Chapter 3: Energy Penalties, Air Emissions, and Cooling Tower Side-Effects

Introduction

This chapter discusses the topics of energy penalties, air emissions, and other environmental impacts of cooling tower systems. The final rule projects that nine new facility power plants will install recirculating closed-cycle wet cooling systems as a result of this rule. These systems, mainly represented by natural-draft wet cooling towers, may present trade-offs in energy efficiency, associated air emissions increases, and some other environmental issues.

The energy penalty is an important and controversial topic for the electricity generation industry. The topic is widely discussed and debated, yet precise theoretical or empirical measures of energy penalties were not readily available to met the Agency's needs. Therefore, the Agency researched and derived energy penalty estimates, based on empirical data and proven theoretical concepts, for a variety of conditions. This chapter presents the research, methodology, public comments, results, and conclusions for the Agency's thorough effort to estimate energy penalties due to the operational performance of power plant cooling systems.

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As a consequence of energy penalties for some cooling systems, increased air pollutant emissions may occur for some power plants as compared to a baseline system. This chapter presents estimates of the increased air emissions for the four key pollutants that are currently well researched and monitored for at power plants in the United States: carbon dioxide (CO2), sulfur dioxide (SO2), nitrogen oxides (NOx), and mercury (Hg).

The remainder of this chapter is organized as follows:

< Section 3.1 presents the energy penalty estimates developed for the final rule and the dry cooling regulatory alternative.

< Section 3.2 presents the air emissions estimates developed for the final rule and the dry cooling regulatory alternative.

< Section 3.3 presents the background, research, and methodology of the energy penalty evaluation. The section focuses on power plants that use steam turbines and the changes in efficiency associated with using alternative cooling systems.

< Section 3.4 presents the methodology for estimation of air emissions increases.

< Section 3.5 discusses side effects of recirculating wet cooling towers, such as vapor plumes, displacement of habitat or wetlands, noise, salt or mineral drift, water consumption through evaporation, and solid waste generation due to wastewater treatment of tower blowdown.

3.1 Energy Penalty Estimates for Cooling

Tables 3-1 through 3-6 present the energy penalty estimates developed for the final rule and the dry cooling regulatory alternative. The Agency presents the methodology for estimation of energy penalties in Section 3.3 of this chapter.

Table 3-1: National Average Annual Energy Penalty, Summary Table

Cooling Type	Percent Maximum Load[a]	Nuclear Percent of Plant Output	Combined-Cycle Percent of Plant Output	Fossil-Fuel Percent of Plant Output
Wet Tower vs. Once-Through	67	1.7	0.4	1.7
Dry Tower vs. Once-Through	67	8.5	2.1	8.6
Dry Tower vs. Wet Tower	67	6.8	1.7	6.9

[a] Average annual penalties occur at non-peak loads..

Table 3-2: National Peak Summer Energy Penalty, Summary Table

Cooling Type	Percent Maximum Load[a]	Nuclear Percent of Plant Output	Combined-Cycle Percent of Plant Output	Fossil-Fuel Percent of Plant Output
Wet Tower vs. Once-Through	100	1.9	0.4	1.7
Dry Tower vs. Once-Through	100	11.4	2.8	10.0
Dry Tower vs. Wet Tower	100	9.6	2.4	8.4

[a] Peak-summer shortfalls occur when plants are at or near maximum capacity.

Table 3-3: Total Energy Penalties at 67 Percent Maximum Load[a]

Location	Cooling Type	Nuclear Annual Average	Combined-Cycle Annual Average	Fossil-Fuel Annual Average
Boston	Wet Tower vs. Once-Through	1.6	0.4	1.6
	Dry Tower vs. Once-Through	7.4	1.8	7.1
	Dry Tower vs. Wet Tower	5.8	1.4	5.5
Jacksonville	Wet Tower vs. Once-Through	1.9	0.4	1.7
	Dry Tower vs. Once-Through	12.0	3.0	12.5
	Dry Tower vs. Wet Tower	10.1	2.5	10.8
Chicago	Wet Tower vs. Once-Through	1.8	0.4	1.8
	Dry Tower vs. Once-Through	7.8	1.9	7.7
	Dry Tower vs. Wet Tower	5.9	1.5	5.9
Seattle	Wet Tower vs. Once-Through	1.5	0.4	1.5
	Dry Tower vs. Once-Through	7.0	1.7	6.9
	Dry Tower vs. Wet Tower	5.5	1.3	5.4

[a] Average annual penalties occur at non-peak loads.

Table 3-4: Total Energy Penalties at 100 Percent Maximum Load[a]

Location	Cooling Type	Nuclear Percent of Plant Output	Combined-Cycle Percent of Plant Output	Fossil-Fuel Percent of Plant Output
Boston	Wet Tower vs. Once-Through	2.1	0.5	1.9
	Dry Tower vs. Once-Through	11.6	2.9	10.2
	Dry Tower vs. Wet Tower	9.5	2.4	8.3
Jacksonville	Wet Tower vs. Once-Through	1.6	0.4	1.4
	Dry Tower vs. Once-Through	12.3	3.1	10.7
	Dry Tower vs. Wet Tower	10.7	2.7	9.3
Chicago	Wet Tower vs. Once-Through	2.2	0.5	2.0
	Dry Tower vs. Once-Through	11.9	2.9	10.4
	Dry Tower vs. Wet Tower	9.6	2.4	8.4
Seattle	Wet Tower vs. Once-Through	1.6	0.4	1.5
	Dry Tower vs. Once-Through	10.0	2.4	8.9
	Dry Tower vs. Wet Tower	8.4	2.0	7.4

[a] Peak-summer shortfalls occur when plants are at or near maximum capacity.

Table 3-5: Annual Penalties (in MW) for the Final Rule by Online Year[a]

Year	Coal-Fired Once-Through Cooling at Baseline	Combined-Cycle, Once-Through Cooling at Baseline
2001		
2002		
2003		
2004		4
2005	70	
2006		
2007	9	4
2008	1	
2009		
2010		4
2011		
2012		
2013		4
2014		
2015		
2016		
2017		4
2018		
2019		
2020		
Total	79	21

[a] The total energy penalty for the final rule is 100 MW, or 0.027 percent of all new generating capacity in the US over the next twenty years.

Table 3-6: Annual Penalties (in MW) for the Dry Cooling-Based Alternative by Online Year[a]

	Coal-Fired			Combined-Cycle		
Year	Recirculating Wet Cooling Baseline		Once-Through Baseline	Recirculating Wet Cooling Baseline		Once-Through Baseline
	Freshwater	Estuary	Freshwater	Freshwater	Estuary	Estuary
2001						
2002						
2003						
2004						22
2005			362	71	8	
2006	164			54	17	
2007	164	56	44	40		22
2008			5	77	8	
2009	108			46		
2010				61		22
2011				102	8	
2012				38		
2013				33		22
2014				54	8	
2015				35		
2016				34		
2017				30		22
2018				37	8	
2019	43			37		
2020	12			31		
Total	491	56	412	779	58	108

[a] The total energy penalty for the dry cooling option (at a total of 83 potentially impacted plants) would be 1900 MW, or 0.5 percent of all new capacity in the US over the next twenty years.

3.2 Air Emissions Estimates for Cooling Systems Upgrades

Tables 3-7 and 3-8 present the incremental air emissions estimates developed for the final rule and the dry cooling regulatory alternative. The Agency presents the methodology for estimation of air emissions increases in section 3.4 of this Chapter.

Table 3-7: Air Emissions Increases for the Final Rule[a]

Fuel Type	Total Effected Capacity (MW)	Annual CO_2 (tons)	Annual SO_2 (tons)	Annual NO_X (tons)	Annual Hg (lbs)
All	9,957	485,860	2,561	1,214	16

[a] These emissions increases represent an increase for the entire US electricity generation industry of approximately 0.02 percent per pollutant.

Table 3-8: Air Emissions Increases for the a Dry Cooling-Based Alternative[a]

Fuel Type	Total Effected Capacity (MW)	Annual CO_2 (tons)	Annual SO_2 (tons)	Annual NO_X (tons)	Annual Hg (lbs)
All	64,070	8,931,056	47,074	22,313	300

[a] These emissions increases represent an increase for the US electricity generation industry of approximately 0.35 percent. For the mercury emissions alone, these emissions are equivalent to the addition of three 800-MW coal-fired power plants operating at near full capacity.

3.3 Background, Research, and Methodology of Energy Penalty Estimates

This energy penalty discussion references the differences in steam power plant efficiency or output associated with the effect of using alternative cooling systems. In particular, this evaluation focuses on power plants that use steam turbines and the changes in efficiency associated with using alternative cooling systems. The cooling systems evaluated include: once-through cooling systems; wet tower closed-cycle systems; and dry cooling systems using air cooled condensers. However, the methodology is flexible as to be extended to other alternative types of cooling systems so long as the steam condenser performance or the steam turbine exhaust pressure can be estimated. A summary and discussion of public comments on EPA's energy penalty analysis is presented in Attachment F to this chapter.

3.3.1 Power Plant Efficiencies

Most power plants that use a heat-generating fuel as the power source use a steam cycle referred to as a "Rankine Engine," in which water is heated into steam in a boiler and the steam is then passed through a turbine (Woodruff 1998). After exiting the turbine, the spent steam is condensed back into water and pumped back into the boiler to repeat the cycle. The turbine, in turn, drives a generator that produces electricity. As with any system that converts energy from one form to another, not all of the energy available from the fuel source can be converted into useful energy in a power plant.

Steam turbines extract power from steam as the steam passes from high pressure and high temperature conditions at the turbine inlet to low pressure and lower temperature conditions at the turbine outlet. Steam exiting the turbine goes to the condenser, where it is condensed to water. The condensation process is what creates the low pressure conditions at the turbine outlet. The steam turbine outlet or exhaust pressure (which is often a partial vacuum) is a function of the temperature maintained at the condensing surface (among other factors) and the value of the exhaust pressure can have a direct effect on the energy available to drive the turbine. The lower the exhaust pressure, the greater the amount of energy that is available to drive the turbine, which in turn increases the overall efficiency of the system since no additional fuel energy is involved.

The temperature of the condensing surface is dependent on the design and operating conditions within the condensing system (e.g., surface area, materials, cooling fluid flow rate, etc.) and especially the temperature of the cooling water or air used to absorb heat and reject it from the condenser. Thus, the use of a different cooling system can affect the temperature maintained at the steam condensing surface (true in many circumstances). This difference can result in a change in the efficiency of the power plant. These efficiency differences vary throughout the year and may be more pronounced during the warmer months. Equally important is the fact that most alternative cooling systems will require a different amount of power to operate equipment such as fans and pumps, which also can have an effect on the overall plant energy efficiency. The reductions in energy output resulting from the energy required to operate the cooling system equipment are often referred to as parasitic losses.

In general, the penalty described here is only associated with power plants that utilize a steam cycle for power production. Therefore, this analysis will focus only on steam turbine power plants and combined-cycle gas plants. The most common steam turbine power plants are those powered by steam generated in boilers heated by the combustion of fossil fuels or by nuclear reactors.

Combined-cycle plants use a two-step process in which the first step consists of turbines powered directly by high pressure hot gases from the combustion of natural gas, oil, or gasified coal. The second step consists of a steam cycle in which a turbine is powered by steam generated in a boiler heated by the low pressure hot gases exiting the gas turbines. Consequently, the combined-cycle plants have much greater overall system efficiencies. However, the energy penalty associated with using alternative cooling systems is only associated with the steam cycle portion of the system. Because steam plants cannot be quickly started or stopped, they tend to be operated as base load plants which are continuously run to serve the minimum load required by the system. Since combined-cycle plants obtain only a portion of their energy from the slow-to-start/stop steam power step, the inefficiency of the start-up/stop time period is more economically acceptable and therefore they are generally used for intermediate loads. In other words, they are started and stopped at a greater frequency than base load steam plant facilities.

One measure of the plant thermal efficiency used by the power industry is the Net Plant Heat Rate (NPHR), which is the ratio of the total fuel heat input (BTU/hr) divided by the net electric generation (kW). The net electric generation includes only electricity that leaves the plant. The total energy plant efficiency can be calculated from the NPHR using the following formula:

$$\text{Plant Energy Efficiency} = 3473 / \text{NPHR} \times 100 \qquad (1)$$

Table 3-9 presents the NPHR and plant efficiency numbers for different types of power plants. Note that while there may be some differences in efficiencies for steam turbine systems using different fossil fuels, these differences are not significant enough for consideration here. The data presented to represent fossil fuel plants is for coal-fired plants, which comprise the majority in that category.

