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Comparison of Alternate Cooling Technologies for California Power Plants
Economic, Environmental and Other Tradeoffs

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Comparison of Alternate Cooling Technologies for California Power Plants

Economic, Environmental and Other Tradeoffs

Final Report, February 2002

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ABSTRACT

This study defines, explains, and documents the cost, performance, and environmental impacts of both wet and dry cooling systems. A survey of the cooling system literature is provided in an annotated bibliography and summarized in the body of the report. Conceptual designs are developed for wet and dry cooling systems as applied to a new, gas-fired, combined-cycle 500-MW plant (170 MW produced by the steam turbine) at four sites chosen to be representative of conditions in California. The initial capital costs range from \$2.7 to \$4.1 million for wet systems using mechanical-draft wet cooling towers with surface steam condensers and from \$18 to \$47 million for dry systems using air-cooled condensers.

Cooling system power requirements for dry systems are four to six times those for wet systems. Dry systems, which are limited by the ambient dry bulb temperature, cannot achieve as low a turbine back pressure as wet systems, which are limited by the ambient wet bulb. Therefore, heat rate penalties and capacity limitations are incurred at some sites depending on local meteorology. A methodology is developed and illustrated that accounts for these several components of cost and performance penalties in selecting an optimized design for a specific site.

A brief review is given of some advanced cooling system technologies currently in development, highlighting an evaporative condenser system with a water-conserving mode that halves the consumptive water use of a conventional wet system. In addition, current research in the power plant cooling field is reviewed with particular attention to concepts for enhancing the performance of dry systems during the peak period (the hottest hours of the year).

ACKNOWLEDGMENTS

The performance of this study was greatly aided and the quality of its results improved by the contributions of many people and organizations.

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These acknowledgements should not be interpreted as meaning that all of the participants are in complete accord with or endorse the results and conclusions of this study. Where differences in opinion or point of view remain, the final report reflects those of the author.

CONTENTS

LIST OF FIGURES

LIST OF TABLES

1
INTRODUCTION

This report sets forth the results of a study conducted for the California Energy Commission (CEC) and EPRI to define, explain, and document the performance, economic, and environmental tradeoffs among the cooling system alternatives—wet, dry, and hybrid (wet/dry)—available for use on new combined-cycle power plants. While much of the information may be more widely applicable, the focus of the analysis and the case study characteristics are specific to conditions in California.

The motivation for investigating this topic is a desire to allocate properly the water resources in the state. Specifically, a conflict may sometimes arise between cooling water requirements for electric power generation and water requirements for agricultural, residential, commercial, industrial, and other needs, as well as for in-stream flow maintenance.

The problem is not new. Regulatory action to maintain a properly balanced allocation goes back at least to 1975 with the promulgation of Resolution #75-58 by the State Water Resources Control Board (SWRCB, 1975). Stating “The use of inland waters for powerplant cooling needs to be carefully evaluated to assure proper future allocation of inland waters considering all other beneficial uses,” the SWRCB established the principle that

> “[w]here the Board has jurisdiction, use of fresh inland waters for powerplant cooling will be approved by the Board only when it is demonstrated that the use of other water supply sources or other methods of cooling would be environmentally undesirable or economically unsound.”

The Drivers

California has experienced high growth rates in many dimensions for decades. The state’s population has more than tripled in the past 50 years and is expected to nearly double again in the next 50. Figure 1-1 displays the historical population growth since 1850 and estimated population up to 2040 (California Department of Finance, 1999, 1998). Concurrently, the state’s economy has grown vigorously. Figure 1-2 shows historical growth as represented by the California Gross State Product expressed in current dollars (U.S. Department of Commerce, 2000). The combined effects of a growing population and expanding agricultural, commercial, and industrial activity have imposed growing demands for both water and electric power.

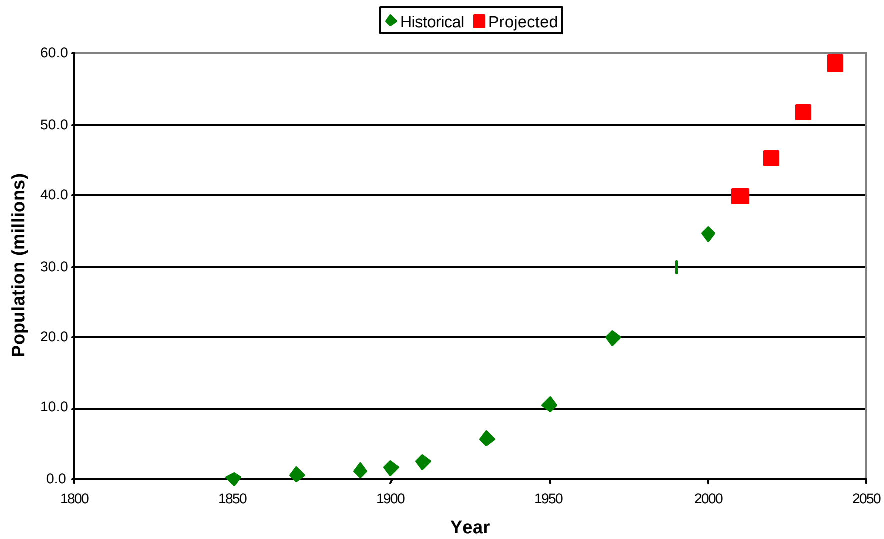

Figure 1-1
California Population: History and Projections

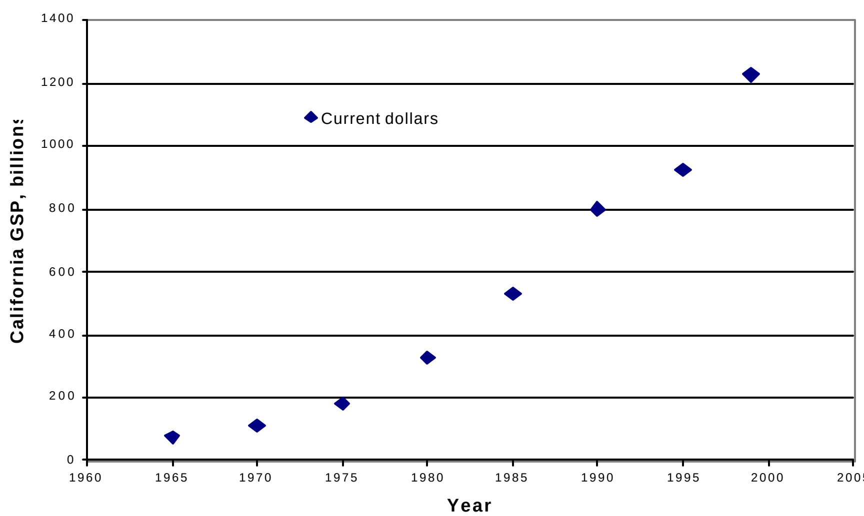

Figure 1-2
California Gross State Product

Figure 1-3 displays urban[1] water use in California, which has risen steadily since the 1940s (SWRCB, 1996) and, in spite of increasing awareness of the need for conservation, is projected

[1] Defined in the California Water Plan (CalDWR, 1998) as "residential, commercial, industrial and institutional" use excluding "agricultural" and "environmental" uses.

to continue to rise for the foreseeable future. While water surpluses and shortages are distributed unevenly across the state, projections through 2020 predict likely shortfalls under average conditions and substantial shortfalls under drought conditions.

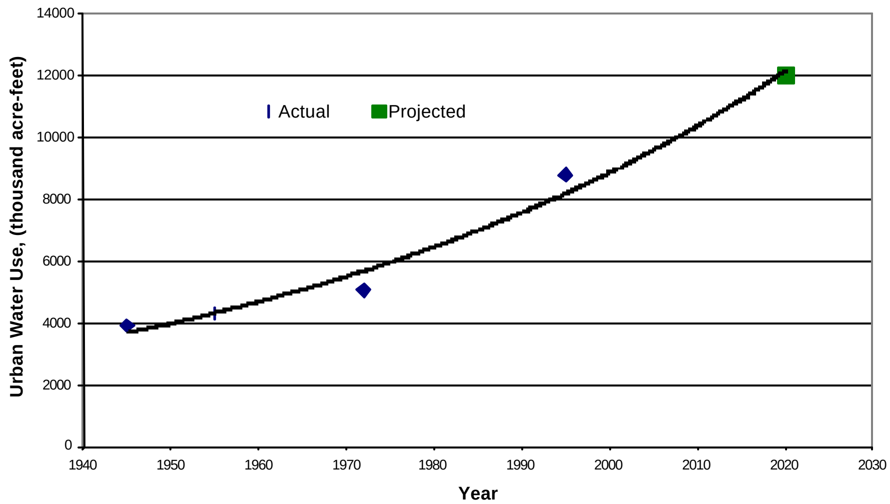

Figure 1-3
California Water Use: Actual and Projected

Electric power generation utilizes water in many ways and in varying amounts depending on the type of generating plant and the type of cooling system employed. The primary use of water is for the condensation of steam, often referred to as power plant cooling. There are several types of power plant cooling systems. These are commonly categorized as once-through cooling, recirculating wet cooling, dry cooling and hybrid, or wet/dry, cooling. These systems, which are described in more detail in later sections of this report, vary widely in the amount of water withdrawn from the environment and in the amount consumed in the plant through evaporation.

In California, the use of once-through systems is largely restricted to existing coastal plants using ocean water for cooling. The focus of this report is on the cooling of inland plants using fresh surface waters or groundwater with either recirculating wet, dry, or hybrid wet/dry systems.

Recirculating wet systems have been the usual method of cooling for inland plants in California and throughout the U.S since the mid-1970s. As will be developed in detail in Section 2, recirculating systems with wet cooling towers consume water at a rate of 10 to 15 gallons per minute (gpm) per MW of electric power *generated with steam-driven turbines*.

A common metric for water use in electric power production used, for example, in the Environmental Performance Report of California's Electric Generation Facilities (CEC, 2001a) is gallons of water per megawatt-hour (gal/MWh) of energy generated. The amount of water used depends not only on the type of cooling system used but also on the type of plant. Simple-cycle plants or combined-cycle plants in which most of the power is generated by gas turbines require no, or much less, water for condensation of steam, but they use water for gas turbine inlet cooling, emissions control, auxiliary equipment cooling, plant maintenance, etc. Stand-alone thermal steam plants and the steam portion of combined-cycle plants use the bulk of their water for steam cycle cooling. This portion of the water use can be eliminated or reduced by the use of dry or wet/dry cooling systems.

Table 1-1 summarizes some nominal values of water use for various plant types with alternative choices of systems for steam cycle cooling.

Table 1-1
Water Requirements for Power Generation (in Gallons per MWh of Plant Output)

Plant Type	Steam Condensing	Auxiliary Cooling and Hotel Load	Total
Stand-alone steam plant	720[(1)]	30[(2)]	750
Simple-cycle gas turbine	0	150[(3)]	150
Combined-cycle plant (2/3 CT + 1/3 steam)	240 (1/3 x 720)	110 (2/3 x 150 + 1/3 x 30)	350
Combined-cycle plant with dry cooling	0	110	110
Stand-alone steam plant with dry cooling	0	30	30

(1) evaporation + blowdown = 12 gpm/MW (see Section 2)
(2) estimated at ~5% of evaporation + blowdown
(3) mid-range of 75 – 200 gal/MWh in Environmental Performance Report (CEC, 2001) for turbine cooling, emissions control and hotel load

The California Energy Commission (CEC, 2001a) estimates that of the 53.2 GW of generating capacity in California, 40% uses once-through cooling and another approximately 30% is provided by hydro or wind facilities. This leaves perhaps 30%, or about 16,000 MW, on evaporative cooling. Assuming a mix of plant types with an average water use of 300 gal/MWh, the total water requirement would be approximately 100,000 acre-feet per year or about 1% of the state’s urban water use. While this is a part of total water consumption in the state, it can be a significant local use in the vicinity of the plant and is frequently a contentious issue during siting discussions.

Requirements for electricity in California have increased steadily for the past several decades (Figure 1-4), at an annual rate of approximately 3.5 to 4% (CEC, 2000)—except for during a period of economic slowdown in the early 1990s (see Figure 1-2). Projected increases of 20% over the next 10 years are consistent with population increase of 20% and economic activity increase of 35% by 2020 (EPRI, 2000).

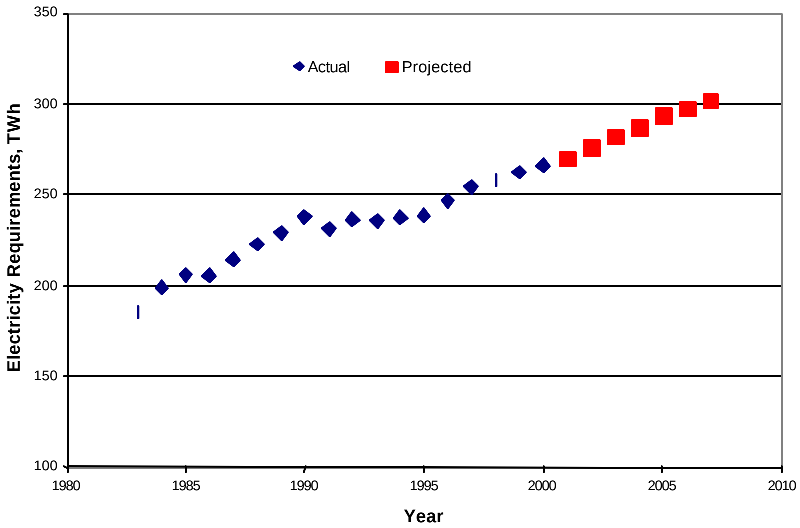

Figure 1-4
Electricity Requirements: Actual and Projected

For a long time, the electricity consumed by California users has included large amounts of power imported from outside the state. In recent years, growth in neighboring states and a hiatus in the construction of new generating facilities within the state have led to increasingly tight supply margins.

For the past few years, increasing awareness of potential power shortages coupled with potentially attractive business opportunities offered by the newly deregulated California electricity market have led to proposals for the licensing and construction of a number of new plants within the state. At present, 50 plants—representing over 21 GWe of new generating capacity—have been approved or are under review by the CEC (Table 1-2).

Table 1-2
California Power Plants Approved or Under Review by the CEC as of October 9, 2001 (CEC, 2001)

Status	Number of Plants	Total New Capacity (MWe)
Approved	31	11,983
—Greater than 300 MWe	17	11,039
—Less than 300 MWe	14	944
Under review	19	9,523

Recognizing the urgent need for both electricity and water, the CEC and EPRI have sponsored this study to address the questions of what technological alternatives might exist to the consumptive use of fresh water for the cooling of electric power plants in the California environment, and what their economic and environmental costs might be.

Cooling System Options

Technological options for power plant cooling are described in detail in Section 2 of this report. From a water consumption standpoint, the major categories of cooling systems can be characterized as follows.

Once-Through Systems

Water is withdrawn from the environment, passed through a steam condenser and returned, slightly heated (typically by 20 to 25ºF), to the source. Withdrawal rates are typically in range of 500 gpm per MWe. No water is consumed or evaporated within the cooling system, but the evaporation rate from the receiving water is slightly higher in the vicinity of the discharge plume.

Recirculating Wet Systems

In recirculating wet systems, smaller amounts (typically 2 to 3% of the amount withdrawn for once-through cooling) are taken into the plant, but the majority is evaporated in the cooling equipment (in mechanical or natural draft cooling towers), with very little water returned to the source. Water withdrawn from a local source is circulated continuously through the cooling system. The cooling system must be replenished with "make-up water" to replace that lost to evaporation and blowdown.

Dry Systems

In dry systems, the ultimate heat rejection to the environment is achieved with air-cooled equipment that discharges heat directly to the atmosphere by heating the air. Dry systems are of two types: direct and indirect. Direct systems duct the steam to air-cooled condensers (ACCs) that can be either mechanical or natural draft units. Indirect systems condense the steam in water-cooled surface condensers, from which the heated water is pumped to air-cooled heat exchangers where it is cooled and then re-circulated to the steam condenser. Dry systems reduce water use at a plant by eliminating the use of water for steam condensation. In most cases, the remaining water use, totaling perhaps 5% of the amount used in recirculating systems, is required for boiler make-up, other cooling applications, and the so-called "hotel load."

Additionally, in some simple-cycle or combined-cycle gas turbine plants, significant amounts of water are injected into the turbines for NO_x reduction. In some systems, spray augmentation of dry system performance has been used or considered. This would require modest additional water use during the hottest times of the year.

Hybrid Wet-Dry Systems

In hybrid wet-dry systems, both wet and dry components are included in the system, and they can be used separately or simultaneously for either water conservation or plume abatement purposes. A combination of wet and dry cooling technology is used; depending on system configuration, water consumption can approach that of recirculating wet systems or be much lower. Design studies have ranged from 30 to 98% reduction in water use compared to all wet recirculating systems (Mitchell, 1989).

In this report, the primary emphasis is on closed-cycle wet cooling using mechanical draft wet cooling towers and on direct dry systems using mechanical draft air-cooled condensers, since these systems are the ones that will be found almost exclusively on new combined-cycle plants in California. However, descriptions of each of the systems mentioned above are included in Section 2 for convenience of reference for readers unacquainted with types of cooling systems and their associated terminology.

Dry cooling has been used rarely in the U.S. until recently. A history of the U.S. and global use of dry cooling is reviewed briefly in Section 3.

Tradeoffs

The choice among the major categories of cooling systems involves a number of tradeoffs. As noted above, the SWRCB policy for allocating fresh water resources in California requires other than fresh inland water to be used for power plant cooling unless the alternatives are shown to be "environmentally undesirable or economically unsound" on the basis that there is a "limited supply of inland water resources in California." Therefore, the primary tradeoffs are between water use and the cost of electric power. Other environmental considerations exist, but they are normally secondary in importance.

The emphasis of the remainder of this report is on an in-depth analysis of the comparative costs and the environmental impacts of alternative cooling systems. However, to provide some background and context for the discussion, qualitative comparisons between a "base" mechanical draft wet cooling system and other systems are given in Table 1-3. The bases for comparison are water consumption; system costs, including capital, O&M, and performance penalty costs; and environmental issues, including water withdrawal and discharge, drift, and visible atmospheric plumes.

Scope of Project

The purpose of this study is to

- Define and compare the current costs and performance in California of dry and hybrid wet/dry cooling towers relative to wet cooling towers;
- Identify the environmental benefits and tradeoffs between wet, dry, and hybrid wet/dry cooling towers;

- Identify future research that can improve the costs and performance of wet, dry, and hybrid wet/dry cooling towers; and
- Identify any other alternative power plant cooling technologies that can improve the environmental and public health costs/risks of California's electricity.

The project has two major elements:

1. Select representative regions in California for a detailed analysis of the cost, performance, and environmental tradeoffs among dry, hybrid, and wet cooling towers; and

2. Identify current research that can improve the relative costs and performance of the dry and hybrid cooling towers and evaluate the relative merits and gaps in this research for applicability in California.

Table 1-3
Tradeoffs Among Various Types of Cooling Systems

Tradeoff	Cooling System Type				
	Once-Through	**Wet Mechanical Draft**	**Dry Air-Cooled Condenser**	**Hybrid Wet/Dry**	
				Plume Abatement	**Water Conservation**
Water Consumption	Minor	8 to 12 gpm/MWe	~ 0 to <5% of wet tower	~ equal to wet tower	20% to 80% of wet tower depending on design
Capital Cost	<< **BASE**	**BASE**	1.5x to 3x base	1.1x to 1.5x base	3x to 5x base
O&M Cost	**< BASE** Pump maintenance, condenser cooling	Highly site specific; fan/pump power; water treatment; tower fill/ condensate cleaning	Finned surface cleaning; gearbox maintenance; fan power	Similar to **BASE**	Similar to **BASE**
Performance Penalty	< **BASE** Base penalty depends on site meteorology	**BASE**	Highly site specific—5% to 20% capacity shortfall on hot and windy days	~ Equal to base	Highly variable, depending on mode of operations
Water Withdrawal	~500gpm/MWe	10 to 15 gpm/MWe	None	~ Equal to base	Variable, but can be reduced from wet by amount of water conservation
Discharge	~500 gpm/MWe; thermal plume and residual chlorine issues	2 to 5 gpm/MWe	None	~ Equal to base	Variable, as with withdrawal
Drift	NA	Negligible;< 0.001% of circulating water flow	None	Small reduction from wet	Similar to wet when used in wet mode
Plume	NA	Visible plume on cold, humid days	None	None	None during normal operating schedules

The work was carried out in several tasks:

1. Define the state-of-the-art of alternative cooling system technologies for gas turbine combined-cycle plants by
   - conducting a literature search and preparing an annotated bibliography, and
   - describing selected existing units and their operating and maintenance experience.
2. Compare the costs, performance, and environmental effects of wet and dry cooling systems by
   - selecting four case study sites representative of the range of operating conditions in California;
   - conducting engineering analyses to determine the capital, operating, and maintenance costs and system performance for each of the alternative cooling systems optimized for conditions at each site;
   - providing bases of comparison, figures of merit, comparisons, and rankings for the alternative systems; and
   - quantifying the environmental effects and describing the advantages and disadvantages of each technology.
3. Identify and discuss current areas of research directed at improving the performance, reducing the installed and operating costs, and minimizing the environmental impacts of power plant cooling systems; and recommend areas worthy of expanded support for the benefit of California citizens.
4. Identify potential alternative technologies and discuss the advantages and disadvantages compared to current systems; review the development status or commercial status and deployment experience; and describe potential barriers to their utilization and opportunities for reducing or removing the barriers.

Organization of Report

The remaining sections of this report describe the methodology, results, and recommendations of the work conducted for the CEC and EPRI under the Tailored Collaboration project, “Wet, Dry, Hybrid Wet/Dry, and Alternative Cooling Technologies.”

Section 2 provides background information on cooling system alternatives for gas turbine combined-cycle power plants. Section 3 defines the state of the art for wet, dry, and hybrid cooling tower technologies on the basis of a survey of the literature, identification of existing and planned installations of alternative cooling systems in the United States and abroad, and discussion of the operating experience of the owner/operators of four dry-cooled plants. Section 4 reviews the selection of four case study sites representative of the range of cooling system operating conditions in California and presents the characteristics of each site. Section 5 develops the cost comparison methodology, assembles the cost information, applies the approach to each of the four case studies, and provides an analysis/synthesis of the results that can be used for more generalized comparisons and for understanding the effect of selected site-specific conditions. Section 6 provides a brief comparison of the environmental effects of the different systems.

The next two sections look to the future: Section 7 reviews current research and development activities with promise for addressing some of the problems associated with water-conserving cooling systems, and Section 8 identifies and describes emerging technologies that may become preferred alternatives to current technology. Section 9 summarizes the conclusions and recommendations resulting from the study. Several appendices contain much of the raw information obtained in the course of the study.

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2
POWER PLANT COOLING SYSTEMS

The following section provides general background information on the typical power plant being proposed for construction in California at the present time and on the cooling system alternatives to be considered.

Combined-Cycle Plants

A simple schematic of a typical gas-fired combined-cycle plant is shown in Figure 2-1.

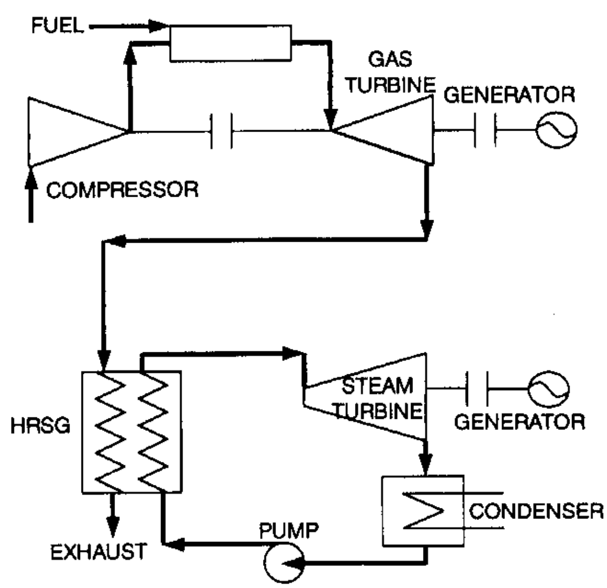


Figure 2-1
Schematic of Typical Combined-Cycle Power Plant

Combined-cycle power plants utilize one or more gas turbines, each driving an electrical generator and providing the heat input to a steam cycle. Hot exhaust gas from the combustion turbine is passed through a heat recovery steam generator (HRSG) to produce steam for a steam turbine. Turbine exhaust steam is condensed in the cooling system and returned to the HRSG. The steam cycle may be more or less complex with single- or multi-pressure HRSG operation. Additionally, in cogeneration applications, steam may be extracted from various points in the cycle for process applications in a partner facility. Condensate may or may not be returned to the power cycle. These characteristics of the steam cycle depend on the project objectives and economics.

The configuration of the steam cycle determines the quantity of steam to be condensed and, therefore, the heat load on the cooling system per unit of electric power generated. The plant configuration chosen for this study is a so-called 2 x 1 arrangement in which two gas turbines feed a single HRSG and steam turbine; no steam is extracted for cogeneration applications.

Approximately two-thirds of the plant output is obtained from the gas turbine side. Nominal overall plant efficiency is about 55%, compared to 38% for newer simple-cycle gas turbines or 35 to 40% for modern fossil steam plants[2] (stand-alone gas-fired boiler and steam turbine). The gas turbines can be either heavy duty or aeroderivative in type. If heavy-duty turbines are used, exhaust gas flow rates and temperatures are higher, enabling the generation of more steam at higher temperatures and hence more output at higher efficiencies from the steam side. However, overall plant efficiency is slightly lower due to the lower efficiency of the gas turbine side of the plant. These tradeoffs are beyond the scope of this discussion, for a more detailed treatment see Drbal (1996).

The base case plant is characterized in Table 2-1, which is typical of modern components operating in combined cycle mode (Leyzerovich, 2001; EPRI TAG, 1998).

Cooling System Alternatives

As in all steam-electric power plants, steam turbine exhaust must be condensed in order to maintain the required sub-atmospheric turbine exit pressure and to return the condensate to the HRSG. Condensing system alternatives, listed in Section 1 under “Cooling System Options,” include once-through cooling, wet cooling towers, dry cooling systems, and hybrid (wet/dry) systems. These are described briefly in the following paragraphs. Once-through cooling is introduced briefly for completeness and for convenience of reference for readers unfamiliar with power plant cooling systems. Since it is not, however, an option for inland power plants in California, it will not be discussed further beyond this section of the report. The discussion will also serve to define some of the terminology and nomenclature used in the rest of the report.

[2] Conventionally referenced to HHV (higher heating value) for steam plants and to LHV (lower heating value) for simple or combined-cycle gas turbine systems. The difference for gas is approximately 10% ($\varsigma_{LHV} \simeq 1.1 \varsigma_{HHV}$).

Table 2-1
Steam Cycle Cooling System Conditions

Quantity	Value
Nominal plant capacity	500 MWe
Configuration	2 x 1
Gas turbine output	165 MW each; (330 MW total)
Steam turbine output	170 MWe
Steam turbine exhaust flow	1,000,000 pounds per hour @ 5% moisture
Turbine back pressure	2.5 in Hga; (T_{cond} = 108.7 ºF)
Cooling system heat load	980 million Btu per hour
Performance Characteristics	
Plant heat input	3.1×10^9 Btu per hour
Overall plant efficiency, LHV	55%
Steam turbine heat rate	9,200 Btu/kwh @ 2.5 in Hga
Steam turbine efficiency, LHV	37%
Gas turbine efficiency, LHV	36%

Once-Through Cooling

In this simplest case, cooling water is drawn from a local source (i.e., ocean, lake, river, or other), passed through the condenser tubes, and returned to the environment at a higher temperature. Steam is condensed in a shell-and-tube surface condenser. The system is shown schematically in Figure 2-2.

