of capacity). Less than 1 percent of the coal power plant capacity in the SERC region has post combustion NOx controls.

5.4.1 Advanced Coal-Fired Replacement Plant

The design of the replacement coal-fired power plant is based on a 400 MW supercritical steam cycle (3500 psig/1050^{o}F/1050^{o}F), which is a power generating facility configured to run under a Rankine cycle where the pressure and temperature of the steam inside the boiler exceed 3,200 psi and 1,100 ^{o}F respectively. The overall net plant efficiency is 39.9 percent, which exceeds the efficiency of the existing coal-fired plants (typically 34 to 38 percent). The maximum coal burn rate is 147 tons per hour with a design margin of 5 percent to get to a burn rate of 154 tons per hour.

The flue gas desulfurization (FGD) system for removing sulfur emissions is comprised of a limestone forced oxidation reactor designed to remove up to 96 percent of the sulfur dioxide in the flue gas. A single module reactor is configured with countercurrent flow of the flue gas and limestone slurry. Formic acid is used as a buffer to enhance the SO_2 removal characteristics.

NO_x control consists of a dual system, low NO_x combustion and selective catalytic reduction (SCR). The low NO_x combustion system is comprised of low NOx burners (LNB) and overfire air (OFA). The SCR system is designed to remove 63 percent of the incoming NOx. Particulate control is achieved with a pulse jet fabric filter capable of removing 99.9 percent of the particulates.

Design conditions for emission control equipment of major pollutants are given in Table 1. The design conditions include controls for SO_2, NOx and particulate emissions. Emission control equipment for mercury and CO_2 are not included for the advanced coal-fired replacement plant.

Table 1 - Design Emission Rate for Airborne Emissions (lb/MWh)

Pollutant	Supercritical Pulverized Coal Plant	Natural Gas Combined Cycle Plant
SO_2	1.47	0
NO_x	1.35	0.189
PM	0.08	0
CO_2	1,740	796

Mercury emissions are estimated from the mercury content in the coal and emission modification factors associated with the pollution control equipment for SO_2, NO_x, and particulates. An approximate control rate is taken as 90 percent removal of the oxidized mercury from coal combustion, based on preliminary field data taken from EPA's Mercury Information Collection Request (ICR) data (EPA 2000). Using these preliminary data, the oxidized fraction of mercury for bituminous and sub-bituminous coal is 70 percent and 35 percent. The estimated overall mercury removal is 31.5 percent and 63 percent for sub-bituminous and bituminous coal, respectively.

Average emission rates from existing coal-fired power generation is higher than for the advanced coal-fired replacement plant. The average SO_2 emission rate for the MAAC and SERC Regions is 19.8 lb/MWh and 14.7 lb/MWh, respectively. This is more than ten times greater than the SO_2 emission rate for the advanced coal power replacement plant. The average NOx emission rate for the MAAC and SERC Regions is 4.27 lb/MWh and 5.5 lb/MWh, respectively. The average NOx emission rate for existing coal-fired power plants is more than three times greater than that for the advanced coal-fired power replacement plant. PM emission rate for the existing coal-fired power plants is more than three times greater than for the advanced coal-fired power plant. CO_2 emission rate is about 25 percent larger for the existing power plants than for the advanced coal-fired replacement plant. The difference in CO_2 emission rate is predominately caused by the higher efficiency of the advanced power plant.

The cooling water system for the coal-fired replacement power plant consists of two 50-percent capacity vertical circulating pumps, a multi-cell mechanical draft evaporative cooling tower, and carbon steel cement-lined interconnect piping.

5.4.2 Advanced Natural Gas Combined Cycle (NGCC) Replacement Plant

The design of the replacement NGCC plant is based on a natural gas combustion turbine (CT) coupled with a heat recovery steam generator (HRSG). The reference plant design for the CT/HRSG technology is based on gas turbine characteristics that are similar to the Westinghouse 501G machine. The combined cycle net efficiency of the plant is 50.6-percent and is capable of producing a net output of 326 MWe. The configuration of the NGCC involves one gas turbine in conjunction with one 1650 psig/1000^{o}F/1000^{o}F steam turbine. The steam turbine is a single multi-stage machine exhausting steam to a single pressure condenser operating at 2 inches of mercury (absolute) when operating at 100 percent design load conditions.

The advanced NGCC system is expected to produce low levels of SO_2 and particulate emissions. For the purposes of this study, the plant is considered to produce negligible SO_2 and particulate emissions as well as no mercury emissions. Low levels of NO_x production from the combustion turbine are achieved by zoning and staging of fuel combustion using dry Low-NO_x can-annular combustion systems. Design conditions for emission control equipment of major pollutants are given in Table 1.

The cooling water system for the NGCC replacement power plant consists of two 50 percent capacity vertical circulating pumps, a multi-cell mechanical draft evaporative cooling tower, and carbon steel cement-lined interconnect piping.

6.0 Model Assumptions

This section of the report outlines the data that were used as inputs to the ASPEN model, where the data came from and why they were selected, any analyses that were made to convert sets of data into single model inputs, and the assumptions that were made. The same model was used for estimating both the peak and the annual energy penalties but some of the inputs varied as appropriate.

6.1 Size and Type of Plant

The objective of this study was not to simulate every possible size and type of steam power plant, but to be representative of a large class of existing plants. Approximately 52 percent of existing net generation in the United States during 2000 was coal-fired (EIA 2001b). Out of 829 existing coal-fired generating units in the United States with capacities greater than or equal to 100 MW, 43 percent fall in the size range of 200 to 600 MW. A 400 MW plant was selected as representative of this range. Because DOE had previous experience using the ASPEN model to simulate various aspects of a hypothetical 400 MW pulverized coal plant, the same model plant was used in this analysis.

6.2 Plant Location

DOE attempted to strike a balance between the number of modeled locations and the number of runs using alternate inputs at each location. Five locations were selected to represent a geographic cross-section of the existing fleet of coal-fired power plants using once-through cooling. Figure 2 plots data from the Edison Electric Institute's (EEI's) Power Statistics Data Base to show that nearly all of the coal-fired power plants with once-through cooling are located in the eastern United States, and particularly in the mid-Atlantic, Appalachian, and Great Lakes regions. Therefore, the first four sites are located in Philadelphia, Pennsylvania, Detroit, Michigan, Indianapolis, Indiana, and Atlanta, Georgia. The fifth site, Yuma, Arizona, is somewhat of an artifact because no once-through cooling plants exist in the southwestern United States except for several coastal California facilities. None of those California plants is coal-fired. Although the southwest Arizona leg of the Colorado River could theoretically support a once-through cooled power plant, this model case was run primarily to get a sense of the potential impact of a hot, dry climate on steam power plant efficiency. In addition, the southwestern site was included to give a projection of the energy penalty for converting from a wet tower to an indirect dry tower.

Site selection was also based on the availability of climatic information (e.g., wet-bulb and dry bulb temperatures, humidity, surface water temperatures) and State and Federal thermal discharge permit data used as input to the Aspen model. Table 2 shows the five site locations and a variety of information about each location for the 1 percent peak summer conditions. Table 3 includes the average monthly wet-bulb, dry-bulb, and surface water temperature for the four sites. The source of these data was the NOAA's 30-year normal temperature records.

The analysis to calculate the incremental air emissions was highly labor intensive and therefore was run for the Delaware River Basin and South sites only.

Table 2 - Locations for Model Runs

Site Name	Location	Water Body	1% Highest Dry Bulb Temp (°F)	1% Highest Wet Bulb Temp (°F)	Humidity (lbs H_2O/lb dry air)	Summer Surface Water Temp. (°F)
Delaware River Basin	Philadelphia, PA	Delaware River	93	79	0.01849	76
Michigan/Great Lakes	Detroit, MI	Lake Erie	92	76	0.01597	73
Ohio River Valley	Indianapolis, IN	White River	94	78	0.01733	76
South	Atlanta, GA	Chattahoochee River	95	78	0.01712	79
Southwest	Yuma, AZ	Colorado River	111	79	0.01469	82

Table 3 – Monthly Average Temperatures

Site Name	Temperatures (° F)	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Delaware River Basin	Wet Bulb	30.3	32.1	38.3	46.9	56.8	65.3	70.0	68.5	62.3	52.3	42.6	33.8
	Dry Bulb	30.3	33.0	42.4	52.3	62.9	71.7	76.7	75.6	68.2	56.4	46.4	35.6
	Surface Water	36.9	34.6	47.3	53.8	63.0	73.9	80.6	80.6	71.2	65.1	53.4	48.0
Michigan/Great Lakes	Wet Bulb	23.7	26.0	32.7	42.5	53.1	61.9	66.1	65.1	57.8	47.3	37.0	27.9
	Dry Bulb	24.6	26.7	36.5	48.2	59.8	69.3	74.1	72.4	65.1	53.5	41.9	29.8
	Surface Water	37.3	35.9	34.5	39.6	51.2	63.9	72.0	73.0	69.1	59.9	51.7	43.9
Ohio River Valley	Wet Bulb	26.5	30.3	37.4	46.6	56.7	65.2	68.8	67.3	60.2	49.7	39.4	27.7
	Dry Bulb	25.5	29.6	41.3	52.4	62.6	71.8	75.3	73.1	66.6	54.6	43.0	30.9
	Surface Water	46.3	45.6	54.1	57.7	68.0	73.3	77.1	76.9	71.3	62.5	51.7	37.3
South	Wet Bulb	39.1	42.4	47.6	53.8	62.5	69.0	72.1	71.2	66.0	53.3	48.4	38.5
	Dry Bulb	41.0	44.9	53.3	61.4	69.1	76.0	78.7	78.0	72.6	62.3	53.1	44.5
	Surface Water	54.0	55.8	57.7	68.2	73.2	75.0	81.7	83.5	75.0	69.4	63.7	58.8
Southwest	Wet Bulb	43.9	46.1	48.5	52.2	56.6	62.0	70.0	70.5	65.7	57.4	48.9	44.0
	Dry Bulb	56.4	60.6	64.8	71.2	78.8	87.6	93.6	92.6	86.7	76.1	64.1	56.4
	Surface Water	51.6	55.6	59.4	65.5	70.7	75.6	77.7	78.4	76.5	70.7	61.5	53.8

6.3 Discharge Temperatures and Range

EPA provided DOE with information on actual and permitted discharge temperatures from commercial coal- and oil-fired power plants in each of the locations selected for analysis (the actual power plants are not identified in the report). The information was compiled from the National Pollutant Discharge Elimination System (NPDES) permit records as found in EPA's Permit Compliance System (PCS) database (http://www.epa.gov/enviro/html/pcs/pcs_userguide.html).

Table 2 indicates that the ambient air and water temperatures are similar for several of the regions. Thus modeling would lead to a corresponding similar energy penalty if the same ranges (difference in temperature between the hot water entering and the cold water leaving the condenser; often referred to as "delta-T") were used. To provide an indication of the energy penalty sensitivity, two or three different ranges were modeled at each location for the peak summer conditions. Additionally, the various condenser ranges were employed to reconcile the differences between permitted and actual discharge temperatures at some of the sites. The primary range case for peak summer conditions at each location is based on the lower of permitted or actual discharge temperatures provided by the EPA. Due to the amount of computer resources required (runs/time) only a single range (15-degrees) was considered for all sites in estimating the annual energy penalty (this was simply chosen as the middle of most of the range assumptions used for the sensitivities at the peak –summer modeling analysis). For some of the locations, surface water conditions in the winter would result in an unacceptably low turbine back pressure below 1 inch of mercury if the range of 15 degrees were used for the once-through cooling option. If this occurs, the range is allowed to increase until the turbine back pressure is approximately 1 inch of mercury. Note that all temperatures referenced in this report are expressed in Fahrenheit degrees. A brief description of the range assumptions derived for each location is provided below.