Table 3-9: Heat Rates and Plant Efficiencies for Different Types of Steam Powered Plants

Type of Plant	Net Plant Heat Rate (BTU/kWh)	Efficiency (%)
Steam Turbine - Fossil Fuel	9,355	37 to 40
Steam Turbine – Nuclear	10,200	34
Combined Cycle – Gas	6,762	51
Combustion Turbine	11,488	30

Source: Analyzing Electric Power Generation under the CAAA. Office of Air and Radiation U.S. Environmental Protection Agency. April 1996 (Projections for year 2000-2004).

Overall, fossil fuel steam electric power plants have net efficiencies with regard to the available fuel heat energy ranging from 37 to 40 percent. Attachment A at the end of this chapter (Ishigai, S. 1999.) shows a steam power plant heat diagram in which approximately 40 percent of the energy is converted to the power output and 44 percent exits the system through the condensation of the turbine exhaust steam, which exits the system primarily through the cooling system with the remainder exiting the system through various other means including exhaust gases. Note that the exergy diagram in Attachment A shows that this heat passing through the condenser is not a significant source of plant inefficiency, but as would be expected it shows a similar percent of available energy being converted to power as shown in Table 3-9 and Attachment A.

Nuclear plants have a lower overall efficiency of approximately 34 percent, due to the fact that they generally operate at lower boiler temperatures and pressures and the fact that they use an additional heat transfer loop. In nuclear plants, heat is extracted from the core using a primary loop of pressurized liquid such as water. The steam is then formed in a secondary boiler system. This indirect steam generation arrangement results in lower boiler temperatures and pressures, but is deemed necessary to provide for safer operation of the reactor and to help prevent the release of radioactive substances. Nuclear reactors generate a near constant heat output when operating and therefore tend to produce a near constant electric output.

Combustion turbines are shown here for comparative purposes only. Combustion turbine plants use only the force of hot gases produced by combustion of the fuel to drive the turbines. Therefore, they do not require much cooling water since they do not use steam in the process, but they are also not as efficient as steam plants. They are, however, more readily able to start and stop quickly and therefore are generally used for peaking loads.

Combined cycle plants have the highest efficiency because they combine the energy extraction methods of both combustion turbine and steam cycle systems. Efficiencies as high as 58 percent have been reported (Woodruff 1998). Only the efficiency of the second stage (which is a steam cycle) is affected by cooling water temperatures. Therefore, for the purposes of this analysis, the energy penalty for combined cycle plants is applicable only to the energy output of the steam plant component, which is generally reported to be approximately one-third of the overall combined-cycle plant energy output.

3.3.2 Turbine Efficiency Energy Penalty

a. Effect of Turbine Exhaust Pressure

The temperature of the cooling water (or air in air-cooled systems) entering the steam cycle condensers affects the exhaust pressure at the outlet of the turbine. In general, a lower cooling water or air temperature at the condenser inlet will result in a lower turbine exhaust pressure. Note that for a simple steam turbine, the available energy is equal to the difference in the enthalpy of the inlet steam and the combined enthalpy of the steam and condensed moisture at the turbine outlet. A reduction in the outlet steam pressure results in a lower outlet steam enthalpy. A reduction in the enthalpy of the turbine exhaust steam, in combination with an increase in the partial condensation of the steam, results in an increase in the efficiency of the turbine system. Of course, not all of this energy is converted to the torque energy (work) that is available to turn the generator, since steam and heat flow through the turbine systems is complex with various losses and returns throughout the system.

The turbine efficiency energy penalty as described below rises and drops in direct response to the temperature of the cooling water (or air in air-cooled systems) delivered to the steam plant condenser. As a result, it tends to peak during the summer and may be substantially diminished or not exist at all during other parts of the year.

The design and operation of the steam condensing system can also affect the system efficiency. In general, design and operational changes that improve system efficiency such as greater condenser surface areas and coolant flow rates will tend to result in an increase in the economic costs and potentially the environmental detriments of the system. Thus, the design and operation of individual systems can differ depending on financial decisions and other site-specific conditions. Consideration of such site-specific design variations is beyond the scope of this evaluation. Therefore, conditions that represent a typical, or average, system derived from available information for each technology will be used. However, regional and annual differences in cooling fluid temperatures are considered. Where uncertainty exists, a conservative estimate is used. In this context, conservative means the penalty estimate is biased toward a higher value.

Literature sources indicate that condenser inlet temperatures of 55 °F and 95 °F will produce turbine exhaust pressures of 1.5 and 3.5 inches Hg, respectively, in a typical surface condenser (Woodruff 1998). If the turbine steam inlet conditions remain constant, lower turbine exhaust pressures will result in greater changes in steam enthalpy between the turbine inlet and outlet. This in turn will result in higher available energy and higher turbine efficiencies.

The lower outlet pressures can also result in the formation of condensed liquid water within the low pressure end of the turbine. Note that liquid water has a significantly lower enthalpy value which, based on enthalpy alone, should result in even greater turbine efficiencies. However, the physical effects of moisture in the turbines can cause damage to the turbine blades and can result in lower efficiencies than would be expected based on enthalpy data alone. This damage and lower efficiency is due to the fact that the moisture does not follow the steam path and impinges upon the turbine blades. More importantly, as the pressure in the turbine drops, the steam volume increases. While the turbines are designed to accommodate this increase in volume through a progressive increase in the cross-sectional area, economic considerations tend to limit the size increase such that the turbine cannot fully accommodate the expansion that occurs at very low exhaust pressures.

Thus, for typical turbines, as the exhaust pressure drops below a certain level, the increase in the volume of the steam is not fully accommodated by the turbine geometry, resulting in an increase in steam velocity near the turbine exit. This increase in steam velocity results in the conversion of a portion of the available steam energy to kinetic energy, thus reducing the energy that could otherwise be available to drive the turbine. Note that kinetic energy is proportional to the square of the velocity. Consequently, as the steam velocity increases, the resultant progressive reduction in available energy tends to offset the gains in available energy that would result from the greater enthalpy changes due to the reduced pressure. Thus, the expansion of the steam within the turbine and the formation of condensed moisture establishes a practical lower limit for turbine exhaust pressures, reducing the efficiency advantage of even lower condenser surface temperatures particularly at higher turbine steam loading rates. As can be seen in the turbine performance curves presented below, this reduction in efficiency at lower exhaust pressures is most pronounced at higher turbine steam loading rates. This is due to the fact that higher steam loading rates will produce proportionately higher turbine exit velocities.

Attachment B presents several graphs showing the change in heat rate resulting from differences in the turbine exhaust pressure at a nuclear power plant, a fossil fueled power plant, and a combined-cycle power plant (steam portion). The first graph (Attachment B-1) is for a GE turbine and was submitted by the industry in support of an analysis for a nuclear power plant. The second graph (Attachment B-2) is from a steam turbine technical manual and is for a turbine operating at steam temperatures and pressures consistent with a sub-critical fossil fuel plant (2,400 psig, 1,000 °F). The third graph (Attachment B-3) is from an engineering report analyzing operational considerations and design of modifications to a cooling system for a combined-cycle power plant.

The changes in heat rate shown in the graphs can be converted to changes in turbine efficiency using Equation 1. Several curves on each graph show that the degree of the change (slope of the curve) decreases with increasing loads. Note that the amount of electricity being generated will also vary with the steam loading rates such that the more pronounced reduction in efficiency at lower steam loading rates applies to a reduced power output. The curves also indicate that, at higher steam loads, the plant efficiency optimizes at an exhaust pressure of approximately 1.5 inches Hg. At lower exhaust pressures the effect of increased steam velocities actually results in a reduction in overall efficiency. The graphs in Attachment B will serve as the basis for estimating the energy penalty for each type of facility.

Since the turbine efficiency varies with the steam loading rate, it is important to relate the steam loading rates to typical operating conditions. It is apparent from the heat rate curves in Attachment B that peak loading, particularly if the exhaust pressure is close to 1.5 inches Hg, presents the most efficient and desirable operating condition. Obviously, during peak loading periods, all turbines will be operating near the maximum steam loading rates and the energy penalty derived from the maximum loading curve would apply. It is also reasonable to assume that power plants that operate as base load facilities will operate near maximum load for a majority of the time they are operating. However, there will be times when the power plant is not operating at peak capacity. One measure of this is the capacity factor, which is the ratio of the average load on the plant over a given period to its total capacity. For example, if a 200 MW plant operates, on average, at 50 percent of capacity (producing an average of 100 MW when operating) over a year, then its capacity factor would be 50 percent.

The average capacity factor for nuclear power plants in the U.S. has been improving steadily and recently has been reported to be approximately 89 percent. This suggests that for nuclear power plants, the majority appear to be operating near capacity most of the time. Therefore, use of the energy penalty factors derived from the maximum load curves for nuclear power plants is reasonably valid. In 1998, utility coal plants operated at an average capacity of 69 percent (DOE 2000). Therefore, use of the energy penalty values derived from the 67 percent load curves would appear to be more appropriate for fossil-fuel plants. Capacity factors for combined-cycle plants tend to be lower than coal-fired plants and use of the energy penalty values derived from the 67 percent load curves rather than the 100 percent load curves would be appropriate.

b. Estimated Changes in Turbine Efficiency

Table 3-10 below presents a summary of steam plant turbine inlet operating conditions for various types of steam plants described in literature. EPA performed a rudimentary estimation of the theoretical energy penalty based on steam enthalpy data using turbine inlet conditions similar to those shown in Table 3-10. EPA found that the theoretical values were similar to the changes in plant efficiency derived from the changes in heat rate shown in Attachment B. The theoretical calculations indicated that the energy penalties for the two different types of fossil fuel plants (sub-critical and super-critical) were similar in value, with the sub-critical plant having the larger penalty. Since the two types of fossil fuel plants had similar penalty values, only one was selected for use in the analysis in order to simplify the analysis. The type of plant with the greater penalty value (i.e., sub-critical fossil fuel) was selected as representative of both types.

Table 3-10: Summary of Steam Plant Operating Conditions from Various Sources

System Type	Inlet Temp. / Pressure	Outlet Pressure	Comments	Source
Fossil Fuel - Sub-critical Recirculating Boiler	Not Given / 2,415 psia	1.5 In. Hg	Large Plants (>500MW) have three (high, med, low) pressure turbines. Reheated boiler feed water is 540 °F.	Kirk-Othmer 1997
Fossil Fuel - Super-critical Once-through Boiler	1,000 °F / 3,515 psia	Not Given		Kirk-Othmer 1997
Nuclear	595 °F / 900 psia	2.5 In. Hg	Plants have two (high, low) pressure turbines with low pressure turbine data at left. Reheated boiler feed water is 464 °F.	Kirk-Othmer 1997
Combined Cycle	Gas - 2,400 °F Steam - 900 °F	Not Given	Operating efficiency ranges from 45-53%	www.greentie.org
Fossil Fuel Ranges	900-1,000 °F / 1,800-3,600 psia	1.0-4.5 In Hg	Outlet pressures can be even higher with high cooling water temperatures or air cooled condensers.	Woodruff 1998.

The three turbine performance curve graphs in Attachment B present the change in heat rate from which changes in plant efficiency were calculated. The change in heat rate value for several points along each curve was determined and then converted to changes in efficiency using Equation 1. The calculated efficiency values derived from the Attachment B graphs representing the 100 percent or maximum steam load and the 67 percent steam load conditions have been plotted in Figure 1. Curves were then fitted to these data to obtain equations that can be used to estimate energy penalties. Figure 1 establishes the energy efficiency and turbine exhaust pressure relationship. The next step is to relate the turbine exhaust pressure to ambient conditions and to determine ambient conditions for selected locations.

Note that for fossil fuel plants the energy penalty affects mostly the amount of fuel used, since operating conditions can be modified, within limits, to offset the penalty. However, the same is not true for nuclear plants, which are constrained by the limitations of the reactor system.