For the base plant defined in Table 2-1, the cooling system load is met with circulating water flows of 500 to 750 gpm/MWe (30,000 to 45,000 gal/MWh) and a water temperature rise of about 15 to 25ºF.

Advantages are

- Highest efficiency,
- Lowest installation and operating costs, and
- Low water consumption.

Disadvantages are

- Highest withdrawal rates,
- Entrainment/impingement, and
- Thermal discharge plume.

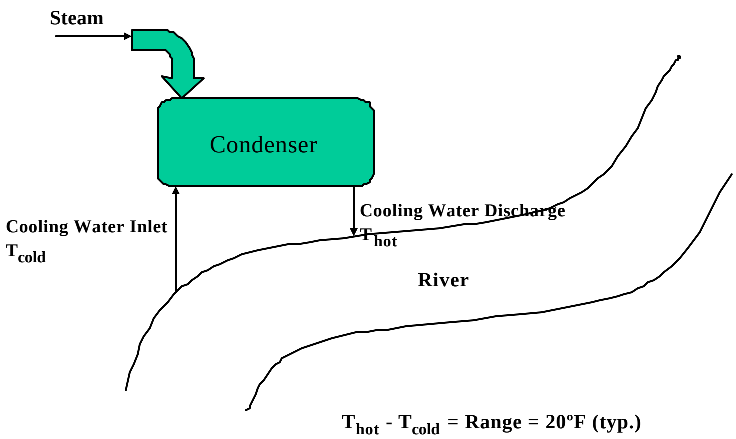


Figure 2-2
Power Plant Cooling System Arrangements: Once-Through Cooling

Recirculating Wet Cooling

Recirculating cooling systems are similar to once-through systems in that the steam is condensed in water-cooled shell-and-tube surface condensers but unlike them in that the heated cooling water is not returned to the environment. Instead, it is cooled in evaporative cooling towers and recirculated to the condenser. A typical system is shown schematically in Figure 2-3.

Advantages are

- Reduced withdrawal rates, and
- Reduced entrainment/impingement.

Disadvantages are

- Decreased plant efficiency,
- Higher capital cost,
- Higher water consumption/evaporation,
- Visible plume/drift emissions,
- Wastewater treatment requirements,

- Chemical treatment programs,
- Emissions of controlled air pollutants or pathogens, and
- Site space.

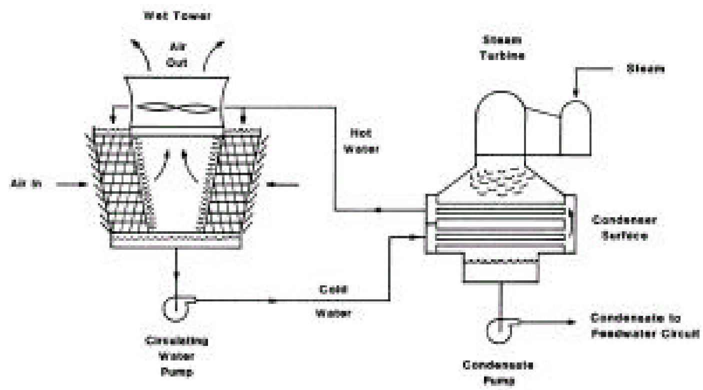


Figure 2-3
Recirculating Wet Cooling Tower

The cooling is achieved by the evaporation of a small fraction (1 to 2%) of the recirculating water flow. Therefore, once the system is filled, the only water withdrawn from the environment is makeup water in amounts sufficient to replace that lost to evaporation, blowdown,[3] and drift.[4] These quantities are discussed in more detail in Section 6, *Environmental Effects*. Therefore, withdrawal rates from the environment are much less than for once-through systems—typically 10 to 15 gpm/MWe (600 to 900 gal/MWh).

There are two common types of wet cooling tower—mechanical draft and natural draft (Figure 2-4). Natural draft towers, in which the airflow through the tower is induced by density differences in a "chimney effect," are less likely to be considered in many areas of California on the basis of aesthetic and seismic concerns. The higher capital costs in comparison to mechanical draft towers are also less favorable for the smaller steam condensation loads (less than 250 MWe on steam) associated with combined-cycle plants. Therefore, they will not be considered further in this study.

Mechanical draft towers are built in both cross-flow and counter-flow designs. Hot water from the condenser is introduced at the top of the tower and flows down through a "fill" section where it is brought into intimate contact with ambient air flowing across, or counter to, the direction of the falling water flow. Both sensible and latent heat transfer to the air cools the bulk of the water,

---

[3] Blowdown is water discharged from the cooling system in order to control the buildup of dissolved and suspended materials that concentrate in the system as a result of the evaporation cycles.

[4] Drift refers to liquid water droplets entrained in the tower exit plume and released to the atmosphere.

which is then collected in a basin and returned to the condenser. The air leaving the tower is heated and humidified to an essentially saturated plume.

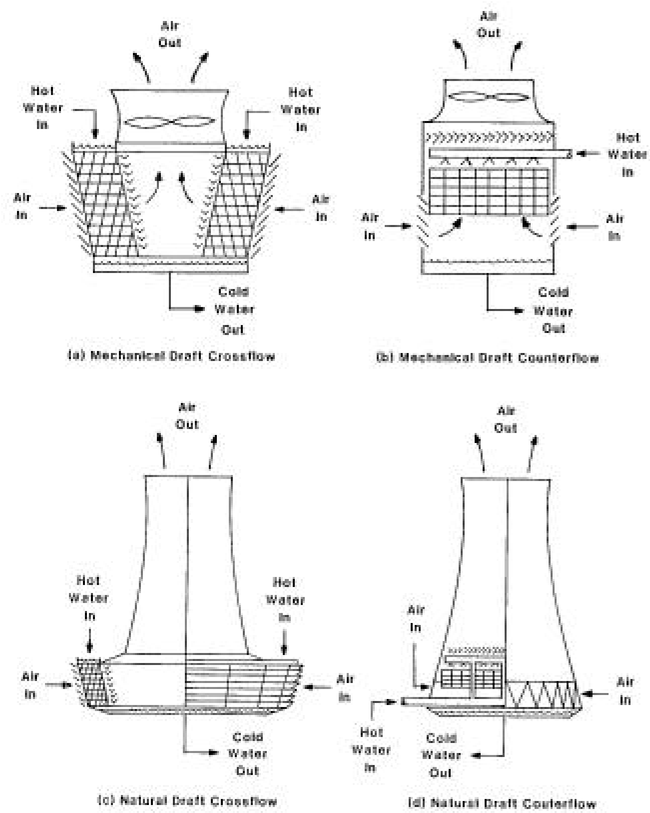


Figure 2-4
Mechanical- and Natural-Draft Cooling Towers

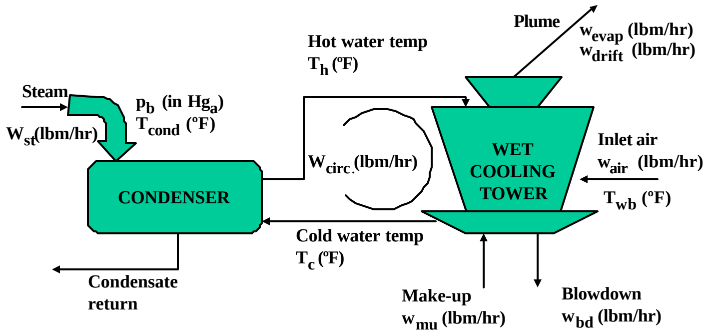
Mass and Heat Balances

The elements of a complete heat balance are shown in Figure 2-5. A comprehensive treatment of the heat and mass transfer processes in the tower is unnecessary for the scope of this study (full treatments are available in many references; e.g., Drbal, 1996; Kroeger, 1998). A summary, to develop some "rules of thumb," is given below for convenience of reference. The symbols used here and elsewhere in this report are listed in Table 2-2 at the end of the section.

With reference to Figure 2-5, the water balance on a wet cooling tower is given by

$$w_{mu} = w_{evap} + w_{bd} + w_d$$

The drift rate, w_d, is neglected in the following approximations.



$T_{cond} - T_h$ = Condenser Terminal Temp = 8 - 10 °F (typ.)
Difference (TTD)
$T_h - T_c$ = Range = 20 - 25 °F (typ.)
$T_c - T_{wb}$ = Approach = 8 - 15 °F (typ.)

$T_{cond} - T_{wb}$ = 35 - 50 °F

Figure 2-5
Cooling Tower Mass and Heat Balance

Evaporation Rate

The rate of evaporation of water from the tower is related to the heat load on the tower, Q_{tower}, which is equal to the heat load on the condenser, Q_{cond}, given by

$$Q_{tower} = Q_{cond} = w_{circ} \text{ x } c_p \text{ x } (T_h - T_c)$$

with the evaporation rate given by

$$w_{evap} = Q_{tower} \text{ x } f_{latent}/h_{fg}$$

where

h_{fg} = latent heat of vaporization (Btu/lbm); ~1000 Btu/lbm

f_{latent} = fraction of total heat rejected by latent heat transfer (0.9 used here; can be lower depending on ambient conditions and design choice)

For the base plant conditions from Table 2-1,

$$Q_{tower} = 980 \text{ x } 10^6 \text{ Btu/hr (for 170 MWe) and}$$

$$w_{evap} = 980 \text{ x } 10^6 \text{ x } 0.9/1000 = 0.88 \text{ x } 10^6 \text{ lbm/hr}$$

On a "per unit output basis,"

w_{evap} = 5.2 lb/kWh or ~ 10 gpm/MW

Blowdown Rate

Blowdown rates are set to control scaling, fouling, and corrosion by limiting the buildup of impurities in the circulating water. This criterion is normally expressed in terms of maximum allowable cycles of concentration (n), defined as the ratio of the concentration of conserved species in the circulating water ($C_{i\ circ}$) to that in the makeup water ($C_{i\ mu}$):

$$n = C_{i\ circ}/C_{i\ mu}$$

The mass balance of species i in the tower requires that

$$w_{mu} \times C_{i\ mu} = w_{bd} \times C_{i\ circ}$$

$$w_{bd} = (w_{evap} + w_{bd}) \times (C_{i\ mu}/C_{i\ circ}) = (w_{evap} + w_{bd}) \text{ x } 1/n$$

Therefore,

$$w_{bd} = w_{evap} / (n - 1)$$

Typical allowable cycles of concentration are from 3 to 6 (DiFilippo, 2001). For n = 6 as a typical value, the required blowdown is

$$w_{bd} = [1/(6 - 1)] \times w_{evap}$$

$$= 2 \text{ gpm/MWe}$$

and the required make-up is

$$w_{mu} = w_{evap} + w_{bd} = 12 \text{ gpm/MWe}$$

Additionally, typical consistent values of tower operating conditions are

Circulating water flow rate, w_{circ}: ~ 500 gpm/MWe
Condenser terminal temperature difference (TTD), $T_{cond} - T_h$: 7 to 10ºF
Tower range, $T_h - T_c$: 20 to 24ºF
Tower approach, T_c - $T_{amb.\ wet\ bulb}$: 8 to 15ºF

Therefore, the achievable steam condensing temperature is given by

$$T_{cond} = T_{amb.\ wet\ bulb} + \text{Approach} + \text{Range} + \text{TTD}$$

For an ambient wet bulb temperature of 70ºF, values in the typical ranges of TTD = 8ºF, Range = 20 ºF, and Approach = 11 ºF would provide a condensing temperature of 109ºF corresponding to a turbine back pressure of 2.5 in. Hga. Tower approach temperature depends on design ambient conditions as well as many other factors including tower type, size, fill choice, and air flow. In general, warmer, more humid conditions lead to lower approach temperatures in the southeastern U.S. and cooler, drier climates lead to higher ones in the northern and western regions.

Dry Cooling

Dry cooling systems may be categorized as direct or indirect. In direct systems, turbine exhaust steam is delivered directly to an air-cooled condenser (ACC), as shown in Figure 2-6. Heat rejection to the environment takes place in a single step, where steam is condensed inside finned tubes, which are typically arranged in an A-frame configuration, and is then cooled by air blown across the finned surfaces. As for wet cooling systems, the dry cooling tower can be either mechanical or natural draft in type. For reasons similar to those discussed for wet systems, this report will restrict attention to mechanical draft units.

Advantages are

- Least water consumption, and
- No entrainment/impingement losses.

Disadvantages are

- Highest installation and operating costs,
- Highest efficiency penalty,
- Increased air emissions/MCU,
- Load limitations on hottest days,
- Site space, and
- Possibility of increased unit trips.

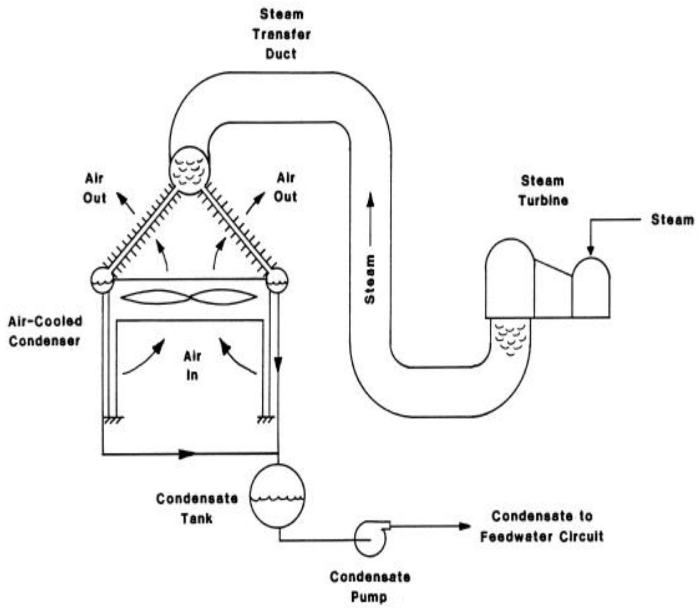


Figure 2-6
Dry (Air-Cooled) Cooling System

Indirect systems have a separate condenser. The condenser may be either a surface condenser of the conventional shell-and-tube type or a so-called barometric condenser, in which the steam is condensed directly on a spray of cooling water. These systems are shown schematically in Figure 2-7. In either case, the water against which the steam is condensed is then circulated to an air-cooled heat exchanger for ultimate heat rejection to the atmosphere.

Systems incorporating a barometric condenser are usually used in conjunction with a natural draft dry cooling tower. Known as Heller systems (Balogh, 1998) and developed and promoted by a Hungarian firm (EGI-Constructing Engineering Ltd., now owned by GEA-AG**,** originally Gesellschaft für Entstaubings-Anlage mbH), they have been used in several installations around the world. They are characterized by a higher initial cost and, in the case of the design using a surface condenser, a higher operating cost as well in most installations. For the plant sizes and business conditions prevalent in California at the present time, indirect systems are not considered further in this report. In the unlikely event that a plant on once-though cooling or using a wet tower should want to retrofit the system to a dry system, an indirect system would likely be the system of choice.

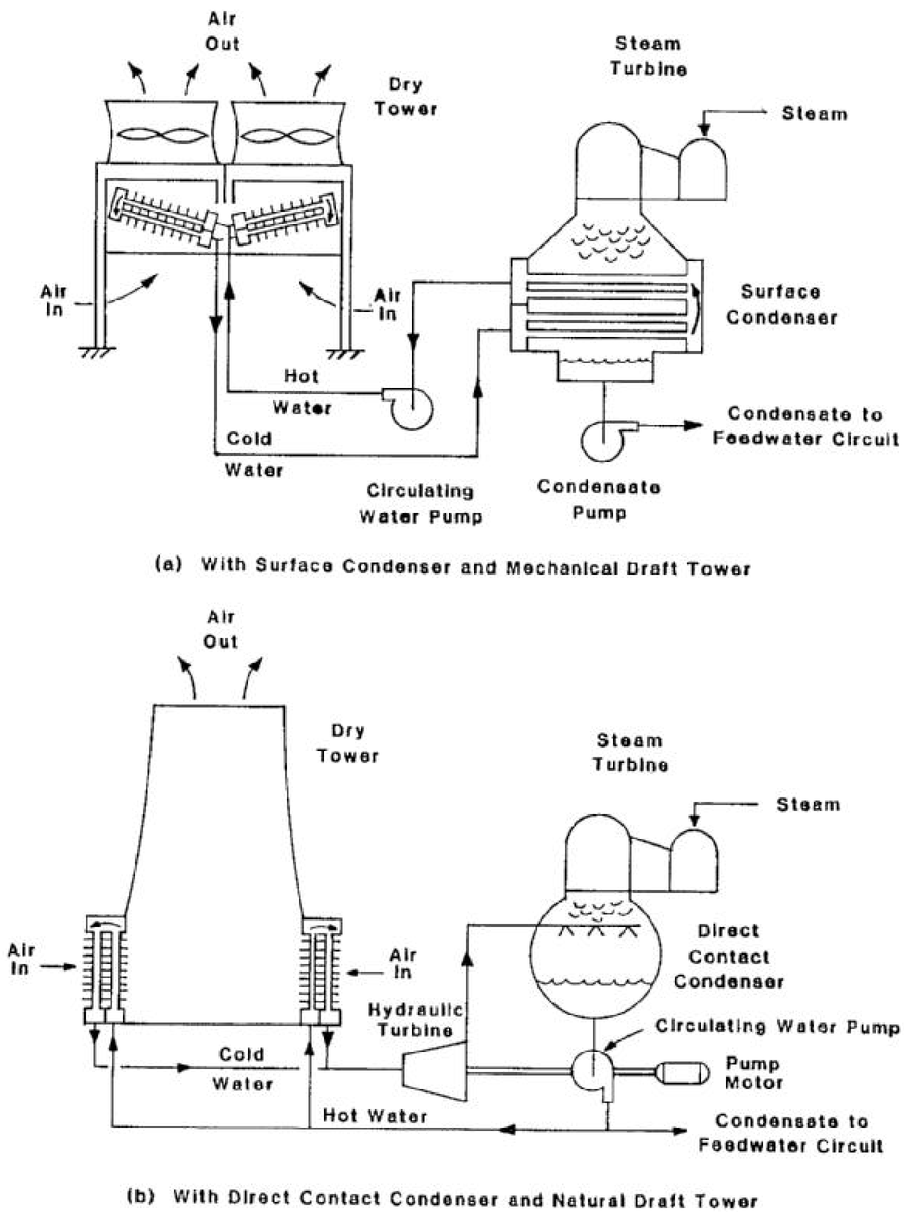


Figure 2-7
Dry Cooling System with Surface and Spray Condensers

Heat Balance

Figure 2-8 shows typical flows and temperatures for a direct dry cooling system with an ACC. The relevant atmospheric variable in this case is the ambient dry bulb temperature. The characteristic quantity for a dry unit is the initial temperature difference (ITD), defined as steam condensing temperature (T_{cond}) minus ambient temperature (T_{amb}).

$$ITD = T_{cond} - T_{amb}$$

Typical design choices range from 20 to 60°F, where a low ITD represents a larger, higher-capital-cost unit that is able to maintain a low condensing temperature on hot days. Conversely, a high ITD represents a smaller and less costly tower that results in higher condensing temperatures, higher turbine back pressures, and lower plant efficiency during hot periods.

Typical designs result in a 25 to 35°F rise as the air passes through the unit.

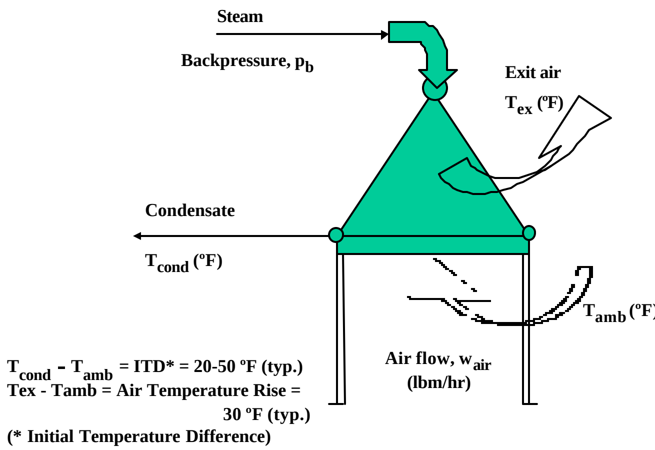


Figure 2-8
Flows and Temperatures in an Air-Cooled Condenser

Hybrid Wet/Dry Systems

Hybrid systems employ a combination of both wet and dry cooling technologies. The two primary types of hybrid systems are water conservation and plume abatement designs.

Water conservation systems are intended to reduce—but not completely eliminate—the use of water for plant heat rejection. A limited amount of water is used during the hottest periods of the year to mitigate the large losses in steam cycle capacity and plant efficiency associated with all-dry operation. These systems can limit annual water use to 2 to 5% (although more typically range from 20 to 80%) of that required for all-wet systems and still achieve substantial efficiency and capacity advantages during the peak load periods of hot weather, as compared to an all-dry system. If sufficient water is available, increasing amounts of plant capacity and efficiency can be attained.

Plume abatement towers, on the other hand, are essentially all-wet systems that employ a small amount of dry cooling to dry out the tower exhaust plume during those cold, high-humidity periods when the plumes is likely to be visible. (Water conservation systems also provide plume abatement because the wet part of the system is normally not used during the colder periods when plumes are likely to form.)

Finally, low-capital-cost hybrid approaches have been considered for use during peak load periods of hot weather to provide short-term enhancement of ACC performance, steam cycle capacity, and plant efficiency. Alternatives include using water to cool the ACC inlet air or deluging the air-cooled surface with water for short periods. Detailed consideration of these approaches is beyond the scope of this report.

Numerous design arrangements exist for hybrid systems. These are discussed in detail in Mitchell (1989)and Lindahl (1992). In brief, possible tower options include the following:

- Single-structure combined tower or separate wet and dry towers
- Series or parallel airflow paths through the wet and dry systems
- Series or parallel connected cooling water circuits

In addition, a number of arrangements are possible for the steam condenser:

- Common condenser
- Divided water box separating the cooling water flows from the wet and dry towers
- Separate condensers

As noted previously, the dry part of the system can use an air-cooled condenser, a direct contact (barometric) condenser, or a conventional shell-and-tube surface condenser.

Schematics of many of these arrangements are shown in Figures 2-9 through 2-12. These systems differ in capital cost and operating flexibility but have similar thermodynamic characteristics. For the purpose of this report, hybrid systems are addressed in a discussion of an evaporative condenser design with the capability for both wet and dry operation.

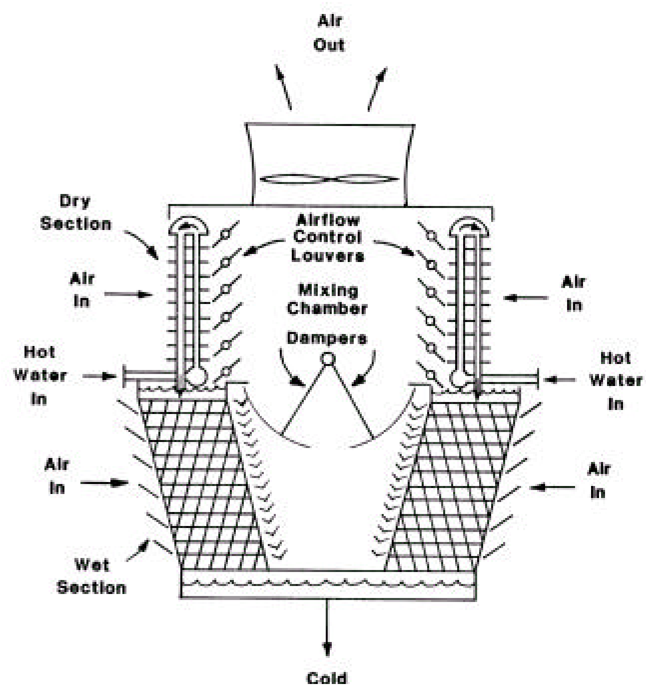


Figure 2-9
Hybrid Systems—Single Tower (Plume Abatement)

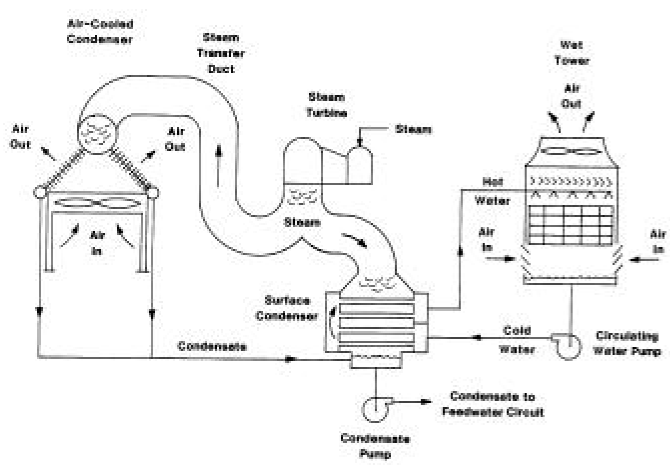


Figure 2-10
Hybrid Systems—Separate Towers (Water Conservation)