6.3.1 - Delaware River Basin

Actual summer average discharge water temperature for the modeled plant with once-through cooling is 96 degrees. As shown on Table 2, the water temperature is 76 degrees, so a 20-degree range was used as the base case. The Delaware River Basin Commission regulations require a five-degree maximum temperature increase at the limit of a mixing zone. This could require a smaller range depending on the design of the mixing zone. Consequently, cases were also analyzed for 10- and 15-degree ranges.

6.3.2 - Michigan/Great Lakes

The modeled plant uses once-through cooling. It operates with a typical 25-degree range that represents the base case for this location. Michigan's regulations for Lake Erie require thermal discharges to the lake not to exceed an average of 80 degrees during summer months. This would require a seven-degree range during the summer season based on Table 2 water temperatures; therefore, that case was developed. An intermediate range of 15 degrees was also analyzed to define the sensitivity of temperature rise and energy loss.

6.3.3 - Ohio River Valley

Information from two plants was used to model this location. Both plants have permits restricting discharges to 90 degrees, thus making a 14-degree range the base case. Both of the plants appear to exceed the permitted temperatures regularly (perhaps due to variances) and therefore a case utilizing a 20-degree temperature increase was also developed. The annual energy penalty modeling was run for a 15-degree range case with the results compared to those that were obtained from a peak energy penalty that also employed a 15-degree range.

6.3.4 - South

The modeled plant discharges to the Chattahoochee River and its permit and operating data show a five-degree increase over receiving water temperature. A five-degree range is too small for a practical cooling system design. Therefore 10-degree and 15-degree range cases were analyzed.

6.3.5 - Southwest

As discussed previously, there are no once-through cooled coal-fired power plants in the southwestern United States. Using EPA's NPDES data, the model run was based on an allowable discharge temperature to the Colorado River in the Yuma area of 92 degrees. Therefore, a 10-degree range would be appropriate for a once-through plant. A 15-degree range case was also developed.

6.4 Approach

In a new installation using dry cooling, steam condensation would occur in a direct, air-cooled exchanger. The plant would be laid out to minimize distance from the steam turbine to the air cooler so that there would not be significant pressure drop between the turbine exhaust and the cooling tower. Direct, air-cooled heat exchangers occupy significant land space. The footprint for a direct, air-cooled condenser integrated with a 400-MW power plant of the type modeled herein would require additional land space on the order of several acres. Since the distance from the steam turbine to the dry tower could not be minimized, the diameter of the piping connecting the two units would be prohibitively large to accommodate minimal pressure losses. Therefore, in a retrofit situation, it is unlikely that a direct air-cooled condenser could be installed. For the purpose of this retrofit study, we have assumed that the existing water-cooled condenser will be retained and the water will be cooled by air in the dry-cooling tower; this is known as an indirect-dry tower.

Most dry-cooling towers existing at or being designed for utility steam power plants today are the direct, air-cooled condenser type and are designed for air-side approach temperatures of 40 degrees or greater. We therefore selected 40 degrees as our first case for dry cooling. This 40-degree air-side approach proved problematic for an indirect-dry system modeled under the one percent highest ambient air conditions because the resultant steam turbine back pressure, determined thermodynamically from the cooling tower approach, condenser range, and terminal temperature difference, was elevated to a level far in excess of the steam turbine's originally designed safe operating conditions based on once-through condenser cooling. Operation of a turbine so far above the design-point back pressure would not be viable without significant levels of modification. The energy penalties resulting from the assumption of a 40-degree air-side approach should be considered optimistic – the penalties actually realized for such a configuration would be higher than predicted here. To get a second, less extreme, set of model outputs, which would be more realistic for the case of an indirect-dry tower, we also evaluated a conservative air-side approach of 20 degrees. The rationale for this is discussed in more detail in Section 10.

For wet cooling towers, the typical commercial design is based on using an approach between the cooling water exiting and the wet-bulb temperature of the entering air of 8 degrees plus 1 to 3 degrees to account for possible plume recirculation. An approach of 10 degrees was used for the model runs used for estimating both the peak and annual energy penalties. For all cooling options (once-through, wet and dry), an 8-degrees approach was specified between the cooling water exiting and the steam entering the condenser. This approach is sometimes referred to as the terminal temperature difference.

6.5 Ambient Air Temperatures

Ambient air dry-bulb and wet-bulb temperatures for all five selected sites modeled under the one percent highest ambient air conditions are from the Marley Company's handbook (Marley 1970). Our estimate of the summer peak performance impact in going from wet to indirect-dry cooling towers was accomplished by evaluating the cooling scenarios at the maximum design point conditions. The industry accepted definition for maximum design point condition in the context of cooling towers is the dry-bulb (indirect-dry cooling towers) and wet-bulb (wet cooling towers) temperatures (^{o}F) that are equaled or exceeded 1 percent of the time, on the average, during the warmest consecutive four months. This is also the period when the demand for electricity is at its peak. In the United States, these are the months of June through September, inclusive. By definition, the maximum design point wet-bulb temperature that would be used to design a wet cooling tower for a steam condensing power plant located in Philadelphia, Pennsylvania is 79° because the ambient air wet-bulb temperature between the months of June through September (inclusive) for that city exceeds 79° less than one percent of the time during that period (see Table 2).

The analysis of annual energy penalty is based on separate estimates made at monthly intervals. The monthly dry-bulb and wet-bulb temperatures are shown in Table 3.

6.6 Ambient Water Temperature

With the exception of the Michigan/Great Lakes site, the ambient water-surface temperatures required to evaluate once-through cooling were provided by the United States Geological Survey (USGS). However, because the USGS does not record water-surface temperatures for the southwest corner of Lake Erie, calendar year 1999 data supplied by the Fermi II nuclear power plant located on the lake's shore in the town of Newport, Michigan was used for peak summer conditions. For the Michigan/Great Lakes site, daily average surface-water temperature data for the years 1997- 2001 from the National Oceanographic and Atmospheric Administration (NOAA) was used in the annual energy penalty analysis. To remain consistent with the accepted definition of ambient-air maximum design point conditions, a mean water-surface temperature was developed by taking the arithmetic average of the 12:00 pm daily temperature recording at the specified sites over the months of June through September for the peak summer conditions.

Water temperature decreases with depth in a water body. The ambient data are assumed to be values taken at the water surface. If a plant withdraws water from a deeper level, that incoming water may have lower temperature than the surface temperature used in the modeling. Monthly ambient water temperature data for each site are shown in Table 3. The site-specific data sources are described in greater detail below:

6.6.1 Delaware River Basin (Philadelphia, Pennsylvania):
USGS Water Resources Data – Pennsylvania, Water Year 2000, Volume 1, Delaware River Basin, Station Number 01474703, Delaware River at Fort Mifflin at Philadelphia, Pennsylvania.

6.6.2 Michigan/Great Lakes (Monroe, Michigan)
Fermi II Nuclear Power Plant
6400 N. Dixie Highway
Newport, Michigan 48166

Also see: http://coastwatch.glerl.noaa.gov/statistics

6.6.3 Ohio River Valley (Indianapolis, Indiana)
USGS Water Resources Data – Indiana, Water Year 2000 (Provisional), Station Number 03353611, White River at Stout Generating Station in Indianapolis, Indiana.

6.6.4 South (Atlanta, Georgia)
USGS Water Resources Data – Georgia, Water Year 2000, Station Number 02336490, Chattahoochee River at State Route 280N, Atlanta, Georgia.

6.6.5 Southwest (Yuma, Arizona)
USGS Water Resources Data – Arizona, Water Year 2000, Station Number 09429490, Above Imperial Dam on the Colorado River, Yuma, Arizona.

7.0 Results of Peak Season Energy Penalty Analysis

7.1 Energy Penalties

The ASPEN model was run for five locations and either two or three ranges at each site for a total of twelve model runs. Each run calculated the energy penalty relative to once-through cooling for a wet tower, and indirect-dry towers at approaches of 20 degrees and 40 degrees. There may be cases in which plants already using wet towers may be asked to convert to indirect dry towers (e.g., plants in the Southwest, none of which use once-through cooling). Further calculations estimated the energy penalty that would be realized in going from a wet tower to an indirect-dry tower. These results are displayed in Table 4. The detailed model output charts are included in Appendix A.

For the purpose of this report, the peak and annual average energy penalties associated with a recirculating cooling system retrofit have been presented as the percent decrease in plant net power output, holding fuel consumption constant, compared to the same facility operating under a once-through cooling scenario. The reduction in plant net power output was determined by the summation of turbine performance loss due to increased steam backpressure and the increase in plant parasitic loads caused by the cooling tower's induced draft fans and head pressure losses. Energy penalties at the 1-percent highest temperature condition relative to once-through cooling systems ranged from 2.41 percent to 3.95 percent for wet towers, from 8.85 percent to 12.14 percent for indirect-dry towers at a 20-degree approach, and 12.67 percent to 15.9 percent for indirect-dry towers at a 40-degree approach. The larger energy penalties that were obtained from the 40-degree approach model results versus the 20-degree approach model results would indicate that the reduction in plant net power was driven to a greater extent by the steam turbine's performance loss than could be overcome by the net gain in parasitic power loss (i.e., the 40-degree approach tower's fans would not consume as much power as those would on the 20-degree approach tower). A sensitivity analysis on condenser ranges, in the case of an indirect-wet cooling tower, shows that increasing the range by 5 degrees tends to decrease the energy penalty, on average, by 0.3 percent. For indirect-dry cooling towers with a 20-degree approach, increasing the condenser range by 5 degrees decreases the energy penalty, on average, by 0.9 percent. Five-degree increases for the condenser range of power plants modeled with 40-degree approach indirect-dry cooling towers decreased the energy penalties, on average, by 1 percent. This trend would seem to indicate that the parasitic power savings that can be had by reducing the cooling water flow rate through the condenser, effectively increasing the condenser range, more than makes up for the minor decrease in turbine performance due to the resultant higher backpressure. Note that the calculated energy penalties for the Southwest location have been omitted from these ranges because there is not likely to be a once-through system located in Yuma. Energy penalties associated with a retrofit from wet towers to indirect dry towers ranged from 6.1 percent to 10.9 percent at a 20-degree approach, and 10.0 percent to 15.2 percent for indirect dry towers at a 40-degree approach.

7.2 Turbine-Back Pressure

Because turbine-back pressure is an important consideration in plant performance, calculated back pressure values are presented in Table 5. The model runs using an approach of 20 degrees calculated a back pressure of the condensing steam between 4.18 and 8.35 inches of mercury and for 40 degrees calculated a back pressure between 7.03 and 13.37 inches of mercury. Steam turbines manufactured in the United States are designed to operate at back-pressures as high as 5.5 inches of mercury. Operation of a steam turbine at backpressures in excess of that recommended by the manufacturer will void the warranty and may cause significant damage to the machine because of adverse aerodynamic effects on the blades. Dry-cooling towers modeled with a 40-degree approach to ambient dry-bulb temperature yielded turbine back pressures that would require prohibitive levels of modification and re-tooling, resulting in even higher energy penalties than shown in this report by virtue of load shedding. For comparison, 6 out of 12 of the dry cooling tower model runs incorporating a 20-degree approach to ambient dry-bulb temperature produced steam-turbine-back pressures in an acceptable, albeit borderline, range.