Figure 1
Plot of Various Turbine Exhaust Pressure Correction Curves for 100% and 67% Steam Loads

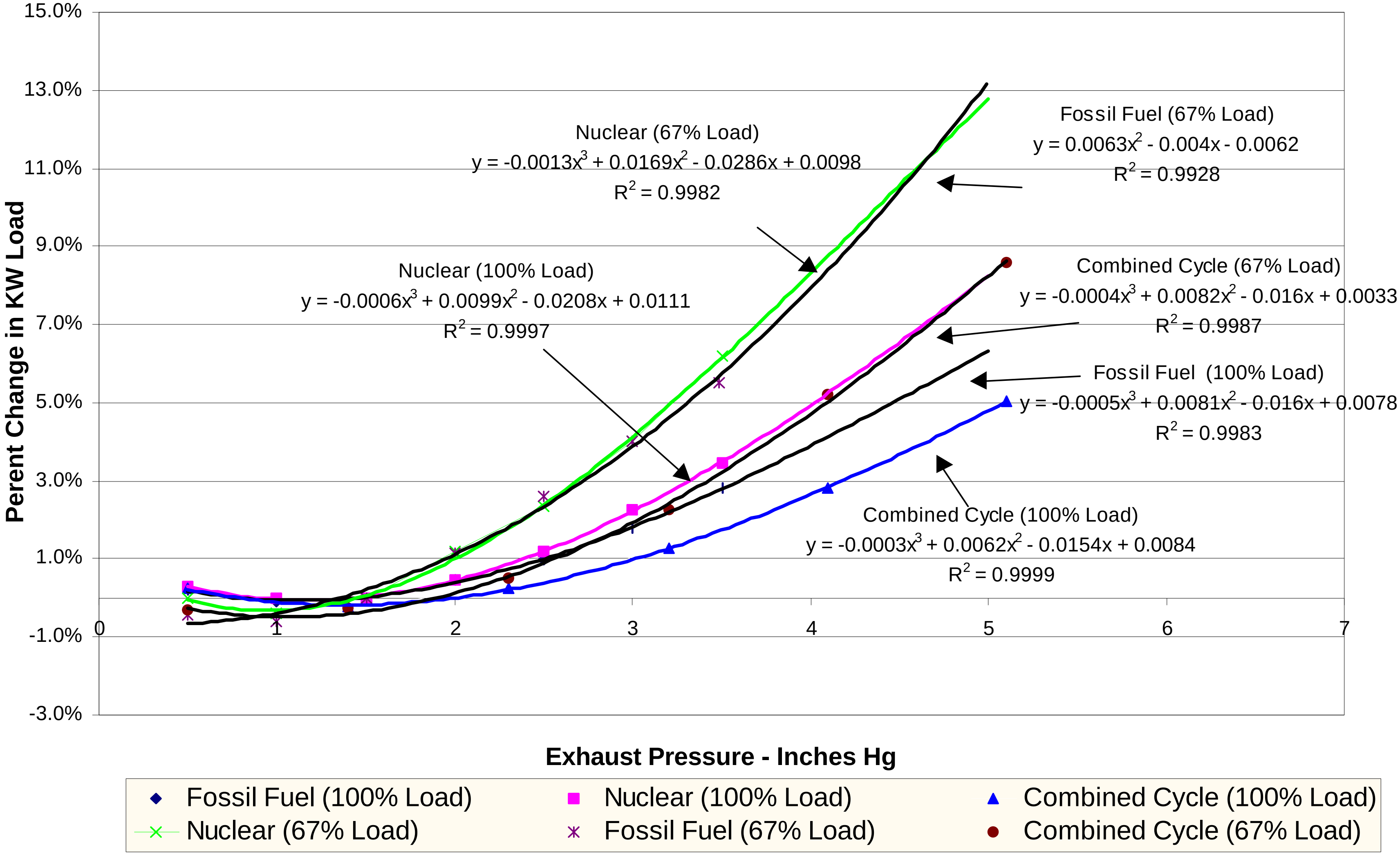
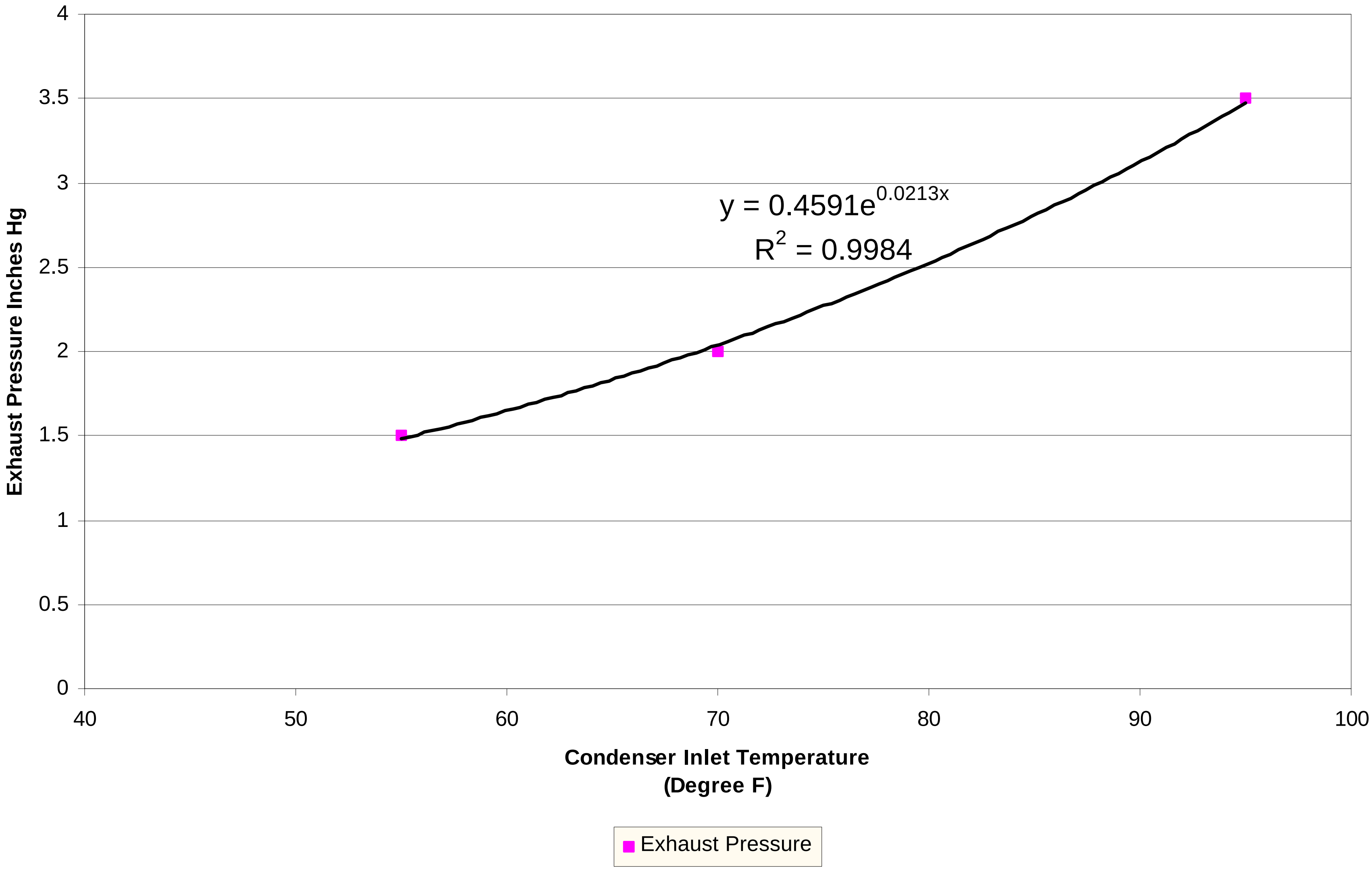


Figure 2
Surface Condenser Cooling Water Inlet Temperature and Steam Pressure Relationship



c. Relationship of Condenser Cooling Water (or Air) Temperature to Steam Side Pressure for Different Cooling System Types and Operating Conditions

˜ *Surface Condensers*

Both once-through and wet cooling towers use surface condensers. As noted previously, condenser inlet temperatures of 55 °F and 95 °F will produce turbine exhaust pressures of 1.5 and 3.5 inches Hg, respectively. Additionally, data from the Calvert Cliffs nuclear power plant showed an exhaust pressure of 2.0 inches Hg at a cooling water temperature of 70 °F. Figure 2 provides a plot of these data which, even though they are from two sources, appear to be consistent. A curve was fitted to these data and was used as the basis for estimating the turbine exhaust pressure for different surface condenser cooling water inlet temperatures. Note that this methodology is based on empirical data that simplifies the relationship between turbine exhaust pressure and condenser inlet temperature, which would otherwise require more complex heat exchange calculations. Those calculations, however, would require numerous assumptions, the selection of which may produce a different curve but with a similar general relationship.

˜ *Once-through Systems*

For once-through cooling systems, the steam cycle condenser cooling water inlet temperature is also the temperature of the source water. Note that the outlet temperature of the cooling water is typically 15 - 20 °F higher than the inlet temperature. This difference is referred to as the "range." The practical limit of the outlet temperature is approximately 100 °F, since many NPDES permits have limitations in the vicinity of 102 - 105 °F . This does not appear to present a problem, since the maximum monthly average surface water temperature at Jacksonville, Florida (selected by EPA as representing warmer U.S. surface waters) was 83.5 °F which would, using the range values above, result in an effluent temperature of 98.5 - 103.5 °F. To gauge the turbine efficiency energy penalty for once-through cooling systems, the temperature of the source water must be known. These temperatures will vary with location and time of year and estimates for several selected locations are presented in Table 3 below.

˜ *Wet Cooling Towers*

For wet cooling towers, the temperature of the cooling tower outlet is the same as the condenser cooling water inlet temperature. The performance of the cooling tower in terms of the temperature of the cooling tower outlet is a function of the wet bulb temperature of the ambient air and the tower type, size, design, and operation. The wet bulb temperature is a function of the ambient air temperature and the humidity. Wet bulb thermometers were historically used to estimate relative humidity and consist of a standard thermometer with the bulb encircled with a wet piece of cloth. Thus, the temperature read from a wet bulb thermometer includes the cooling effect of water evaporation.

Of all of the tower design parameters, the temperature difference between the wet bulb temperature and the cooling tower outlet (referred to as the "approach") is the most useful in estimating tower performance. The wet cooling tower cooling water outlet temperature of the systems that were used in the economic analysis for the final §316(b) New Facility Rule had a design approach of 10 °F. Note that the design approach value is equal to the difference between the tower cooling water outlet temperature and the ambient wet bulb temperature only at the design wet bulb temperature. The actual approach value at wet bulb temperatures other than the design value will vary as described below.

The selection of a 10 °F design approach is based on the data in Attachment C for recently constructed towers. Moreover, a 10 °F approach is considered conservative. As can be seen in Attachment D, a plot of the tower size factor versus the approach shows that a 10 °F approach has a tower size factor of 1.5. The approach is a key factor in sizing towers and has significant cost implications. The trade-off between selecting a small approach versus a higher value is a trade-off between greater capital cost investment versus lower potential energy production. In states where the rates of return on energy investments are fixed (say between 12% and 15%), the higher the capital investment, the higher the return.

For the wet cooling towers used in this analysis, the steam cycle condenser inlet temperature is set equal to the ambient air wet bulb temperature for the location plus the estimated approach value. A design approach value of 10 °F was selected as the common design value for all locations. However, this value is only applicable to instances when the ambient wet bulb temperature is equal to the design wet bulb temperature. In this analysis, the design wet bulb temperature was selected as the 1 percent exceedence value for the specific selected locations.