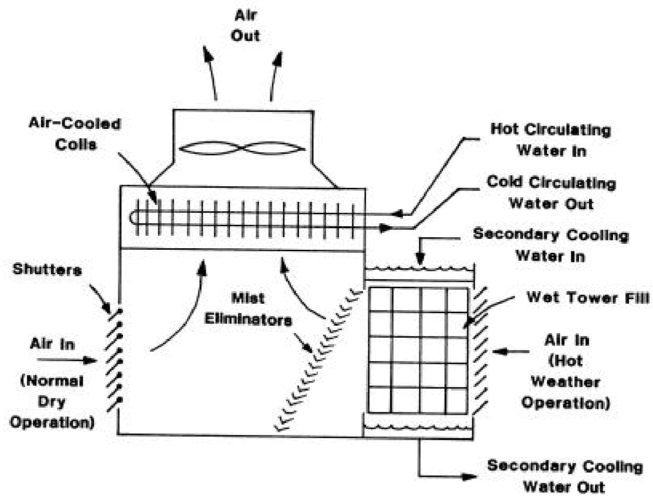
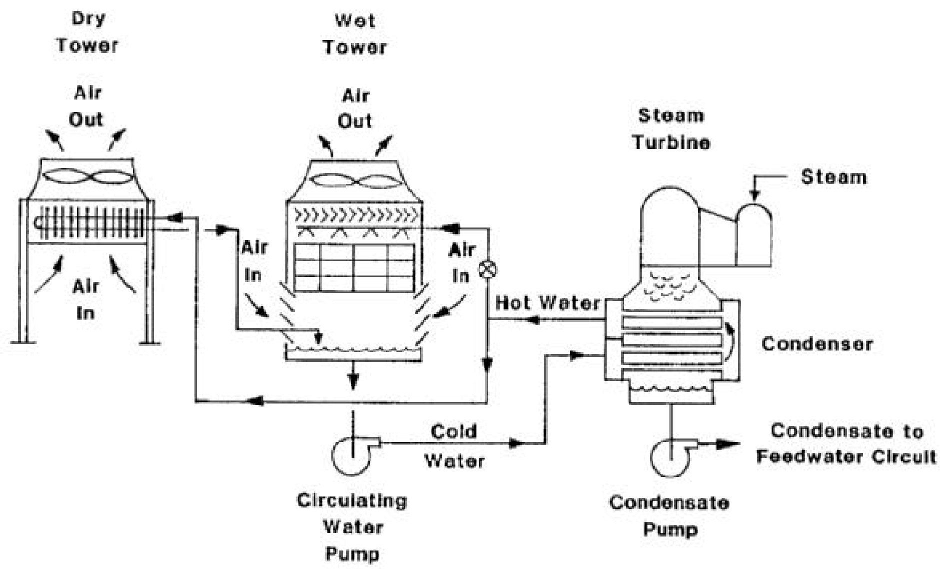
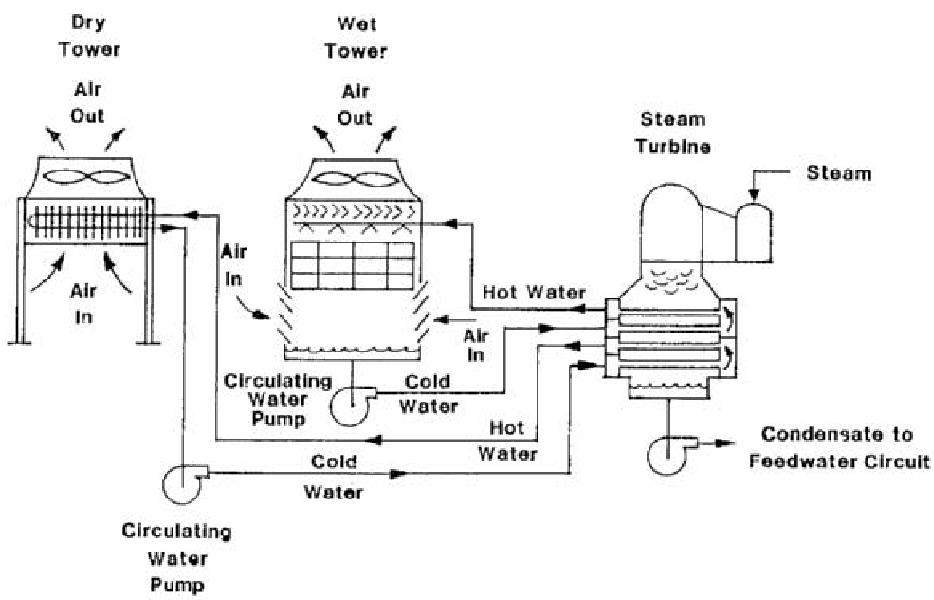


Figure 2-11
Hybrid Systems—Pre-Cooling Section (Water Conservation)



(a) With Common Condenser



(b) With Divided Waterbox Condenser

Figure 2-12
Separate Towers and Surface Condensers (Water Conservation)

Nomenclature

Table 2-2
Nomenclature

Symbol	Quantity	Units
c_i	Concentration (of i-th species) in water stream	lb of i-th species/lb water
c_p	Specific heat	Btu/lb-ºF
f_{latent}	Fraction of heat rejected as latent heat (from wet tower)	--
h_{fg}	Heat of vaporization	Btu/lb
n	Cycles of concentration	--
p_b	Back pressure (turbine exhaust pressure)	in. Hga or lb/in^2
Q	Heat load	Btu/hr
T	Temperature	ºF
w	Flow rate	lb/hr
	Subscripts/Abbreviations	
air	air-side conditions	--
amb.	ambient conditions	--
c	cold	--
circ.	circulating water	--
cond	condensing steam conditions	--
drift	cooling tower drift	--
evap	evaporated flow	--
h	hot	--
mu	make-up (to cooling tower)	--
st.	steam	--
wb	wet bulb	--
ITD	initial temperature difference	ºF
TTD	terminal temperature difference	ºF

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3
COOLING TECHNOLOGIES: THE STATE OF THE ART

This section presents information that serves to define the state of the art for dry, hybrid wet-dry, and wet cooling tower technologies. The material consists of three parts: first, a survey of the literature in the form of an annotated bibliography (Appendix A) supplemented in this section with some categorization and brief summaries of the important citations in each category; second, a listing of existing and planned installations around the world (Appendix B), supplemented here with some brief observations on trends in the adoption of water-conserving cooling systems; and third, information gathered during interviews with owner/operators of some existing plants with dry cooling or hybrid wet/dry systems (Appendix C), supplemented here with a brief discussion regarding the ability to generalize the experience in the use of these systems from these cases.

Literature Survey

Appendix A lists over 125 references on the subject of dry, wet/dry, and wet cooling tower technologies. The citations are drawn from books, journals, the trade press, conference and workshop reports and academic theses. The bulk of the material falls into four categories: technical background and description, design and optimization methodology, cost analysis, and operating experience. Most of the references contain information from more than one of these categories.

Many of the citations in this survey are old by technical literature standards. Over half were published before 1990 and one-third before 1980. This is particularly true for wet cooling towers, which have been standard fixtures at thermal power plants for over half a century. Much of the material, even when published 30 or more years ago, is still relevant and accurate. Some classical references are highlighted for their enduring usefulness. Special attention is given to publications of the past 5 years that update subjects undergoing recent changes or reflect changing economic, regulatory, and business conditions in the power industry and its supplier community.

A brief summary of the highlights from the survey follows for the major types of cooling systems. A few broad subjects of primary importance are identified for each type, as are major sources of information.

Wet Cooling

The thermodynamics, fluid mechanics, and many of the mechanical design aspects of wet cooling towers have been well established and thoroughly documented in the literature for a long time. Much of this literature was organized and reviewed by in the *Handbook of Heat Transfer*

Applications (Maulbetsch and Bartz, 1985). General references that summarize the computational and design procedures for wet towers include Cooper (2000), Hamilton (2000), and older work by DesJardins (1992), LeFevre (1977), and Feltzin (1991). Data compilations for various fill types and tower configurations are available from the Cooling Technology Institute (CTI 1999c), papers by Fulkerson (1999), and other references such as Kelly's Handbook (Kelly 1976) and from vendors. A good introduction to the practical aspects of tower configurations, the advantages and disadvantages of particular designs, and operation and maintenance issues was presented in 1973 in *POWER* (Special Report: "Cooling Towers" 1973), which is still valuable as an introduction to the subject.

Other topics of major interest are performance testing of towers; tower maintenance with particular emphasis on film fill performance and its deterioration due to fouling and plugging; structural problems including some catastrophic failures; and retrofit to improve or restore performance. The main sources of current and archival information on these topics are the *Journal of the Cooling Technology Institute*, the CTI's Bibliography of Technical Publications (CTI 1999a) and their website (CTI, 1999b), and a series of EPRI-sponsored workshops (Proceedings: International Cooling-Tower and Spray Pond Symposium, 1990; Proceedings of the 9th IAHR Cooling Tower and Spraying Pond Symposium, 1994; Proceedings: Cooling Tower And Advanced Cooling Systems Conference, 1995; Tsou, 1997). The EPRI Proceedings, while part of the gray literature, can be found on www.epri.com and may still be readily available.

Some more recent material provides descriptive information on the most modern designs, available most easily from vendor sources (e.g., Balogh, 1998; Streng, 2000) and online at the major vendors' websites (BDT Engineering; Hamon Cooling Systems; Marley Cooling Tower); on high-performance fill designs (Aull, 2000; Hobson, 1995); and on changing practices in cooling tower water treatment (Richardson, 2000; Howarth, 2000; Gill, 1997; DiFilippo, 2001).

Environmental issues of particular current interest are plume visibility, drift control (Missimer, 1998; Suptic, 1999), and emissions of controlled air pollutants or pathogens such as *Legionella* (Adams, 1978; DiFilippo, 2001).

Dry Cooling

The best summary of the field is a comprehensive text by D. G. Kroeger (1998). Over 600 pages long and containing over 900 references, it is a fully up-to-date reference work on the field containing all important citations as of 1997.

Dry cooling first emerged as a subject of technological interest in the early 1970s with the construction of the first dry cooling towers in the U.S. and Europe. Literature at the time consisted of descriptions of these early units (see sections below and Table 3-1) —somewhat as engineering curiosities—plus a body of work on economic comparisons of dry units with wet towers or once-through cooling with the emphasis on developing a methodology for making appropriate comparisons. Much of this work was sponsored by the U.S. Department of Energy (USDOE), the Atomic Energy Commission (AEC), the Environmental Protection Agency (USEPA) and EPRI. It is documented primarily in the gray report literature (Mitchell, 1989).

Interesting work on the development of an indirect system that uses a direct contact barometric condenser (the so-called Heller system; see Figure 2-7 and accompanying text) was carried out in Hungary, widely reported, and implemented at a number of installations in the former USSR and the Middle East (Balogh, 1998). Major advances resulted from the adoption of dry cooling on a very large scale in South Africa at locations of rich energy resources and virtually no water. Again, trade press articles provided descriptions of these units (Goldschagg, 1999; Von Cleve, 1984; Van der Walt *et al.*, 1974; Trage, 1990), but there was little new research literature until the 1990s. Then, there was renewed interest in the technology in the U.S. and elsewhere, stimulated by concern over water consumption and plumes from wet cooling towers. At this time, a number of papers appeared, detailing optimization schemes for selecting the best finned tube geometries (Bonger, 1995; Buys, 1989b; Buys, 1989a; Ecker, 1978) and revisiting the question of how to best compensate for reduced performance during the hottest periods (Conradie, 1991a; Oosthuizen 1995).

There is little usable, quantitative cost information in the open literature. A few references contain qualitative comparisons and “typical” costs but give little or no insight into the basis of the costs, the breakdown among the many system components, or the sensitivity of the costs to important design and environmental factors. Some information is available in the gray literature such as submissions for siting approvals (Ledford, 1999; Miller, 2000) or embedded in various engineering/economic design packages such as EPRI’s GATE program (EPRI, 2000) or Thermoflow (Thermoflow, 1999).

Little material has been published by the CTI over the past several decades. (Their bibliography contains only three references on dry systems since 1969.) Their recent change of name from the Cooling *Tower* Institute to the Cooling *Technology* Institute reflects, in part, an awareness of the importance of dry and hybrid systems. The 2001 Annual CTI Meeting at Corpus Christi, TX, included a one-half day educational session on comparing wet and dry cooling and, predictably, more papers will be forthcoming on the topic in the future. Finally, some general descriptive system information is available from vendor websites (Marley Cooling Tower; BDT Engineering; Hamon Cooling Systems; Niagara Blower Company).

Hybrid Wet/Dry Cooling

As was the case for dry cooling, the subject of hybrid wet/dry cooling received some attention of a “system analysis/cost comparison” nature in the 1970s, with very little research literature since that time. Most of the same reports that presented methods and results on comparative costs of wet and dry systems included material on hybrid systems as well (Mitchell, 1989). Most of the hybrid systems were chosen for purposes of plume abatement rather than for water conservation in the early days of the technology, and this is still largely the case. A notable exception was a large water conservation tower installed on Unit #4 (550 MWe) of the San Juan Plant in San Juan, NM, in 1977.

The best treatment of the technology covering the several general design configurations and describing the thermodynamics and psychometrics of plume formation and abatement is given in Lindahl and Jameson (1993) Almost no cost information is available in the open literature. General technology descriptions can be obtained from vendor information and from their websites (Marley Cooling Tower; DT Engineering; Hamon Cooling Systems; Niagara Blower Company).

One variation on the wet/dry theme is the use of some simple wet enhancement methods on dry towers for peak shaving the cooling load on the hottest days. This has been the subject of a few papers and academic theses (Conradie, 1991b), but very little hard economic or engineering information is available.

History of Dry Cooling Technology

The first use of dry cooling at power plants was on a few small units in Germany in the 1930s (Kroeger, 1998; Miliaris, 1974). However, the real history of dry cooling on units of substantial size and its evolution to today's technology began about 40 years ago with an indirect, natural draft system at Rugeley in the U.K. in 1962. Originally, the use of dry cooling was more common in Europe, the former USSR, and the Middle East and South Africa than in the U.S. Table 3-1 lists 13 units from around the world installed in the three decades from 1962 to 1993 representing many of the important milestones and seminal technology in the field.

Table 3-1
Early Major Dry Cooling Installations

Unit	Size (MWe)	Country	Date	References
Direct Systems				
Neil Simpson-(Black Hills P&L)	20	USA (Wyoming)	1968	Simpson 1970
Utrillas	160	Spain	1970	March 1970
Wyodak	330	USA (Wyoming)	1977	Schulenberg 1977; Kosten 1981
Touss	4 x 150	Iran	1987	Kroeger 1998
Matimba	6 x 665	South Africa	1991	Goldschagg 1995, 1999
Majuba	6 x 665 (3 dry cooled)	South Africa	1998	Varley 1999
Indirect, Natural Draft Systems				
Rugeley	120	UK	1962	Christopher 1969
Ibbenbüren	150	Germany	1967	Scherf 1969
Gagarin	2x100; 2x220	Hungary	1969/1972	Kroeger 1998
Grootvlei #5	200	South Africa	1971	Van der Walt *et al.* 1974
Razdan	2x200; 2x210	Armenia	1971/1974	Trage 1990
Grootvlei #6	200	South Africa	1978	Kroeger 1998
Schmehausen	300 (nuclear)	Germany	1977	Hirschfelder 1973

In the U.S., the earliest installations were in Wyoming [Neil Simpson, Wyodak], where the desire to locate the plants in an area with very low-cost surface deposits of Western coal but with very limited water resources made dry cooling both the environmental and economic system of choice. The 330-MWe unit at Wyodak near Gillette, WY, was for many years the largest direct air-cooled system in the world. South Africa was the world leader in the use of dry cooling on large installations, again motivated by the need for power close to fuel recovery and refining activities in an arid region. Constructed in 1991, the 4000-MWe Matimba plant was 10 times the size of Wyodak and remains the largest plant on dry cooling in the world today.

A number of units beginning with Rugeley in 1962 were of the indirect type of a Hungarian design, utilizing a direct-contact condenser and a natural draft tower. This technology was used at several installations at about the 200-MWe size range (Miliaris, 1974). The Kendal plant in

South Africa, which uses a conventional surface condenser, is the largest indirect system in the world with 6 x 686 units on dry cooling. Brief descriptions of these units are found in Kroeger (1998), and more detailed information for some of them is found in the references listed in Table 3-1.

Trends in Dry Cooling Use

Over the past 40 years, the technology has become more widely used. Appendix B contains recent lists of existing and planned installations from four major vendors of dry and hybrid cooling systems, BDT Engineering (Balcke-Dürr), GEA (GEA), Hamon Cooling Systems, and Marley Cooling Tower Company (Marley Cooling Tower). Not all of the listed installations are at electric power plants, and the lists are not complete (e.g., several installations in Table 3-1 are not included). However, the lists are sufficient to identify two noteworthy trends in the use of the technology.

First, use has expanded rapidly at a continually increasing rate. Figure 3-1 shows the increase in the installed MWe on dry cooling for both the U.S. and the world for the periods prior to 1980, during the 1980s, during the 1990s, and for future planned units. These figures, totaled from sources including the vendors' lists in Appendix B, show a clear increase with time even though new plant construction was slow (at least in the U.S.) during much of the 1990s. It is estimated that there are over 2500 MWe of U.S. power generation on dry cooling and from 15 to 20 GWe worldwide. Nonetheless, the existing capacity on dry cooling in the U.S. and around the world is, of course, quite small compared to that on wet recirculating or once-through systems.

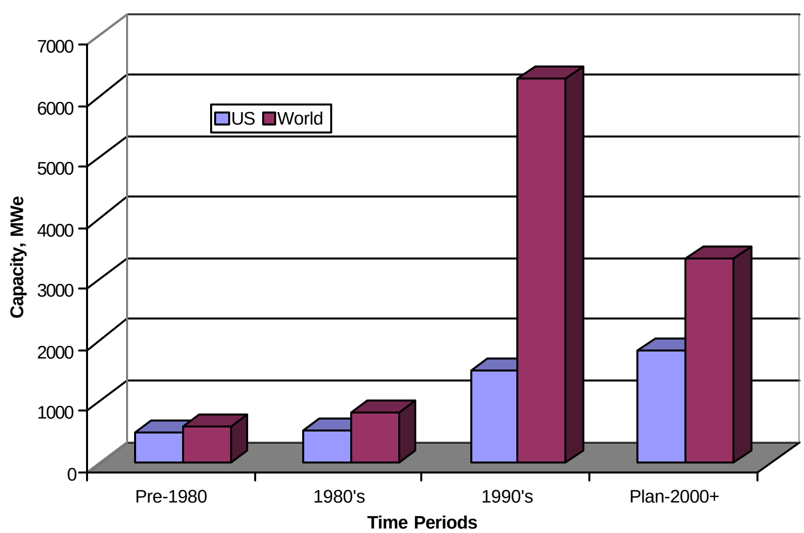

Figure 3-1
Trends in Dry Cooling

Second, the use of dry cooling is not restricted to arid regions. Table 3-2 lists the distribution of U.S. dry cooling units installed and planned by state expressed both as the number of units and as the total generating capacity. Twenty-three states plus Washington, DC, have some amount of dry cooling, including many in the Northeast where rainfall and water supply are, at least on average, plentiful.

Table 3-2
Dry Cooling in the United States (Source: Vendor Lists—Appendix B)

State	Dry Cooling		Hybrid, Wet/Dry	
	Number of Units	Capacity (MWe)	Water Conservation	Plume Abatement
Alaska	2	75		
California	6	273		1
Colorado				1
Connecticut	2	280		
Hawaii	1	20		
Iowa	1	40		2
Idaho	1	40		2
Illinois	1	9		1
Massachusetts	8	454		
Maine	2	100		
Michigan	1	9		
Minnesota	1	6		
Montana	1	50		
New Jersey	2	385		
New Mexico			1 (660 MWe)	
Nevada	1	150		
New York	6	205		2
Pennsylvania	2	88		2
Rhode Island	1	80		
Texas	2	450		3
Virginia	2	100		1
Washington, DC				1
West Virginia	1	80		
Wyoming	5	1180		
Total US	49	4074	1 (660 MWe)	16

While the early use of dry cooling was usually motivated by plentiful, low-cost fuel in regions of little water, as in the case of Wyodak or the South African installations, the reasons for choosing dry cooling now include other environmental considerations such as plume visibility and wastewater discharge. In many cases, the decision has been driven by the desire to shorten licensing approval times for proposed plants by removing any water related issues from the discussions. Even in a competitive, deregulated industry shortening the “time to market” may outweigh future reductions in generating efficiency (this view is not universally held; see discussion below under “Calpine”).

Recent Developments

As noted in Section 1, as early as 1975, California’s State Water Resources Control Board promulgated a policy (SWRCB, 1975) mandating the use of waters other than fresh surface water for power plant cooling, unless doing so could be shown to be “environmentally undesirable or economically unsound.”

Recently, the U.S. Environmental Protection Agency in their support document, “Economic and Engineering Analyses of the Proposed §316(b) New Facility Rule” (USEPA, 2000) identified dry cooling as an “alternative option” which would “impose more stringent compliance requirements on the electric generating segment of the industry” based on a “zero-intake-flow (or nearly zero, extremely low-flow) requirement commensurate with levels achievable through the use of dry cooling systems.”

In summer 2000, New York State’s Department of Environmental Conservation required the use of dry cooling at the Athens Generating Station (a 1080-MW gas-fired plant) as the preferred Best Technology Available for “minimizing adverse environmental impact” of a cooling water intake structure (Cahill, 2000).

Operating Experience

Table 3-3 identifies five plants that use dry or hybrid wet/dry cooling systems at which operating personnel were contacted and interviewed to learn about the operating and maintenance experience with the systems. Appendix C contains an outline of the subjects covered in the interviews and notes from several of the meetings.

In addition, comments were received from Prof. D. G. Kröger, University of Stellenbosch, who consults for dry cooling manufacturers and users worldwide. His experience is consistent with that reported below by the operators. He further notes that there are occasional problems with the dephlegmators (air removal sections) at some operating conditions. These were not mentioned in any of the interviews with U.S. users.

Table 3-3
Plants Interviewed for O&M Experience

Plant	Location	Type/Size	Date Installed	System Vendor	Owner	Contact
Crockett	Crockett, CA	CC/220 MWe	1996	BDT	NRG et al.	Don Curran
El Dorado	Boulder City, NV	CC/480 MWe	1999	GEA	Reliant/ Sempra	George Tater
Sutter; Acadia	Yuba City, CA; Eunice, LA	CC/560 MWe; CC/1080 MWe	Under construction		Calpine	Kim Stucki
MassPower	Springfield, MA	CC/240 MWe	1993	Niagara Blower/ Resorcon Inc.	El Paso *et al.*	Sal Paolucci
Chinese Station	Jamestown, CA	Wood waste fired/ 25 MWe	1984	GEA	Pacific Ultrapower	Ron Brown

Crockett Co-Generation

The Crockett Co-Generation Plant in Crockett, CA, was visited on June 12, 2000. The host for the site visit was Peter H. So, plant engineer. Current contact point is John Walsh, plant operations manager. Detailed contact information is given in Appendix C.

The plant began operation in 1996. It is a 240-MWe combined cycle plant in a 2 x 1 configuration (two 80-MW combustion turbines; one 80-MW steam turbine, single shaft). The plant sells electricity to the grid and also provides process steam to the C&H Sugar plant on a neighboring site. Nominally 250,000 lb/hr of steam are supplied to C&H at 450 psi.

The air-cooled condenser is a BDT Engineering unit consisting of three banks of five cells each—four cells for steam condensation and an extra cell in each bank (three extra cells total) for auxiliary cooling. The unit is equipped with Alpina low-noise fans. No capital cost information was provided.

No serious start-up or maintenance issues were reported. Specific comments include the following:

- There have been no air in-leakage problems, and the vacuum systems and de-aerators have worked well.
- The water chemistry is easy to control, although there are very high make-up rates (complete cycle water turnover every 2 or 3 days) because condensate from the sugar plant is either not returned or has to be discharged because of contamination.
- Cleaning of the finned tube surfaces was last performed 18 months ago. The high-pressure cleaning system works well in removing the "sugar dust" coating from the fins. There were conflicting reports on the need for and the effectiveness of the cleaning. Some felt that there was no improvement in performance as a result of the cleaning. Others reported that the automatic controls had cut the fans back from full to half speed at similar operating conditions after cleaning.
- No noise problems exist. Limits of 50 dB at the property boundary are easily maintained.
- No corrosion problems have occurred on the condenser surfaces.
- Maintenance costs are not accounted for separately but have not appeared to be excessive. The ACC unit is maintained as part of the normal plant routine by the regular staff.

The performance of the unit and the plant has been satisfactory. No quantitative performance data were provided (there is no systematic monitoring), and acceptance test results were not available. Anecdotal information includes the following:

- A "consensus estimate" by plant personnel indicates that the hottest summer days incur a 3- to 5-MWe reduction of output from the steam side (and a reduction of ~7 to 8 MWe on the combustion turbine side).
- Fan power is the biggest plant load. There are 15 fans (each 150 hp), which in total consume about 2 MW.
- No problems with wind effects have been experienced.
- Freeze protection has not been a problem. (However, cold weather operating controls sometimes act strangely, i.e., fans will turn off and operators don't know why. This is thought to be a conflict between the BDT operating system and the plant's Foxboro I&C package, but is not considered a serious problem.)
- The plant manager and engineers would like to explore methods for wet enhancement of ACC performance on the hottest days.

The complete set of meeting notes is provided in Appendix C, along with extensive information on the air-cooled condenser including the following:

- Schematic of the ACC flow system
- ACC design data
- ACC general arrangement drawings
- ACC process data sheet
- ACC performance summary

- ACC performance characteristics (turbine back pressure vs. % heat load for a range of ambient air temperature and various fan settings)

El Dorado

El Dorado Energy's Boulder City Power Plant was visited on October 28, 2000. The host for the visit was George Tatar, facility manager. Detailed contact information is available in Appendix C.

The plant began operation in May 2000. It is a gas-fired 540-MWe 2 x1 combined cycle (two 180-MW combustion turbines; one 180-MW steam turbine). The ACC is a GEA unit. Auxiliary cooling (lube oil, etc) is provided by a closed-loop fin-fan cooler from BDT Engineering. The steam turbine, which is of Westinghouse high back pressure design, operates over a range of 2 to 8.5 in. Hga. No capital cost information was provided.

No start-up, maintenance or cleaning problems had yet been encountered on the cooling system. Some unrelated start-up problems with the steam turbine thrust bearing had been experienced.

No performance data or acceptance test data were made available. Performance issues include

- A "dramatic" performance drop-off when ambient temperature exceeds 110°F; and
- A "distinct loss" of cooling during periods of gusting winds.

The reasons given for the choice of dry cooling include

- The "politics" of fresh water use in the Las Vegas area,
- Unavailability of reclaimed municipal water for wet tower make-up,
- Inadequate area on plant property for evaporation ponds to dispose of cooling tower blowdown, and
- An economic advantage for dry cooling when compared to wet cooling with vapor compression evaporators (VCE) brine concentrators

The facility manager wants to explore the use of spray cooling or some other form of wet enhancement to mitigate the performance deterioration on the hottest days.

Additional information about the plant, including the original meeting notes, is included in Appendix C.

Calpine

On September 18, 2000, a meeting was held at the Calpine offices with Kim N. Stucki, manager, plant engineering. Contact information is in Appendix C.

In advance of the meeting, a list of discussion topics (see Appendix C) was prepared focusing primarily on how cooling systems are compared and on operating issues associated with dry cooling systems. In summary, the responses were as follows:

- Dry cooling is never the system of choice on economic grounds.
- Calpine had little operating experience to share since they had no dry-cooled plants in operation.

Although the Sutter Plant, a 516-MWe 2 x 1 combined-cycle plant in Yuba City, California that went on-line in 2001, uses dry cooling, the system was chosen on other than economic grounds. A comparative analysis was carried out for the Acadia Plant (1080-MWe 4 x 2 combined cycle), which is currently under construction near Eunice, Louisiana, and is scheduled to begin operation in June 2002. A summary of the analysis indicated additional project costs of over $23 million for the use of dry cooling and an additional cost of $30 million over the life of the project in lost revenue from reduced power production attributable to the higher plant heat rate and lower capacity with dry cooling.

Some rules of thumb included in the consideration of dry cooling are provided below:

- Lost capacity for dry cooling equals 16 MWe on an average day and 28 MWe on a hot day, equivalent to 4 to 8% of the plant's steam-side output.
- Cooling systems are designed and compared at the design back pressure at the 1% temperature (temperature exceeded for 1% [88 hours] of the year).
- Dry cooling saves approximately 80% of makeup water and 85% of wastewater discharge over a typical year.
- The loss of 1 kW is approximately worth $1500 over the life of a project.
- The capital cost of the dry cooling system is approximately three times that of a wet cooling system.
- There is no value in reduced licensing time in the current business climate since licensing is not the pacing issue: Delivery times on combustion turbines ordered today are more than 3 years.