Table 4 - Energy Penalty Results from ASPEN Model at One Percent Highest Temperature Conditions

Location	Range (°F)	Energy Penalty Relative to Once-Through Cooling System (%)			Energy Penalty Relative to Wet Tower (%)	
		Wet Tower	Indirect Dry Tower - 20 °F Approach	Indirect Dry Tower - 40 °F Approach	Indirect Dry Tower - 20 °F Approach	Indirect Dry Tower - 40 °F Approach
Delaware River	10	3.48	10.41	14.19	7.2	11.1
Delaware River	15	3.12	9.27	13.23	6.3	10.4
Delaware River	20	2.95	8.85	12.67	6.1	10.0
Michigan/Great Lakes	7	3.95	12.14	15.9	8.5	12.4
Michigan/Great Lakes	15	3.08	9.75	13.21	6.9	10.4
Michigan/Great Lakes	25	2.94	8.92	12.77	6.2	10.1
Ohio River Valley	14	2.98	9.61	13.42	6.8	10.8
Ohio River Valley	20	2.77	9.03	12.82	6.4	10.3
South	10	2.78	10.27	14.52	7.7	12.1
South	15	2.41	9.33	13.14	6.9	10.7
Southwest	10	2.46	13.06	17.25	10.9	15.2
Southwest	15	2.06	12.04	15.79	10.0	13.7

Note: Shaded cells represent hypothetical cases in which a once-through cooling plant is located in the southwestern United States.

Table 5 - Turbine-Back Pressure Results from ASPEN Model at One Percent Highest Temperature Conditions

Location	Range (°F)	Turbine Back Pressure (inches of mercury)		
		Wet Tower	Indirect Dry Tower 20 °F Approach	Indirect Dry Tower - 40 °F Approach
Delaware River	10	2.38	4.65	7.77
Delaware River	15	2.75	5.3	8.77
Delaware River	20	3.17	6.04	9.88
Michigan/Great Lakes	7	1.99	4.18	7.03
Michigan/Great Lakes	15	2.52	5.17	8.56
Michigan/Great Lakes	25	3.35	6.68	10.87
Ohio River Valley	14	2.60	5.30	8.77
Ohio River Valley	20	3.08	6.19	10.1
South	10	2.31	4.9	8.15
South	15	2.67	5.58	9.20
Southwest	10	2.38	7.39	11.92
Southwest	15	2.75	8.35	13.37

Note: Shaded cells represent pressures greater than 5.5 inches of mercury.

8.0 Results of Annual Energy Penalty Modeling

8.1 Energy Penalties

The energy penalties were calculated on a monthly basis for each site. Monthly penalty values were arithmetically averaged to estimate the annual energy penalty. Note that this is not exact because the use of averaging conditions may underestimate penalties occurring during the very hot and very cold times of the year when electricity demand is greater than the periods with more moderate temperatures. Table 6 shows the estimated annual energy penalties for each site. The penalty associated with retrofitting wet towers ranges between 0.8 and 1.5 percent while retrofits to dry towers are considerably higher. For the sake of comparison, the one percent highest temperature energy penalties, assuming a 15-degree range, are shown in parentheses.

Table 6 – Estimated Annual Energy Penalty

Site Location	Energy Penalty Relative to Once-Through Cooling System (%)			Energy Penalty Relative to Wet Tower (%)	
	Wet Tower	Indirect Dry Tower - 20 °F Approach	Indirect Dry Tower - 40 °F Approach	Indirect Dry Tower - 20 °F Approach	Indirect Dry Tower - 40 °F Approach
Delaware River Basin	1.18 (3.12)	4.71 (9.27)	8.23 (13.23)	3.57	7.13
Michigan/Great Lakes	1.47 (3.08)	4.17 (9.75)	8.05 (13.21)	3.29	6.68
Ohio River Valley	1.14 (2.98)	4.50 (9.61)	7.91 (13.42)	3.39	6.84
South	0.82 (2.41)	5.20 (9.33)	8.82 (13.14)	4.41	8.07
Southwest	0.80 (2.06)	7.70 (12.04)	11.37 (15.79)	6.96	10.66

( )- Peak energy penalty model results run using a 15 degree F range.

8.2 Turbine Back Pressure

None of the monthly pressure values at any of the sites exceeds the critical threshold of 5.5 inches of mercury design point for the turbine's safe operation for the once-through, wet cooling, or 20-degree approach dry cooling options. Some exceptions are present primarily for the 40-degree approach dry cooling option for the Southwest (Yuma) site. It should be noted that using temperature averaging misses the peak summer conditions when this threshold of 5.5 inches of mercury is often exceeded for the dry cooling options at both approach assumptions (see section 7.2).

9.0 Results of Air Emissions Modeling

Increased air emissions of SO_2, NOx, PM, Hg, and CO_2 have been estimated under three scenarios for two power pool regions. Air emissions increase as the energy penalty increases Increased air emissions are a function of baseline operating conditions and increased fuel consumption. Increased fuel consumption is a result of the energy penalty associated with conversion of an existing cooling system to a closed cycle cooling system employing a cooling tower. Baseline air emissions are contained in the NETL database for coal power plant operations.

For the three scenarios developed in this study, increasing fuel consumption at existing coal power plants yields the largest increase in air emissions because existing systems are both less efficient at producing power and therefore burn more coal and , on average, have less emissions control equipment. The higher the energy penalty, the larger the fuel consumption and increase in air emissions. In this study, the largest energy penalty is associated with conversion of a once-through cooling system to a dry tower.

The capacity of coal power plants in the MAAC region is about one quarter of that in the SERC region. Since the SERC region has a larger power generation capacity, the baseline air emissions are consistently higher than that for the MAAC region.

The emission rate, expressed in mass per unit of power generation, varies with the installed control equipment and coal properties (e.g., sulfur content). For the SERC region, the average SO_2 emission rate (14.7 lb/MWh) is slightly lower than for the MAAC region (19.8 lb/MWh). Even though there are a similar percentage of SO2 controls in both regions, the SO_2 emission rate is lower in the SERC region because a lower sulfur coal is used in that region. The average sulfur content of coal in the SERC region is 0.95 lb/MMBtu as compared to an average sulfur content of 1.22 lb/MMBtu for the MAAC region.

The NOx emission rate for the MAAC region (4.27 lb/MWh) is lower than for the SERC region (5.50 lb/MWh). One of the principal reasons for the lower emission rate in the MAAC region is the more frequent installation of post-combustion NOx controls.

The particulate emission rate for the MAAC and SERC region is similar, 0.29lb/MWh and 0.27 lb/MWh, respectively. The dominant particulate control device is cold-side ESPs for both regions.

The mercury emission rate for the MAAC region is 7.2×10^{-5} lb/MWh and is more than 70 percent greater than in the SERC region 4.1×10^{-5} lb/MWh. The dominant reason for the higher mercury emission rate in the MAAC region is the higher mercury content in the coal.

CO_2 emissions are not controlled at existing coal power plants. Emission rates are primarily a function of the efficiency of power generation while fuel properties play a minor role. The CO_2

emission rates are similar for the MAAC and SERC region, 2,190 lb/MWh and 2,200 lb/MWh, respectively. Increased CO_2 emissions that yield no significant economic benefit to the gross domestic product will negatively affect this Administration's carbon intensity reduction goal to mitigate the threat of climate change associated with increased emissions of greenhouse gases.

For a given amount of power generation, fuel consumption will increase proportionally with an increase in energy penalty. For the three scenarios developed in this study, the energy penalty and increased fuel consumption as a result of conversion of a power plant to a closed-cycle cooling system is lowest for a wet cooling tower and highest for conversion to a dry cooling tower with a high range.

Annual coal consumption for existing power plants in the MAAC and SERC region is 45 million tons and 201 million tons, respectively. For the scenario where existing coal power plants will makeup lost power generation by increasing coal feed rate, the increase in coal consumption is equal to the energy penalty multiplied by the baseline coal consumption for each affected facility.

If a replacement plant is used to make up lost power generation from an energy penalty, the replacement plant will use less fuel to produce an equivalent amount of power. This is because the replacement plants are designed to be more efficient than the existing plants. The NETL database contains fuel consumption and associated power generation for each coal power plant. It also contains power generation losses associated with each energy penalty scenario developed in the present study. Replacement plant fuel consumption is calculated from lost power generation from the existing plant and the efficiency of the replacement plant.

The baseline air emissions for the two regions modeled in this study are provided in Section 9.1 and 9.2.

9.1 – MAAC Region (Region for Delaware River Basin Site)

The baseline generating and emission conditions for the MAAC in 1998 are outlined in Table 7. The additional emissions associated with making up electricity lost to the energy penalty under three different generating scenarios are shown in Table 8.

Table 7 – Baseline Conditions for MAAC – 1998

Parameter	Quantity
Electric Power Generation	115,396,147,673 kWh
SO_2 Emissions	1,142,434 tons
NO_x Emissions	246,143 tons
PM Emissions	16,501 tons
Hg Emissions	8,259 lbs
CO_2 Emissions	126,121,852 tons

Table 8 – Increased Annual Emissions for MAAC

Pollutant	Supercritical Pulverized Coal Plant	Natural Gas Combined Cycle Plant	Increase Output at Existing Coal Plants
Convert Once-through Cooling to Wet Towers			
SO_2 (tons)	467	0	6,781
NO_x (tons)	428	60	1653
PM (tons)	24	0	89
Hg (lbs)	32	0	47
CO_2 (tons)	552,352	304,849	719,290
Convert Once-Through Cooling to Dry Towers (20-degree range)			
SO_2 (tons)	3,629	0	49,254
NO_x (tons)	3,333	466	10,751
PM (tons)	188	0	705
Hg (lbs)	258	0	354
CO_2 (tons)	4,296,083	1,965,335	5,416,344
Convert Once-Through Cooling to Dry Towers (40-degree range)			
SO_2 (tons)	6,609	0	89,377
NO_x (tons)	6,069	850	19,395
PM (tons)	341	0	1,284
Hg (lbs)	471	0	645
CO_2 (tons)	7,822,121	3,678,395	9,846,019

9.2 – SERC Region (Region for Southern Site)

The baseline generating and emission conditions for the SERC in 1998 are outlined in Table 9. The additional emissions associated with making up electricity lost to the energy penalty are shown in Table 10.

Table 9 – Baseline Conditions for SERC – 1998

Parameter	Quantity
Electric Power Generation	443,571,690,271 kWh
SO_2 Emissions	3,249,891 tons
NO_x Emissions	1,226,216 tons
PM Emissions	59,686 tons
Hg Emissions	18,381 lbs
CO_2 Emissions	487,485,740 tons

Table 10 – Increased Annual Emissions for SERC

Pollutant	Supercritical Pulverized Coal Plant	Natural Gas Combined Cycle Plant	Increase Output at Existing Coal Plants
Convert Once-through Cooling to Wet Towers			
SO_2 (tons)	1,524	0	17,143
NO_x (tons)	1,399	196	6,558
PM (tons)	66	0	312
Hg (lbs)	57	0	91
CO_2 (tons)	1,803,355	824,984	2,275,368
Convert Once-Through Cooling to Dry Towers (20-degree range)			
SO_2 (tons)	12,551	0	127,024
NO_x (tons)	11,527	1,614	48,012
PM (tons)	552	0	2,330
Hg (lbs)	459	0	712
CO_2 (tons)	14,856,682	6,796,505	18,770,251
Convert Once-Through Cooling to Dry Towers (40-degree range)			
SO_2 (tons)	24,552	0	246,736
NO_x (tons)	22,548	3,157	93,184
PM (tons)		0	4,529
Hg (lbs)	898	0	1,388
CO_2 (tons)	29,061,904	13,294,986	36,715,227

As seen in Tables 8 and 10, the largest increase in annual emissions is for CO_2. There are no controls for carbon dioxide at power plants so emissions increase proportionally with increased fuel consumption. If increased coal consumption is used to compensate for lost power production associated with an energy penalty, SO_2 and NOx emission will increase. Both of these pollutants have adverse health and welfare impacts. These pollutants contribute to acid rain formation that causes acidification of lakes and streams and can damage trees at high elevations. Based on health concerns, SO_2 and NOx have historically been regulated under the Clean Air Act. These pollutants interact with the atmosphere to form fine sulfate and nitrate particles. Scientific studies have identified a relationship between elevated levels of fine particles and increased illness and premature death from heart and lung disorders, such as asthma and bronchitis.