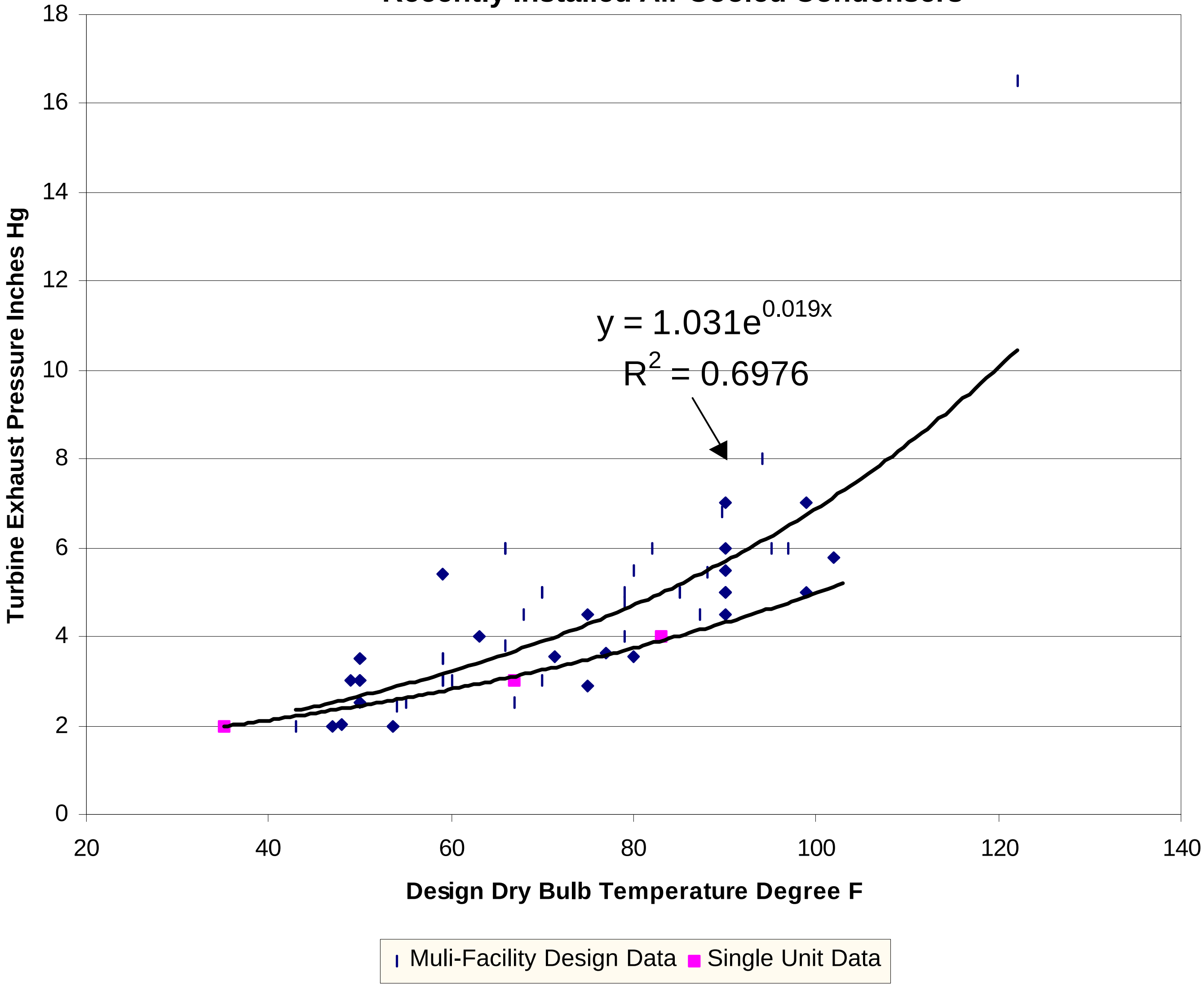
Attachment E provides a graph showing the relationship between different ambient wet bulb temperatures and the corresponding approach for a “typical” wet tower. The graph shows that as the ambient wet bulb temperature decreases, the approach value increases. The graph in Attachment E was used as the basis for estimating the change in the approach value as the ambient wet bulb temperature changes from the design value for each location. Differences in the location-specific design wet bulb temperature were incorporated by fitting a second order polynomial equation to the data in this graph. The equation was then modified by adjusting the intercept value such that the approach was equal to 10 °F when the wet bulb temperature was equal to the design 1 percent wet bulb temperature for the selected location. The location-specific equations were then used to estimate the condenser inlet temperatures that correspond to the estimated monthly values for wet bulb temperatures at the selected locations.

˜ *Air Cooled Condensers*

Air cooled condensers reject heat by conducting it directly from the condensing steam to the ambient air by forcing the air over the heat conducting surface. No evaporation of water is involved. Thus, for air cooled condensers, the condenser performance with regard to turbine exhaust pressure is directly related to the ambient (dry bulb) air temperature, as well as to the condenser design and operating conditions. Note that dry bulb temperature is the same as the standard ambient air temperature with which most people are familiar. Figure 3 presents a plot of the design ambient air temperature and corresponding turbine exhaust pressure for air cooled condensers recently installed by a major cooling system manufacturer (GEA Power Cooling Systems, Inc.). An analysis of the multiple facility data in Figure 3 did not find any trends with respect to plant capacity, location, or age that could justify the separation of these data into subgroups. Three facilities that had very large differences (i.e., >80 °F) in the design dry bulb temperature compared to the temperature of saturated steam at the exhaust pressure were deleted from the data set used in Figure 3.

A review of the design temperatures indicated that the design temperatures did not always correspond to annual temperature extremes of the location of the plant as might be expected. Thus, it appears that the selection of design values for each application included economic considerations. EPA concluded that these design data represent the range of condenser performance at different temperatures and design conditions. A curve was fitted to the entire set of data to serve as a reasonable means of estimating the relationship of turbine exhaust pressure to different ambient air (dry bulb) temperatures. To validate this approach, condenser performance data for a power plant from an engineering contractor report (Litton, no date) was also plotted. This single plant data produced a flatter curve than the multi-facility plot. In other words, the multi-facility curve predicts a greater increase in turbine exhaust pressure as the dry bulb temperature increases. Therefore, the multi-facility curve was selected as a conservative estimation of the relationship between ambient air temperatures and the turbine exhaust pressure. Note that in the case of air cooled condensers, the turbine exhaust steam pressure includes values above 3.5 inches Hg.

Figure 3
Design Dry Bulb and Design Exhaust Pressure for Recently Installed Air Cooled Condensers



˜ *Regional and Seasonal Data*

As noted above, both the source water temperature for once-through cooling systems and the ambient wet bulb and dry bulb temperatures for cooling towers will vary with location and time of year. To estimate average annual energy penalties, EPA sought data to estimate representative monthly values for selected locations. Since plant-specific temperature data may not be available or practical, the conditions for selected locations in different regions are used as examples of the range of possibilities. These four regions include Northeast (Boston, MA), Southeast (Jacksonville, FL), Midwest (Chicago, IL) and Northwest (Seattle, WA). The Southwest Region of the US was not included, since there generally are few once-through systems using surface water in this region.

Table 3-11 presents monthly average coastal water temperatures at the four selected locations. Since the water temperatures remain fairly constant over short periods of time, these data are considered as representative for each month.

Table 3-11: Monthly Average Coastal Water Temperatures (°F)

Location	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
Boston, MA[a]	40	36	41	47	56	62	64.5	68	64.5	57	51	42
Jacksonville, FL[a]	57	56	61	69.5	75.5	80.5	83.5	83	82.5	75	67	60
Chicago, IL[b]	39	36	34	36	37	48	61	68	70	63	50	45
Seattle, WA[a]	47	46	46	48.5	50.5	53.5	55.5	56	55.5	53.5	51	49

[a] Source: NOAA Coastal Water Temperature Guides, (www.nodc.noaa.gov/dsdt/cwtg).

[b] Source: Estimate from multi-year plot "Great Lakes Average GLSEA Surface Water Temperature" (http://coastwatch.glerl.noaa.gov/statistics/).

˜ *Wet and Dry Bulb Temperatures*

Table 3-12 presents design wet bulb temperatures (provided by a cooling system vendor) for the selected locations as the wet bulb temperature that ambient conditions will equal or exceed at selected percent of time (June through September) values. Note that 1 percent represents a period of 29.3 hours. These data, however, represent relatively short periods of time and do not provide any insight as to how the temperatures vary throughout the year. The Agency obtained the *Engineering Weather Data Published by the National Climatic Data Center* to provide monthly wet and dry bulb temperatures. In this data set, wet bulb temperatures were not summarized on a monthly basis, but rather were presented as the average values for different dry bulb temperature ranges along with the average number of hours reported for each range during each month. These hours were further divided into 8-hour periods (midnight to 8AM, 8AM to 4PM, and 4PM to midnight).

Unlike surface water temperature, which tends to change more slowly, the wet bulb and dry bulb temperatures can vary significantly throughout each day and especially from day-to-day. Thus, selecting the temperature to represent the entire month requires some consideration of this variation. The use of daily maximum values would tend to overestimate the overall energy penalty and conversely, the use of 24-hour averages may underestimate the penalty, since the peak power production period is generally during the day.

Since the power demand and ambient wet bulb temperatures tend to peak during the daytime, a time- weighted average of the hourly wet bulb and dry bulb temperatures during the daytime period between 8AM and 4PM was selected as the best method of estimating the ambient wet bulb and dry bulb temperature values to be used in the analysis. The 8AM - 4PM time-weighted average values for wet bulb and dry bulb temperatures were selected as a reasonable compromise between using daily maximum values and 24-hour averages. Table 3-13 presents a summary of the time-weighted wet bulb and dry bulb temperatures for each month for the selected locations. Note that the highest monthly 8AM - 4PM time-weighted average tends to correspond well with the 15 percent exceedence design values. The 15 percent values represent a time period of approximately 18 days which are not necessarily consecutive.

Table 3-12: Design Wet Bulb Temperature Data for Selected Locations

Location	Wet Bulb Temp (°F)			Corresponding Cooling Tower Outlet Temperature (°F)		
	% Time Exceeding			% Time Exceeding		
	1%	5%	15%	1%	5%	15%
Boston, MA	76	73	70	86	83	80
Jacksonville, FL	80	79	77	90	89	87
Chicago, IL	78	75	72	88	85	82
Seattle, WA	66	63	60	76	73	70

Source: www.deltacooling.com

Table 3-13: Time-Weighted Averages for Eight-Hour Period from 8am to 4pm (°F)

Location		Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Design 1%
Boston	Wet Bulb	27.5	29.3	36.3	44.6	53.9	62.7	67.9	67.4	61.5	52.0	42.6	32.6	74.0
	Dry Bulb	33.0	35.3	43.2	53.5	63.8	73.9	80.0	78.2	70.4	59.9	49.5	38.4	88.0
Jacksonville	Wet Bulb	52.9	55.3	59.6	64.5	70.3	75.1	77.1	77.1	75.1	69.1	63.1	55.9	79.0
	Dry Bulb	59.8	63.6	70.3	76.6	83.0	87.2	89.3	88.1	85.1	77.8	70.6	62.6	93.0
Chicago	Wet Bulb	23.3	27.0	37.2	46.6	56.6	64.9	69.8	69.3	62.2	51.2	39.1	27.9	76.0
	Dry Bulb	27.6	31.8	43.9	55.7	67.9	77.4	82.5	80.6	72.4	59.9	45.0	32.2	89.0
Seattle	Wet Bulb	39.4	41.8	44.2	47.2	52.0	56.0	59.2	59.6	57.2	51.0	44.0	39.7	65.0
	Dry Bulb	44.3	47.8	51.5	55.6	61.8	67.2	71.6	71.6	67.3	58.1	49.0	44.3	82.0

c. Calculation of Energy Penalty

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 would best be performed by combining (integrating) the results of individual calculations performed on a periodic basis. For this analysis, a monthly basis was chosen.

The estimated monthly turbine exhaust pressure values for alternative cooling system scenarios were derived using the curves in Figures 2 and 3 in conjunction with the monthly temperature values in Tables 3-11 and 3-13. These turbine exhaust pressure values were then used to estimate the associated change in turbine efficiency using the equations from Figure 1. EPA then calculated the energy penalty for each month. Annual values were calculated by averaging the 12 monthly values.

Tables 3-14 and 3-15 present a summary of the calculated annual average energy penalty values for steam rates of 100 percent and 67 percent of maximum load. These values can be applied directly to the power plant output to determine economic and other impacts. In other words, an energy penalty of 2 percent indicates that the plant output power would be reduced by 2 percent. In addition, Tables 3-14 and 3-15 include the maximum turbine energy penalty associated with maximum design conditions such as once-through systems drawing water at the highest monthly average, and wet towers and air cooled condensers operating in air with a wet bulb and dry bulb temperature at the 1 percent exceedence level. EPA notes that the maximum design values result from using the maximum monthly water temperatures from Table 3-11 and the 1% percent exceedence wet bulb and dry bulb temperatures from Table 3-12.

EPA notes that the penalties presented in Tables 3-14 and 3-15 **do not** comprise the total energy penalties (which incorporate all three components of energy penalties: turbine efficiency penalty, fan energy requirements, and pumping energy usage) as a percent of power output. The total energy penalties are presented in section 3.1 above. The tables below only present the turbine efficiency penalty. Section 3.3.3 presents the fan and pumping components of the energy penalty.