The following anecdotal information on operational issues was provided by Calpine:

- They have no experience with high back pressure turbines.
- Wet enhancement with spray cooling was tried at a wood-burning plant near Sonoma, California, with attendant O&M problems. No details or explanation were available.
- Calpine routinely uses fogging systems to cool gas turbine inlet air. They use de-mineralized water and have no problems.

MassPower

A telephone interview was held with Sal Paolucci, plant operations manager of MassPower, on March 2, 2001. Contact information is provided in Appendix C.

The plant, located in Springfield, MA, is equipped with an evaporative condenser cooling system provided by Resorcon, Inc., a subsidiary of Niagara Blower. It is the largest power generation application of this technology in the world.

Basic background information on the plant and the cooling system was presented in a 1995 paper at the ASME Industrial Power Conference (Basile, 1995). A brief description of the plant and the cooling system follows.

The plant is a 240-MWe gas-fired co-generation plant, generating power for the grid and supplying steam to Monsanto's Indian Orchard plant in Springfield, MA. Operation began in 1993. The power train is comprised of two 84-MW gas turbine/generators, a triple pressure heat recovery boiler, and a 72-MW steam turbine/generator. Evaporative cooling of the gas turbine inlet air is used for enhanced power production at ambient temperatures above 59ºF.

Plant cooling is provided by a wet surface air-cooled steam condenser and an auxiliary water cooler. The condenser consists of five cells condensing 661,000 lb/hr of steam and maintaining a back pressure of 2.8 in. Hga at ambient conditions of 20ºF dry bulb and 19ºF wet bulb. Hot weather specifications are 4.8 in. Hga back pressure at 97ºF dry bulb and 80ºF wet bulb for a steam flow of 582,000 lb/hr. Total power requirements are approximately 900 HP, and make-up water requirements range from 690 gpm at 7ºF ambient to 1076 gpm at 97ºF/80ºF ambient.

The unit can be operated in a plume abatement mode by isolating the spray water flow to the interior bundles of each cell with a plume reduction of 75% during "normal ambient conditions."

Plant operations personnel are quite satisfied with unit performance. Startup and shutdown are easily accomplished. Hot day operation has not been limited: The turbine alarms at 6 in. Hga and trips at 7.5 in. Hga. Back pressure on the hottest days is maintained at 4 to 4.5 in. Hga. There have been no water treatment problems, and the unit is easily cleaned on-line to maintain clean tubes.

Chinese Station

Chinese Station was visited on March 23, 2001. The plant is equipped with an ACC and a separate air-cooled auxiliary cooler. The condenser has been outfitted with a performance enhancement spray system to maintain plant output during hot weather. The visit was hosted by Ron Brown, plant operations manager; contact information is provided in Appendix C.

The facility is operated by Constellation Services and owned by Pacific Ultrapower, a company jointly owned by Baltimore Gas & Electric and Ogden (now Covanta). It is located at Enterprise Drive, Hatler Industrial Park, Chinese Camp, California (1.5 miles south of the Rte. 108/120 split approximately 45 miles east of Modesto).

The plant is a 25-MW (gross)/22-MW (net) unit fueled with wood waste derived both from agriculture and forestry waste and urban wood waste. The fuel has a heat content of ~ 8,000 Btu/lb, a moisture content that varies from 20 to 60% depending on the source, and an ash content of up to 17%.

The plant has been in operation since 1986. The plant serves two functions: wood waste disposal and electric power generation for purchase by PG&E under a 30-year contract. The waste is purchased for a nominal charge, which essentially covers the suppliers' cost of transporting the waste to the site while enabling them to avoid storage, disposal, or tipping costs elsewhere. The ash is sold to local concrete plants or to agricultural interests for mixing with fertilizer and cattle feed.

The ACC is a GEA 4-cell unit (3 condensing cells; 1 reflux [or dephlegmator] cell) sized for 189,500 lb/hr of turbine exhaust steam with 8.7% moisture. The design point is 8.9 in. Hga at 97°F ambient. At temperatures above 90°F (up to summer peak temperatures of 110-115°F), capacity falls as minimum achievable back pressure rises to 10 in. Hga and higher. Turbine alarm is set at 12 in. Hga.

The unit fouls frequently due to the high dust levels from the waste wood fuel piles. Dust screens have been installed to reduce the blown dust, but cleaning of the finned tube surfaces is required as often as once a week. Cleaning is done with a fire hose to wash down the surfaces.

Plume recirculation is a problem on windy days. "Wing-like" structures have been added to the top of the ACC along the sides to deflect plume recirculation.

The spray system was installed about 5 years ago. The motivation for installing the augmentation sprays was to avoid penalty charges incurred when the contracted-for power level cannot be delivered. The sprays are located along one side of the condenser at heights of 8 and 10 ft. Spray nozzles are located about every 2 ft along the pipes and supplied with high-pressure water (up to 450 psi) that has been treated by reverse osmosis. Additional arrays placed across the tower between the fans are used only on the hottest days. Water flow needed to saturate the inlet air at peak summer temperatures is about 40 gpm. The spray system is capable of 150% overspray or 100 gpm; 60 gpm is collected in a basin under the ACC and recycled.

Full spray lowers the back pressure by about 2 in. Hg, resulting in 1 to 1.5 MW additional output. Maximum augmentation is achieved when the finned tube surfaces become wet. This has not resulted in any reported corrosion damage. There may have been some scaling or fouling, but this is difficult to determine in the high-dust environment.

Nozzle fouling has been a problem. In 5 years, nozzles have been replaced twice. A variety of nozzles are used (BETE, Delavan, and others). Some are the pin impingement type, others the swirl type. It does not appear that complete evaporation is achieved even in the absence of overspray.

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4
CASE STUDY SITE SELECTION

Within California, there are great variations in climate, hydrology, and elevation—all important considerations in the siting of power plants and in the choice of cooling systems. The approach taken in this project was to select four case study sites that span the range of conditions likely to be encountered in siting deliberations for new power plants using other than once-through ocean cooling for steam-cycle heat rejection.

Figure 4-1 shows the locations of all existing power plants in the state. In addition to numerous coastal sites, there are large concentrations of plants in the San Francisco Bay Area, the Los Angeles Basin, the Central Valley, and the Delta Region. The many plants in the northern and eastern parts of the state are mostly hydroelectric units. Figure 4-2 indicates the sites of future plants—approved, expected, and currently in the application process.

The case studies are intended to span the range of important site characteristics for locations likely to be chosen for future gas-fired plants in the state. This section introduces key characteristics and identifies case study sites.

Important Site Characteristics

The comparative performance and cost of wet, dry, and hybrid cooling systems for combined-cycle power plants are determined by a number of characteristics of the site. The most important are

- Site meteorology,
- Site elevation,
- Water availability,
- Wind conditions,
- Local environmental constraints, and
- Proximity to other activities.

These characteristics are described below.

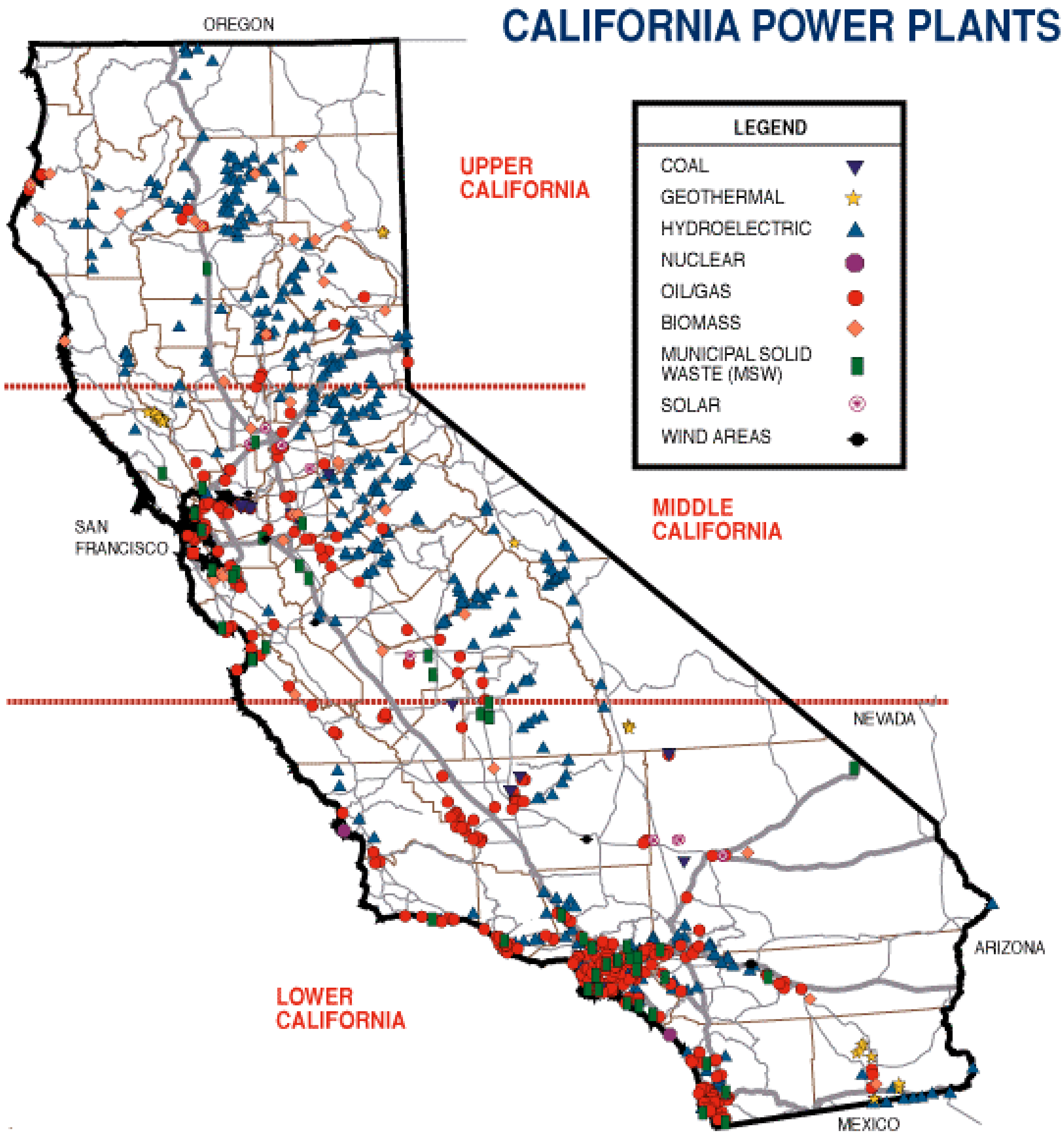

Figure 4-1
California Power Plants (Source: CEC website)

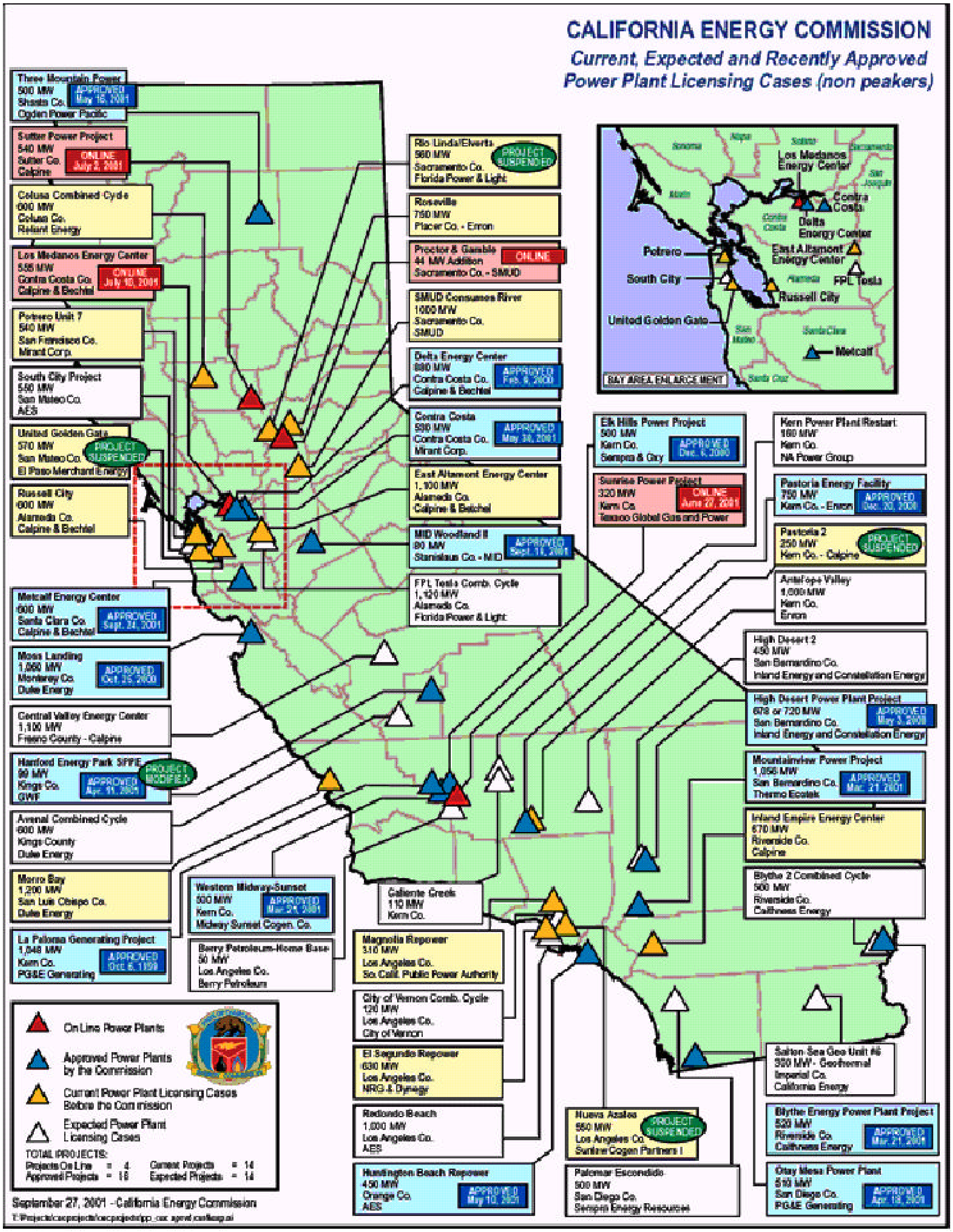

Figure 4-2
Current, Expected, and Recently Approved Licensing Cases (Source: CEC website)

Site Meteorology

The primary determinant of the relative performance and cost of a wet, dry, or hybrid cooling system is the meteorology at the plant site. As discussed in Section 2, dry systems are limited by their approach to ambient dry bulb temperature and wet systems by their approach to ambient wet bulb temperature. The wet bulb temperature is always (except in the rare instance of 100% relative humidity) less than the dry bulb temperature, often significantly. Therefore, for reasonably designed wet or dry systems, the wet system will nearly always perform better.

Wet system performance increases as wet bulb temperature decreases and hence the technology is particularly favored by hot, dry conditions. Dry system performance suffers as dry bulb temperature increases and therefore is favored by cool, but humid conditions. Of particular importance are extreme high-temperature conditions that are typically coincident with peak summer loads on the electric power system driven by high air-conditioning demand. At these conditions, dry cooling can impose a substantial capacity penalty on the system by limiting the achievable back pressure on the steam turbine and hence the available power output.

To illustrate the importance of this effect on the comparative costs of wet and dry systems, sites were chosen with a range of peak temperatures and relative humidities.

Another element of site meteorology is winter conditions. Under a range of colder conditions, a visible plume can be produced by wet towers and by the wet portions of hybrid towers. Under some conditions, the plume can be quite voluminous and persistent. These conditions, when accompanied by the right wind patterns, can keep the plume at low levels and blow it toward local features such as highways and airports, impairing essential visibility. This may require preventive measures such as plume abatement capability on the towers or interrupted operation during some periods with attendant higher cost and lost revenue. The situation is obviously highly site specific.

Under more extreme winter conditions, the potential for freeze-ups exists. Both wet and dry towers are subject to damage from freezing, and the problem for most wet systems has received considerable attention in the literature (Michell, 1997; Fabre, 1994). The problem can normally be addressed through prudent operating practice, such as fan control and taking cells out of service to increase the heat load on the others. At likely plant sites in California, it is not expected to be a problem and was not considered in the site selection.

Site Elevation

Site elevation, *per se*, affects the performance of cooling systems through the effect of reduced air density at higher elevations on fan power requirements. A fixed amount of heat rejection at a given set of temperatures requires a fixed mass flow of cooling air. At lower densities, an increased volume flow must be maintained to provide the same mass flow; thus, a correspondingly higher fan power is consumed. This increase in fan power is unfavorable to dry cooling since the technology requires substantially higher air flow than do wet systems for the same cooling load. While a range of elevations was chosen for illustrative purposes, it should be noted that the difference in fan power with elevation is readily calculable, and the application of the appropriate energy cost penalty to the economic analysis is straightforward.

Secondary effects of site elevation result from the fact that higher elevations are less likely to experience summer temperatures and humidity as high as those encountered at lower elevations. These considerations were discussed explicitly in the previous sections on site meteorology.

Water Availability

Water availability is obviously a crucial factor in the evaluation of the relative costs of wet and dry cooling systems. As will be discussed in Section 5, the cost of water for make-up to a wet cooling tower can be a significant item in cost comparisons at some locations. This is particularly true if water must pumped over long distances and big changes in elevation, requiring large capital investments in water supply facilities and high operating costs for pumping power. In such situations, the capital cost ratios of dry to wet systems can be significantly reduced, as indicated in the studies conducted for the Elk Hills plant (Miller, 2000).

However, this consideration is entirely site specific. It is readily accounted for parametrically to determine what the breakeven cost of water would have to be in order to alter the economic ranking of dry vs. wet systems at a particular site with a given meteorology. Therefore, no explicit use of water availability was made in the case study selection process.

Wind Conditions

Wind can have a serious detrimental effect of cooling tower performance in either wet or dry systems. These effects usually are the result of plume recirculation where hot exhaust air (for dry systems) or hot, moist exhaust air (for wet systems) is blown down near the tower inlet and entrained with the inlet air. This raises the inlet temperature (dry or wet bulb) and degrades tower performance. In dry towers, where the fan inlet is below the heat exchangers and more exposed to the open atmosphere, it is possible for gusts of wind to disrupt the incoming air flow patterns and partially starve some or all of the fans. For this reason, the consequences of wind-related effects on performance are likely to be more severe for dry systems than for wet.

All these effects can be exacerbated by local topographic features or nearby structures that alter wind patterns by creating vortices, downdrafts, or other flow perturbations. The situation can often be mitigated or corrected by the construction of wind barriers (Duvenhage, 1996; Goldschagg, 1995).

While important, wind data were not used in the selection of case study sites for two reasons. First, it is difficult to determine the relative effect on wet vs. dry systems without site-specific details. Second, the occurrence of wind-related performance degradation is intermittent and seasonal and, hence, difficult to quantify in economic terms. Therefore, it would be impossible to generalize from any case study other than to suggest that careful attention should be paid to local wind patterns for either system.

Local Environmental Constraints

Wet systems in particular require the disposal of sometimes substantial quantities of blowdown in order to maintain suitable water quality in the tower and condenser. Local regulations based on conditions in surface receiving waters, aquifers, or injection wells may limit disposal options, requiring alternatives such as brine concentration or evaporation to dryness that may have a potentially significant effect on the system cost. Again, these are site-specific considerations that are difficult to generalize but relatively easy to account for in any particular economic comparison study. They were not included in the case study selection process.

Proximity to Other Activities

A final consideration is the presence of nearby activities, facilities, or neighborhoods and the constraints that they may impose on the design and operation of cooling systems. Examples include the proximity of population centers, highways, airports, agricultural operations (sensitive crops or livestock), and designated scenic areas—any and all of which may require significant system modifications to control noise, drift, visible plumes, other visual impact, or other considerations. Plume abatement capability and low-noise fans in particular can add substantially to the cost of both wet and dry systems. These items will be identified in the cost discussions of Section 5 but were not explicitly included in the case study site selections.

Chosen Sites

Four sites were selected based on two primary considerations:

1. They are from locations in California where a significant number of plants are planned and/or are expected to be proposed. Coastal sites were not included on the basis that ocean cooling is likely to be the preferred choice, and this approach is sanctioned by Resolution 75-58.

2. The sites represent a range of the important meteorological conditions discussed in the section above on "Site Meteorology."

The chosen sites are as follows:

- A high desert site characterized by conditions at Blythe, California;
- A northern mountain site characterized by conditions at Burney, California (near Redding);
- A Central Valley site characterized by conditions at McKittrick, California (near Bakersfield); and
- A Bay Area/Delta Region site characterized by conditions at Pittsburg, California.

The sites are proximate to locations designated on Figure 4-2 as sites for current or proposed plants in the Blythe, Three Mountain, LaPaloma, and Contra Costa areas (note that the case study sites are hypothetical, and have no relation to these proposed plants). Table 4-1 summarizes important site characteristics and meteorological information. Appendix D provides detailed information about each site.

Plant/steam conditions at each site are as follows:

- **Plant type**: gas-fired combined cycle; 500 MW in 2 x 1 configuration (typical of current plant designs in California)
- **Steam flow**: 1,000,000 lb/hr
- **Steam quality**: 95%
- **Enthalpy**: 1057 Btu/lb
- **Back pressure**: 2.5 in. Hga (T_{sat} = 109°F)

Table 4-1
Site Specifications

Characteristic	Site 1 (Blythe)	Site 2 (Three Mountain)	Site 3 (LaPaloma)	Site 4 (Contra Costa)
Location				
Longitude	-114.596	-121.660	-119.622	-121.805
Latitude	33.610	40.833	35.306	38.005
Elevation (ft)	390	967	320	10
Temperature, dry bulb				
T_{avg}	72	62	67	60
T_{max}	117	118	106	104
$T_{1\%}$	111	103	103	85
$T_{2\%}$	109	101	101	81
$T_{5\%}$	106	98	99	77
Temperature, wet bulb				
$T_{wb\ avg}$	63	58	60	56
$T_{wb\ max}$	79	77	75	75
$T_{wb\ 1\%}$	78	70	72	66
$T_{wb\ 2\%}$	77	69	71	64
$T_{wb\ 5\%}$	76	67	70	63

References

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2. Fabre, L. Wet Cooling Towers: The Different Antifreeze Systems---Experience Feedback. Proceedings of the 9th IAHR Cooling Tower and Spray Pond Symposium; von Karman Institute Brussels, Belgium. International Association for Hydraulic Research; 1994.

3. Goldschagg, H. B. *et al.* Air-Cooled Steam Condenser Performance in the Presence of Crosswinds Palo Alto, CA: Electric Power Research Institute; 1995.

4. Michell, F. L. and D. H. Drew. Cold Weather Operating Guidelines and Experience for Natural Draft Cooling Towers. Proceedings: Cooling Tower Technology Conference; St. Petersburg, FL. Palo Alto, California: EPRI; 1997: pp. 4.1 - 4.10.

5. Miller, T. O. and J. E. Luckart. Elk Hills Power, LLC's Opening Brief on Part Two Subject Areas. California Energy Commission: Siting Committee; 2000(Docket No. 99-AFC-1).

5
COMPARATIVE COST ANALYSES

Introduction and Guidelines

In order to make appropriate decisions in choosing between wet, dry, and hybrid cooling systems, an accurate estimate of the total cost for each alternative is necessary. Comparisons are often difficult to make, and the findings are impossible to generalize with sufficient precision to be useful in design or licensing procedures.

This section is intended to provide a set of guidelines for the purpose of evaluating the completeness and credibility of cost estimates and of understanding the differences between allegedly comparable estimates. At the outset, a methodology will be developed for making fair comparisons. The use of the methodology will be illustrated through four case studies representative of the likely range of conditions in California. Differences among the resulting cost comparisons will be analyzed in order to understand the influence of site-specific conditions on the relative costs. Finally, a series of parametric studies will be presented to illustrate the sensitivity of the overall cost comparisons to uncertainties in individual cost components or to unusual conditions causing some cost elements to be significantly different from average values.

Two guidelines for comparison are paramount:

1. The estimates being compared must include all costs that are affected by the choice of the cooling systems, particularly the costs associated with the effects of cooling system capability on plant heat rate and generation capacity.

2. Comparisons must be made between optimized systems, in which the appropriate choice of cooling system size and capability balances initial capital cost against the cost of performance penalties that an undersized cooling system might impose on the generating plant.

Cautions and Limitations

The cost information and comparisons presented in this report are based on vendor-supplied data for four case studies chosen to be representative of sites and conditions in California. Therefore, the results should not be extrapolated or applied to situations that differ substantially from these situations. The following specifics should be noted:

- The plant type and size were the same for all cases, namely, a new, 500-MWe gas-fired, combined-cycle plant. The steam portion of the cycle delivers one-third or about 170 MWe. The results should not be used in applications such as
  - Retrofit situations,
  - Stand-alone fossil-fired or nuclear steam plants, or
  - Steam cycles substantially different in size from 170 MWe.
- The sites were all inland sites with fresh water supply. The results should not be applied to ocean sites using salt water for cooling tower make-up.

Methodology

As described in some detail in Section 2, cooling systems are required to condense the steam at the turbine exhaust and to maintain the design turbine back pressure. For a given ambient temperature and humidity, the size and effectiveness of the cooling system determines how low a condensing temperature can be maintained for a specified water flow. Figure 5-1 illustrates the qualitative variation in steam turbine performance and steam turbine heat rate with varying back pressure.

Two curves and one data set are displayed:

- A “conventional” turbine designed to operate with once-through or wet recirculating cooling systems: These turbines are highly efficient at low (1 to 2.5 in. Hga) back pressure, incur large heat rate penalties as the back pressure rises, and are usually limited to operation below 5 in. Hga.
- A “modified” turbine: While somewhat less efficient at lower back pressures, these designs maintain their performance better as back pressure rises and can be designed to operate at levels well above 5 in. Hga. For the subsequent analyses, an upper limit of 8 in. Hga is assumed.
- Points from an operating turbine: Points taken from a heat rate curve for a turbine currently installed and operating on a gas-fired, combined-cycle plant using an air-cooled condenser (ACC) are plotted in Figure 5-1 and seen to agree well with the “modified” case.