The range of results for increased air emissions from the three scenarios indicates that widespread installation of wet cooling towers on coal power plants would likely stress the power industry's ability to meet demand and regulatory requirements but would likely not impact the ability for the electric generation sector to meet more stringent air emission caps, such as limits for NOx emissions under the NOx State Implementation Plan (SIP) Call or for limits of SO_2 and

NOx emissions under the Acid Rain Program (Title IV of the Clean Air Act Amendments of 1990). A likely scenario for mitigation of increased emissions (with the exception of CO_2 emissions) would be installation of environmental control equipment at existing plants. The extent to which controls would be added to offset increased emissions has not been investigated in this report.

10.0 Discussion

10.1 Energy Penalties

10.1.1 One Percent Highest Temperature Conditions

The one-percent highest temperature energy penalties estimated in this study are significant in light of the large number of coal-fired and other-fueled plants that currently operate under once-through cooling systems. If new regulations for existing facilities cause more than a few such plants to retrofit wet- or dry-cooling towers, the loss of available energy to the nation as a whole or to certain regions of the country where once-through cooling is widely used could be very important to utilities' ability to supply abundant and affordable electricity.

For the sake of discussion, consider that in 1996, 258,906 MW of electric generating capacity consisted of plants employing once-through cooling systems (EEI 1996). Assume that 10 percent, 25 percent, 50 percent, or 100 percent of the once-through plants producing that power may be required to retrofit to wet cooling towers. Using the low (2.4 percent), arithmetic average (3.0 percent) and the high (4.0 percent) energy penalties generated by the ASPEN model, the resulting loss in power generating capacity would be significant and is illustrated below (Table 11). Keep in mind that these energy penalties were generated in five different locations during the peak energy demand period of the summer months. Many of the locations are in the Eastern half of the country (see Figure 2) and heat waves could affect the entire area where the once-through cooled plants are concentrated. Note that in developing these tables, we have extrapolated these energy penalties to the whole United States at the same time of peak demand.

The energy, time, and expense required to make up for these losses is significant. The energy lost to the energy penalty could range from 621 MW to more than 10,000 MW. This represents from 0.24 to 4 percent of the quantity of power currently generated by once-through cooled plants. For example in the "average" case, 19 additional 400-MW plants would have to be built to replace the generating capacity lost by replacing once-through cooling with wet cooling towers in 100 percent of existing steam plants. The amount of additional fuel that would be required to generate the lost power is huge.

The second reason involves whether the existing turbines in place at most U.S. once-through cooled power plants could operate at such high levels of back pressure. The results of the modeling suggest that, under the climatic inputs assumed, dry cooling will not be a feasible option as a retrofit during peak summer conditions for many plants originally designed to operate with once-through cooling.

Table 11 - Wet Cooling Tower Energy Penalties and Impact at One Percent Highest Temperature Conditions

Once-through Cooling Systems Required to Retrofit (%)	Wet, Recirculating Cooling Tower Retrofit Penalty (%)					
	Low Value* 2.4		Average Value* 3.0		High Value* 4.0	
	Energy Penalty (MW)	% of Total Steam Electric Capacity	Energy Penalty (MW)	% of Total Steam Electric Capacity	Energy Penalty (MW)	% of Total Steam Electric Capacity
10	621	.24	777	0.30	1,036	0.40
25	1,553	.60	1,942	0.75	2,589	1.00
50	3,106	1.20	3,883	1.50	5,178	2.00
100	6,212	2.40	7,766	3.00	10,356	4.00

* The energy penalties calculated for the Southwest site are not used here because once-through cooled plants are not likely to be found in that region.

A similar analysis was performed for those plants that might be required to retrofit their once-through cooling systems to indirect-dry towers using the conservative 20-degee approach; the resulting energy penalty impacts would be over three times higher (see Table 12), ranging from more than 2,000 MW to over 33,000 MW. This represents from 0.9 to more than 13 percent of the quantity of power currently generated by once-through cooled plants. There are two contributors to the increased impacts. First of all, the magnitude of the energy penalties for indirect-dry towers is several times higher than the penalty for wet towers due to much higher turbine back pressures associated with dry towers. To accommodate the energy lost to the energy penalty, the "average" case would require 66 new 400-MW plants to be built to replace the generating capacity lost by replacing once-through cooling with indirect dry cooling towers with a 20-degree air-side approach in 100 percent of existing steam plants. If the 40-degree approach had been analyzed here, the results would have been significantly higher.

Table 12 - Indirect-Dry Cooling Tower Energy Penalties and Impact at One Percent Highest Temperature Conditions

Once-through Cooling Systems Required to Retrofit (%)	Indirect-Dry (20° F Approach) Cooling Tower Retrofit Penalty (%)					
	Low Value* 8.8		Average Value* 10.2		High Value* 13.1	
	Energy Penalty (MW)	% of Total Steam Electric Capacity	Energy Penalty (MW)	% of Total Steam Electric Capacity	Energy Penalty (MW)	% of Total Steam Electric Capacity
10	2,278	0.88	2,641	1.02	3,392	1.31
25	5,696	2.20	6,602	2.55	8,479	3.28
50	11,392	4.4	13,204	5.10	16,958	6.55
100	22,784	8.80	26,408	10.20	33,917	13.10

* The energy penalties calculated for the Southwest site are not used here because once-through cooled plants are not likely to be found in that region.

As noted earlier, 6 of the 12 model runs using a 20-degree approach resulted in pressures that exceeded the 5.5 inches of mercury threshold. For situations in which the turbine backpressure falls within 1 to 1.5 inches of mercury of that threshold, it would be theoretically possible to reduce the back pressure to safe levels through load shedding. This possibility may be applicable for 3 of the 6 model runs that exceeded the pressure threshold. The load shedding would be accomplished through a reduction in the temperature range of the condenser cooling water, which is directly proportional to plant output. For example, if the condenser cooling water temperature range were 20 degrees at full load, reducing the plant output by 50 percent would reduce the condensing temperature by 10 degrees. This would provide about 1.2 inches of mercury reduction in turbine back pressure. However, the severity of the energy penalty (in terms of reduced power output) associated with this small amount of back pressure relief makes it an unattractive option that would only be selected in an emergency situation for a short time. Peer review comments received on this report indicated that a 20-degree approach was too conservative and cooling systems designed on that basis would not function properly. Even if a retrofitted dry tower with a 20-degree approach is hypothetically assumed, it cannot operate safely under the peak temperature conditions at most U.S. locations.

Many plants would face costly modifications to enable existing steam turbines to operate safely at back pressures so far removed from their original design point. The standard steam turbine in most water-cooled plants is designed for optimal operation at 1.5 inches of mercury back pressure. These units are designed to normally operate between 1.0 and 6.5 inches of mercury without significant loss of turbine performance. At extreme off-design operation, pressures significantly above 6.5 inches of mercury, standard turbines will experience degraded turbine performance, and physical damage to the turbines is possible.

All of the model runs using an approach of 40 degrees calculated a back pressure of the condensing steam of between 7.03 and 13.37 inches of mercury. This range of turbine back pressures would not allow for safe operation of the system. In order to operate above 5.5 inches of mercury, existing standard steam turbines would either have to be replaced by new turbines with different designs, or at the very least, would need to be significantly modified by removal of stages (this means that the last few stages of the turbine would actually be removed, causing even greater losses in efficiency). Therefore, the turbines would not be fully operational at the assumed approach of 40 degrees and the estimated energy penalties would likely be greater than forecasted in this study. We concluded that the cost of such modification would be unacceptable. Retrofit of once-through cooled plants to dry cooling using a 40-degree approach (the current industry norm) is not a practical or acceptable option for peak season conditions.

It is interesting to note that attempts to retrofit a dry tower at the Southwest location, using either a 20-degree or a 40-degree approach, would exceed the 5.5 inches of mercury threshold. According to the model results, dry towers could not be retrofit onto once-through plants in the Southwest. Two points can be made on this issue. First, since few if any plants in the Southwest are likely to use once-through cooling, the case is purely hypothetical. On the other hand, the calculations point out that dry cooling combined with traditional boilers and turbines will not be an effective cooling remedy in very warm climates. If dry cooling is to be used in hot climates, the entire power generating system – boiler, turbine, condenser, and cooling – must be built to a different set of specifications in a coordinated fashion.

The cost to convert/retrofit an existing plant to operate using a dry tower would be very high and could cause utilities to evaluate various options. The requirement and design details would be site specific and we have not analyzed this option. In one example reported in the literature (http://www.glencanyon.net/navajo.htm), the Navajo power plant would have to reduce its output by 30 percent to utilize dry cooling.

10.1.2 Annual Average Temperature Conditions

The annual average energy penalties are lower than those calculated for the one-percent highest temperature conditions and the impacts to national and regional energy supply are consequently less extreme. Nevertheless, in parts of the country with many once-through cooled plants, even the annual energy penalty can have a significant impact. Using an average of the annual average energy penalties from the four sites excluding the Southwest site (1.15 percent) and following the same types of calculations outlined in Tables 11 and 12, requirements to retrofit wet cooling towers at 100 percent of once-through cooled plants would result in 2,984 MW of lost energy. Requirements to retrofit indirect-dry towers with a 20-degree approach at 100 percent of once-through cooled plants would result in 12,375 MW of lost energy. These composite losses are not huge, but they could have an undesirable affect on some regions of the country.

Apparently, the issue of excessive turbine backpressure for dry tower retrofits does not apply during most temperature conditions. Nevertheless, the periods of highest electricity demand typically occur during those one-percent highest temperature conditions, such that dry cooling is most likely not a retrofit option.

10.2 Dry Cooling Tower Footprint Area

Another significant issue limiting the viability of a potential retrofit of an indirect dry tower to an existing plant is the amount of land area required by the cooling tower. A commonly used measure of plant equipment land use is called the battery limit footprint area, which is defined as the amount of land area "inside the fence" occupied by the actual equipment, plus any additional space needed for maintenance access. For power plants, the footprint area is usually expressed in square feet of land needed per megawatt of generating capacity.

To determine what the footprint area would be for indirect dry cooling towers, a representative site was selected (Delaware River Basin) and the indirect dry towers that would be needed for those design conditions were sized. This involved: performing design calculations for the actual heat exchanger modules that would be needed at that site to accomplish the degree of cooling required; calculating the amount of space these units would occupy; and then adding an allowance for the additional space that would be needed for maintenance access determined the footprint area.

An indirect dry cooling tower is simply a heat exchanger in which hot water rejects heat to the atmosphere by conduction and convection. The hot water flows inside a series of tubes that are cooled by ambient air. To enhance the heat transfer efficiency of the tubes, helical fins are attached to the outside of the tubes to provide additional surface area on the air side. As is typical of other large heat exchangers, these finned tubes are arranged in large factory-assembled modules that can be trucked to the plant site and connected together to complete the cooling tower assembly.

The most critical parameter in sizing an indirect dry cooling tower is the total surface area required for the heat exchanger, which may be readily determined from simple engineering relationships, depending upon the hot water flow rate and temperature, the ambient dry-bulb temperature and the desired approach temperature. The most important of these is the desired approach temperature. As the approach temperature is reduced, the cooling tower is forced to operate closer to the thermodynamic limits on how much heat can be theoretically transferred from the water to the air and the surface area increases significantly.

Once the total surface area is determined, it is a straightforward matter for the designer to select a tube configuration, module layout and then calculate the number of tubes and modules required to complete the cooling tower heat transfer array.

The heat exchanger sizing calculations for this example were based on the following design parameters:

- 1-inch diameter finned tubes made of hot-dipped galvanized steel,
- 50 feet long tubes,
- 0.375-inch high helical fins,
- 10 fins per inch,
- spaced at 2.2-inch pitch laterally and longitudinally, and
- an optimized tube array configured in an A-frame overall arrangement.