Table 3-14: Calculated Energy Penalties for the Turbine Efficiency Component at 100 Pecent of Maximum Steam Load

Location	Cooling Type	Percent Maximum Load	Nuclear Maximum Design	Nuclear Annual Average	Combined Cycle Maximum Design	Combined Cycle Annual Average	Fossil Fuel Maximum Design	Fossil Fuel Annual Average
Boston	Wet Tower vs. Once-through	100%	1.25%	0.37%	0.23%	0.05%	1.09%	0.35%
	Dry Tower vs. Once-through	100%	9.22%	2.85%	2.04%	0.55%	7.76%	2.48%
	Dry Tower vs. Wet Tower	100%	7.96%	2.48%	1.81%	0.50%	6.66%	2.13%
Jacksonville	Wet Tower vs. Once-through	100%	0.71%	0.54%	0.14%	0.10%	0.61%	0.38%
	Dry Tower vs. Once-through	100%	9.86%	6.21%	2.30%	1.35%	8.22%	5.16%
	Dry Tower vs. Wet Tower	100%	9.14%	5.68%	2.16%	1.25%	7.61%	4.78%
Chicago	Wet Tower vs. Once-through	100%	1.39%	0.42%	0.26%	0.05%	1.21%	0.40%
	Dry Tower vs. Once-through	100%	9.47%	3.09%	2.12%	0.60%	7.96%	2.68%
	Dry Tower vs. Wet Tower	100%	8.08%	2.67%	1.85%	0.55%	6.75%	2.28%
Seattle	Wet Tower vs. Once-through	100%	0.77%	0.29%	0.12%	0.03%	0.70%	0.28%
	Dry Tower vs. Once-through	100%	7.60%	2.63%	1.61%	0.49%	6.46%	2.30%
	Dry Tower vs. Wet Tower	100%	6.83%	2.34%	1.48%	0.45%	5.76%	2.02%
Average	Wet Tower vs. Once-through	100%	1.03%	0.40%	0.19%	0.06%	0.90%	0.35%
	Dry Tower vs. Once-through	100%	9.04%	3.70%	2.02%	0.75%	7.60%	3.15%
	Dry Tower vs. Wet Tower	100%	8.00%	3.29%	1.83%	0.69%	6.70%	2.80%

Note: See Section 3-1 for the total energy penalties. This table presents only the turbine component of the total energy penalty.

Table 3-15: Calculated Energy Penalties for the Turbine Efficiency Component at 67% Pecent of Maximum Steam Load

Location	Cooling Type	Percent Maximum Load	Nuclear Maximum Design	Nuclear Annual Average	Combined Cycle Maximum Design	Combined Cycle Annual Average	Fossil Fuel Maximum Design	Fossil Fuel Annual Average
Boston	Wet Tower vs. Once-through	67%	2.32%	0.73%	0.42%	0.14%	2.04%	0.88%
	Dry Tower vs. Once-through	67%	13.82%	4.96%	3.20%	0.98%	15.15%	4.69%
	Dry Tower vs. Wet Tower	67%	11.50%	4.23%	2.78%	0.84%	13.11%	3.81%
Jacksonville	Wet Tower vs. Once-through	67%	1.22%	1.03%	0.24%	0.18%	1.08%	0.93%
	Dry Tower vs. Once-through	67%	13.61%	9.63%	3.50%	2.14%	16.96%	10.06%
	Dry Tower vs. Wet Tower	67%	12.39%	8.60%	3.27%	1.96%	15.88%	9.14%
Chicago	Wet Tower vs. Once-through	67%	2.53%	0.98%	0.47%	0.16%	2.23%	1.02%
	Dry Tower vs. Once-through	67%	14.03%	5.39%	3.30%	1.07%	15.67%	5.30%
	Dry Tower vs. Wet Tower	67%	11.50%	4.41%	2.83%	0.91%	13.44%	4.27%
Seattle	Wet Tower vs. Once-through	67%	1.60%	0.67%	0.27%	0.11%	1.50%	0.74%
	Dry Tower vs. Once-through	67%	12.16%	4.60%	2.60%	0.90%	12.31%	4.50%
	Dry Tower vs. Wet Tower	67%	10.56%	3.93%	2.33%	0.79%	10.81%	3.75%
Average	Wet Tower vs. Once-through	67%	1.92%	0.85%	0.35%	0.15%	1.71%	0.89%
	Dry Tower vs. Once-through	67%	13.41%	6.14%	3.15%	1.27%	15.02%	6.14%
	Dry Tower vs. Wet Tower	67%	11.49%	5.29%	2.80%	1.12%	13.31%	5.24%

Note: See Section 3-1 for the total energy penalties. This table presents only the turbine component of the total energy penalty.

3.3.3 Energy Penalty Associated with Cooling System Energy Requirements

This analysis is presented to evaluate the energy requirements associated with the operation of the alternative types of cooling systems. As noted previously, the reductions in energy output resulting from the energy required to operate the cooling system equipment are often referred to as parasitic losses. In evaluating this component of the energy penalty, it is the differences between the parasitic losses of the alternative systems that are important. In general, the costs associated with the cooling system energy requirements have been included within the annual O&M cost values developed in Chapter 2 of this document. Thus, the costs of the cooling system operating energy requirements do not need to be factored into the overall energy penalty cost analysis as a separate value, but may have been in some instances as part of a conservative approach.

Alternative cooling systems can create additional energy demands primarily through the use of fans and pumps. There are other energy demands such as treatment of tower blowdown, but these are insignificant compared to the pump and fan requirements and will not be included here. Some seasonal variation may be expected due to reduced requirements for cooling media flow volume during colder periods. These reduced requirements can include reduced cooling water pumping for once-through systems and reduced fan energy requirements for both wet and dry towers. However, no adjustments were made concerning the potential seasonal variations in cooling water pumping. The seasonal variation in fan power requirements is accounted for in this evaluation by applying an annual fan usage rate. The pumping energy estimates are calculated using a selected cooling water flow rate of 100,000 gpm (223 cfs).

a. Fan Power Requirements

˜ ***Wet Towers***

In the reference *Cooling Tower Technology* (Burger 1995), several examples are provided for cooling towers with flow rates of 20,000 gpm using 4 cells with either 75 (example #1) or 100 Hp (example #2) fans each. The primary difference between these two examples is that the tower with the higher fan power requirement has an approach of 5 ^{o}F compared to 11 ^{o}F for the tower with the lower fan power requirement. Using an electric motor efficiency of 92 percent and a fan usage factor of 93 percent (Fleming 2001), the resulting fan electric power requirements are equal to 0.236 MW and 0.314 MW for the four cells with 75 and 100 Hp fan motors, respectively. These example towers both had a heat load of 150 million BTU/hr. Table 3-16 provides the percent of power output penalty based on equivalent plant capacities derived using the heat rejection factors described below. Note that fan gear efficiency values are not applicable because they do not affect the fan motor power rating or the amount of electricity required to operate the fan motors.

A third example was provided in vendor-supplied data (Fleming 2001), in which a cooling tower with a cooling water flow rate of 243,000 gpm had a total fan motor capacity brake-Hp of 250 for each of 12 cells. This wet tower had a design temperature range of 15 °F and an approach of 10 °F. The percent of power output penalty shown in Table 7 is also based on equivalent plant capacities derived using the heat rejection factors described below.

A fourth example is a cross-flow cooling tower for a 35 MW coal-fired plant in Iowa (Litton, no date). In this example, the wet tower consists of two cells with one 150 Hp fan each, with a cooling water flow rate of 30,000 gpm. This wet tower had a design temperature range of 16 °F, an approach of 12 °F, and wet bulb temperature of 78 °F. The calculated energy penalty in this example is 0.67 percent.

Example #2, which has the smallest approach value, represents the high end of the range of calculated wet tower fan energy penalties presented in Table 3-16. Note that smaller approach values correspond to larger, more expensive (both in capital and O&M costs) towers. Since the fossil fuel plant penalty value for example #4, which is based mostly on empirical data, is just below the fossil fuel penalty calculated for example #2, EPA has chosen the calculated values for example #2 as representing a conservative estimate for the wet tower fan energy penalty.

EPA notes that the penalties presented in Tables 3-16 **do not** comprise the total energy penalty (which incorporates all three components of energy penalties: turbine efficiency penalty, fan energy requirements, and pumping energy usage) as a percent of power output. The total energy penalties are presented in section 3.1 above. The table below only presents the fan component of the penalty.

Table 3-16: Wet Tower Fan Power Energy Penalty

Example Plant	Range/ Approach (Degree F)	Flow (gpm)	Fan Power Rating (Hp)	Fan Power Required (MW)	Plant Type	Plant Capacity (MW)	Percent of Output (%)
#1	15/11	20,000	300	0.236	Nuclear	35	0.68%
					Fossil Fuel	43	0.55%
					Comb. Cycle	130	0.18%
#2	15/5	20,000	400	0.314	Nuclear	35	0.91%
					Fossil Fuel	43	0.73%
					Comb. Cycle	130	0.24%
#3	15/10	243,000	3,000	2.357	Nuclear	420	0.56%
					Fossil Fuel	525	0.45%
					Comb. Cycle	1574	0.15%
#4	16/12	30,000	300.0	0.236	Fossil Fuel	35	0.67%

Note: See Section 3-1 for the total energy penalties. This table presents only the fan component of the total energy penalty.

˜ *Air Cooled Condensers*

Air cooled condensers require greater air flow than recirculating wet towers because they cannot rely on evaporative heat transfer. The fan power requirements are generally greater than those needed by wet towers by a factor of 3 to 4 (Tallon 2001). While the fan power requirements can be substantial, at least a portion of this increase over wet cooling systems is offset by the elimination of the pumping energy requirements associated with wet cooling systems described below.

The El Dorado power plant in Boulder, Nevada which was visited by EPA is a combined-cycle plant that uses air cooled condensers due to the lack of sufficient water resources. This facility is located in a relatively hot section of the U.S. Because the plant has a relatively low design temperature (67 ^{o}F) in a hot environment, it should be considered as representative of a conservative situation with respect to the energy requirements for operating fans in air cooled condensers. The steam portion of the plant has a capacity of 150 MW (1.1 million lb/hr steam flow).

The air cooled condensers consist of 30 cells with a 200 Hp fan each. A fan motor efficiency of 92 percent is assumed. Each fan has two operating speeds, with the low speed consuming 20 percent of the fan motor power rating.

The facility manager provided estimates of the proportion of time that the fans were operated at low or full speed during different portions of the year (Tatar 2001). Factoring in the time proportions and the corresponding power requirements results in an overall annual fan power factor of 72 percent for this facility. In other words, over a one year period, the fan power requirement will average 75 percent of the fan motor power rating. A comparison of the climatic data for Las Vegas (located nearby) and Jacksonville, Florida shows that the Jacksonville mean maximum temperature values were slightly warmer in the winter and slightly cooler in the summer. Adjustments in the annual fan power factor calculations to address Jacksonville's slightly warmer winter months resulted in a projected annual fan power factor of 77 percent. EPA chose a factor of 75 percent as representative of warmer regions of the U.S. Due to lack of available operational data for other locations, this value is used for facilities throughout the U.S. and represents an conservative value for the much cooler regions.

Prior to applying this factor, the resulting maximum energy penalty during warmer months is 3.2 percent for the steam portion only. This value is the maximum instantaneous penalty that would be experienced during high temperature conditions. When the annual fan power factor of 75 percent is applied, the annual fan energy penalty becomes 2.4 percent of the plant power output. An engineer from an air cooled condenser manufacturer indicated that the majority of air cooled condensers being installed today also include two-speed fans and that the 20 percent power ratio for the low speed was the factor that they used also. In fact, some dry cooling systems, particularly those in very cold regions, use fans with variable speed drives to provide even better operational control. Similar calculations for a waste-to-energy plant in Spokane, Washington resulted in a maximum fan operating penalty of 2.8 percent and an annual average of 2.1 percent using the 75 percent fan power factor. Thus, the factor of 2.4 percent selected by EPA as a conservative annual penalty value appears valid.

b. Cooling Water Pumping Requirements

The energy requirements for cooling water pumping can be estimated by combining the flow rates and the total head (usually given in feet of water) that must be pumped. Estimating the power requirements for the alternative cooling systems that use water is somewhat complex in that there are several components to the total pumping head involved. For example, a once-through system must pump water from the water source to the steam condensers, which will include both a static head from the elevation of the source to the condenser (use of groundwater would represent an extreme case) and friction head losses through the piping and the condenser. The pipe friction head is dependent on the distance between the power plant and the source plus the size and number of pipes, pipe fittings, and the flow rate. The condenser friction head loss is a function of the condenser design and flow rate.