The data have been presented as “heat rate ratio” (HRR), normalized to unity at 2.5 in Hga; that is,

$$\text{HRR} = (\text{Heat Rate})/(\text{Heat Rate @ 2.5 in. Hga})$$

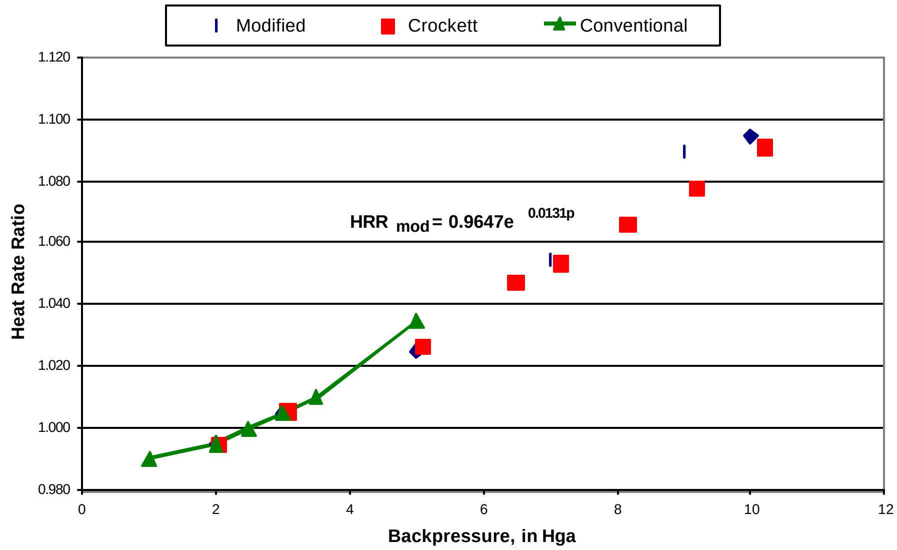

Figure 5-1
Heat Rate Ratio vs. Back Pressure

As the back pressure increases above the design value, the turbine heat rate increases, requiring increased steam mass flow and higher fuel consumption for comparable power generation. At some level, the turbine cannot tolerate further increases and steam flow must be reduced, leading to a decrease in plant output. The system suffers penalty costs at high ambient temperatures or humidity in the form of higher heat rates and, as the turbine back pressure limit is approached, capacity losses as the turbine steam flow is reduced to keep the back pressure within operating limits. Since wet evaporative cooling systems cool to the wet bulb temperature, they can maintain a lower back pressure than a dry system at nearly all ambient conditions. Additionally, larger and, hence, more costly systems of either type will provide higher plant efficiency and output.

Therefore, the economic design tradeoff for selecting the optimum cooling system is between the capital cost of the cooling system, which increases with system size, and the performance penalties—both in heat rate and capacity—that decrease as the cooling system size increases. Figure 5-2 illustrates the tradeoffs schematically. The following sections will deal with the determination of the various elements of these costs.

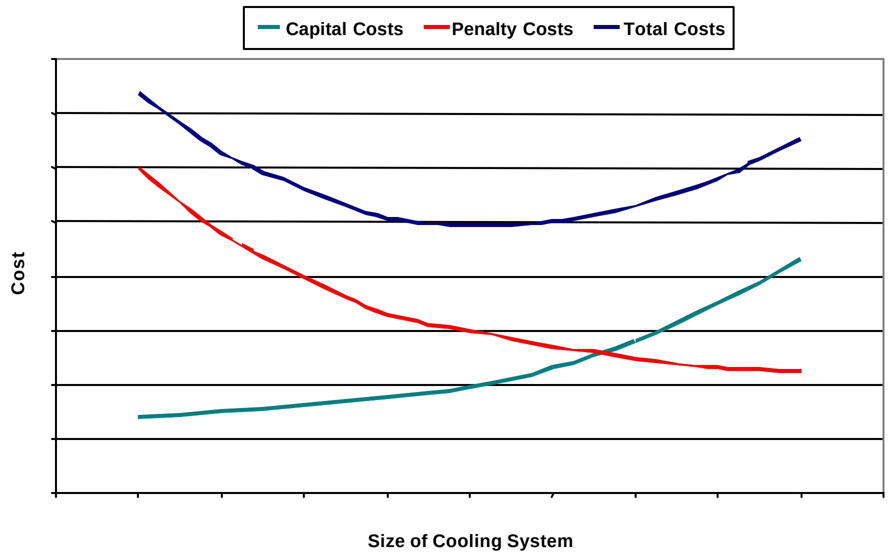

Figure 5-2
Schematic of Tradeoffs

Previous Analyses

As part of work done on water-conserving cooling systems during the 1970s and 1980s, a number of studies were conducted to compare the costs of wet, dry and wet/dry cooling. They represent many different approaches to determining a fair estimate of the cost of using dry cooling and comparing it to that of conventional cooling methods, either once-through or wet recirculating systems. They differ from the current study in many important ways:

- They are for large, standalone steam power pants, both nuclear and fossil (coal fired).
- They were evaluated for sites all around the country---many with meteorology very different from any found in California;
- They are based in large measure on economic assumptions and criteria appropriate for a regulated power industry; and
- The costs of equipment, fuel, electricity, water, and other components of the total cost of power production have changed dramatically since most of these evaluations—and these changes have not necessarily been in proportion to one another.

As a result, the absolute costs, the $/kW values, and even the cost ratios from these older studies are no longer directly useful for current estimates and system choice. However, some things can be learned from previous work:

- Because the systems being compared were designed and priced in a consistent way at the time of the study, the cost ratios remain a good indicator of the sensitivity of total costs to variations in individual cost elements.

- The effect can be seen of different approaches to the cost comparison methodology; effects arise from
  - Different methods of setting baselines and determining the amount of reduced capacity or generated energy resulting from the use of dry cooling;
  - Different methods of determining the cost of replacing these shortfalls;
  - Differences in site meteorology; and
  - Differences in plant characteristics, such as the effect of different steam turbine designs (conventional, high back pressure, extended range, etc.).

The studies were done by steam turbine vendors (GE and Westinghouse), architect-engineering firms (R. W. Beck, Gilbert Associates, United Engineers and Constructors) and consulting firms (Dynatech R/D Company). A comprehensive survey of these studies was conducted by Mitchell (Mitchell 1989) to put them on a common basis and understand the differences among them. Detailed discussions of each study are found in that reference.

The studies summarized in this section were performed by R.W. Beck (Rossie, 1970; Rossie, 1972; Rossie, 1973; Mitchel,l 1978), United Engineers and Constructors (Hu, 1976; Hu, 1977 ; UEC, 1975), GE (Sebald, 1976), Westinghouse (Oleson, 1972), Gilbert Associates (Clukey, 1976), and Dynatech (Guyer, 1980). Summary results are presented here in Figures 5-3 to 5-6 as ratios of the important cost indicators:

- Capital cost ratios (Figure 5-3)
- Capital cost + capacity replacement cost ratios (Figure 5-4)
- Total evaluated cost ratios (Figure 5-5)
- Busbar power production cost ratios (Figure 5-6).

Cost factors are defined as follows:

- Capital cost: includes everything from the turbine flange outward plus, where appropriate, incremental steam supply costs, turbine-generator cost adders, and cost of supplemental system for providing cooling for plant auxiliaries.
- Capacity replacement cost: cost of equipment needed to deliver to the power system that amount of generating capacity lost on the hottest hour of the year as a result of using dry cooling as compared to a reference (baseline system), usually wet cooling.
- Total evaluated cost: includes only costs affected by the choice of cooling system (cooling system capital cost, fixed O&M costs, replacement capacity and energy costs, cooling system make-up water costs, and incremental fuel costs in cases where the steam supply was scaled up to account for the poorer heat rates of high back-pressure turbines).
- Busbar production cost: includes all the costs of generating electricity and the plants being compared (since the cooling system costs are a small fraction of the total plant capital, operating, and fuel costs, these ratios are normally close to unity, even when the total cooling system cost ratios are 3 to 5 or higher).

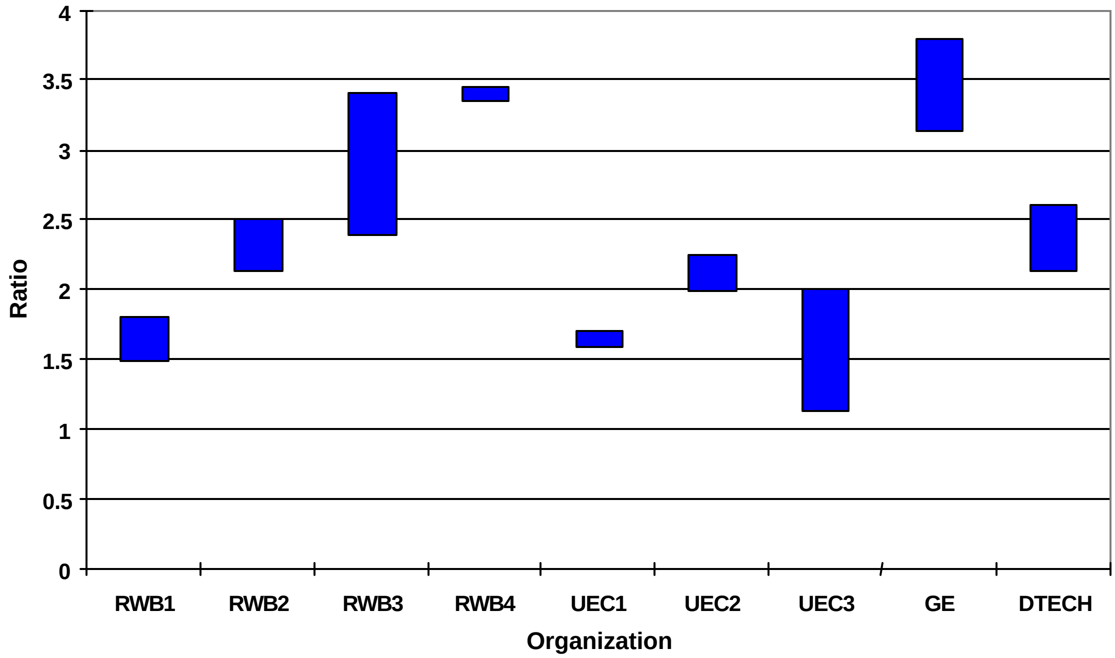

Figure 5-3
Capital Cost Ratio

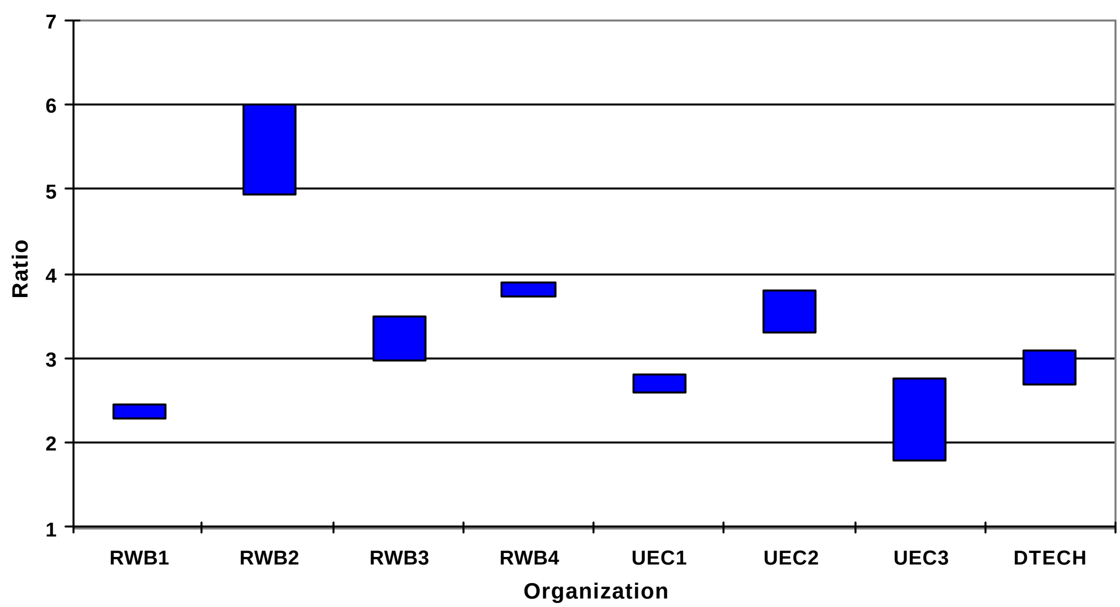

Figure 5-4
Capital Plus Capacity Replacement Cost Ratio

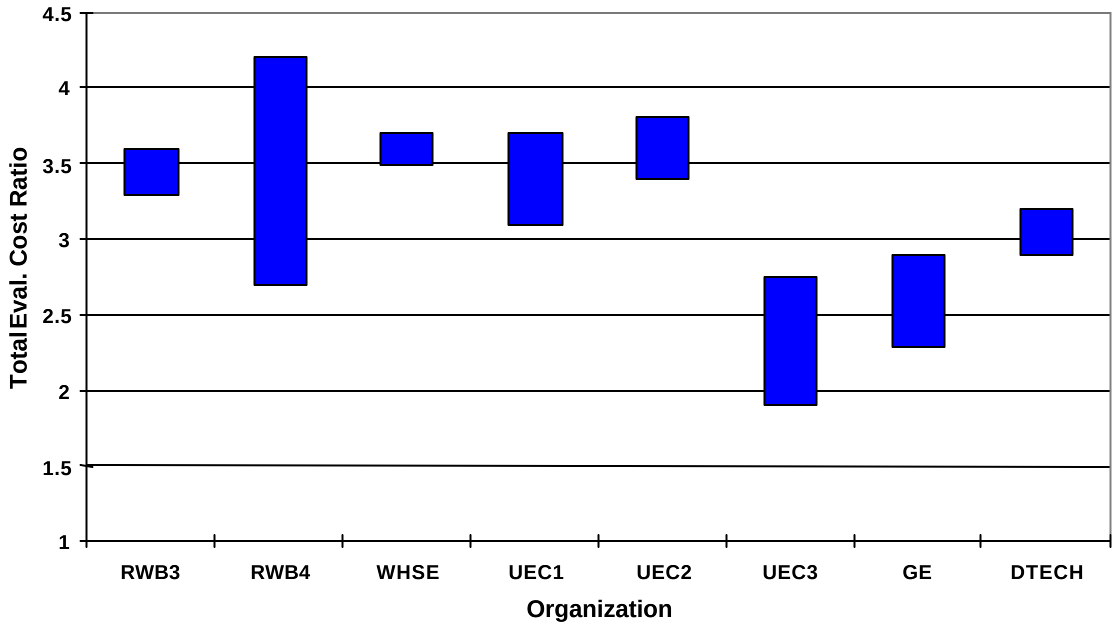

Figure 5-5
Total Evaluated Cost Ratio

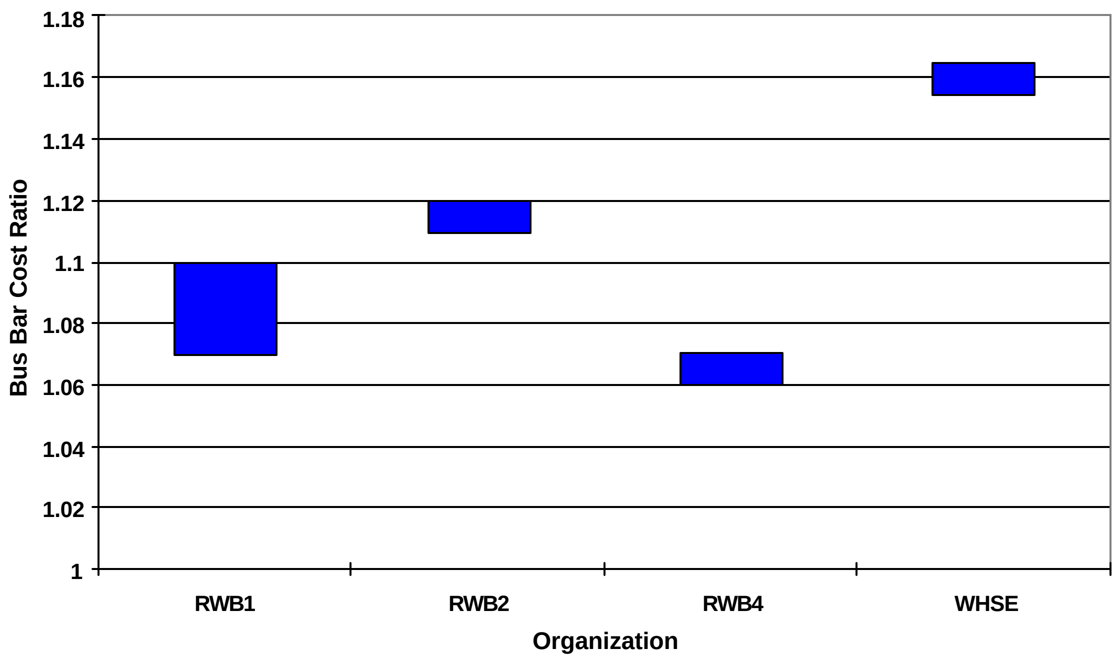

Figure 5-6
Busbar Power Cost Ratio

Cost Development

The following section provides a detailed basis for the estimated cost used in this study to compare alternative cooling systems. Costs are developed for three different systems:

- Recirculating wet cooling
- Direct dry cooling
- Evaporative condensers

For each of the systems, information is provided for each cost element influenced by the choice of cooling system and important to the economic analysis and system selection. These elements are

- Capital cost,
- Cost of energy required to run the cooling system,
- Cost of plant capacity reduction,
- Cost of efficiency reduction (reduced output or increased fuel cost),
- O&M costs (excluding energy), and
- Other miscellaneous cost considerations.

The capital cost analysis for this study was based on information assembled from a number of sources. The primary information was provided by several of the major equipment vendors in the form of budget price estimates for four specified sites, each representative of a California meteorology and location. In addition, information was obtained from licensing application material supplied by the CEC, energy industry studies, interviews with system owner/operators, and the open literature.

In some cases, the details and sources of the specific information were not for attribution. Therefore, the costs in the following sections are presented as ranges or as qualitative estimates. It is significant, however (as discussed in connection with some of the individual items), that there is excellent agreement among most of the estimates. There still may be substantial differences among cost estimates for an installation at a particular site. However, the source of such differences, as will be discussed in many of the following sections, is the choice of design point and not differences in the estimated cost of a system of a particular size and capability.

The equipment to be included in the cost estimate is everything downstream of the turbine flange and includes the costs of engineering, site preparation, erection, installation, and testing. Estimates of this kind cannot include the level of detail that is normally used in actual design calculations. General cost categories of the sort found in engineering "handbooks" are used in this analysis.

Recirculating Wet Cooling

The two major elements of a recirculating wet cooling system are the cooling tower and the shell-and-tube surface condenser. The system arrangement is shown schematically in Figure 2-3. The cost development for recirculating wet cooling is presented in the following sections:

- Surface condenser costs
- Cooling tower costs
- Capital cost elements
- Example costs
- Alternative designs: low first cost vs. total evaluated cost
- Case studies and cost correlations

Surface Steam Condenser

The steam surface condenser is one of the major cost components of a once-through or closed-cycle wet cooling system. For this study, a range of costs was developed for a conventional shell-and-tube steam condenser to be used in conjunction with a wet cooling tower.

The condenser specifications were taken from the plant operating conditions given in Section 2, specifically,

Turbine exhaust flow: 1,000,000 lb/hr @ 5% moisture
Heat load: 980×10^6 Btu/hr
Turbine back pressure: 2.5 in. Hga (T_{cond} = 108.7°F).

Condenser design guidelines were chosen as

Tube material: 304 stainless steel; 1” OD/20 gauge
Terminal temp. difference (TTD): 5–10°F
Range: 15–25°F
Cold water temperature: 70–90°F

Condenser area requirements and costs were determined by calculations based on Heat Exchanger Institute Handbook (HEI 1995) procedures and by vendor estimates.

HEI Procedures

Base heat transfer coefficients (U_o) for a typical range of tubeside water velocities were obtained from the HEI Handbook. Correction factors were also obtained for the following:

Cleanliness coefficient: $F_c = 0.85$
Tube material factor: $F_m = 0.86$ (for 304 stainless steel)
Temperature correction factor: $F_w = 1.04$ (at 80°F)

Table 5-1 provides corrected overall heat transfer coefficients (U), given by

$$U\{Btu/ft^2\text{-hr-}°F\} = U_o \times F_c \times F_m \times F_w$$

Table 5-1
Overall Heat Transfer Coefficient vs. Tubeside Water Velocity in Surface Steam Condensers (source: HEI Handbook)

Velocity, V (ft/sec)	**Base Heat Transfer Coefficient, U_o** (Btu/ft^2-°F)	**Corrected Heat Transfer Coefficient, U** (Btu/ft^2-hr-°F)
5	588	447
6	645	489
7	696	529
8	745	565
9	783	595

An example calculation of the required area for a mid-range of expected operating conditions is given by

$$A_{req'd} = Q/U \times \Delta T_{ln\ mean},$$

where

$$\Delta T_{ln\ mean} = \{(T_{cond} - T_{cw\ in}) - (T_{cond} - T_{cw\ ex})\}/\ln\{(T_{cond} - T_{cw\ in})/(T_{cond} - T_{cw\ ex})\}$$

For tube-side water velocity of 8 ft/sec, TTD of 7°F, range of 20°F, and T_{cond} of 109°F, it follows that $T_{cw\ in}$ = 82°F, $T_{cw\ ex}$ = 102°F, $\Delta T_{ln\ mean}$ = 15°F, and $A_{req'd}$ = 117,075 sq ft.

Supplier Quotes

A major condenser supplier was asked to provide budget price estimates for the specified heat load, condensing temperature, and tube material for five cold water inlet temperatures from 70 to 90°F. Vendor-selected design guidelines included the following:

Range = 20°F (if available)
TTD = 5°F minimum
Tube side Δp_{max} = 10 psi

All cases resulted in two tube pass designs, divided water box, carbon steel shell, tube support plates, tube sheets, air cooling shrouds, and water boxes. Tube side velocities ranged from 7.5 to 10 ft/sec and the number of tubes from 9000 to 16,000. The prices included an air ejector air removal package. The prices were quoted “ex-works,” meaning ready for shipping on the supplier’s loading dock. Shipping, unloading, assembly, and testing are extra to be provided by others.

The estimated areas and budget prices are shown in Table 5-2. The sample case, based on HEI Handbook (HEI, 1995) procedures, is included in the table for comparison purposes and shows excellent agreement.

Table 5-2
Surface Steam Condensers, Supplier-Quoted Prices (Equipment "Ex-Works")

Cold Water Temp. (°F)	Surface Area (sq ft)	Budget Price ($)
70	60,578	790,000
75	75,887	990,000
80	102,843	1,240,000
82 (HEI example)	*117.075*	--
85	147,952	1,630,000
90	163,386	1,800,000

For a total cost estimate (including installation), factors of 15 to 20% were suggested by architect/engineer personnel (Shaw/Brock). A 20% adjustment was chosen. Table 5-3 displays the costs, normalized against area and turbine output (in kWe), along with the log mean temperature difference ($\Delta T_{\text{ln mean}}$) for each case. Figure 5-7 displays the costs in $/kWe plotted against $\Delta T_{\text{ln mean}}$ showing a smooth relationship that is used for scaling condenser costs for the cost comparisons.

Table 5-3
Surface Steam Condensers—Normalized Budget Prices

Cold Water Temp. (°F)	$DT_{\text{ln mean}}$ (°F)	Base Price	Installed Price (+ 20%)	Price/Area ($/sq ft)	Price/Output ($/kWe)
70	27.5	$ 790,000	$ 948,000	15.6	5.58
75	22.2	$ 990,000	$ 1,190,000	15.7	7.00
80	16.7	$ 1,240,000	$ 1,490,000	15.4	8.76
85	12.0	$ 1,630,000	$ 1,960,000	13.2	11.5
90	10.4	$ 1,800,000	$ 2,160,000	13.2	12.7

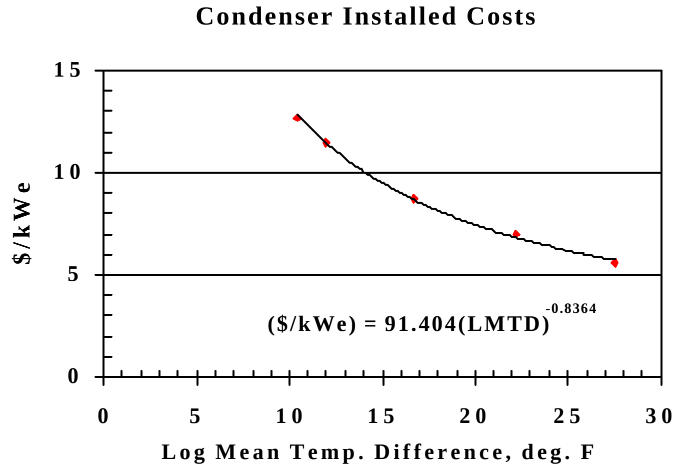

Figure 5-7
Supplier Quote Correlation

Comparison with Published Costs

As a final "sanity check," a comparison was made with some values found in the technical literature and in submissions to licensing hearings. The three sources used were

- Material submitted as part of the Elk Hills (Miller, 2000) licensing hearings,
- Material submitted as part of High Desert Power Project licensing hearings (Ledford, 1999), and
- Hutton (1999), in which wet towers and surface condensers are compared with evaporative condensers.

Detailed information on the particular design values used in each case is not necessarily available, nor is it always known whether installation, delivery, engineering, or other contingencies are included. Therefore, these comparisons are simply to determine whether the values are in the general range that would be predicted by the correlation used in this study.

Costs given for the 170-MWe (steam) Elk Hills plant (Miller, 2000) and the 240-MWe (steam) High Desert Power Project (Ledford, 1999) are $1.9 and $2.9 million respectively, corresponding to $11/kWe and $12.1/kWe. No information was available to calculate either the log mean temperature difference or the cost per unit area, but the points fall within the range of costs shown in Table 5-3 and Figure 5-7.

The case studies presented by Hutton (1999) range from $8 to $12.5/kWe, corresponding to the middle to high end of the range shown in Figure 5-7 using a conversion of 1,000,000 lb/hr steam flow being equivalent to 170MWe. However, the cost per square foot in those case studies was apparently fixed at $25, considerably higher than the values developed above.

Two differences may account for the lack of agreement. The tube materials in the analysis by Hutton (1999) were copper (presumably Cu-Ni or Admiralty, but not specified) rather than 304 stainless steel, and the unit sizes ranged from 100,000 to 400,000 lb/hr in steam flow, as compared to 1,000,000 lb/hr. An informal consultation with the vendor that supplied the estimates in Table 5-2 suggests that these differences might result in “budget prices” in the $15 to $20/sq ft range. This translates to an “installed price” range of $18 to $25/sq ft, approaching that of Hutton (1999).

Wet Cooling Towers

The base system chosen to represent recirculating wet cooling is the mechanical draft, cross-flow wet cooling tower in the traditional in-line arrangement of cells to form a rectangular tower (as shown schematically in Figure 2-4). This choice, while giving up some performance and efficiency advantage to counter-flow designs, has initial cost, maintenance, and operability advantages, certainly in comparison with natural draft or circular mechanical draft designs. This is particularly true for the modest sizes required for the steam-cycle portion of combined-cycle plants, typical of the current California market.

The major capital cost elements are characterized in Table 5-4. Table 5-5 provides a typical cost breakdown by cost element for a selected case study example, the Central Valley Site (designed for 2.5 in. Hga at average seasonal temperature and humidity: $T_{avg.\ dry\ bulb}$ = 67°F; $T_{avg.\ wet\ bulb}$ = 60°F).