The calculated battery limit footprint for the indirect dry tower for the 40-degree approach Delaware Valley case was 186 square feet per megawatt of generating capacity. This is nearly equal to the footprint typical of an entire coal-fired power plant with once-through cooling (200 square feet per megawatt). Retrofitting such a cooling tower to an existing plant would mean nearly doubling the battery limit footprint area.

The indirect dry tower footprint for the 20-degree case approach was 57 percent larger than the 40-degree approach case, 292 square feet per megawatt. Retrofitting that tower to an existing plant would entail about a 150 percent increase in the plant footprint area.

To observe a visual example of the large battery footprint of a dry cooling system for a new facility, readers are directed to a website describing a direct-dry tower system at a 40-MW geothermal power plant operated by Steamboat Geothermal near Reno, Nevada. The plant employs 240 fans covering a significant area of land relative to the overall size of the plant. Photographs of that facility are available at http://home.nvbell.net/sbgeo/steamboat.html. Note that these towers are <u>direct</u>-dry towers that were built as part of the original construction. Also note the large surface area of fans for just 40-MW of generating capacity.

10.3 Incremental Air Emissions

The incremental air emissions associated with the annual average energy penalty are proportional to the energy penalty. For each increase in energy penalty, the air emissions will increase at a constant rate. When once-through cooled plants are converted to wet cooling towers, the incremental air emissions are not large on a percentage basis (generally less than one percent), but the absolute increase in pounds or tons of air pollutants is large nonetheless. If once-through cooled plants are converted to dry towers, however, the incremental air emissions can be significant. For dry towers with a 20-degree approach, the percentage increase in air emissions can exceed 4 percent depending on how the power company makes up the lost energy. For dry towers with a 40-degree approach, the percentage increase in air emissions can approach 8 percent and the number of additional pounds or tons is quite large.

It is logical to assume that numerous power plants are located in highly populated areas. By virtue of their high populations, many of those same areas are likely to experience less than ideal air quality. If some or many of the power plants in areas with diminished air quality must modify their cooling systems, thereby imposing an energy penalty, they will need to generate additional power. Assuming that plants in the same highly populated areas generate the additional power, air quality will be further diminished by virtue of the additional air emissions resulting from burning additional fuel. Of the five air pollutants evaluated in this report, three (SO_2, NOx, and PM) are of national concern to human health and welfare. EPA's National Ambient Air Quality Standards (NAAQS) set concentration limits for those pollutants to maintain suitable air quality. NOx also contributes to the formation of ozone, which is yet another criteria pollutant having standards set under NAAQS. Mercury is an air toxic and can bioaccumulate into the food chain. Carbon dioxide (CO_2) is a greenhouse gas and is the largest contributor to global warming potential.

EPA refers to those parts of the country where air pollution levels consistently exceed the NAAQS as nonattainment areas. As of January 2002, the EPA website for listing nonattainment areas, http://www.epa.gov/air/oaqps/greenbk/index.html, has identified a significant number of nonattainment areas in the MAAC and SERC power pools. The MAAC region is in nonattainment for ozone in 18 counties located in New Jersey, 36 counties in Pennsylvania, and 14 counties in Maryland. A smaller area is in nonattainment for SO_2 and PM. The SERC region is in nonattainment for ozone in 13 counties located in Georgia, and 2 counties in Alabama.

Increased emissions of NOx, SO_2, and PM will stress areas that are currently in nonattainment. Although the impact of increased emissions on air quality is not part of this study, it should be a consideration for at least nonattainment areas. Increased air emissions associated with increased fuel usage required to offset the energy penalty from a cooling tower is of special concern in Class I areas. The Clean Air Act defines mandatory Class I federal areas as certain national parks (over 6,000 acres), wilderness areas (over 5,000 acres), national memorial parks (over 5,000 acres), and international parks that were in existence as of August 1977. These sensitive areas have undergone significant change in air quality. For example, without the effects of pollution, a natural visual range is approximately 140 miles in the West and 90 miles in the East. When considering our current air quality, in the West, the range is 33-90 miles, and in the East, the range is only 14-24 miles.

Through the 1977 amendments to the Clean Air Act, Congress set a national goal for visibility as "the prevention of any future, and the remedying of any existing impairment of visibility in mandatory Class I Federal areas which impairment results from manmade air pollution." The amendments required EPA to issue regulations to assure "reasonable progress " toward meeting the national goal. Furthermore, the Clean Air Act requires all states to develop and implement an operating permit program that meets minimum Federal requirements. The operating permit program covers a variety of significant operations, including sources required to have pre-construction or new source permits under New Source Review or Prevention of Significant Deterioration (PSD) requirements. Although the calculated increase in emissions is relatively small, for plants located near a national park, even a small increase in emissions would not be allowed under the Clean Air Act's PSD requirements. Increased air emissions associated with the installation of cooling towers may not be permitted under PSD requirements if these emissions were shown to affect the quality of air in Class I areas.

There is one Class I area in the MAAC power pool and fifteen Class I areas in the SERC power pool. This represents about 10 percent of the 163 federally mandated Class I areas designated in the United States. Special analyses are required when a proposed new emissions source may impact any of the Class I areas. The Class I areas in the SERC regional power pool include the Shenandoah and Great Smokey Mountain National Parks, both of which have seen a statistically significant upward trend in ozone concentration from 1990 to 1999.

All increased emissions presented in this study are estimated on an annual basis. However, air quality can be sensitive to climatic conditions that are seasonal in nature. For instance, NO_x emissions (a precursor to ozone formation) can be especially damaging to air quality during hot summer days. This period of time encompasses the period in which electricity frequently peaks to its greatest demand. Peak energy penalties for the MAAC and SERC regions can be 2.6 to 2.9 times greater than for annual energy penalties based on model plant estimates for conversion of a once-through cooling system to a wet cooling tower (see Table 6). The increase in air emissions during peak temperatures (1 percent highest temperature condition) would be 2.6 to 2.9 times greater than the annual average increase in air emissions.

11.0 Acknowledgments

The following persons directly contributed to this report:

National Energy Technology Laboratory
Robert W. Gross
Dennis N. Smith
Edward L. Parsons, Jr.
Walter W. Shelton (employed by EG&G Inc., an NETL contractor)
Thomas J. Feeley, Jr.
Gilbert V. McGurl

DOE - Office of Fossil Energy
Debra J. Littleton

Argonne National Laboratory
John A. Veil (Mr. Veil's contribution to the project was funded through DOE contract W-31-109-Eng-38)

The authors acknowledge and express appreciation for assistance from the EPA, the Electric Power Research Institute (EPRI), the Utility Water Act Group (UWAG), the Edison Electric Institute (EEI), and the Hudson Riverkeeper in reviewing the assumptions and data for the model runs.

10.0 References

Buchanan, T.L., M.R. DeLallo, H.N. Goldstein, G.W. Grubbs and J.S. White, 1998, "Market-Based Advanced Coal Power System," (Section 3.1, Pulverized Coal-Fired Subcritical Plant , 400 MWe), prepared For: The U.S. Department of Energy, Office of Fossil Energy, Contract No. DE-AC01-94FE62747, by Parsons Infrastructure & Technology, Reading, PA, December.

Burns, J.M., and W.C. Micheletti, 2000, "Comparison of Wet and Dry Cooling Systems for Combined Cycle Power Plants," prepared for Hunton & Williams, November 4.

DOE, 1999, "Market-Based Advanced Coal Power Systems," DOE/FE-0400, U.S. Department of Energy, Office of Fossil Energy, May.

EIA, 2001a, "Annual Energy Outlook 2000 with Projections to 2020," DOE/EIA-0383(2002), U.S. Department of Energy, Energy Information Administration, December.

EIA, 2001b, "Electric Power Annual 2000, Volume 1," DOE/EIA-0348(2000)/1, U.S. Department of Energy, Energy Information Administration, August.

EIA, 2000, "Inventory of Electric Utility Power Plants in the United States 1999, DOE/EIA-0095(99)/1, U.S. Department of Energy, Energy Information Administration, September.

EEI, 1996, *Environmental Directory of US Powerplants,* Edison Electric Institute, Washington, D.C.

Enick, R., S. Klara, J. Marano, 1994, "High Pressure Humidification Columns: Design Equations, Algorithm, And Computer Code," DOE/PETC/TR-94-05 (DE94014756), July.

EPA, 2000, "Control of Mercury Emissions from Coal-Fired Utility Boilers," memorandum from F.T. Princiotta to J. Seitz, U.S. Environmental Protection Agency, October 25.

EPA, 2002, "Technical Development Document for the Proposed Section 316(b) Phase II Existing Facilities Rule," EPA 821-R-02-003, U.S. Environmental Protection Agency, April.

Marley, 1970, "Cooling Towers Fundamentals and Application Principles - Engineering Weather Data," Kansas City, MO.

Micheletti, W.C., and J.M. Burns, 2001, "Technical Review of Tellus Institute Report - Comments on the EPA's Proposed Regulations on Cooling Water Intake Structures for New Facilities," prepared for Hunton & Williams, June 16.

SAIC, 2001, "Steam Plant Energy Penalty Evaluation," draft report by Science Applications International Corporation, April 20.

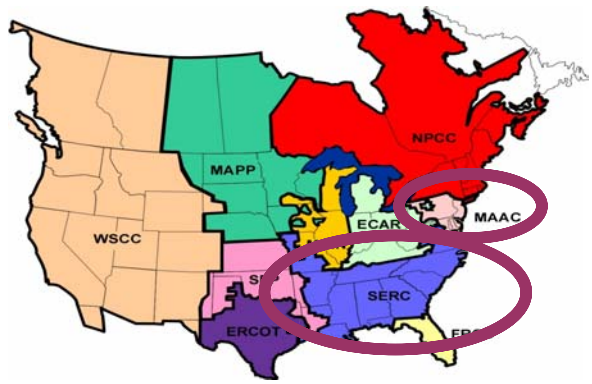
Shah, M., L. Bool and W. Shelton, 2001, "Oxygen Transport Membranes (OTM)" (Task 3: OTM Integrated Combustion Processes), CRADA (99-F032) between NETL and Praxair, Inc., April.

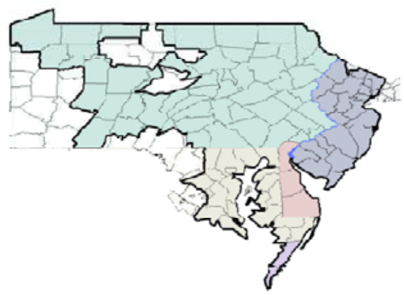
USGS, 1998, "Estimated Use of Water in the United States in 1995," U.S. Geological Survey.

Veil, J.A., J.C. VanKuiken, S. Folga, and J.L. Gillette, 1993, "Impact on the Steam Electric Power Industry of Deleting Section 316(a) of the Clean Water Act: Energy and Environmental Impacts," Argonne National Laboratory Report ANL/EAIS-5, January.

Veil, J.A., M.G. Puder. D.J. Littleton, and N. Johnson, 2001, "A Holistic Look at Minimizing Adverse Environmental Impact under Section 316(b) of the Clean Water Act," Proceedings of American Fisheries Society Annual Conference, Special Symposium on Defining and Assessing Adverse Environmental Impacts, Phoenix, AZ, August 19-23.