Wet cooling towers must also pump water against both a static and friction head. A power plant engineering consultant estimated that the total pumping head at a typical once-through facility would be approximately 50 ft (Taylor 2001). EPA performed a detailed analysis of the cooling water pumping head that would result from different combinations of piping velocities and distances. The results of this analysis showed that the pumping head was in many scenarios similar in value for both once-through and wet towers, and that the estimated pumping head ranged from approximately 40 to 60 feet depending on the assumed values. Since EPA's analysis produced similar values as the 50 ft pumping head provided by the engineering consultant, this value was used in the estimation of the pumping requirements for cooling water intakes for both once-through and wet tower systems. The following sections describe the method for deriving these pumping head values.

˜ *Friction Losses*

In order to provide a point of comparison, a cooling water flow rate of 100,000 gpm (223 cfs) was used. A recently reported general pipe sizing rule indicating that a pipe flow velocity of 5.7 fps is the optimum flow rate with regards to the competing cost values was used as the starting point for flow velocity (Durand et al. 1999). Such a minimum velocity is needed to prevent sediment deposition and pipe fouling. Using this criterion as a starting point, four 42-inch steel pipes carrying 25,000 gpm each at a velocity of 5.8 fps were selected. Each pipe would have a friction head loss of 0.358 ft/100 ft of pipe (Permutit 1961), resulting in a friction loss of 3.6 ft for every 1,000 ft of length. Since capital costs may dictate using fewer pipes with greater pipe flow rates, two other scenarios using either three or two parallel 42-inch pipes were also evaluated. Three pipes would result in a flow rate and velocity of 33,000 gpm and 7.7 fps, which results in a friction head loss of 6.1 ft/1000ft. Two pipes would result in a flow rate and velocity of 50,000 gpm and 11.6 fps, which results in a friction head loss of 12.8 ft/1000ft. The estimated 50 ft total pumping head was most consistent with a pipe velocity of 7.7 fps (three 42-inch pipes).

The relative distances of the power plant condensers to the once-through cooling water intakes as compared to the distance from the plant to the alternative cooling tower can be an important factor. In general, the distances that the large volumes of cooling water must be pumped will be greater for once-through cooling systems. For this analysis, a fixed distance of 300 ft was selected for the cooling tower. Various distances ranging from 300 ft to 3,000 ft are used for the once-through system. The friction head was also assumed to include miscellaneous losses due to inlets, outlets, bends, valves, etc., which can be calculated using equivalent lengths of pipe. For 42-in. steel pipe, each entrance and long sweep elbow is equal to about 60 ft in added pipe length. For the purposes of this analysis, both systems were assumed to have five such fittings for an added length of 300 ft. The engineering estimate of 50 ft for pumping head was most consistent with a once-through pumping distance of approximately 1,000 ft.

˜ *Static Head*

Static head refers to the distance in height that the water must be pumped from the source elevation to the destination. In the case of once-through cooling systems, this is the distance in elevation between the source water and the condenser inlet. However, many power plants eliminate a significant portion of the static head loss by operating the condenser piping as a siphon. This is done by installing vacuum pumps at the high point of the water loop. In EPA's analysis, a static head of 20 ft produced a total pumping head value that was most consistent with the engineering consultant's estimate of 50 feet.

In the case of cooling towers, static head is related to the height of the tower, and vendor data for the overall pumping head through the tower is available. This pumping head includes both the static and dynamic heads within the tower, but was included as the static head component for the analysis. Vendor data reported a total pumping head of 25 ft for a large cooling tower sized to handle 335,000 gpm (Fleming 2001). The tower is a counter-flow packed tower design. Adding the condenser losses and pipe losses resulted in a total pumping head of approximately 50 feet.

˜ *Condenser Losses*

Condenser design data provided by a condenser manufacturer, Graham Corporation, showed condenser head losses ranging from 21 ft of water for small condensers (cooling flow <50,000 gpm) to 41 ft for larger condensers (Hess

2001). Another source showed head losses through the tubes of a large condenser (311,000 gpm) to be approximately 9 ft of water (HES. 2001). For the purposes of this analysis, EPA estimated condenser head losses to be 20 ft of water. For comparable systems with similar cooling water flow rates, the condenser head loss component should be the same for both once-through systems and recirculating wet towers.

˜ *Flow Rates*

In general, the cooling water flow rate is a function of the heat rejection rate through the condensers and the range of temperature between the condenser inlet and outlet. The flow rate for cooling towers is approximately 95 percent that of once-through cooling water systems, depending on the cooling temperature range. However, cooling tower systems also still require some pumping of make-up water. For the purposes of this analysis, the flow rates for each system will be assumed to be essentially the same. All values used in the calculations are for a cooling water flow rate of 100,000 gpm. Values for larger and smaller systems can be factored against these values. The total pump and motor efficiency is assumed to be equal to 70 percent.

c. Analysis of Cooling System Energy Requirements

This analysis evaluates the energy penalty associated with the operation of cooling system equipment for conversion from once-through systems to wet towers and for conversion to air cooled systems by estimating the net difference in required pumping and fan energy between the systems. This penalty can then be compared to the power output associated with a cooling flow rate of 100,000 gpm to derive a percent of plant output figure that is a similar measure to the turbine efficiency penalty described earlier. The power output was determined by comparing condenser heat rejection rates for different types of systems. As noted earlier, the cost of this energy penalty component has already been included in the alternative cooling system O&M costs discussed in Chapter 2 of this document, but was derived independently for this analysis.

Table 3-17 shows the pumping head and energy requirements for pumping 100,000 gpm of cooling water for both once-through and recirculating wet towers using the various piping scenario assumptions. In general, the comparison of two types of cooling systems shows offsetting energy requirements that essentially show zero pumping penalty between once-through and wet towers as the pumping distance for the once-through system increases to approximately 1,000 ft. In fact, it is apparent that for once-through systems with higher pipe velocities and pumping distances, more cooling water pumping energy may be required for the once-through system than for a wet cooling tower. Thus, when converting from once-through to recirculating wet towers, the differences in pumping energy requirements may be relatively small.

As described above, wet towers will require additional energy to operate the fans, which results in a net increase in the energy needed to operate the wet tower cooling system compared to once-through. Note that the average calculated pumping head across the various scenarios for once-through systems was 54 ft. This data suggests that an average pumping head of 50 feet for once-through systems appears to be a reasonable assumption where specific data are not available.

EPA notes that the penalties presented in Tables 3-17 and 3-18 **do not** comprise the total energy penalties (which incorporate all three components of energy penalties: turbine efficiency penalty, fan energy requirements, and pumping energy usage) as a percent of power output. The total energy penalties are presented in section 3.1 above. The tables below only present the pumping components.

Table 3-17: Cooling Water Pumping Head and Energy for 100,000 gpm System Wet Towers Versus Once-through At 20' Static Head

Cooling System Type	Distance Pumped ft.	Static Head ft.	Condenser Head ft	Equiv. Length Misc. Losses ft.	Pipe Velocity fps	Friction Loss Rate ft/1,000ft	Friction Head ft.	Total Head ft.	Net Difference ft	Flow Rate gpm	Hydraulic-Hp Hp	Brake-Hp Hp	Power Required kW	Energy Penalty kW
Once-through at 20' Static Head Using 4: 42" Pipes at 300' Length														
Once-through	300	20	21	300	5.8	3.6	2	43		100,000	1089	1556	1161	
Wet Tower	300	25	21	300	5.8	3.6	2	48	5	100,000	1216	1737	1296	135
Once-through at 20' Static Head Using 3: 42" Pipes at 300' Length														
Once-through	300	20	21	300	7.7	6.1	4	45		100,000	1127	1610	1201	
Wet Tower	300	25	21	300	7.7	6.1	4	50	5	100,000	1254	1791	1336	135
Once-through at 20' Static Head Using 2: 42" Pipes at 300' Length														
Once-through	300	20	21	300	11.6	12.8	8	49		100,000	1229	1755	1310	
Wet Tower	300	25	21	300	11.6	12.8	8	54	5	100,000	1355	1936	1444	135
Once-through at 20' Static Head Using 4: 42" Pipes at 1000' Length														
Once-through	1000	20	21	300	5.8	3.6	5	46		100,000	1153	1647	1229	
Wet Tower	300	25	21	300	5.8	3.6	2	48	2	100,000	1216	1737	1296	67
Once-through at 20' Static Head Using 3: 42" Pipes at 1000' Length														
Once-through	1000	20	21	300	7.7	6.1	8	49		100,000	1235	1764	1316	
Wet Tower	300	25	21	300	7.7	6.1	4	50	1	100,000	1254	1791	1336	20
Once-through at 20' Static Head Using 2: 42" Pipes at 1000' Length														
Once-through	1000	20	21	300	11.6	12.8	17	58		100,000	1455	2079	1551	
Wet Tower	300	25	21	300	11.6	12.8	8	54	-4	100,000	1355	1936	1444	-107
Once-through at 20' Static Head Using 4: 42" Pipes at 3000' Length														
Once-through	3000	20	21	300	5.8	3.6	12	53		100,000	1335	1907	1423	
Wet Tower	300	25	21	300	5.8	3.6	2	48	-5	100,000	1216	1737	1296	-127
Once-through at 20' Static Head Using 3: 42" Pipes at 3000' Length														
Once-through	3000	20	21	300	7.7	6.1	20	61		100,000	1543	2204	1644	
Wet Tower	300	25	21	300	7.7	6.1	4	50	-11	100,000	1254	1791	1336	-309
Once-through at 20' Static Head Using 2: 42" Pipes at 3000' Length														
Once-through	3000	20	21	300	11.6	12.8	42	83		100,000	2101	3002	2239	
Wet Tower	300	25	21	300	11.6	12.8	8	54	-30	100,000	1355	1936	1444	-795

Note: Wet Towers are assumed to always be at 300' distance and have the same tower pumping head of 25' in all scenarios shown.
The same flow rate of 100,000gpm (223 cfs) is used for all scenarios.
See Section 3-1 for the total energy penalties. This table presents only the pumping component of the total energy penalty.

˜ Cooling System Energy Requirements Penalty as Percent of Power Output

One method of estimating the capacity of a power plant associated with a given cooling flow rate is to compute the heat rejected by the cooling system and determine the capacity that would match this rejection rate for a "typical" power plant in each category. In order to determine the cooling system heat rejection rate, both the cooling flow (100,000 gpm) and the condenser temperature range between inlet and outlet must be estimated. In addition, the capacity that corresponds to the power plant heat rejection rate must be determined. The heat rejection rate is directly related to the type, design, and capacity of a power plant. The method used here was to determine the ratio of the plant capacity divided by the heat rejection rate as measured in equivalent electric power.

An analysis of condenser cooling water flow rates, temperature ranges and power outputs for several existing nuclear plants provided ratios of the plant output to the power equivalent of heat rejection ranging from 0.75 to 0.92. A similar analysis for coal-fired power plants provided ratios ranging from 1.0 to 1.45. Use of a lower factor results in a lower power plant capacity estimate and, consequently, a higher value for the energy requirement as a percent of capacity. Therefore, EPA chose to use values near the lower end of the range observed. EPA selected ratios of 0.8 and 1.0 for nuclear and fossil-fueled plants, respectively. The steam portion of a combined cycle plant is assumed to have a factor similar to fossil fuel plants of 1.0. Considering that this applies to only one-third of the total plant output, the overall factor for combined-cycle plants is estimated to be 3.0.

In order to correlate the cooling flow energy requirement data to the power output, a condenser temperature range must also be estimated. A review of data from newly constructed plants in Attachment C showed no immediately discernable pattern on a regional basis for approach or range values. Therefore, these values will not be differentiated on a regional basis in this analysis. The data did, however, indicate a median approach of 10 °F (average 10.4 °F) and a median range of 20 °F (average 21.1 °F). This range value is consistent with the value assumed in other EPA analyses and therefore a range of 20 °F will be used. Table 3-18 presents the energy penalties corresponding to the pumping energy requirements from Table 3-17 using the above factors.