Table 5-4
Capital Cost Elements for Wet Cooling System Equipment for New Plants

Element	Comment	Cost
Wet cooling tower	Erected tower including structure, fans, circulating pumps, fill, drift eliminators, etc.	Strongly dependent on materials, assumed Douglas Fir; typically 35 to 45% of system cost
Installation/erection	Included in bases price	--
Surface steam condenser	Major cost element (see previous subsection)	Typical range of $5 to $12/kWe; approx. 35 to 45% of system cost
Tower basin	Including typical site preparation	Significant cost item; function of tower size; estimated at $25/ft of basin perimeter plus $10/sq ft of basin area; typically 3 to 6% of system cost
Electricals and controls	Fan/pump motor wiring and controls, etc.	Important cost item; estimated at $25,000 per cell
Circulating water system	Pumps, piping, valves, etc.	Can be significant cost item; dependent on site layout; assumed at 5% of total installed cost
Water supply/intake structure	Highly site dependent; minor if source is nearby; major, if water supply is far from site or at much lower elevation	Estimated at 1 to 2% total installed cost
Water treatment/blowdown discharge	Usually minor; may be significant if in zero-discharge region where evaporation ponds or brine concentrators may be required	Estimated at 1% total installed cost
Auxiliary cooling	Typically 5% additional heat load	Estimated at 5% additional cost
Additional elements	Typically minor and site dependent	Not included in case study estimates of comparisons
– Site preparation/ access provision	– Highly site dependent; likely minor; not likely to be affected significantly by system choice	
– Winter operation; freeze protection	– Location dependent	
– Low-noise fans	– Significant cost if required; more important for dry systems than for wet	
– Painting	– Typically minor costs	
– Fire and lightning protection	– Typically minor costs	
– Acceptance testing	– Typically minor costs	

Table 5-5
Capital Cost Breakdown for Wet Cooling System Equipment at California Central Valley Location, "Low First Cost" Design

Element	Cost
Wet cooling tower	1,377,000
Installation/erection	(included in above)
Surface steam condenser	1,486,000
Tower basin	165,500
Electricals and controls	125,000
Circulating water system (@ ~5–6%)	170,000
Water supply/intake structure (@ 2%)	70,000
Water treatment/blowdown discharge (@ 1%)	35,000
Auxiliary cooling (@ ~5–6%)	170,000
Total	3,600,000

Wet System Cost Analysis

Budget prices were obtained for wet cooling towers for each of the four reference sites described in Section 4. For each site, tower size and power requirements were determined for a series of operating conditions corresponding to a range of circulating water flow rates and, hence, tower ranges. In all cases, the hot water (or "tower on") temperature was assumed to be 109ºF. Assuming a condenser terminal temperature difference (TTD) of 7ºF, this corresponds to a condensing temperature of 116ºF and a turbine back pressure of 3 in. Hga. The circulating water flows for the specified design heat load of 980 x 10^6 Btu/hr ranged from a maximum of 130,800 gpm (tower range = 15ºF) to a minimum of 65,400 gpm (tower range = 30ºF). The flow was never reduced to a level where the approach to wet bulb would be less than 5ºF.

Two criteria were used for each site:

- A "low first cost" case in which the capital cost of the tower was minimized at the expense of additional fan power; and
- An "evaluated cost" case in which the sum of the capital cost and the cost of power evaluated over the assumed 30-year life of the tower was minimized.

The latter method results in a more expensive tower but a lower lifetime cost.

The budget price for the tower included the erected/installed cost of the tower itself, the basin costs, and the fan/motor costs, but not other items such as the condenser. Those elements of the cost that were or were not included—and how they were accounted for to develop the capital cost for a complete wet recirculating cooling system—are described in detail in Tables 5-2 through 5-4 and the accompanying text.

The tower and system costs are detailed in Table 5-6, and the correlations developed from the initial vendor data are presented in Figures 5-8 through 5-15.

Table 5-6
Site-to-Site Cost Estimates—Wet Cooling Tower and Surface Condenser for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle

	Desert Site	Mountain Site	Valley Site	Bay Area Site
Low First Cost Design				
Total Cost	2,924,000	2,710,000	2,820,000	2,680,000
Total BHP (in hp)	1723	1851	1794	1505
-fans (in hp)	1234	1498	1441	1198
-pumps (in hp)	489	353	353	307
Minimum Evaluated Cost Design				
Total Cost	3,331,000	3,118,000	3,405,000	2,960,000
Total BHP (in hp)	964	978	1030	987
-fans (in hp)	377	645	713	680
-pumps (in hp)	587	333	317	307

Figure 5-8 presents the raw data for the capital cost of the tower alone for each of the four sites over the range of acceptable circulating water flow rates plotted against tower approach ($T_{\text{circ. cold}} - T_{\text{amb. wet bulb}}$) for the "Low First Cost" design case. It can be seen that while the expected general trend of lower cost/smaller tower at higher approach temperatures is evident, there is scatter and discontinuity even within the points for each site. This results from the fact that towers are a collection of individual cells, and designs cannot vary continuously as certain limits on pressure drop, height, and other design quantities are encountered. Therefore, for general studies of this type it is appropriate to develop smoothed curves to approximate the costs for given conditions. The results will vary, but not excessively so, for the price that would be developed in a site-specific, detailed design estimate.

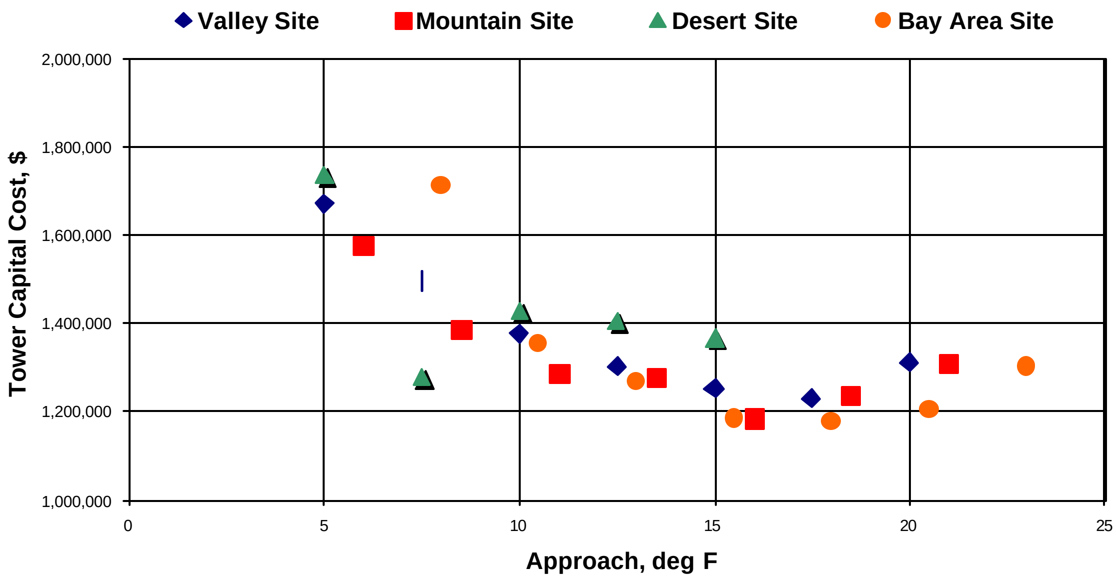

Figure 5-8
Wet Cooling Tower Capital Cost vs. Approach for Low First Cost Design (for New 500-MWe Facilities with 170-MWe Steam Cycle)

Figure 5-9 presents data in a similar form for the capital cost of the complete wet recirculating cooling system for new facilities. Two items are noteworthy: First, the costs now exhibit a minimum, rather than a monotonic, decline with increasing approach temperature. This results from the higher costs of the condenser at the higher circulating water flows, smaller ranges, and higher condenser inlet water temperatures associated with the higher approaches. Second, the significantly higher wet bulb temperatures at the Desert site again result in higher condenser inlet temperatures than the other sites and substantially higher condenser costs.

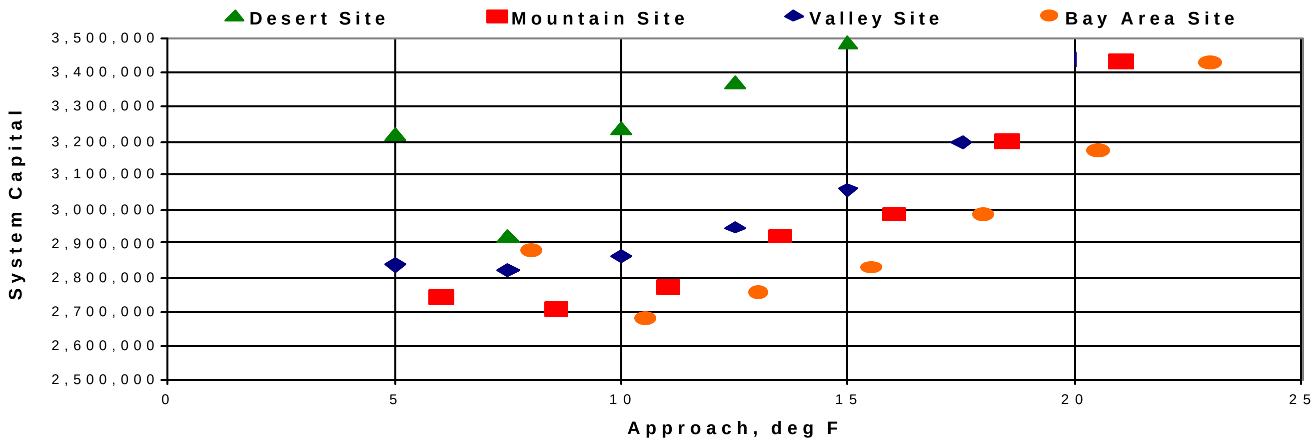

Figure 5-9
Wet Cooling System Capital Cost vs. Approach for Low First Cost Design (for New 500-MWe Facilities with 170-MWe Steam Cycle)

Figures 5-10 and 5-11 present the same data in the same form for the cases in which the towers were sized for a minimum evaluated cost that includes the cost of power over the life of the tower. A comparison of the two cases shows substantially higher capital costs for the latter case.

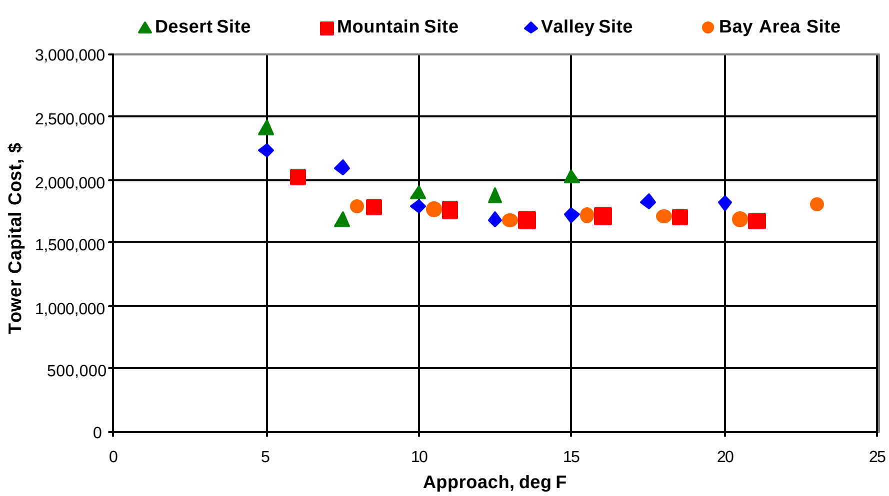

Figure 5-10
Wet Cooling Tower Capital Cost vs. Approach for Minimum Evaluated Cost Design (for New 500-MWe Facilities with 170-MWe Steam Cycle)

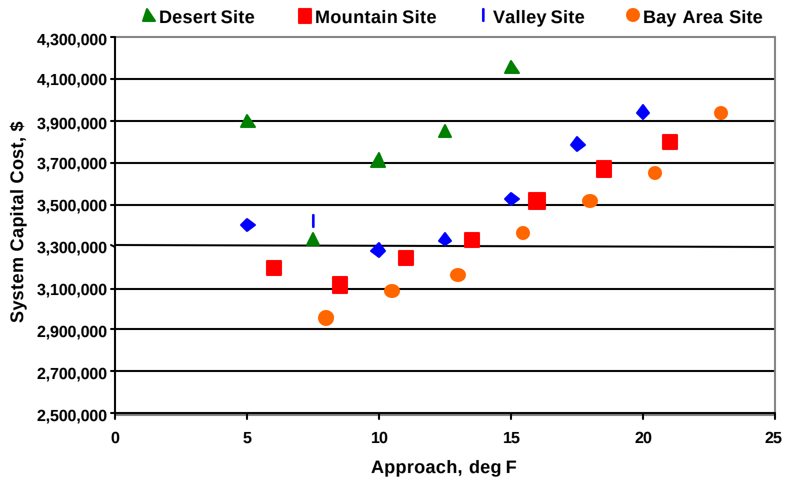

Figure 5-11
Wet Cooling System Capital Cost vs. Approach for Minimum Evaluated Cost Design (for New 500-MWe Facilities with 170-MWe Steam Cycle)

Figure 5-12 displays the power requirements for each of the designs for the two cost cases. It is seen that the power for the minimum evaluated cost designs is typically one-half to two-thirds that of the low first cost designs.

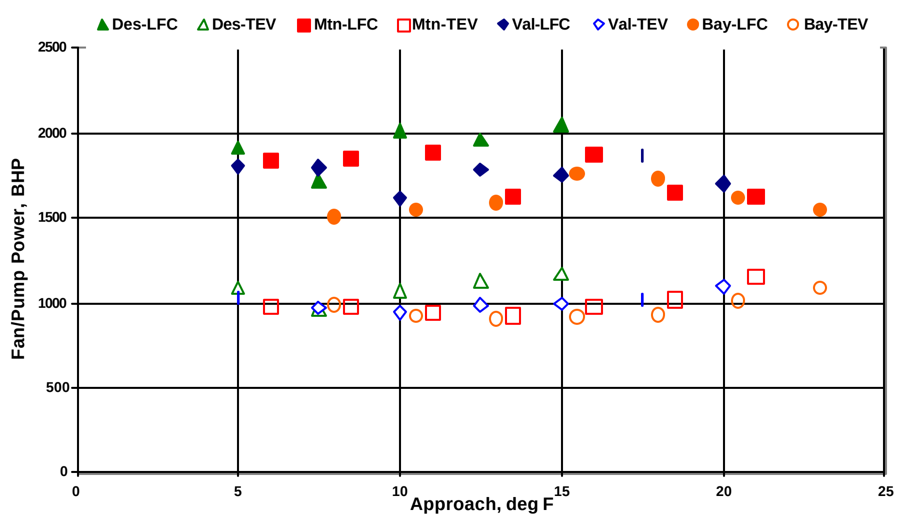

Figure 5-12
Wet Cooling System Power Requirement (Pumps and Fans) vs. Approach (for New 500-MWe Facilities with 170-MWe Steam Cycle)

The cost of the power required to operate the pumps and fans of the cooling system is borne continuously for the life of the plant. In order to put these continuing future costs on a common basis with the initial capital costs, they have been converted into an “evaluated cost per kW.” The evaluated cost of a future lost kW depends on many factors—the value of the energy it would have generated, an anticipated escalation or inflation rate, an expected discount rate, and a marginal tax rate and the life of the plant over which the costs are to be borne.

The values chosen for evaluating the cooling system power costs are an energy cost of \$60/MWh, a 6.7% discount rate, a 3% escalation, a 50% tax rate, and a 30-year plant life. These result in an evaluated power cost of \$3625/kW. These parameters were selected in discussions with vendors, users, and the CEC as reasonable values for the power industry situation in California at the present time. All of these values are subject to significant variability and debate, and a complete parametric treatment would be valuable. It is beyond the scope of this study at this time, however. Therefore, this method for evaluating power costs for comparison and optimization purposes will be used throughout the study, with energy cost varied in the treatment of penalty costs.

Figures 5-13 and 5-14 show the actual total evaluated cost of each of the cases, where the power costs are evaluated as described above.

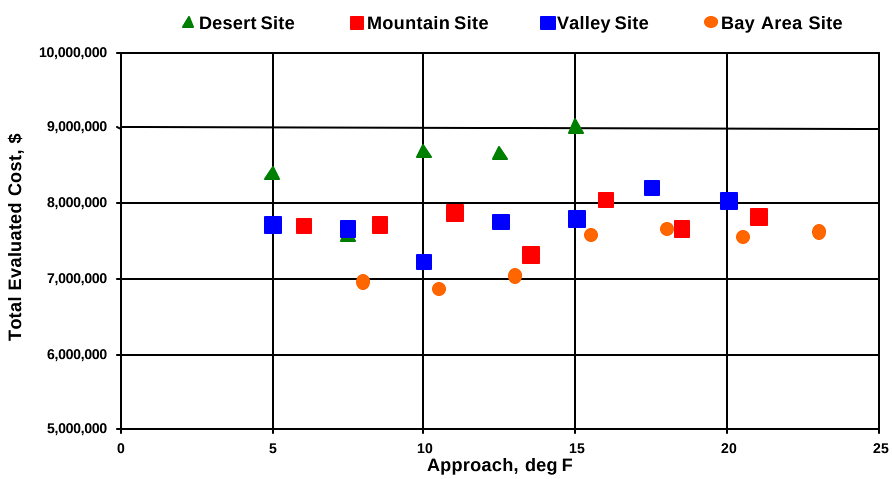

Figure 5-13
Wet Cooling System Total Evaluated Cost vs. Approach for Low First Cost Design (for New 500-MWe Facilities with 170-MWe Steam Cycle)

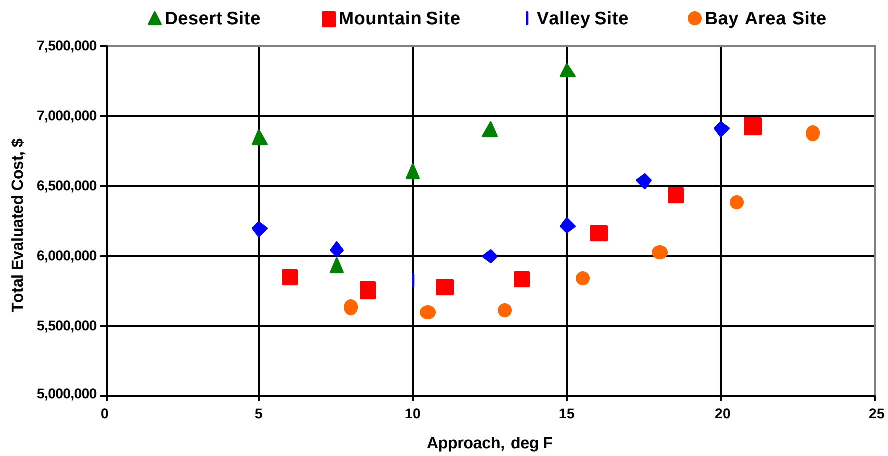

Figure 5-14
Wet Cooling System Total Evaluated Cost vs. Approach for Minimum Evaluated Cost Design (for New 500-MWe Facilities with 170-MWe Steam Cycle)

Figure 5-15 simply summarizes the comparison of Figures 5-13 and 5-14 and illustrates that the total costs for the low first cost designs are substantially higher than those for the minimum evaluated cost designs in all cases.

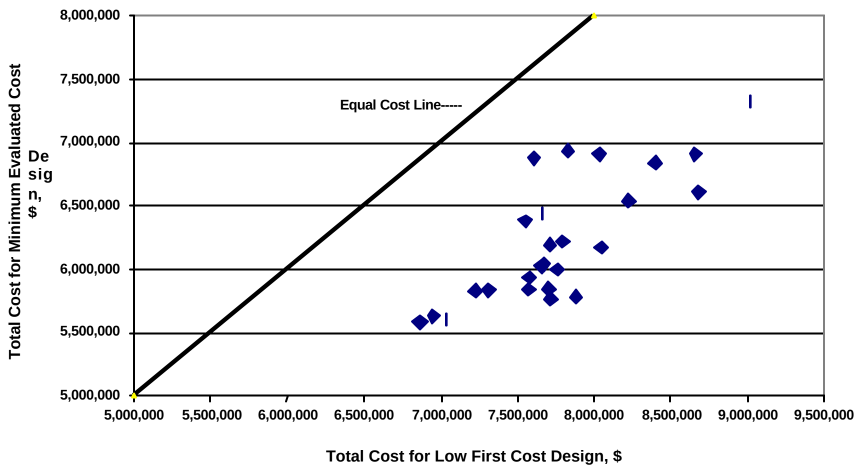

Figure 5-15
Comparison of Total Evaluated Costs for Wet Cooling Systems, Minimum Evaluated Cost Design vs. Low First Cost Design (for New 500-MWe Facilities with 170-MWe Steam Cycle)

Performance Penalties for Wet Systems

As discussed earlier, cooling systems that limit the attainable back pressure during hotter or more humid periods of the year incur penalties in the form of lost energy output from the plant. As discussed later in the section on dry systems, these penalties can significantly influence the size and cost of the optimum system for dry cooling. However, for wet cooling systems their influence is far less, particularly in climates typical of California.

Wet systems face performance limits during periods of high humidity. While sustained periods of high humidity during peak load (air conditioning) seasons are commonplace, the climate of typical California sites characteristically exhibits low humidity during peak load (high dry bulb temperatures) seasons. Table 5-7 illustrates this for the four sites used in this study.

Table 5-7
Relative Humidity at Peak Temperature Periods

Site	RH @ T_{max}	RH @ $T_{1\%}$	RH @ $T_{2\%}$	RH @ $T_{5\%}$
Desert	19	22	24	26
Mountain	16	19	19	19
Valley	24	22	23	23
Bay Area	26	36	39	46

As seen from the wet bulb duration curves in Figure 5-16, at none of the sites does the wet bulb temperature exceed the average wet bulb temperature for more than 1000 hours per year and, at all but the desert site, for more than 500 hours per year.

Assuming a back pressure of 2.5 in. Hga at the average wet bulb temperature, a 10°F increase in wet bulb temperature would correspond roughly to a back-pressure rise to 3.35 in. Hga (well below the operating limit for conventional turbines; see Figure 5-1) and a corresponding heat rate ratio of 1.005. This is equivalent to lost power generation of less than 1 MW for less than 1000 hours per year. At an energy cost of $100/MWh, the penalty would be less than $100,000 per year, which is insufficient to significantly affect the choice of an optimum-size wet cooling system. Therefore, no further consideration of energy or capacity penalty costs is provided for wet systems. This conclusion would have to be reexamined carefully for other locations with other climates.

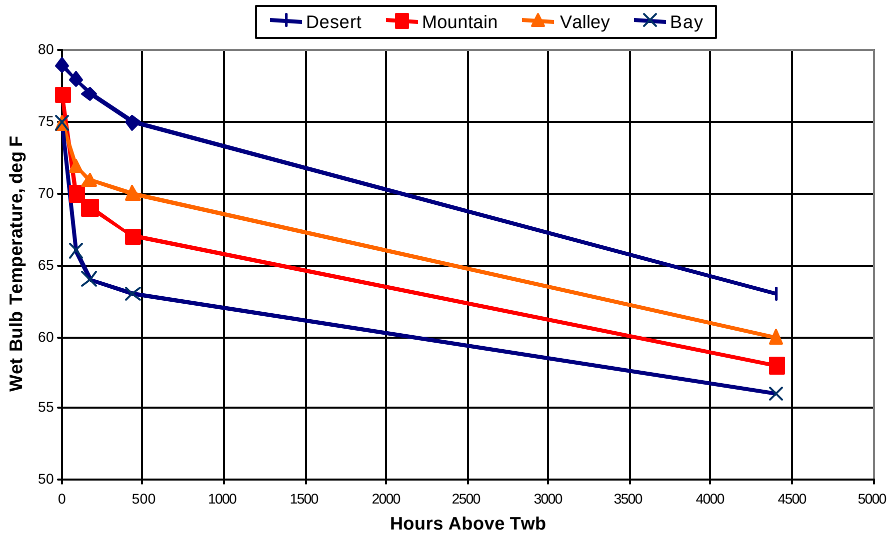

Figure 5-16
Wet Bulb Duration Curves

Dry Cooling

The base system for dry cooling is a direct system with a mechanical draft air-cooled condenser (ACC) as shown schematically in Figure 2-6. The major items of the capital cost of the system are displayed in Table 5-8 and discussed below.

Cost Elements

Typically included in base ACC cost are the basic heat transfer and flow components from the turbine exhaust to the condensate tank, specifically

- Finned tube heat exchanger elements,
- Fans and motors,
- ACC support structure,
- Steam exhaust duct,
- Piping and valves,
- Air removal equipment, and
- Support for start-up, training, and testing.
- Typically not included are erection/installation; electrical wiring, switches, etc., and hook-up; site preparation; foundation support; steam duct support; and fire and lightning protection.

Table 5-8
Capital Cost Elements for Dry Cooling System Equipment for New Facilities

Element	Comment	Cost
Air-cooled condenser	See discussion in text	Strongly dependent on choice of design point expressed as ITD (T_{cond}-T_{amb}); ranges from $100 to $250/kWe
Installation/erection	Significant cost item; quoted in different ways; see discussion in text	Ranged from $175,000 to $225,000 per cell; $200,000 used in comparisons
Steam duct support; column foundations	Installation dependent	Estimated for 10^6 lb/hr unit at $120,000 to $160,000; $150,000 used for costs and comparisons
Electricals and controls	Fan/pump motor wiring and controls, etc.; see discussion in text	Important cost item; estimates ranged from at $20,000 to $35,000 per cell; used 5.5% of installed base cost
Auxiliary cooling	Typically 5% additional heat load; typically handled with separate unit (usually wet) but occasionally as extra cells on ACC; see discussion in text	Estimated at 7.5% additional cost without specifying choice of auxiliary unit
Cleaning system for finned tube surfaces	Minor but required in most locations	Estimated at $150,000
Low-noise fans	Included in base costs (far-field sound pressure levels of ~65dBa at 400 feet)	---
Additional elements	Typically minor and site dependent	Not included in case study estimates of comparisons
- Water supply/intake structure	Minor (but not zero) for dry systems	
- Water treatment/blowdown discharge	Minor (but not zero) for dry systems	
- Site preparation/access provision	Highly site dependent and likely minor; not affected significantly by system choice	
- Finish paint; fire/lightning protection	Typically minor costs	
- Winter operation; freeze protection	Location dependent and relevant to both wet and dry systems; typically 2 to 4% of total installed cost; not included for California estimates	

Budget prices for a dry cooling tower and associated heat transfer and flow components were obtained from several major vendors for each of the four sites described in Section 4. The vendors were provided with information on the site location and meteorology and plant characteristics. A brief discussion of some of the cost items follows.

Installation/Erection

Installation and erection costs were quoted separately, with the usual caution that they could be quite site specific. In some cases, they were expressed as “per cell” costs ranging from ~ $175,000 to $225,000. In other cases, they were quoted as a total cost for the entire tower, based apparently on an estimated percentage of the ACC capital cost. When the per cell costs were translated into a total tower cost, they too were approximately a fixed percentage of the capital costs. However, the calculated percentages varied considerably from source to source, ranging from under 30% to over 50%. Nonetheless, the total costs (capital plus installation/erection) shown in Figure 5-17 lie on a single curve. The good agreement among overlapping points from the several sources suggests that there may be compensating differences among what is included in which part of the estimate.

Electrical Wiring/Hook-Up

Electrical hook-up was normally not included in the base price. Again, a per cell estimate was given, with a fairly wide range of $20,000 to $35,000 per cell. When applied to the individual cases, this worked out to a percentage of the base installed cost ranging from 3.5 to 7.5%. An intermediate value of 5.5% was used in the comparisons.