Figure 1 – Power Pools

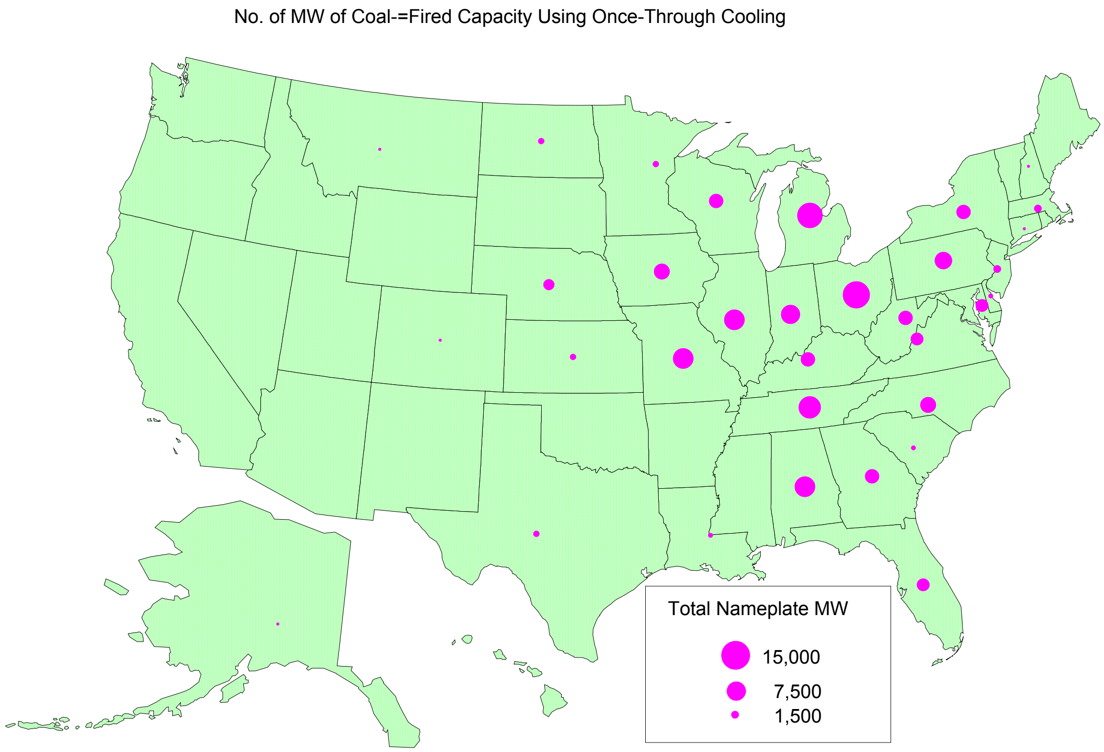




Mid-Atlantic Area Council (MAAC) Regional Power Pool

Southeastern Electric Reliability Council (SERC) Regional Power Pool

Figure 2 - Generating Capacity of Coal-Fired Units Using Once-Through Cooling by State
Source for Data: Power Statistics Data Base, Edison Electric Institute



Appendix A - Model Outputs from ASPEN Model under the One-Percent Highest Temperature Conditions

Delaware River Basin (Philadelphia, Pa - Delaware River) - 10-degree range				
Climate Conditions				
	degree F	Humidity		
Temperature - surface water	76			
Temperature - dry bulb : T_{DB}	93	0.01849	(lbs H2O/ lbs Dry Air)	
Temperature - wet bulb : T_{WB}	79			
Thermal Discharge Limit	86			
Cooling Method	Once Through	Indirect Wet	Indirect Dry (40°approach)	Indirect Dry (20°approach)
Power (MWe)				
Gross Power	431.13	422.03	390.32	404.91
Plant/Prep Auxiliaries	17.07	17.07	17.07	17.07
Steam Cycle Pumps	10.47	10.47	10.47	10.47
Cooling tower Fans	0.00	2.98	14.66	14.30
Circulating Water pumps	3.51	5.35	4.83	4.65
Net Power	400.08	386.15	343.30	358.42
Results				
Net Plant Efficiency (HHV, %)	38.48	37.14	33.02	34.47
Penalty Power (MWe)	*	13.93	56.79	41.66
Penalty Power (%)	*	3.48	14.19	10.41
Water Usage (MM Lbs/Hr)				
(Condenser + Cooling Tower)				
Cooling Water to Condenser	170.35	173.96	185.97	180.10
CT Blowdown and Drift Loss	-	3.45	1.86	1.80
CT Evaporation Loss	-	1.54	-	-
Total Water Usage (non-recir)	170.35	4.98	1.86	1.80
Condenser Conditions				
Q Heat Duty (MMBtu/Hr)	1687.62	1720.28	1834.94	1782.01
Steam to Condenser				
- Pressure (psia) / (In. Hg)	0.79 / 1.61	1.17 / 2.38	3.81 / 7.77	2.28 / 4.65
- Inlet Temp (F) : T_1	94	107	151	131
- Exit Temp (F) : T_2	91	104	148	128
Cooling Water (CW) Temperature				
- Entering (F) : T_4	76	89	133	113
- Exiting (F) : T_5	86	99	143	123
- CW Range (F) : $T_5 - T_4$	10	10	10	10
Terminal Temp Difference				
- Inlet (F) : $T_1 - T_5$	8	8	8	8
- Outlet (F) : $T_2 - T_4$	15	15	15	15

Cooling Tower Conditions	n/a	Wet Tower	Dry Tower	Dry Tower
Temperatures (F):				
- Approach to Wet Bulb : $T_3 - T_{WB}$		10	n/a	n/a
- Approach to Ambient : $T_4 - T_{DB}$		n/a	40	20
- Entering Hot Water : T_5		99	143	123
- Exiting Cool Water : T_3		89	133	113
- Entering Air : T_6		93	93	93
Entering Air Flowrate (MM Lbs/Hr)		141.74	696.65	679.57
(* - Penalties calculated relative to this case,				

Delaware River Basin (Philadelphia, Pa - Delaware River) - 15-degree range				
Climate Conditions				
	degree F			
Temperature - surface water	76			
Temperature - dry bulb : TDB	93	0.01849	(lbs H2O/ lbs Dry Air)	
Temperature - wet bulb : TWB	79			
Thermal Discharge Limit	86			
Cooling Method	Once Through	Indirect Wet	Indirect Dry (40°approach)	Indirect Dry (20°approach)
Power (MWe)				
Gross Power	430.07	420.86	388.84	403.55
Plant/Prep Auxiliaries	17.17	17.17	17.17	17.17
Steam Cycle Pumps	10.53	10.53	10.53	10.53
Cooling tower Fans	0	2.01	10.8	9.77
Circulating Water pumps	2.36	3.61	3.24	3.13
Net Power	400.02	387.56	347.1	362.95
Results				
Net Plant Efficiency (HHV, %)	38.25	37.07	33.2	34.71
Penalty Power (MWe)	*	12.46	52.91	37.07
Penalty Power (%)	*	3.12	13.23	9.27
Water Usage (MM Lbs/Hr)				
(Condenser + Cooling Tower)				
Cooling Water to Condenser	114.88	117.2	124.69	121.32
CT Blowdown and Drift Loss	-	2.31	1.25	1.21
CT Evaporation Loss	-	1.54	-	-
Total Water Usage (non-recir)	114.88	3.85	1.25	1.21
Condenser Conditions				
Q Heat Duty (MMBtu/Hr)	1709.58	1742.63	1858.62	1805.14
Steam to Condenser				
- Pressure (psia) / (In. Hg)	0.92 / 1.88	1.35 / 2.75	4.31 / 8.77	2.61 / 5.30
- Inlet Temp (F) : T1	99	112	156	136
- Exit Temp (F) : T2	96	109	153	133
Cooling Water (CW) Temperature				
- Entering (F) : T4	76	89	133	113
- Exiting (F) : T5	91	104	148	128
- CW Range (F) : T5 - T4	15	15	15	15
Terminal Temp Difference				
- Inlet (F) : T1 - T5	8	8	8	8
- Outlet (F) : T2 - T4	20	20	20	20

Cooling Tower Conditions	n/a	Wet Tower	Dry Tower	Dry Tower
Temperatures (F):				
- Approach to Wet Bulb : T3 - TWB		10	n/a	n/a
- Approach to Ambient : T4 - TDB		n/a	40	20
- Entering Hot Water : T5		104	148	128
- Exiting Cool Water : T3		89	133	113
- Entering Air : T6		93	93	93
Entering Air Flowrate (MM Lbs/Hr)		95.48	513.4	464.25
(* - Penalties calculated relative to this case,				

Delaware River Basin (Philadelphia, Pa - Delaware River) - 20-Degree Range				
Climate Conditions				
	degree F			
Temperature - surface water	76			
Temperature - dry bulb : TDB	93	0.01849	(lbs H2O/ lbs Dry Air)	
Temperature - wet bulb : TWB	79			
Thermal Discharge Limit	86			
Cooling Method	Once Through	Indirect Wet	Indirect Dry (40°approach)	Indirect Dry (20°approach)
Power (MWe)				
Gross Power	429.79	420.46	388.06	402.96
Plant/Prep Auxiliaries	17.3	17.3	17.3	17.3
Steam Cycle Pumps	10.61	10.61	10.61	10.61
Cooling tower Fans	0	1.53	8.27	7.98
Circulating Water pumps	1.8	2.74	2.47	2.38
Net Power	400.08	388.28	349.41	364.69
Results				
Net Plant Efficiency (HHV, %)	37.97	36.85	33.16	34.61
Penalty Power (MWe)	*	11.8	50.67	35.39
Penalty Power (%)	*	2.95	12.67	8.85
Water Usage (MM Lbs/Hr)				
(Condenser + Cooling Tower)				
Cooling Water to Condenser	87.37	89.14	95.25	92.25
CT Blowdown and Drift Loss	-	1.75	0.95	0.92
CT Evaporation Loss	-	1.54	-	-
Total Water Usage (non-recir)	87.37	3.3	0.95	0.92
Condenser Conditions				
Q Heat Duty (MMBtu/Hr)	1735.54	1769.05	1886.62	1832.34
Steam to Condenser				
- Pressure (psia) / (In. Hg)	1.07 / 2.18	1.56 / 3.17	4.86 / 9.88	2.97 / 6.04
- Inlet Temp (F) : T1	104	117	161	141
- Exit Temp (F) : T2	101	114	158	138
Cooling Water (CW) Temperature				
- Entering (F) : T4	76	89	133	113
- Exiting (F) : T5	96	109	153	133
- CW Range (F) : T5 - T4	20	20	20	20
Terminal Temp Difference				
- Inlet (F) : T1 - T5	8	8	8	8
- Outlet (F) : T2 - T4	25	25	25	25

Cooling Tower Conditions	n/a	Wet Tower	Dry Tower	Dry Tower
Temperatures (F):				
- Approach to Wet Bulb : T3 - TWB		10	n/a	n/a
- Approach to Ambient : T4 - TDB		n/a	40	20
- Entering Hot Water : T5		109	153	133
- Exiting Cool Water : T3		99	133	113
- Entering Air : T6		93	93	93
Entering Air Flowrate (MM Lbs/Hr)		72.63	393.12	379.24
(* - Penalties calculated relative to this case,				