Table 3-18: Comparison of Pumping Power Requirement and Energy Penalty to Power Plant Output

Cooling system Type	Distance Pumped	Static Head	Power Required	Flow Rate	Range	Nuclear Power/ Heat	Nuclear Equiv. Output	Nuclear Pumping	Fossil Fuel Power/ Heat	Fossil Fuel Equiv. Output	Fossil Fuel Pumping	Comb.-Cycle Power/ Heat	Comb.-Cycle Equiv.	Comb.-Cycle Pumping
	ft.	ft.	kW	gpm	ºF	Ratio	(MW)	% of Output	Ratio	(MW)	% of Output	Ratio	Output (MW)	% of Output
Once-through at 20' Static Head Using 4: 42" Pipes at 300' Length														
Once-through	300	20	1161.1	100,000	20	0.8	235	0.49%	1	294	0.39%	3	882	0.13%
Wet Tower	300	25	1295.6	100,000	20	0.8	235	0.55%	1	294	0.44%	3	882	0.15%
Once-through at 20' Static Head Using 3: 42" Pipes at 300' Length														
Once-through	300	20	1201.4	100,000	20	0.8	235	0.51%	1	294	0.41%	3	882	0.14%
Wet Tower	300	25	1335.9	100,000	20	0.8	235	0.57%	1	294	0.45%	3	882	0.15%
Once-through at 20' Static Head Using 2: 42" Pipes at 300' Length														
Once-through	300	20	1309.6	100,000	20	0.8	235	0.56%	1	294	0.45%	3	882	0.15%
Wet Tower	300	25	1444.1	100,000	20	0.8	235	0.61%	1	294	0.49%	3	882	0.16%
Once-through at 20' Static Head Using 4: 42" Pipes at 1000' Length														
Once-through	1000	20	1228.8	100,000	20	0.8	235	0.52%	1	294	0.42%	3	882	0.14%
Wet Tower	300	25	1295.6	100,000	20	0.8	235	0.55%	1	294	0.44%	3	882	0.15%
Once-through at 20' Static Head Using 3: 42" Pipes at 1000' Length														
Once-through	1000	20	1316.3	100,000	20	0.8	235	0.56%	1	294	0.45%	3	882	0.15%
Wet Tower	300	25	1335.9	100,000	20	0.8	235	0.57%	1	294	0.45%	3	882	0.15%
Once-through at 20' Static Head Using 2: 42" Pipes at 1000' Length														
Once-through	1000	20	1550.6	100,000	20	0.8	235	0.66%	1	294	0.53%	3	882	0.18%
Wet Tower	300	25	1444.1	100,000	20	0.8	235	0.61%	1	294	0.49%	3	882	0.16%
Once-through at 20' Static Head Using 4: 42" Pipes at 3000' Length														
Once-through	3000	20	1422.5	100,000	20	0.8	235	0.60%	1	294	0.48%	3	882	0.16%
Wet Tower	300	25	1295.6	100,000	20	0.8	235	0.55%	1	294	0.44%	3	882	0.15%
Once-through at 20' Static Head Using 3: 42" Pipes at 3000' Length														
Once-through	3000	20	1644.5	100,000	20	0.8	235	0.70%	1	294	0.56%	3	882	0.19%
Wet Tower	300	25	1335.9	100,000	20	0.8	235	0.57%	1	294	0.45%	3	882	0.15%
Once-through at 20' Static Head Using 2: 42" Pipes at 3000' Length														
Once-through	3000	20	2239.3	100,000	20	0.8	235	0.95%	1	294	0.76%	3	882	0.25%
Wet Tower	300	25	1444.1	100,000	20	0.8	235	0.61%	1	294	0.49%	3	882	0.16%

Note: Wet Towers are assumed to always be at 300' distance and have the same tower pumping head of 25' in all scenarios shown. The same flow rate of 100,000gpm (223 cfs) is used for all scenarios. Power/Heat Ratio refers to the ratio of Power Plant Output (MW) to the heat (in equivalent MW) transferred through the condenser. See Section 3-1 for the total energy penalties. This table presents only the pumping component of the total energy penalty

d. Summary of Cooling System Energy Requirements

EPA chose the piping scenario in Table 3-17 where pumping head is close to 50 ft for both (i.e., once-through at 1,000 ft and 3-42 in. pipes in Table 3-17). Thus, the cooling water pumping requirements for once-through and recirculating wet towers are nearly equal using the chosen site-specific conditions. Table 3-19 summarizes the fan and pumping equipment energy requirements as a percent of power output for each type of power plant. Table 3-20 presents the net difference in energy requirements shown in Table 3-19 for the alternative cooling systems. The net differences in Table 3-20 are the equipment operating energy penalties associated with conversion from one cooling technology to another.

EPA notes that the penalties presented in Tables 3-19 and 3-20 **do not** comprise the total energy penalties (which incorporate all three components of energy penalties: turbine efficiency penalty, fan energy requirements, and pumping energy usage) as a percent of power output. The total energy penalties are presented in section 3.1 above. The tables below only present the pumping and fan components. Section 3.3.2 presents the turbine efficiency components of the energy penalty.

Table 3-19: Summary of Fan and Pumping Energy Requirements as a Percent of Power Output

	Wet Tower Pumping	Wet Tower Fan	Wet Tower Total	Once-through Total (Pumping)	Dry Tower Total (Fan)
Nuclear	0.57%	0.91%	1.48%	0.56%	3.04%
Fossil Fuel	0.45%	0.73%	1.18%	0.45%	2.43%
Combined-Cycle	0.15%	0.24%	0.39%	0.15%	0.81%

Note: See Section 3.1 for the total energy penalties.

Table 3-20: Fan and Pumping Energy Penalty Associated with Alternative Cooling System as a Percent of Power Output

	Wet Tower Vs Once-through	Dry Tower Vs Wet Tower	Dry Tower Vs Once-through
Nuclear	0.92%	1.56%	2.48%
Fossil Fuel	0.73%	1.25%	1.98%
Combined-Cycle	0.24%	0.42%	0.66%

Note: See Section 3.1 for the total energy penalties.

3.4 AIR EMISSIONS INCREASES

Due to the cooling system energy penalties, as described in section 3.3 and presented in section 3.1 above, EPA estimates that air emissions will marginally increase from power plants which upgrade cooling systems. The energy penalties reduce the efficiency of the electricity generation process and thereby increase the quantity of fuel consumed per unit of electricity generated. In estimating annual increases in air emissions, the Agency based its calculations on the mean annual energy penalties provided in Table 3-1 above. EPA presents the annual air emissions increases for the final rule and the dry cooling regulatory alternative in Tables 3-7 and 3-8 in section 3.2 above.

EPA developed estimates of incremental air emissions estimates for the two types of power plants projected to upgrade cooling systems as a result of this rule (or a regulatory alternative): combined-cycle and coal-fired power plants. Generally, combined-cycle plants produce significantly less air emissions per kilowatt-hour of electricity generated than coal-fired plants. Because the combined-cycle plant requires cooling for approximately one-third of its process (on a megawatt capacity basis) and because of the differences in combustion products from natural gas versus coal, the combined-cycle plant produces less air emissions, even after coal-fired plants are equipped with state-of-the-art emissions controls. However, for the case of the air emissions estimates for the final rule and regulatory alternatives considered, EPA estimates that plants incurring an energy penalty will not increase their fuel consumption on-site to overcome incurred energy penalties. Instead, the Agency estimates that energy penalties at facilities affected by the requirements of this rule (or the regulatory alternatives) would purchase replacement power from the grid and the air emissions increases associated with a particular energy penalty at an effected plant would be released by the rest of the grid as a whole (thereby comprising negligible increases at a large number and variety of power plants). EPA received comments asserting that not all facilities, especially during times of peak demand, would be able to increase their fuel consumption to overcome energy penalties. Therefore, the air emissions increases presented in section 3.2 of this chapter represent uniform national air emissions increases per unit of energy penalty, regardless of the plant at which the energy penalty is occurring. For the final rule and regulatory alternatives considered, the key difference between air emissions increases estimated at facilities projected to upgrade cooling systems is directly related to the size of the energy penalty that the plant will incur. For the sake of comparison, EPA also calculated the air emissions increases for the final rule and regulatory alternatives in the case where the effected plants would increase fuel consumption to overcome the penalties. The comparative results are presented in Tables 3-21 and 3-22. EPA found small national differences between increased air emissions as calculated on the plant versus grid basis. For more information on the supporting calculations see DCN 3-3085.

The data source for the Agency's air emissions estimates of CO_2, SO_2, NO_X, and Hg is the EPA developed database titled E-GRID 2000. This database is a compendium of reported air emissions, plant characteristics, and industry profiles for the entire US electricity generation industry in the years 1996 through 1998. The database relies on information from power plant emissions reporting data from the Energy Information Administration of the Department of Energy. The database compiles information on every power plant in the United States and includes statistics such as plant operating capacity, air emissions, electricity generated, fuel consumed, etc. This database provided ample data for the Agency to conduct air emissions increases analyses for this rule. The emissions reported in the database are for the power plants' actual emissions to the atmosphere and represent emissions after the influence of air pollution control devices. To test the veracity of the database for the purposes of this rule, the Agency compared the information to other sources of data available on power plant capacities, fuel-types, locations, owners, and ages. Without exception, the E-GRID 2000 database provided accurate estimates of each of these characteristics versus information that EPA was able to obtain from other sources.

As noted above, the E-GRID 2000 database contains data on existing power plants. For the national analysis presented in section 3.2 above, EPA estimated that the annual generation of electricity would not increase over the life of the rule. Therefore, the emissions increases as a percent of national capacity presented in Tables 3-7 and 3-8 above are conservatively estimated and ignore projected growth rates of power plant capacity. For the comparative analysis of plant versus grid based emissions the Agency purposefully chose, when analyzing specific power plants (and not just the grid as a whole), to focus on the most recently constructed plants with multiple years of operating data (where possible). In addition, the Agency selected a variety of plants from different regions of the country with different urban versus rural locations. The capacity of the model plants was chosen as closely as possible to the average size plant within scope of the rule. Therefore, the Agency's comparative estimates of the air emissions increases from the scenario where individual plants are able to consume more fuel to overcome the energy penalties present nationally applicable results for the variety of plants and locations expected for the new facility rule. The model facility plant information along with the supporting calculations for this analysis can be found in DCN 3-3085.

Because the Agency estimates that the air emissions increases for the final rule (and regulatory alternatives) will come from the mix of plant types across the nation, the issue of baseline cooling systems is moot. However, for the scenario where EPA estimated (for the sake of comparison) that plants would increase fuel consumption to overcome energy penalties, and the air emissions would occur at the site, the issue of cooling system is more relevant. EPA attempted to consider baseline cooling systems when selecting the model facilities upon which to base the air emissions profiles for combined-cycle and coal-fired plants. However, because the emissions would be used to estimate changes in cooling systems from once-through to wet towers and, for the case of regulatory alternatives, from once-through to dry towers and wet towers to dry towers, the Agency ultimately determined that age, size, and location of the plant were more important factors to consider than the baseline cooling system. The effect is such, for the comparative example of plants increasing fuel consumption to overcome energy penalties as a result of the final rule, the Agency may have marginally overestimated the air emissions increases due to cooling system changes. EPA reiterates that this has no bearing on the estimated air emissions for the final rule and is relevant only for the comparative analysis presented in Tables 3-21 and 3-22. The basis for the Agency stating that it may have overestimated emissions in this comparative case for the final rule is due to the fact that several of the plants used as model facilities in the air emissions analysis actually utilize wet-cooling towers at baseline. Therefore, the baseline energy efficiency would be lower than a once-through system and the related baseline air emissions rates per unit of fuel consumed would be higher. Thus, for the case of the upgrades from once-through to wet cooling towers, EPA likely is overestimating the compliance air emissions rates per unit of fuel consumed in this comparative case. For the case of the dry cooling alternative, the effect is less pronounced and the Agency may be underestimating, in the end, the comparative air emissions increases. This is due to the fact that the majority of power plants have wet cooling towers at baseline. For the case of 90 percent of the plants to be upgraded to dry cooling in this regulatory alternative, the proper baseline cooling system is wet cooling towers. Therefore, the baseline air emissions rates per unit of electricity generated are lower than would represent a majority of plants employing wet cooling at baseline.