Auxiliary Cooling

Auxiliary cooling requirements are usually but not always met with separate fin-fan units. For a typical estimate of a plant requirement for an auxiliary cooling of 5 to 10% of the condenser heat load, an allocation of 7.5% of base price was added. This is consistent with the one other study in which the auxiliary cooling cost was broken out and discussed as a separate item. At one of the sites visited with an operating system (the Crockett Co-Generation facility), the auxiliary cooling was provided by extra cells on the ACC. Three of 15 cells, or 20% of the tower capacity, were dedicated to auxiliary cooling. This approach was characterized by one of the estimators as “quite costly” and represents more than the generally allocated amount.

Additional Items

A number of additional items, not included in the base price, are normally necessary for a complete, installed system. These include sensors and controls, finned surface cleaning equipment, fire and lightning protection, and finish painting. The costs are mostly minor with the exception of the sensors and controls. Several sources estimated the cleaning system as $100,000 to $150,000.

Dry System Cost Analysis

In estimating costs of dry systems for the four case study sites, vendors were free to select the design point of their own choosing, based presumably on their engineering and commercial experience with the appropriate tradeoff between initial cost and performance penalties. As a result, the design ambient temperature for a particular site differed significantly from estimate to estimate, leading to towers of significantly different size and cost for each location.

Figure 5-17 gives the range of capital cost vs. ITD. These costs vary by a factor of two for ACC sizes ranging from an ITD of 20°F (large surface area) to 55°F (small surface area).

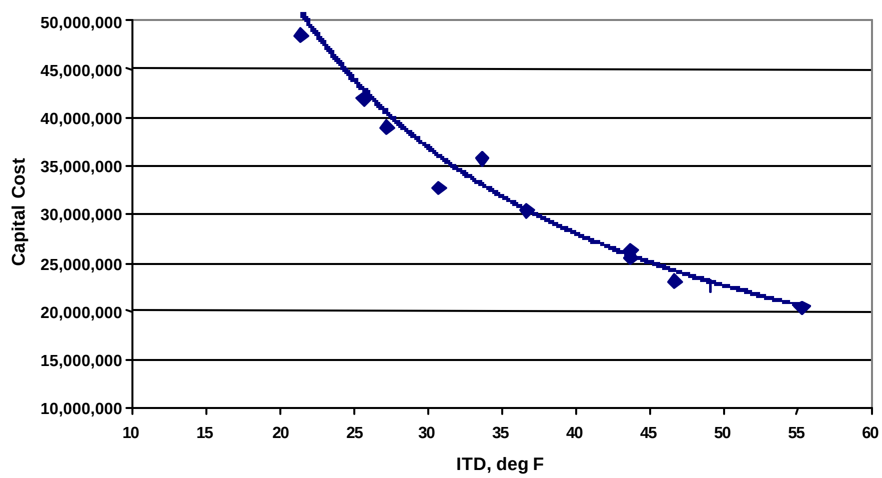

Figure 5-17
Capital Cost vs. ITD for Air-Cooled Condenser (for New 500-MWe Facilities with 170-MWe Steam Cycle)

Figure 5-18 displays the same data with costs expressed in $/kWe, since this is a common metric for estimating the cost of power plants and their components. It should be noted that all the case studies were for the same heat load, steam flow, and turbine back pressure, differing only in site elevation (slightly) and in site temperature and humidity profiles; extrapolating these costs on a $/kWe basis to units of very different size or operating conditions should be done with caution. For the cases in this study, the costs lie on a single curve over a wide range of ITD (from 20 to 55°F), with a corresponding cost range of $290 to $125/kWe.

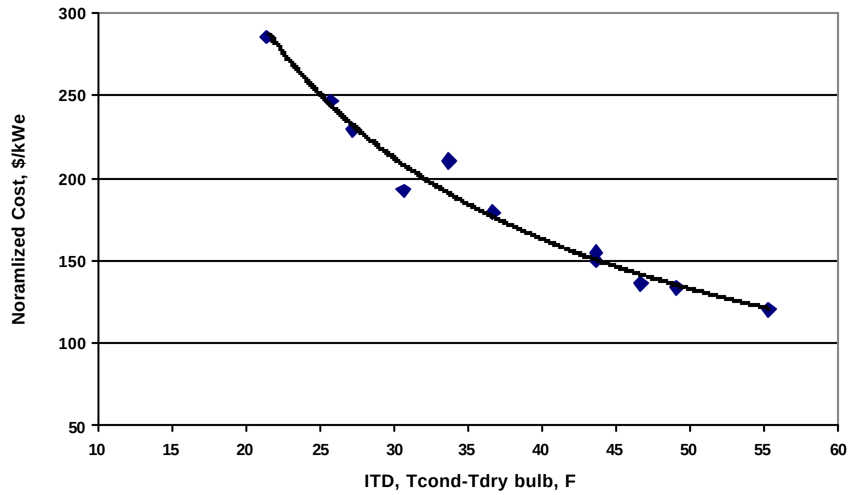

Figure 5-18
Normalized Cost vs. ITD for Air-Cooled Condenser (for New 500-MWe Facilities with 170-MWe Steam Cycle)

Power Requirements

Figure 5-19 displays the power requirements at the motor terminals for the range of design ITDs. Using the evaluated cost for power of $3625 kW developed earlier, the total evaluated costs are shown in Figure 5-20. The values were normally provided as power required at the motor terminals. In those cases where the required power was specified as fan shaft power, a motor efficiency of 95% was assumed.

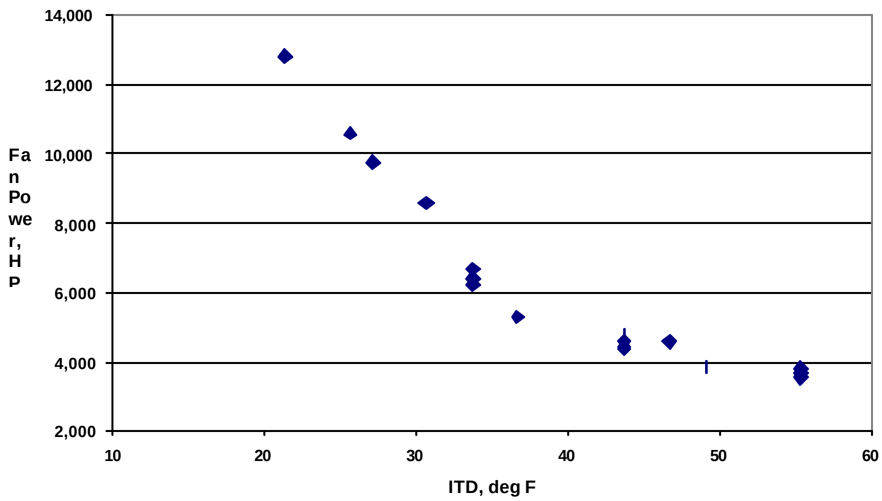

Figure 5-19
Fan Power vs. ITD

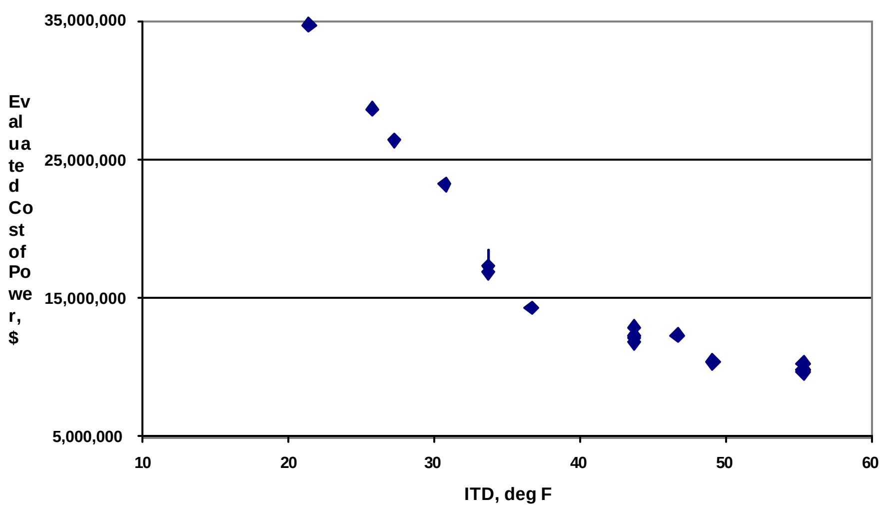

Figure 5-20
Evaluated Cost of Power

The capital costs (Figure 5-17) and the evaluated cost of power (Figure 5-20) sum to the total evaluated cost shown in Figure 5-21.

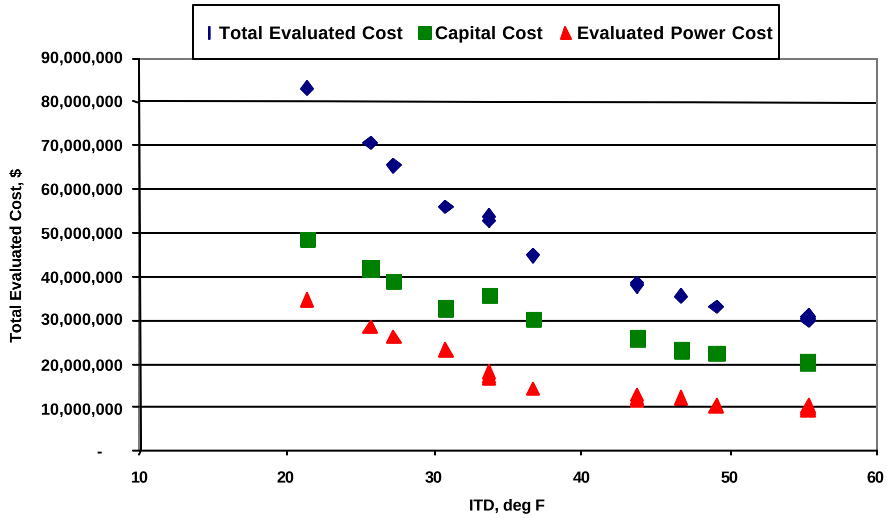

Figure 5-21
Total Evaluated Costs vs. ITD

Determination of Penalty Costs

As discussed in earlier sections, the selection of the optimum cooling system requires a determination of the effects on plant performance. A dry cooling system is designed to maintain a certain back pressure for a given heat load at a given ambient temperature. Figure 5-22 shows the variation in turbine back pressure with ambient temperature for ACCs of differing size.

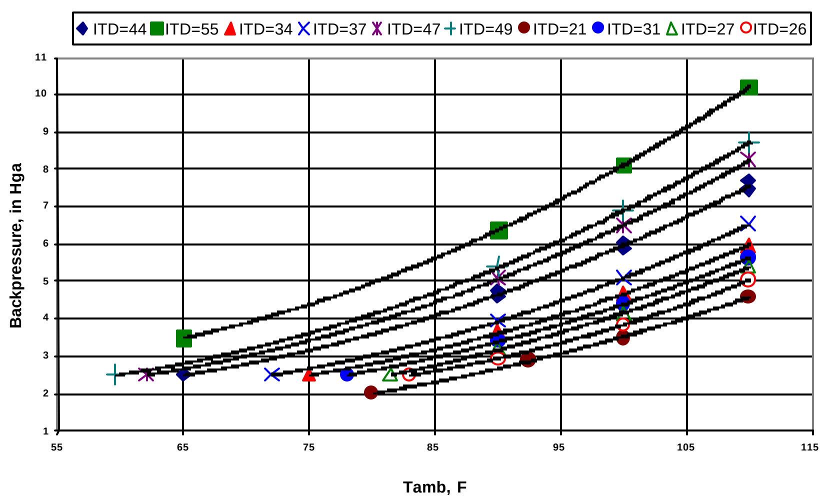

Figure 5-22
ACC Performance—Back Pressure vs. T_{amb} (for New 500-MWe Facilities with 170-MWe Steam Cycle)

The design ambient temperature is normally selected to be some value well below the maximum temperature expected at the site during the hottest periods of the year. Therefore, during periods in which the ambient temperature exceeds the design temperature, the back pressure will be higher than design resulting in a higher plant heat rate. For a steam cycle with a fixed heat input, this translates to a lower power output. Alternatively, if the heat input can be increased, as with supplementary duct firing, the plant output may be maintained, but the fuel costs will increase.

Additionally, as discussed earlier, steam turbines are designed with upper allowable limits on the back pressure. If the ambient temperature reaches a high enough level, this back-pressure limit may be approached. The steam flow must then be reduced to avoid the risk of damage to the turbine. This can result in a significant reduction in the power output from the steam cycle. In the case of some combined-cycle units, the only method of reducing steam flow may be to reduce the exhaust gas flow from the gas turbines to the heat recovery steam generator (HRSG), which will limit the output from the combustion turbine side of the plant as well—with even greater loss of energy output.

The following section presents a methodology for determining these energy and capacity penalties. The discussion uses the performance of five different ACCs, covering a wide range of size and cooling capacity, to illustrate the method, the significance of the result on the selection of the dry cooling system, and the contribution of these penalties to the total cost of a dry cooling system. The systems are compared for the Desert site described in Section 4.

Penalty Evaluation Methodology

The five design cases to be analyzed for the Desert site are characterized in Table 5-9, which summarizes their design points and size (expressed as design ITD).

Table 5-9
Cases for Penalty Evaluation Analysis

Case	Design Back Pressure, in. Hga	Design $T_{ambient}$, ºF	Design ITD, ºF	Capital Cost	Evaluated Power Cost	Total Eval. Cost
1	2.9	92.4	21.4	48,545,495	34,636,875	83,182,370
2	2.5	75.0	33.7	35,724,938	17,300,675	53,025,613
3	2.5	72.0	36.7	30,376,813	14,318,750	44,695,563
4	2.5	65.0	43.7	25,517,813	12,357,625	37,875,438
5	3.5	65.0	55.3	20,414,250	9,886,100	30,300,350

The annual temperature profile for the Desert site is given in Figure 5-23, expressed as the hours for which a given ambient temperature is exceeded.

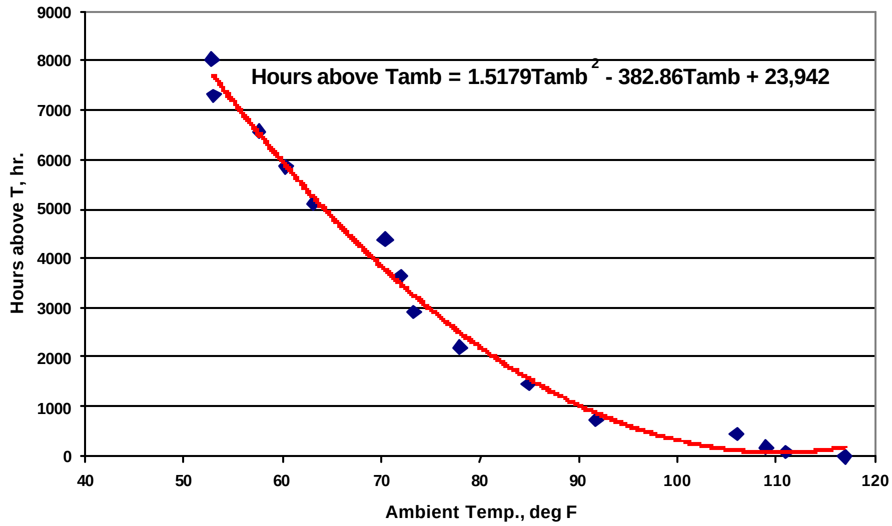

Figure 5-23
Desert Site Annual Temperature Profile

Performance curves for the ACC, shown in Figure 5-24, display the back pressure that can be achieved at a given ambient temperature for the design heat load.

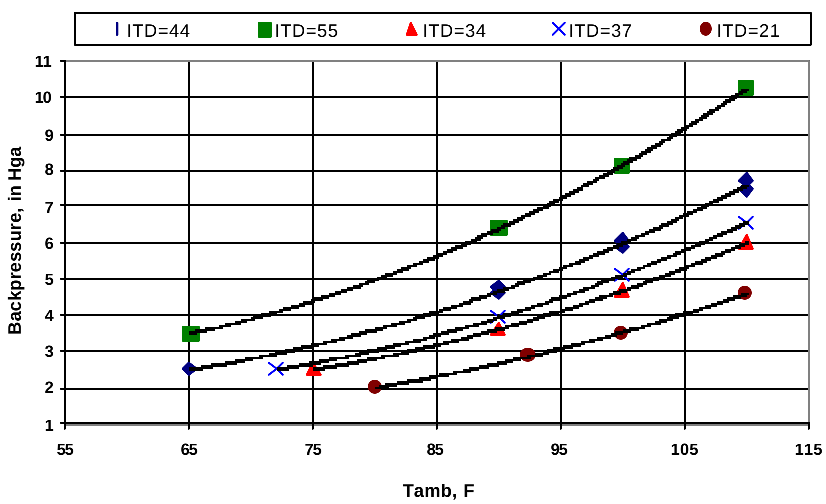

Figure 5-24
ACC Performance—Back Pressure vs. T_{amb} (for New 500-MWe Facilities with 170-MWe Steam Cycle)

From these two curves, one can calculate, for each of the five ACC designs, the number of hours per year the system would operate at or above a given back pressure. The data are given in Table 5-10.

Table 5-10
Operating Hours Above Given Back Pressure

Case	2.5 in. Hga	3 in. Hga	5 in. Hga	7 in. Hga	8 in. Hga (max)
1	2,242	1,464	102	0	0
2	3,766	2,687	683	0	0
3	4,245	3,028	835	102	0
4	5,469	4,163	1,515	354	81
5	8,594	6,435	3,028	1,464	916

The heat rate curve for the "modified" turbine displayed in Figure 5-1 can be re-plotted, using the definition of heat rate ratio to yield the lost output, in kW, at a given back pressure. This is displayed in Figure 5-25 below.

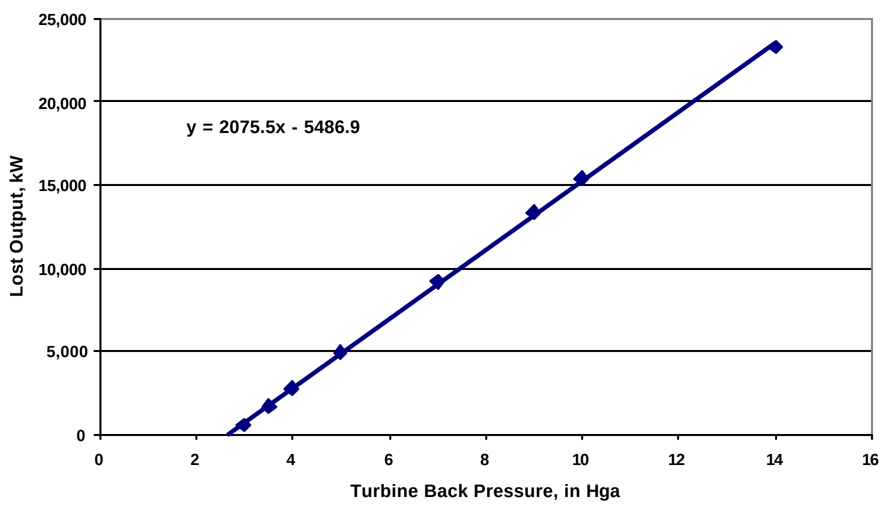

Figure 5-25
Lost Output vs. Turbine Back Pressure

This relationship, combined with the operating times from Table 5-10, provides the amount of energy lost each year as a result of increased heat rate at conditions up to the point where steam flow must be reduced to maintain back pressure below 8 in. Hga. These energy losses are tabulated in Table 5-11 below.

Table 5-11
Lost Energy from Heat Rate Penalty

Case	Lost Output, MWh
1	4,718
2	10,636
3	12,576
4	19,400
5	35,718

Additional penalties are incurred when the ambient temperature rises enough for the back pressure to approach the turbine operating limit, assumed here to be 8 in, Hga. At temperatures between this level ($T_{8"}$) and the maximum ambient temperature, the steam flow must be reduced to maintain the back pressure below this limit to avoid turbine trip.

The average reduction is steam flow is calculated at the average temperature during that period, estimated as $T_{av.} = (T_{8"} + T_{max})/2$. Operation at the higher ambient temperature (T_{av}) while maintaining the 8" limit would require an ITD greater than the available ITD by $\{(ITD_{8"} - ITD_{av})/ITD_{av}\}$x 100% or an equivalent % reduction in steam flow. Tables 5-12 and 5-13 present the reductions and lost output associated with the capacity limit conditions. Note that only Cases 4 and 5 encounter the 8" limit at the maximum expected site temperature of 117ºF.

Table 5-12
Lost Output from Capacity Reduction Penalty—Steam Side Only

Case	T_{max}	$T_{8"\ Hga}$	T_{cond} @ 8 " Hga	T_{av}	Required Reduction in ITD, %	Average Reduction in Output, MW	Duration, hours	Lost Output, MWh
4	117	112.5	152	114.8	5.7	20.2	81	1,632
5	117	99	152	108	17	38.1	916	34,884

Table 5-13
Lost Output from Capacity Reduction Penalty—Whole Plant

Case	T_{max}	$T_{8\text{” Hga}}$	T_{cond} @ 8 “ Hga	T_{av}	Required Reduction in ITD, %	Average Reduction in Output, MW	Duration, hours	Lost Output, MWh
4	117	112.5	152	114.8	5.7	39.6	81	3,205
5	117	99	152	108	17	95.8	916	87,786

The lost output associated with a reduced steam flow depends on what must be done to achieve it. If it is possible to reduce the steam flow without reducing the gas turbine output, by, for example, opening a hot gas exhaust bypass around the HRSG, then the reduction in output applies only to the steam side of the plant. If, however, as is more often the case, exhaust gas bypass is not allowed for environmental reasons, the % output reduction will apply to the entire plant as the gas turbine flow must be reduced as well.

The cost of these penalties is evaluated in a manner similar to that applied to the fan power requirements. However, since these penalties are typically incurred at the hottest times of the year, the value of the lost energy may well be higher than the yearly average value assumed for the fan power. Tables 14-16 present the evaluated cost of the penalties for a range of energy values, with the other parameters (discount rate, escalation rate, tax rate, plant life) held the same.

Table 5-14
Lost Energy from Heat Rate Penalty

Case	Lost Output, MWh	Value @ $50/MWh	Value @ $60/MWh	Value @ $100/MWh	Value @ $250/MWh	Value @ $500/MWh
1	4,718	1,620,416	1,952,308	3,240,832	8,102,080	16,204,160
2	10,636	3,652,977	4,401,177	7,305,954	18,264,885	36,529,770
3	12,576	4,319,278	5,203,949	8,638,556	21,596,390	43,192,780
4	19,400	6,663,008	8,027,720	13,326,016	33,315,040	66,630,080
5	35,718	12,267,490	14,780,108	24,534,980	61,337,450	122,674,900

Table 5-15
Evaluated Cost of Lost Output from Capacity Reduction Penalty—Steam Side Only

Case	Lost Output, MWh	Value @ $50/MWh	Value @ $60/MWh	Value @ $100/MWh	Value @ $250/MWh	Value @ $500/MWh
1	0	0	0	0	0	0
2	0	0	0	0	0	0
3	0	0	0	0	0	0
4	1,632	560,604	675,427	1,121,208	2,803,020	5,606,040
5	34,884	11,981,065	14,435,018	23,962,130	59,905,325	119,810,650

Table 5-16
Evaluated Cost of Lost Output from Capacity Reduction Penalty—Whole Plant

Case	Lost Output, MWh	Value @ $50/MWh	Value @ $60/MWh	Value @ $100/MWh	Value @ $250/MWh	Value @ $500/MWh
1	0	0	0	0	0	0
2	0	0	0	0	0	0
3	0	0	0	0	0	0
4	3,205	1,100,747	1,326,201	2,201,494	5,503,735	11,007,470
5	87,768	30,150,328	36,325,696	60,300,656	150,751,640	301,503,280

Figures 5-26 and 5-27 display the sum of the total evaluated cost from Figure 5-21 and the penalty costs from Tables 5-15 and 5-16.

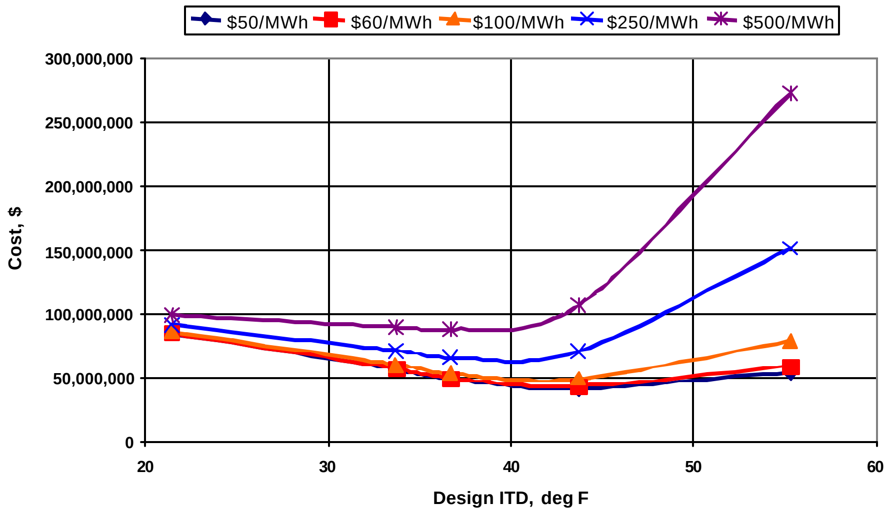

Figure 5-26
Total Evaluated Cost and Penalties—Steam Side Only

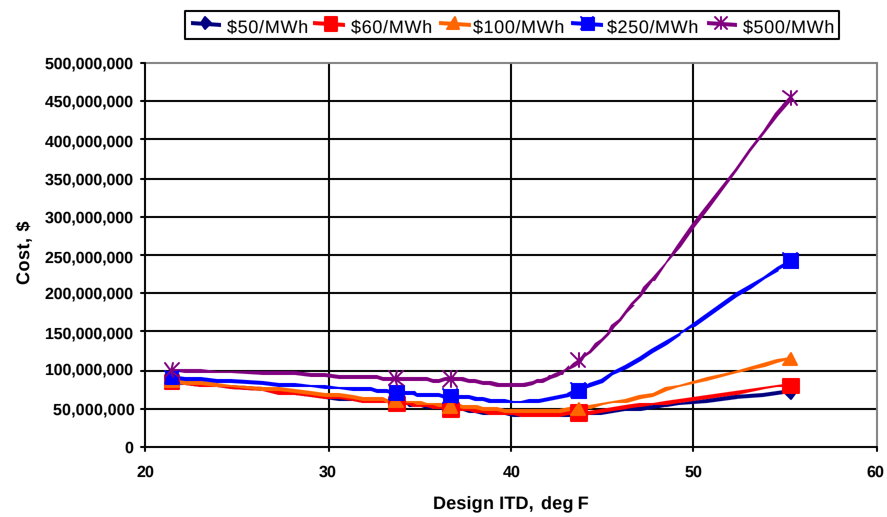

Figure 5-27
Total Evaluated Cost and Penalties—Whole Plant

Extension of Penalty Evaluation Procedure to Other Sites

The results of applying this penalty evaluation procedure to other sites differs only because of differing site meteorology. Figure 5-28 displays the temperature duration curves for each of the sites. Several features are noteworthy:

- The Mountain and Valley sites have temperature duration curves that are very similar in spite of their widely different locations and environment.
- While the general shape of the Mountain and Valley curves are also similar to that of the Desert curve, they differ from the Desert profile importantly at the high temperature periods. Both exhibit a sharp drop in the number of hours above a certain temperature at about 80°F. As a result, the hours during which they incur large heat rate or capacity penalties are substantially less than at the Desert site (less than half as many hours above 90°F, for example). As a result, the cost curves corresponding to those displayed for the Desert site in Figures 5-26 and 5-27 would have similar shapes at the low ITD end, and the location of the minima would be nearly the same. However, they would not exhibit the sharp upturn at the high ITD end since the penalties associated with the choice of smaller ACCs would be far less. Therefore, the selection of a unit with an ITD in the range of 40ºF would be appropriate for all three sites, but the consequences of choosing a lower first cost unit would be less at the Mountain or Valley sites than at the Desert site.
- The Bay Area site exhibits a very different climate. Over 8000 hours are below 70°F. This essentially eliminates the influence of the heat rate and capacity penalties in the selection of the optimum unit and suggests that the choice of a lowest first cost ACC would be appropriate.