Michigan/Great Lakes (Detroit, Michigan - Lake Erie) 7-degree range				
Climate Conditions				
	degree F			
Temperature - surface water	73			
Temperature - dry bulb : T_{DB}	92	0.01597	(lbs H2O/ lbs Dry Air)	
Temperature - wet bulb : T_{WB}	76			
Thermal Discharge Limit	80			
Cooling Method	Once Through	Indirect Wet	Indirect Dry (40°approach)	Indirect Dry (20°approach)
Power (MWe)				
Gross Power	432.30	423.29	390.55	404.97
Plant/Prep Auxiliaries	16.95	16.95	16.95	16.95
Steam Cycle Pumps	10.40	10.40	10.40	10.40
Cooling tower Fans	-	4.20	20.00	19.59
Circulating Water pumps	4.94	7.55	6.81	6.58
Net Power	400.00	384.19	336.40	351.45
Results				
Net Plant Efficiency (HHV, %)	38.74	37.21	32.58	34.04
Penalty Power (MWe)	*	15.82	63.60	48.55
Penalty Power (%)	*	3.95	15.90	12.14
Water Usage (MM Lbs/Hr)				
(Condenser + Cooling Tower)				
Cooling Water to Condenser	240.37	245.46	262.21	254.82
CT Blowdown and Drift Loss	-	4.88	2.62	2.55
CT Evaporation Loss	-	1.54	-	-
Total Water Usage (non-recir)	240.37	6.42	2.62	2.55
Condenser Conditions				
Q Heat Duty (MMBtu/Hr)	1661.08	1693.33	1811.57	1759.3
Steam to Condenser				
- Pressure (psia) / (In. Hg)	0.66 / 1.33	0.98 / 1.99	3.45 / 7.03	2.05 / 4.18
- Inlet Temp (F) : T_1	88	101	147	127
- Exit Temp (F) : T_2	85	98	144	124
Cooling Water (CW) Temperature				
- Entering (F) : T_4	73	86	132	112
- Exiting (F) : T_5	80	93	139	119
- CW Range (F) : $T_5 - T_4$	7	7	7	7
Terminal Temp Difference				
- Inlet (F) : $T_1 - T_5$	8	8	8	8

- Outlet (F) : $T_2 - T_4$	12	12	12	12
Cooling Tower Conditions	n/a	Wet Tower	Dry Tower	Dry Tower
Temperatures (F):				
- Approach to Wet Bulb : $T_3 - T_{WB}$		10	n/a	n/a
- Approach to Ambient : $T_4 - T_{DB}$		n/a	40	20
- Entering Hot Water : T_5		93	139	119
- Exiting Cool Water : T_3		86	132	112
- Entering Air : T_6		92	92	92
Entering Air Flowrate (MM Lbs/Hr)		199.50	950.41	931.30
(* - Penalties calculated relative to this case,				

Michigan/Great Lakes (Detroit, Michigan - Lake Erie) 15-degree range				
Climate Conditions				
	degree F			
Temperature - surface water	73			
Temperature - dry bulb : TDB	92	0.01597	(lbs H2O/ lbs Dry Air)	
Temperature - wet bulb : TWB	76			
Thermal Discharge Limit	80			
Cooling Method	Once Through	Indirect Wet	Indirect Dry (40°approach)	Indirect Dry (20°approach)
Power (MWe)				
Gross Power	429.93	420.82	387.59	402.21
Plant/Prep Auxiliaries	17.08	17.08	17.07	17.07
Steam Cycle Pumps	10.47	10.48	10.48	10.48
Cooling tower Fans	0	1.99	9.64	10.52
Circulating Water pumps	2.34	3.57	3.22	3.11
Net Power	400.04	387.72	347.19	361.03
Results				
Net Plant Efficiency (HHV, %)	38.45	37.27	33.38	34.71
Penalty Power (MWe)	*	12.32	52.85	39.01
Penalty Power (%)	*	3.08	13.21	9.75
Water Usage (MM Lbs/Hr)				
(Condenser + Cooling Tower)				
Cooling Water to Condenser	113.78	116.07	123.96	120.53
CT Blowdown and Drift Loss	-	2.29	1.24	1.21
CT Evaporation Loss	-	1.54	-	-
Total Water Usage (non-recir)	113.78	3.83	1.24	1.21
Condenser Conditions				
Q Heat Duty (MMBtu/Hr)	1693.4	1726.11	1846.4	1793.26
Steam to Condenser				
- Pressure (psia) / (In. Hg)	0.84 / 1.71	1.24 / 2.52	4.20 / 8.56	2.54 / 5.17
- Inlet Temp (F) : T1	96	109	155	135
- Exit Temp (F) : T2	93	106	153	132
Cooling Water (CW) Temperature				
- Entering (F) : T4	73	86	132	112
- Exiting (F) : T5	88	101	147	127
- CW Range (F) : T5 - T4	15	15	15	15
Terminal Temp Difference				
- Inlet (F) : T1 - T5	8	8	8	8
- Outlet (F) : T2 - T4	20	20	20	20

Cooling Tower Conditions	n/a	Wet Tower	Dry Tower	Dry Tower
Temperatures (F):				
- Approach to Wet Bulb : T3 - TWB		10	n/a	n/a
- Approach to Ambient : T4 - TDB		n/a	40	20
- Entering Hot Water : T5		101	147	127
- Exiting Cool Water : T3		86	132	112
- Entering Air : T6		92	92	92
Entering Air Flowrate (MM Lbs/Hr)		94.34	457.97	499.99
(* - Penalties calculated relative to this case,				

Michigan/Great Lakes (Detroit, Michigan - Lake Erie)			25-degree range	
Climate Conditions				
	degree F			
Temperature - surface water	73			
Temperature - dry bulb : TDB	92	0.01597	(lbs H2O/ lbs Dry Air)	
Temperature - wet bulb : TWB	76			
Thermal Discharge Limit	80			
Cooling Method	Once Through	Indirect Wet	Indirect Dry (40°approach)	Indirect Dry (20°approach)
Power (MWe)				
Gross Power	429.73	419.91	385.7	400.85
Plant/Prep Auxiliaries	17.35	17.34	17.34	17.34
Steam Cycle Pumps	10.64	10.63	10.63	10.63
Cooling tower Fans	0	1.23	6.55	6.37
Circulating Water pumps	1.45	2.2	1.98	1.92
Net Power	400.29	388.5	349.18	364.58
Results				
Net Plant Efficiency (HHV, %)	37.87	36.77	33.05	34.51
Penalty Power (MWe)	*	11.79	51.11	35.71
Penalty Power (%)	*	2.94	12.77	8.92
Water Usage (MM Lbs/Hr)				
(Condenser + Cooling Tower)				
Cooling Water to Condenser	70.27	71.59	76.41	74.32
CT Blowdown and Drift Loss	-	1.4	0.76	0.74
CT Evaporation Loss	-	1.55	-	-
Total Water Usage (non-recir)	70.27	2.95	0.76	0.74
Condenser Conditions				
Q Heat Duty (MMBtu/Hr)	1746.19	1777.97	1900.27	1846.96
Steam to Condenser				
- Pressure (psia) / (In. Hg)	1.13 / 2.31	1.65 / 3.35	5.34 / 10.87	3.28 / 6.68
- Inlet Temp (F) : T1	106	119	165	145
- Exit Temp (F) : T2	103	116	162	142
Cooling Water (CW) Temperature				
- Entering (F) : T4	73	86	132	112
- Exiting (F) : T5	98	111	157	137
- CW Range (F) : T5 - T4	25	25	25	25
Terminal Temp Difference				
- Inlet (F) : T1 - T5	8	8	8	8
- Outlet (F) : T2 - T4	30	30	30	30

Cooling Tower Conditions	n/a	Wet Tower	Dry Tower	Dry Tower
Temperatures (F):				
- Approach to Wet Bulb : T3 - TWB		10	n/a	n/a
- Approach to Ambient : T4 - TDB		n/a	40	20
- Entering Hot Water : T5		111	157	137
- Exiting Cool Water : T3		86	132	112
- Entering Air : T6		92	92	92
Entering Air Flowrate (MM Lbs/Hr)		58.18	311.5	302.96
(* - Penalties calculated relative to this case,				

Ohio River Valley (Indianapolis, Indiana - White River)			**14-degree range**	
Climate Conditions				
	degree F			
Temperature - surface water	76			
Temperature - dry bulb : T_{DB}	94	0.01733	(lbs H2O/ lbs Dry Air)	
Temperature - wet bulb : T_{WB}	78			
Thermal Discharge Limit	90			
Cooling Method	Once Through	Indirect Wet	Indirect Dry (40°approach)	Indirect Dry (20°approach)
Power (MWe)				
Gross Power	430.23	421.77	388.38	403.07
Plant/Prep Auxiliaries	17.15	17.15	17.15	17.15
Steam Cycle Pumps	10.52	10.52	10.52	10.52
Cooling tower Fans	0.00	2.14	10.89	10.46
Circulating Water pumps	2.53	3.85	3.47	3.35
Net Power	400.03	388.11	346.34	361.58
Results				
Net Plant Efficiency (HHV, %)	38.30	37.17	33.17	34.63
Penalty Power (MWe)	*	11.92	53.69	38.45
Penalty Power (%)	*	2.98	13.42	9.61
Water Usage (MMLbs/Hr)				
(Condenser + Cooling Tower)				
Cooling Water to Condenser	122.78	125.08	133.62	129.87
CT Blowdown and Drift Loss	-	2.47	1.34	1.30
CT Evaporation Loss	-	1.56	-	-
Total Water Usage (non-recir)	122.78	4.03	1.34	1.30
Condenser Conditions				
Q Heat Duty (MMBtu/Hr)	1705.11	1735.47	1856.41	1802.99
Steam to Condenser				
- Pressure (psia) / (In. Hg)	0.89 / 1.82	1.28 / 2.60	4.31 / 8.77	2.61 / 5.30
- Inlet Temp (F) : T_1	98	110	156	136
- Exit Temp (F) : T_2	95	107	153	133
Cooling Water (CW) Temperature				
- Entering (F) : T_4	76	88	134	114
- Exiting (F) : T_5	90	102	148	128
- CW Range (F) : $T_5 - T_4$	14	14	14	14
Terminal Temp Difference				
- Inlet (F) : $T_1 - T_5$	8	8	8	8

- Outlet (F) : $T_2 - T_4$	19	19	19	19
Cooling Tower Conditions	n/a	Wet Tower	Dry Tower	Dry Tower
Temperatures (F):				
- Approach to Wet Bulb : $T_3 - T_{WB}$		10	n/a	n/a
- Approach to Ambient : $T_4 - T_{DB}$		n/a	40	20
- Entering Hot Water : T_5		102	148	128
- Exiting Cool Water : T_3		88	134	114
- Entering Air : T_6		94	94	94
Entering Air Flowrate (MM Lbs/Hr)		101.80	517.57	497.19
(* - Penalties calculated relative to this case,				

Ohio River Valley (Indianapolis, Indiana - White River)			20-degree range	
Climate Conditions				
	degree F			
Temperature - surface water	76			
Temperature - dry bulb : TDB	94	0.01733	(lbs H2O/ lbs Dry Air)	
Temperature - wet bulb : TWB	78			
Thermal Discharge Limit	90			
Cooling Method	Once Through	Indirect Wet	Indirect Dry (40°approach)	Indirect Dry (20°approach)
Power (MWe)				
Gross Power	429.71	421.1	387.24	402.15
Plant/Prep Auxiliaries	17.3	17.3	17.3	17.3
Steam Cycle Pumps	10.61	10.61	10.61	10.61
Cooling tower Fans	0	1.52	8.13	7.97
Circulating Water pumps	1.8	2.74	2.47	2.39
Net Power	400	388.93	348.73	363.88
Results				
Net Plant Efficiency (HHV, %)	37.97	36.92	33.11	34.54
Penalty Power (MWe)	*	11.07	51.27	36.12
Penalty Power (%)	*	2.77	12.82	9.03
Water Usage (MMLbs/Hr)				
(Condenser + Cooling Tower)				
Cooling Water to Condenser	87.41	88.99	95	92.36
CT Blowdown and Drift Loss	-	1.75	0.95	0.92
CT Evaporation Loss	-	1.56	-	-
Total Water Usage (non-recir)	87.41	3.31	0.95	0.92
Condenser Conditions				
Q Heat Duty (MMBtu/Hr)	1735.24	1766.15	1889.05	1834.71
Steam to Condenser				
- Pressure (psia) / (In. Hg)	1.07 / 2.18	1.51 / 3.08	4.97 / 10.1	3.04 / 6.19
- Inlet Temp (F) : T1	104	116	162	142
- Exit Temp (F) : T2	101	113	159	139
Cooling Water (CW) Temperature				
- Entering (F) : T4	76	88	134	114
- Exiting (F) : T5	96	108	154	134
- CW Range (F) : T5 - T4	20	20	20	20
Terminal Temp Difference				
- Inlet (F) : T1 - T5	8	8	8	8
- Outlet (F) : T2 - T4	25	25	25	25