Table3-21. Comparison of Calculation Techniques for Net Air Emissions Increases of the Final Rule

Compensation Technique	Total Energy Penalty MW	Annual CO_2 (tons)	Annual SO_2 (tons)	Annual NO_x (tons)	Annual Hg (lbs)
Increased Fuel Consumption	100	712,886	1,543	1,518	23
Market Power Replacement	100	485,860	2,561	1,214	16

Table3-22. Comparison of Calculation Techniques for Net Air Emissions Increases of Dry Cooling

Compensation Technique	Total Energy Penalty MW	Annual CO_2 (tons)	Annual SO_2 (tons)	Annual NO_x (tons)	Annual Hg (lbs)
Increased Fuel Consumption	1,900	11,427,552	18,649	23,432	272
Market Power Replacement	1,900	8,931,036	47,074	22,313	300

3.5 OTHER ENVIRONMENTAL IMPACTS

Recirculating wet cooling towers can produce side effects such as vapor plumes, displacement of habitat or wetlands, noise, salt or mineral drift, water consumption through evaporation, and increased solid waste generation due to wastewater treatment of tower blowdown. The *Generic Environmental Impact Statement for License Renewal of Nuclear Plants* (NUREG-1437 Vol. 1, Nuclear Regulatory Commission) addresses the majority of these issues in depth, and the Agency refers to the detailed research contained therein several times in this discussion.

The Agency considered non-aquatic impacts of recirculating cooling towers for the proposal. While the Agency did not present quantified information regarding these side effects in the proposal, the Agency discussed the effects of both wet and dry cooling towers in the proposal. Specifically, the Agency discussed discharge water quality, salt drift, water conditioning chemicals and biocides, vapor plumes, energy efficiency, land use, and air emissions increases (65 FR 49080-49081). The Agency invited comments to the proposal on the subject of adverse environmental impact and whether or not it should consider non-aquatic impacts such as salt/mineral drift and reductions in the efficiency of electricity generation leading to increased air emissions as examples of adverse environmental impact (65 FR 49075). In turn, the Agency received no usable data (only anecdotal information) from commenters supporting assertions that these "side effects" pose significant environmental problems. The Agency researched the subjects further after proposal and provided some of the information in the notice of data availability and has cited other information from NUREG-1437.

The vast majority (90 percent) of power plants projected within the scope of this rule would install recirculating wet cooling towers in absence of this rule. Of these 74 power plants, the Agency projects that the cooling towers to be constructed will be of the mechanical draft type. (Stone & Webster 1992). For the other nine power plants for which EPA has projected the compliance costs associated with wet cooling towers, the Agency projects that the towers to be installed would be of the mechanical draft type, also.

3.5.1 Vapor Plumes

Natural draft or mechanical draft cooling towers can produce vapor plumes. Plumes can create problems for fogging and icing, which have been recorded to create dangerous conditions for local roads and for air and water navigation. Plumes are in some cases disfavored for reasons of aesthetics. Generally, mechanical draft cooling towers have significantly shorter plumes than those for natural draft towers (by approximately 30 percent). A "treatment" technique for these plumes in very rare cases is the installation of plume abatement (wet/dry hybrid cooling towers) on the tower. This is currently practiced at a small portion of recently constructed facilities (See DCN #2-037). As such, EPA's capital costs are not adjusted to reflect this type of plume abatement for this nationally applicable rule in which only 9 facilities are projected to install wet cooling towers.

Regarding aesthetics of cooling tower plumes, the Agency points to the Track II compliance option as an alternative for new facility power plants, in addition to the plume abatement controls, which are an option for new plants that choose to site where plume aesthetics are a public nuisance. The Agency notes that land area buffers may also be a simple means for reducing the effects of visible plumes, though this would be highly site-specific. As such, EPA has considered the subject of visible plumes to be a small issue when weighed against the serious aquatic environmental impacts of once-through cooling.

In the development of the final rule, the Agency considered the land area required for installation of cooling towers at new power plants. The Agency examined the sensitivity of costs to new power plants of purchasing additional land for (1) installing mechanical draft cooling towers in lieu of once-through cooling (for those power plants expected to incur the costs of cooling towers only) and (2) providing land area buffers for plumes at a portion of facilities. The Agency determined the final annualized costs were not sensitive to the described changes in land costs. The Agency also understands that the costs of these land acquisitions as a portion of total project costs for new power plants are negligible. In addition, because this rule applies to new facilities which have the ability, in the majority of cases, to alter the design and location of their facilities without encountering most of the hurdles associated with retrofitting existing facilities, the issue of additional land acquisition is not as significant.

The Agency considers the issue of plume "re-entrainment" to be an issue that has been well addressed by designers and operators of wet cooling towers. The technology is mature and well designed after many decades of use throughout the world in a variety of climates. The Agency considers plume re-entrainment at the nine power plants projected to upgrade their cooling system to be a small effect. For wet cooling towers, the plume re-entrainment value occasionally referenced is 2 percent (Burns & Micheletti 2000). This value, in the Agency's estimates would not appreciably impact cooling tower performance, nor have a discernable environmental impact.

3.5.2 Displacement of Wetlands or Other Land Habitats

Mechanical draft cooling towers can require land areas (footprints) approaching 1.5 acres for the average sized new cooling tower projected for this rule. When determining the area needed for wet cooling towers, plants generally consider the possible plume effects, and plan for the amount of space needed to minimize the effects of local fogging and icing and to minimize re-entrainment of the plume by the tower. The land requirements of mechanical draft wet cooling towers at new combined-cycle power plants generally do not approach the size of the campus. Dry cooling towers generally require approximately 3 to 4 times the area of a wet tower for a comparable cooling capacity. In consideration of displacement of wetlands or other land and habitat due to the moderate plant size increases due to cooling tower installations at nine facilities, the Agency determined that existing 404 programs would more than adequately protect wetlands and habitats for these modest land uses.

3.5.3 Salt or Mineral Drift

The operation of cooling towers using either brackish water or salt water can release water droplets containing soluble salts, including sodium, calcium, chloride, and sulfate ions. Additionally, salt drift may occur at fresh water systems that operate recirculating cooling water systems at very high cycles of concentration. Salt drift from such towers may be carried by prevailing winds and settle onto soil, vegetation, and waterbodies. Commenters expressed the concern that salt drift may cause damage to crops through deposition directly on the plants or accumulation of salts in the soil. The cooling tower system design and the salt content of the source water are the primary factors affecting the amount of salt emitted as drift. In addition, modern cooling towers utilize advanced fill materials that have been developed to minimize salt or mineral drift effects. The Agency estimates that the typical plant installing a cooling tower as a result of the requirements of this rule will equip the tower with modern splash fill materials. As such, the Agency has applied capital costs for the abatement of drift in the compliance costs of this rule.

In the cases where it is necessary, salt drift effects (if any) may also be mitigated by additional means that are similar to those used to minimize migrating vapor plumes (that is, through acquisition of buffer land area surrounding the tower). Additionally, modern cooling towers are designed as to minimize drift through the use of drift elimination technologies such as those costed by the Agency. NUREG-1437 states the following concerning salt/mineral drift from cooling towers: "generally, drift from cooling towers using fresh water has low salt concentrations and, in the case of mechanical draft towers, falls mostly within the immediate vicinity of the towers, representing little hazard to vegetation off-site. Typical amounts of salt or total dissolved solids in freshwater environments are around 1000 ppm (ANL/ES-53)." The Agency projects that four of the nine power plants which will upgrade their cooling system from once-through to recirculating closed-cycle will utilize freshwater sources, where salt drift will not be an issue. The Agency anticipates that the other five plants (each a combined-cycle design) will utilize estuarine/tidal water sources for cooling and that the issue of salt drift at these plants is of small significance and can be mitigated. This conclusion is supported by those reached in NUREG about salt-drift upon extensive study at existing nuclear plants: "monitoring results from the sample of [eighteen] nuclear plants and from the coal-fired Chalk Point plant, in conjunction with the literature review and information provided by the natural resource agencies and agricultural agencies in all states with nuclear power plants, have revealed no instances where cooling tower operation has resulted in measurable productivity losses in agricultural crops or measurable damage to ornamental vegetation. Because ongoing operational conditions of cooling towers would remain unchanged, it is expected that there would continue to be no measurable impacts on crops or ornamental vegetation as a result of license renewal. The impact of cooling towers on agricultural crops and ornamental vegetation will therefore be of small significance. Because there is no measurable impact, there is no need to consider mitigation. Cumulative impacts on crops and ornamental vegetation are not a consideration because deposition from cooling tower drift is a localized phenomenon and because of the distance between nuclear power plant sites and other facilities that may have large cooling towers."

3.5.4 Noise

Noise from mechanical draft cooling towers is generated by falling water inside the towers plus fan or motor noise or both. However, power plant sites generally do not result in off-site levels more than 10 dB(A) above background (NUREG-1437 Vol. 1). Noise abatement features are an integral component of modern cooling tower designs, and as such are reflected in the capital costs of this rule, which were empirically verified against real-life, turn-key costs of recently installed cooling towers. A very small fraction of recently constructed cooling towers also further install noise abatement features associated with low noise fans. The Agency collected data on recently constructed cooling tower projects from cooling tower vendors. The Agency obtained detailed project descriptions for these 20 projects and none utilize low noise fans. In addition, the cost contribution of low noise fans, in the rare case in which they may be installed at a new facility, would comprise a very small portion of the total installed capital cost of the cooling system. As such, the Agency is confident that the issue of noise abatement is not critical to the evaluation of the environmental side-effects of cooling towers. In addition, this issue is primarily in terms of adverse public reactions to the noise and not environmental or human health (i.e., hearing) impacts. The NRC adds further, "Natural-draft and mechanical-draft cooling towers emit noise of a broadband nature...Because of the broadband character of the cooling towers, the noise associated with them is largely indistinguishable and less obtrusive than transformer noise or loudspeaker noise."

3.5.5 Solid Waste Generation

For cooling towers, recirculation of cooling water increases solid wastes generated because some facilities treat the cooling tower blowdown in a wastewater treatment system, and the concentrated pollutants removed from the blowdown add to the amount of wastewater sludge generated by the facility.

EPA has accounted for solid waste disposal from cooling tower blow-down wastewater treatment in the operation and maintenance costs of this rule. EPA reiterates that only nine power plants would incur the costs to install wet cooling towers as a result of this rule. The associated solid waste disposal increases for these plants would be extremely small compared to the scope of facilities covered by the rule and negligible for the industry as a whole.

3.5.6 Evaporative Consumption of Water

Cooling tower operation is designed to result in a measurable evaporation of water drawn from the source water. Depending on the size and flow conditions of the affected waterbody, evaporative water loss can affect the quality of aquatic habitat and recreational fishing. Once-through cooling consumes water, in and of itself. According to NUREG-1437, "water lost by evaporation from the heated discharge of once-through cooling is about 60 percent of that which is lost through cooling towers." NUREG-1437 goes on to further state, "with once-through cooling systems, evaporative losses...occur externally in the adjacent body of water instead of in the closed-cycle system." Therefore, evaporation does occur due to heating of water in once-through cooling systems, even though the majority of this loss happens down-stream of the plant in the receiving water body.

The Agency has considered evaporation of water and finds these issues not to be significant for this rule. The Agency notes, again, that 90 percent of the in-scope power plants will install cooling towers regardless of the requirements of this rule. The nine other facilities, which may comply with the rule either through installation of flow reduction technologies similar to cooling towers (such as recirculating cooling lakes, cooling canals, or hybrid wet-dry cooling towers) or compliance with track II, are expected to consume approximately 127,000 gallons per minute (evaporative loss) when all new plants are operating. This represents less than three (3) percent of the baseline intake flow of the power plants within the scope of the rule. As a percentage of the total flow of water used for electricity generation in the US, this represents 0.1 percent. See DCN 3-3085.

3.5.7 Manufacturers

The Agency notes that the discussion thus far concerning side effects has focused exclusively on power plants. The Agency expects that 29 manufacturers will incur costs equivalent to installations of closed-cycle wet cooling towers as a result of this rule. However, even though these costs reflect cooling tower installations, the Agency projects that manufacturing facilities will comply, in the majority of cases, with this rule through the adoption of recycling and reuse design changes and operational practices at their plants. Therefore, the majority of issues discussed in this section are not of concern to manufacturing facilities for the final rule nor is the issue of energy penalties.

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