Cooling Tower Conditions	n/a	Wet Tower	Dry Tower	Dry Tower
Temperatures (F):				
- Approach to Wet Bulb : T3 - TWB		10	n/a	n/a
- Approach to Ambient : T4 - TDB		n/a	40	20
- Entering Hot Water : T5		108	154	134
- Exiting Cool Water : T3		88	134	114
- Entering Air : T6		94	94	94
Entering Air Flowrate (MM Lbs/Hr)		72.43	386.44	378.93
(* - Penalties calculated relative to this case,				

South (Atlanta, Georgia - Chattahoochee River)			10-degree range	
Climate Conditions				
	degree F			
Temperature - surface water	79			
Temperature - dry bulb : TDB	95	0.01712	(lbs H2O/ lbs Dry Air)	
Temperature - wet bulb : TWB	78			
Thermal Discharge Limit	84			
Cooling Method	Once Through	Indirect Wet	Indirect Dry (40°approach)	Indirect Dry (20°approach)
Power (MWe)				
Gross Power	431.26	424.95	390.89	405.59
Plant/Prep Auxiliaries	17.18	17.17	17.17	17.17
Steam Cycle Pumps	10.52	10.53	10.53	10.53
Cooling tower Fans	0	2.99	16.41	14.26
Circulating Water pumps	3.54	5.37	4.85	4.69
Net Power	400.01	388.88	341.92	358.93
Results				
Net Plant Efficiency (HHV, %)	38.27	37.21	32.72	34.34
Penalty Power (MWe)	*	11.13	58.09	41.08
Penalty Power (%)	*	2.78	14.52	10.27
Water Usage (MM Lbs/Hr)				
(Condenser + Cooling Tower)				
Cooling Water to Condenser	172.05	174.62	186.81	181.57
CT Blowdown and Drift Loss	-	3.461	1.868	1.816
CT Evaporation Loss	-	1.587	-	-
Total Water Usage (non-recir)	172.05	5.048	1.868	1.816
Condenser Conditions				
Q Heat Duty (MMBtu/Hr)	1704.15	1726.83	1850.04	1796.65
Steam to Condenser				
- Pressure (psia) / (In. Hg)	0.87 / 1.76	1.13 / 2.31	4.00 / 8.15	2.41 / 4.90
- Inlet Temp (F) : T1	97	106	153	128
- Exit Temp (F) : T2	94	103	150	125
Cooling Water (CW) Temperature				
- Entering (F) : T4	79	88	135	115
- Exiting (F) : T5	89	98	145	125
- CW Range (F) : T5 - T4	10	10	10	10
Terminal Temp Difference				

- Inlet (F) : T1 - T5	8	8	8	8
- Outlet (F) : T2 - T4	15	15	15	15
Cooling Tower Conditions		Wet Tower	Dry Tower	Dry Tower
Temperatures (F):				
- Approach to Wet Bulb : T3 - TWB	n/a	10	n/a	n/a
- Approach to Ambient : T4 - TDB		n/a	40	20
- Entering Hot Water : T5		98	145	125
- Exiting Cool Water : T3		88	135	115
- Entering Air : T6		95	95	95
Entering Air Flowrate (MM Lbs/Hr)		142.09	779.82	677.97
(* - Penalties calculated relative to this case,				

Region 4 - South (Atlanta, Georgia - Chattahoochee River) **15-degree range**

Climate Conditions

	degree F	
Temperature - surface water	79	
Temperature - dry bulb : T_{DB}	95	0.01712 (lbs H2O/ lbs Dry Air)
Temperature - wet bulb : T_{WB}	78	
Thermal Discharge Limit	84	

Cooling Method	**Case 1** Once Thru	**Case 2** Indirect Wet	**Case 3a** Indirect Dry	**Case 3b **** Indirect Dry
Power (MWe)				
Gross Power	430.26	423.88	389.47	404.29
Plant/Prep Auxiliaries	17.28	17.27	17.27	17.27
Steam Cycle Pumps	10.59	10.59	10.59	10.59
Cooling tower Fans	-	2.01	10.89	10.57
Circulating Water pumps	2.39	3.62	3.27	3.16
Net Power	400.01	390.38	347.44	362.69
Results				
Net Plant Efficiency (HHV, %)	38.05	37.14	33.05	34.50
Penalty Power (MWe)	*	9.63	52.57	37.32
Penalty Power (%)	*	2.41	13.14	9.33
Water Useage (MM Lbs/Hr)				
(Condenser + Cooling Tower)				
Cooling Water to Condenser	116.04	117.65	125.8	122.38
CT Blowdown and Drift Loss	-	2.321	1.258	1.224
CT Evaporation Loss	-	1.582	-	-
Total Water Useage (non-recir)	116.04	3.903	1.258	1.224
Condenser Conditions				
Q Heat Duty (MMBtu/Hr)	1726.69	1749.64	1874.31	1820.37
Steam to Condenser				
- Pressure (psia) / (In. Hg)	1.01 / 2.05	1.31 / 2.67	4.52 / 9.20	2.74 / 5.58
- Inlet Temp (F) : T_1	102	111	158	128
- Exit Temp (F) : T_2	99	108	155	125
Cooling Water (CW) Temperature				
- Entering (F) : T_4	79	88	135	115

- Exiting (F) : T_5	94	103	150	130
- CW Range (F) : $T_5 - T_4$	15	15	15	15
Terminal Temp Difference				
- Inlet (F) : $T_1 - T_5$	8	8	8	8
- Outlet (F) : $T_2 - T_4$	20	20	20	2-
Cooling Tower Conditions		Wet Tower	Dry Tower	Dry Tower
Temperatures (F):				
- Approach to Wet Bulb : $T_3 - T_{WB}$	n/a	10	n/a	n/a
- Approach to Ambient : $T_4 - T_{DB}$		n/a	40	20
- Entering Hot Water : T_5		103	150	130
- Exiting Cool Water : T_3		88	135	115
- Entering Air : T_6		95	95	95
Entering Air Flowrate (MM Lbs/Hr)		95.73	517.56	502.27

(* - Penalties calculated relative to this case,

** - Case 3b changes the "approach to ambient temperature from Case 3a)

Southwest (Yuma, Arizona - Colorado River)			**10-degree range**	
Climate Conditions				
	degree F			
Temperature - surface water	82			
Temperature - dry bulb	111	0.01469	(lbs H2O/ lbs Dry Air)	
Temperature - wet bulb	79			
Thermal Discharge Limit	92			
Cooling Method	Once Through	Indirect Wet	Indirect Dry (40°approach)	Indirect Dry (20°approach)
Power (MWe)				
Gross Power	431.54	426.54	380.98	395.37
Plant/Prep Auxiliaries	17.38	17.38	17.38	17.38
Steam Cycle Pumps	10.58	10.58	10.58	10.58
Cooling tower Fans	n/a	3.01	16.99	14.78
Circulating Water pumps	3.58	5.4	5.04	4.87
Net Power	400	390.17	330.99	347.76
Results				
Net Plant Efficiency (HHV, %)	38.08	37.15	31.52	33.11
Penalty Power (MWe)	*	9.84	69.01	52.24
Penalty Power (%)	*	2.46	17.25	13.06
Water Usage (MM Lbs/Hr)				
(Condenser + Cooling Tower)				
Cooling Water to Condenser	173.64	175.5	192.97	187.58
CT Blowdown and Drift Loss	-	3.473	1.929	1.876
CT Evaporation Loss	-	1.816	-	-
Total Water Usage (non-recir)	173.64	5.289	1.929	1.876
Condenser Conditions				
Q Heat Duty (MMBtu/Hr)	1720.86	1738.75	1901.91	1849.89
Steam to Condenser				
- Pressure (psia) / (In. Hg)	0.95 / 1.93	1.17 / 2.38	5.86 / 11.92	3.63 / 7.39
- Inlet Temp (F) : T1	100	107	169	149
- Exit Temp (F) : T2	97	104	166	146
Cooling Water (CW) Temperature				
- Entering (F) : T4	82	89	151	131
- Exiting (F) : T5	92	99	161	141
- CW Range (F) : T5 - T4	10	10	10	10
Terminal Temp Difference				
- Inlet (F) : T1 - T5	8	8	8	8
- Outlet (F) : T2 - T4	15	15	15	15

Cooling Tower Conditions		Wet Tower	Dry Tower	Dry Tower
Temperatures (F):				
- Approach to Wet Bulb : T3 - TWB	n/a	10	n/a	n/a
- Approach to Ambient : T4 - TDB		n/a	40	20
- Entering Hot Water : T5		99	161	141
- Exiting Cool Water : T3		89	151	131
- Entering Air : T6		111	111	111
Entering Air Flowrate (MM Lbs/Hr)		142.47	807.5	702.5
(* - Penalties calculated relative to this case,				

15-degree range

Climate Conditions

	degree F		
Temperature - surface water	82		
Temperature - dry bulb	111	0.01469	(lbs H2O/ lbs Dry Air)
Temperature - wet bulb	79		
Thermal Discharge Limit	92		

Cooling Method	**Case 1** Once Thru	**Case 2** Indirect Wet	**Case 3a** Indirect Dry	**Case 3b **** Indirect Dry
Power (MWe)				
Gross Power	430.58	425.59	379.64	394.18
Plant/Prep Auxiliaries	17.49	17.48	17.48	17.48
Steam Cycle Pumps	10.65	10.65	10.65	10.65
Cooling tower Fans	0.00	2.03	11.27	10.92
Circulating Water pumps	2.41	3.65	3.37	3.26
Net Power	400.04	391.78	336.86	351.86
Results				
Net Plant Efficiency (HHV, %)	37.85	37.08	31.88	33.30
Penalty Power (MWe)	*	8.26	63.17	48.18
Penalty Power (%)	*	2.06	15.79	12.04
Water Useage (MM Lbs/Hr)				
(Condenser + Cooling Tower)				
Cooling Water to Condenser	117.12	118.46	129.24	125.685
CT Blowdown and Drift Loss	-	2.333	1.292	1.257
CT Evaporation Loss	-	1.788	-	-
Total Water Useage (non-recir)	117.12	4.121	1.292	1.257
Condenser Conditions				
Q Heat Duty (MMBtu/Hr)	1744.33	1762.28	1926.91	1874.26
Steam to Condenser				
- Pressure (psia) / (In. Hg)	1.10 / 2.24	1.35 / 2.75	6.57 / 13.37	4.10 / 8.35
- Inlet Temp (F) : T_1	105	112	174	154
- Exit Temp (F) : T_2	102	109	171	151
Cooling Water (CW) Temperature				
- Entering (F) : T_4	82	89	151	131
- Exiting (F) : T_5	97	104	166	146

- CW Range (F) : $T_5 - T_4$	15	15	15	15
Terminal Temp Difference				
- Inlet (F) : $T_1 - T_5$	8	8	8	8
- Outlet (F) : $T_2 - T_4$	20	20	20	20
Cooling Tower Conditions		Wet Tower	Dry Tower	Dry Tower
Temperatures (F):				
- Approach to Wet Bulb : $T_3 - T_{WB}$	n/a	10	n/a	n/a
- Approach to Ambient : $T_4 - T_{DB}$		n/a	40	20
- Entering Hot Water : T_5		104	166	146
- Exiting Cool Water : T_3		89	151	131
- Entering Air : T_6		111	111	111
Entering Air Flowrate (MM Lbs/Hr)		96.16	535.82	519.13

(* - Penalties calculated relative to this case,
** - Case 3b changes the "approach to ambient temperature from Case 3a)