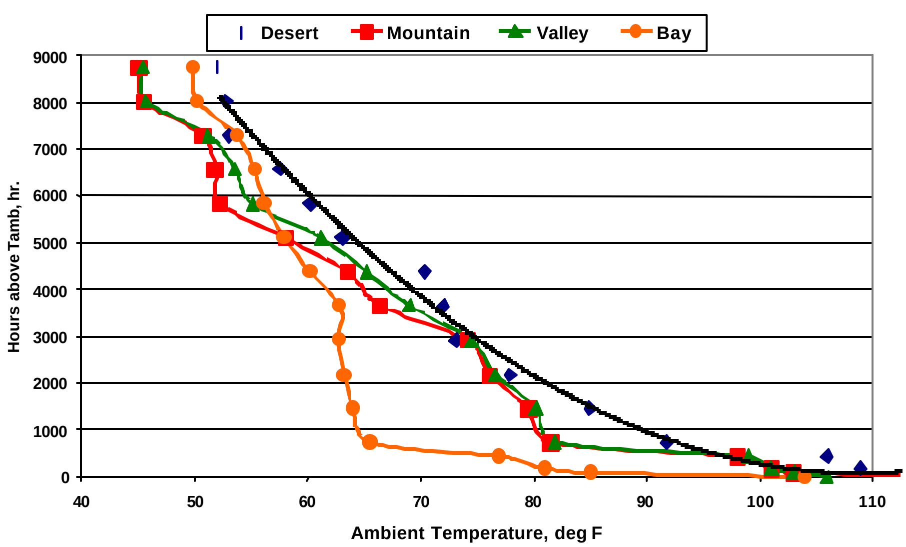

Figure 5-28
Site Temperature Profiles

The cost breakdown for a chosen ACC at the Valley site is tabulated in Table 5-17 (system designed for 2.5 in. Hga at average seasonal temperature and humidity, $T_{avg.\ dry\ bulb}$ = 67°F; $T_{avg.\ wet\ bulb}$ = 6 °F). The capital cost, fan power, and total evaluated cost for each of the four sites are shown in Table 5-18.

Table 5-17
Capital Cost Breakdown for Dry Cooling System Equipment at California Central Valley Location—Optimized Design (for New 500-MWe Facilities with 170-MWe Steam Cycle)

Element	Cost
Air-cooled condenser	17,200,000
Installation/erection	5,000,000
Steam duct support; column foundations	150,000
Electricals and controls	1,240,000
Cleaning system for finned tube surfaces	150,000
Auxiliary cooling (@ 7.5%)	1,780,000
Total	25,520,000

Table 5-18
Site-to-Site Cost Estimates—Air-Cooled Condenser ((for New 500-MWe Facilities with 170-MWe Steam Cycle)

	Desert Site ITD = 37	**Mountain Site** ITD = 44	**Valley Site** ITD = 44	**Bay Area Site** ITD = 55
Capital Cost	30,300,000	25,500,000	25,500,000	20,400,000
Fan BHP (in hp)	5300	4770	4570	3560
Total Evaluated Cost	44,700,000	38,400,000	37,900,000	30,000,000

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6
ENVIRONMENTAL IMPACTS

This section reviews and compares the environmental impacts of wet, dry, and hybrid cooling systems on new gas-fired, combined-cycle power plants. While the primary focus of this report is on the issue of consumptive water use, there are other environmental impacts from each of the different types of cooling systems. To bound the scope of the discussion, the material in "Power Plant Cooling Systems: Requirements for Approval" found in the California Code of Regulations, Title 20, Division 2, Section 5—§2012, Appendix B, is used as a guide (CalCodeRegs). The topics to be covered are listed in Table 6-1.

Regulatory Matters

Discussions of the environmental impacts are often inseparable from discussions of the rules and regulations that control them. It must be assumed that any cooling system selected will be designed and operated in such a way as to conform to all relevant environmental regulations. A comprehensive economic comparison, therefore, would include the costs of the environmental control systems required for compliance.

However, the regulatory requirements are complex. Power generation facilities, including their cooling systems, have been the object of legislative and regulatory attention at both the federal and state levels for decades. Furthermore, the regulatory philosophy has changed, particularly (but not only) in California. Historically, environmental regulations either invoked quantitative measurable limits on specific emitted chemicals or on the parameters of environmental releases, or alternatively, mandated the use of certain technologies, designated as "best available" or the like. Cost of compliance, while sometimes capped in the case of impacts considered less severe, was reasonably easy to anticipate.

Recent shifts in philosophy focus on the establishment of goals and objectives for particular regions that may vary with present condition and intended use. Effluent limits and technology requirements are then set on a case-by-case basis to achieve the local goals and objectives.

A detailed description of the regulatory framework and the statutory basis for environmental regulations is available in the report from a companion EPRI-CEC project on the use of degraded water for power plant cooling (DiFilippo 2001). The discussion here is limited to the nature of the impacts of the alternative cooling systems.

Table 6-1
Power Plant Cooling Systems: Requirements for Approval (from California Code of Regulations, Title 20, Div. 2, Chap. 5---§2012, App. B)

Subsection (of App. B)	Subject	Requirement	Relevant Code/Regulation
(g)(14)	Water Resources		Waste Discharge Req'ts; NPDES; Policy 75-58
(g)(12)	Waste Management		Cal. Code, title 22, §66261.20 *et seq.*
(g)(10)	Hazardous Materials Handling		Cal. Code, title 22, §66261.20 *et seq.* Also, Health and Safety Code, Section 25531.
(g)(8)(A)	Air Quality	Information necessary for air pollution control district to complete Determination of Compliance	None cited
(g)(4)	Noise		None cited
(g)(6)(F)	Visual Resources	Assessment of impact of visible plumes	None cited
(g)(9)	Pubic Health		Health and Safety Code, Section 25294.8
(g)(13)	Biological Resources		Cal. Code, title 20 Sects. 1702 (q) and (v)
(g)(15)	Agriculture and Soils	Effect of emissions on surrounding soil-vegetation	None cited

Types of Cooling System Impacts

Impacts of cooling systems include the effects of emissions to all of the environmental media (air, water, land) as well as societal effects including public health and public nuisance. Specific items to be addressed, following the guidance of Table 6-1, include

- Water resources,
- Waste management (including water discharge),

- Hazardous materials,
- Air emissions,
- Noise,
- Visual resources,
- Public health,
- Biological impacts, and
- Agriculture and soils.

Water Resources

As discussed in Sections 1 and 2, the primary consumptive use of water in gas-fired, combined-cycle power plants is in evaporative cooling towers for the condensation of turbine exhaust steam. Additional minor uses include the following:

- Steam cycle make-up: This can vary considerably with the operating profile of the plant. Co-generation plants, for example, generate electricity and provide process steam to industrial partners. In some instances, condensate is not returned to the plant but rather disposed of in the industrial process or, if returned, is no longer usable due to contamination.
- Auxiliary cooling load: In addition to turbine steam condensation, there are cooling loads in a power plant (e.g., oil cooling, generator hydrogen coolers) that typically amount to about 5% of the turbine condenser heat load. If this cooling is also provided with wet evaporative cooling, the evaporation losses are increased correspondingly.
- Hotel load: This refers to plant service water, such as cleaning water, sanitary water, drinking water, and air-conditioning condenser loads.

In some cases, modest amounts of water are used for plant performance enhancement:

- Gas turbines: Some combined-cycle units use evaporative or spray cooling on the inlet air of the gas turbine units to maintain output on hot days. This consumption rate can range from 0.6 to 1.2 gpm/MW of gas turbine capacity during the periods when it is being used (Molis 1997).
- ACC: While not commonly used, similar systems for cooling of the inlet air to the air-cooled condenser can maintain steam cycle capacity on hot days. The usage rates on the steam side would be higher than on the gas turbine side since the air flow per MW is higher, and might range from 2 to 6 gpm/MW of steam turbine capacity.

Dry Cooling

In comparison to recirculating wet cooling, a dry cooling system reduces the water consumption by 90 to 95% by

- Eliminating the evaporative cooling tower for the turbine steam condensing load, and

- Sometimes eliminating the auxiliary cooling load (in some systems, the auxiliary cooling is accomplished by adding cells to the ACC; in others, a separate fin-fan or evaporative cooler is used, in which case water consumption is still required).

The hotel load remains essentially the same. Water for performance enhancement is unchanged by the choice of cooling system. If inlet air cooling is used on the ACC, the water consumption rate can be 25% of that for a full wet system during the period that the enhancement is required. On an annual basis, depending on the duration of the hot periods, it is typically a small fraction (1 to 5%) of the annual consumption of a wet cooling system (Kroeger, 1998).

Hybrid System

A hybrid system reduces the water requirements for steam condensation in proportion to the fraction of the load carried by the dry part of the system. If the hybrid system is used primarily for plume abatement, it operates as a wet system except for cold periods when plumes would form. During those times when the dry system is in use, it would typically carry 25% of the load with a corresponding 25% reduction in water use. However, on an annual basis, since the dry system operates only intermittently, the water consumption would be nearly the same—95 to 99% that of a conventional wet cooling system.

Alternatively, a hybrid system intended for water conservation operates dry during the colder seasons, using water only when required to maintain plant efficiency and capacity during the hot periods when the dry system's limitations become severe. The water savings relative to an all-wet system can vary widely depending on the economic criteria used in system selection and design. The unit at the 500-MW San Juan plant was designed to carry approximately 75% of the heat load on the wet portion at the highest-temperature conditions (Kroeger, 2000). On an annual basis, design studies estimate water savings ranging from 30 to 98% (Mitchell, 1989).

Waste Management (Including Water Discharge)

Wet cooling systems require the discharge of cooling tower blowdown. The SWRCB has adopted a State Implementation Policy for implementing the receiving water standards in EPA's California Toxics Rule. Under the section on intake water credits, a facility that takes water from an impaired water body may discharge back to that water body if the concentration of the pollutants has not been increased. This offers relief to plants using once-through cooling; however, for plants that use cooling towers, blowdown treatment is required. This may also require consideration of the disposal of solid waste, such as basin sludge or water treatment system sludges from evaporation ponds, brine concentrators, side-stream softeners, or other blowdown reduction processes.

A detailed discussion of the effects, regulations, and technological options for treatment or reduction of the impact in the case of wet cooling systems is given by (DiFilippo 2001) Obviously, dry systems eliminate blowdown discharge and waste disposal related to blowdown treatment. Hybrid systems reduce these effects in proportion to the fraction of the condensing load carried by the dry part of the system. The water and wastewater treatment and discharge facilities are included as line items in the capital cost of cooling systems but are not major cost elements (see Section 5).

Hazardous Materials

The handling of hazardous materials is a worker safety issue, normally regulated under OSHA guidelines. For the operation of a wet cooling system, the most relevant concerns are over water treatment chemicals and waste streams. Chlorine and bromine compounds for biological fouling control are used and stored on site in large quantities. Scaling and fouling control operations involve specialty chemicals of a wide range of composition, as well as acids and bases (such as sulfuric acid, sodium hydroxide, hydrated lime, etc.) for pH control. In comparison to many chemical and petrochemical plant operations, the chemicals used at power plants are not severe hazards and can be dealt with through routine hazardous operations (HAZOP) procedures, appropriate worker training, and reasonable plant design and maintenance.

Hybrid towers require the same operations, materials, and precautions as do wet towers in proportionally reduced scales. Dry systems effectively eliminate major water treatment problems. However, this is not an issue of sufficient concern to affect the decision between cooling system types.

Air Emissions

Air-borne emissions from cooling towers are primarily associated with the drift and volatile compounds stripped from the water by the air flow. Drift consists of the small droplets entrained by the air passing through the tower as it flows past falling films or droplets of water. The smallest droplets are carried out of the tower with the air. Drift eliminators keep these losses to very low levels, typically less than 0.005% of the circulating water flow rate. For a tower on a 500-MWe combined-cycle plant, this corresponds to less than 5 gpm or approximately one-half the flow from a hand-held garden hose. However, even this small amount of discharge to the atmosphere invokes air quality control regulations.

Federal Regulations

Airborne emissions from cooling towers are regulated under the Clean Air Act, specifically the provisions of the National Emissions Standards for Hazardous Air Pollutants (NESHAPS). Listed pollutants under NESHAPS with relevance to wet cooling towers include asbestos (in the case of older towers using cement-asbestos [CAB] fill), chromium, zinc and zinc oxide, and the trihalomethanes. Title 40, Section 1, Part 63 (NESHAPS for Source Categories) Subpart Q (NESHAPS for Industrial Process Cooling Towers{IPCT}), Section 63.402 states:

> "No owner…shall use chromium-based water treatment chemicals in any affected IPCT."

Under Part 749, Water Treatment Chemicals; Subpart D, Air-Conditioning and Cooling Systems; §749.68, Hexavalent Chromium-based Water Treatment Chemicals in Cooling Systems, chromium-based compounds are prohibited in commerce for “comfort cooling towers” and for new units of IPCTs (but not for existing IPCTs).

State Regulations

Title 17, Public Health, Division 3, Air Resources, Section 1, Air Resources Board, SubSection 7.5, Airborne Toxic Control Measures; §93103, Regulation for Chromate Treated Cooling Towers bans the use of hexavalent chromium containing compounds in cooling tower circulating water. For existing towers, especially wood towers, that have used such compounds in the past, a period of time is permitted to allow the chemicals to desorb from the tower and be eliminated—so long as the level in the circulating water does not go above 0.15 mg/L (8 mg/L for wood towers), and tests show a continuous decrease over time.

While the drift amounts are relatively small, the droplets contain all of the impurities contained in the circulating water. Furthermore, as a result of droplet evaporation in the tower, plume, and atmosphere, impurities are often found at significantly higher concentrations in drift. When the drift droplets evaporate completely, the contaminants remain in the air as fine particulate matter (PM10) and constitute a source of PM10 emissions from these gas-fired, combined-cycle plants.

The magnitude of PM10 emissions from cooling towers is a subject of debate and discussion at this time. Total drift rates are historically difficult to measure accurately. Modern high-performance drift eliminators are specified to control drift to 0.002% of the circulating water flow or less. For a 500-MW combined-cycle plant with ~170 MW generation from the steam portion and an assumed circulating water flow of 500 gpm/MW, this corresponds to a total drift rate of 1.7 gpm or about 20,000 lb/day. At a circulating water solids concentration of 10,000 mg/L, this corresponds to a potential drift mass of solid particulate of 200 lb/day. Although there is considerable uncertainty in these estimates, a recent study (DiFilippo, 2001) reports a similar result of 300 lb/day for a comparably sized plant.

By comparison, estimates developed by the California Air Resources Board and presented at the CEC/EPRI Workshop (EPRI 2000) of PM10 emissions from state-of-the-at gas turbines are 0.0001 lb/kWh, amounting to about 800 lb/day from the combustion turbine side of a 500-MW combined-cycle plant.

In the event that the circulating water contains bacterial or pathogenic species, drift is a potential transport pathway from the plant to the surrounding areas. The usual example in this area is *Legionella.* References include (CTI 2000). This is discussed in somewhat greater length later in this Section under “Public Health.”

Other air emissions include volatile species that are formed in the cooling tower as a result of water treatment processes such as chlorination and then stripped from the water by the cooling air. Trace amounts of volatile organic compounds (VOCs) and particularly trihalomethanes (THMs) have been detected. A complete discussion of this area is found in DiFilippo (2001).

Second-Order Effects

The primary air emissions from combined-cycle plants are, of course, from the combustion of the gas fuel for the combustion turbines. As noted in earlier sections, the choice of cooling system can affect the overall plant heat rate and capacity. Therefore, to meet a given total system load, more fuel must be burned if dry cooling is used—with a corresponding increase in emissions of NO_x, particulate matter, SO_2, and CO_2 in amounts and proportions that depend on where and in what equipment the additional fuel is used.

A detailed treatment of this issue is beyond the scope of this report. However, on a system-wide basis, one can estimate that if 10% of system generation were to be equipped with dry cooling and each of those units incurred an annual average 10% performance penalty, neither of which is likely, the effect on overall system heat rate would be, to first order, only about 1%—with a correspondingly small impact on state-wide air emissions. On the other hand, a case-by-case analysis of these emissions would be needed to determine what the local environmental impact of each cooling option would be.

Noise

The effect of noise on residences, hospitals, libraries, schools, places of worship, and other neighboring places must be considered both during construction and operation. While no regulatory limits were found, the "Information Requirements" (CalCodeRegs) specifically define areas where a "potential of 5db (A) or more over existing background levels" might occur as affected areas. In addition, noise levels within the project site boundary and their impact on workers at the site must be considered.

The primary noise from cooling facilities is fan noise and "fill" noise caused by the flow of water down over the tower fill. Dry cooling would eliminate the "fill" noise perhaps at the expense of somewhat more fan noise since the quantity of air flowing through a dry tower is greater than that through a wet tower of the same capacity.

In the case of the Crockett Co-Generation Plant (see Appendix C), the plant is located on the outskirts of the town of Crockett in a mixed commercial/residential area with private homes located across the street to the south. On the northern boundary are the Carquinez Straits of the San Francisco Bay, a recreational and commercial boating area. Site-specific noise limits of 50 dB at the nearest residence and at a distance of 300 ft into the Bay were imposed and met with no difficulty through the use of Alpina low-noise fans.

In cases where ultralow-noise fans are required, the cost can be substantial (see Section 5). Recent discussions with one vendor suggest a 10% premium on the base cost of the ACC. In another study, noise control was estimated to account for an $8 million cost differential on a 750-MW combined-cycle plant, amounting to nearly 20% of the ACC capital cost (Micheletti 2001;UWAG 2000).

Visual Resources

An assessment of the visual impact of a project is required, including a description of the dimensions, color, and material of each major visible component. Specific reference is made to light, glare, and visible plumes.

Of the several cooling systems, once-through cooling is the least visually intrusive, but is a likely system of choice only for coastal locations. Cooling towers are among the larger components at gas-fired, combined-cycle plants.

Large hyperbolic natural draft towers, such as were used at the Rancho Seco Nuclear Plant, are the largest structures, but these are also unlikely choices for the cooling system sizes needed for the steam portion of a typical 500-MWe combined-cycle plant. Mechanical draft wet towers and dry towers are more generally comparable in overall configuration. Dry towers are larger, typically with a ground area or footprint twice that of a wet tower of the same capacity. They are also typically taller because of the large ground clearance needed to deliver air to the fans. However, in some cases, the absence of a large holding basin, which requires that wet towers be built on grade, can allow creative location of an air-cooled condenser. This was the case at Crockett, where the ACC is placed on top of the turbine building to accommodate a tight site and to blend into the site less obtrusively.

On cold days, wet towers can produce a large visible plume as the warm saturated air leaving the tower mixes with the cold ambient air and the water vapor condenses. In some locations, these plumes may obscure visibility, creating dangerous conditions on roadways, or may lead to local icing on neighboring roads or structures.

Obviously, a dry tower never produces a visible plume. Hybrid wet/dry towers are often designed specifically for plume abatement. In these designs, dry warm air from the dry portion of the tower, when mixed with the saturated air from the wet portion, results in an unsaturated mixture that will not lead to condensation of water vapor when mixed with ambient air.

Public Health

The most frequently cited public health issue in the context of cooling towers is the possibility of Legionnaire’s Disease, so-called because of an outbreak at an American Legion convention in Philadelphia in 1976, attributed to pathogens (*Legionella pneumophilia*) in the cooling tower for the HVAC system in the hotel. While the frequency of occurrence of Legionnaire’s Disease is small (approximately 1400 cases reported to the Center for Disease Control annually) and the number of these attributable to cooling towers (at power plants or anywhere else) is even fewer, the question has been investigated extensively in the U.S. and abroad. An extensive discussion of the transmission pathways, monitoring methods, and control procedures is given by DiFilippo (2001) in the context of the use of degraded or recycled water in wet cooling towers. More complete treatments are found in the CTI and ASHRAE literature and references therein.

While the consequences of exposure can be very severe and even fatal—particularly to at-risk populations (elderly, smokers, individuals with chronic respiratory problems or with suppressed immune systems)—the evidence of harm is sparse and largely anecdotal. Cooling towers are a common element of our industrial, commercial, and residential scenes in high-density population areas in all climates. No compelling epidemiology has established a significant threat.

Reasonable monitoring and control procedures should be followed, but it is unlikely that the choice of cooling system would be significantly influenced by public health concerns.

It should be noted that Legionnaire's Disease should also be viewed in the context of worker safety and health. This has been addressed in OSHA guidelines, and the topic is reviewed by DiFilippo (2001).

Biological Impacts

The biological impacts usually considered in relationship to cooling systems are entrainment and impingement losses. These result from aquatic life—from fish to microorganisms—being drawn into the cooling system (entrainment) or onto the intake screen (impingement) along with the cooling water. These effects, which have long been recognized and regulated under Section 316(b) of the Clean Water Act (EPA 2000), were usually considered in connection with once-through cooling systems. In once-through cooling, intake requirements of 500 gpm/MW can result in high withdrawals from the source water body even for modest-size plants.

The use of wet recirculating systems reduces the withdrawal rates to perhaps 2.5 to 3% of those in once-through systems. In recirculating evaporative systems, however, the entrained organisms have no chance of survival, and the water is not returned to the source water body. In the case of the smallest water bodies with low in-stream flows, it may be possible that even this reduced amount could have an important biological impact.

The use of dry systems will result in another 20-fold reduction in withdrawal rates. However, in comparison to once-through cooling, the further reduction over what wet recirculating systems achieve is small—from 2.5 to 3% down to perhaps 0.1%.

The EPA has recently proposed (EPA 2000) that power plants requiring NPDES permits and operating cooling water intake structures (CWIS) that withdraw more than 2 million gpd must consider alternative technologies to determine the Best Available Technology for the site. To put this in perspective, 2 million gpd is the withdrawal required for 3 MW on once-through cooling or approximately 100 MW on wet cooling towers. Therefore, a 500-MW combined-cycle plant with one-third of its capacity (~170 MW) from the steam cycle that uses wet cooling towers would withdraw over 3 million gpd and would, therefore, be required to consider the Best Available Technology for water intake.

In the support document issued as part of the Notice of Proposed Rulemaking, EPA introduced dry cooling as one the "Alternative Regulatory Options" (EPA 2000). In summer 2000, New York State's Department of Environmental Conservation required the use of dry cooling at the Athens Generating Station (a 1080-MW gas-fired plant) as the preferred Best Available Technology for "minimizing adverse environmental impact" of a cooling water intake structure (Cahill, 2000).

The question has been posed as to whether dry cooling may also have a biological impact in the form of "atmospheric entrainment" losses in which insects, or even birds, may be drawn into the large air stream passing through an ACC. There is no research or information on this question currently available, but further investigation may be warranted in the future.

Agriculture and Soils

Deleterious impacts of power plant cooling systems on surrounding agriculture have not been an issue except in a few special circumstances. One notable study was conducted in the mid-1970s at the Potomac Electric Power Company's Chalk Point Station in Maryland. In that case, the towers were run on brackish make-up water with a circulating water salinity comparable to sea water (35,000 ppm TDS); the towers were hyperbolic natural draft towers with a plume exit plane elevation of about 400 feet; and the plant was located in a tobacco-growing region with a specialty crop of leaves intended for use as the outer wrappers of cigars. High-salinity droplet deposition on the leaves could create small, discolored spots, making a leaf unusable without in any way affecting the health of the plant or the quality of the soil. Even under these conditions, the risk was eventually determined to be negligibly small, and the plant and towers continued to operate with no special controls and no adverse impact on the region's agricultural activity.

Summary

On balance, the environmental effects of dry and wet/dry systems are reduced in comparison to wet systems, but case-by-case analyses should be conducted to identify important site-specific effects for any system chosen.

The most important environmental effects of wet systems arise from water consumption, water and waste discharge, intake losses (entrainment or impingement), drift, and visible plumes. The use of dry cooling essentially eliminates all of these effects; wet/dry systems can be operated to eliminate the visible plume and to reduce the other effects by an amount proportional to the reduction in water consumption relative to all-wet systems.

In some other cases, the effects of dry cooling are greater than for wet systems. Specifically, dry cooling towers are physically larger and hence create a correspondingly greater visual impact than wet towers; the reduced plant efficiency which occurs during certain hotter periods of the year from the use of dry cooling results in higher fuel consumption and correspondingly higher air emissions from the combustion processes in the plant; and the noise from air-cooled systems is comparable to or greater than that from wet towers unless special provisions for low-noise fans are made. In addition, dry systems entrain more air than wet systems, creating the possibility of an increased "atmospheric entrainment" impact.

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7
CURRENT RESEARCH AND DEVELOPMENT

As noted in Section 3, in spite of renewed interest in dry cooling as a technology, there has been relatively little published in the research literature in recent years. It can be assumed that development work of the product improvement variety has been, and is, under way at major vendor facilities but is not yet published in the open literature.

The limited amount of published work falls in four categories:

- Heat exchanger design (including fan development for noise reduction)
- Performance enhancement
- System design
- Analytical methods

Some of the work cited in these categories is old by research definitions of "current" R&D but is included here either because it has yet to be exploited and may still be relevant or because it was the basis for current and presumably still evolving designs.

A summary of "the latest worldwide technology in environmentally designed cooling towers," including some attention to dry and wet/dry technologies, was presented a few years ago (Mirsky *et al*., 1992).

Heat Exchanger Design

The primary objective of heat exchanger design work on the finned tube coils for air-cooled condensers has been to reduce the air-side pressure drop while maintaining heat transfer effectiveness. The major advance in this area has been through modification of the tube geometry from circular to elliptical to flattened shapes that permit the use of a single tube row. This eliminates the losses associated with flow transitions from row to row encountered in either in-line or staggered arrangements of round tubes. This approach, actively promoted by Hamon (Bonger and R. Chandron, 1995), was summarized in a 1999 article in *Modern Power Systems* (Staff report, 1998) stating the following advantages:

- Freeze-proof operation (due to uniform steam flow distribution)
- Increased power generation during winter operation (without risk of freezing)
- Lower fan power consumption
- Increased thermal efficiency

Detailed performance calculations for these exchangers using single row, flattened tubes are available in Sections 5 and 8 of Kroeger's text (Kroeger, 1998).

It is noted in the *Modern Power Systems* article (Staff report, 1998) that the use of this design in natural draft units is promising. While this is of little interest to the California situation, it may expand advances in the design more widely.

Work also continues on the development of ultralow-noise fans. Some developments have been reported in EPRI workshops in St. Petersburg, FL (van der Spek and P. J. M. Nelissen, 1995), and in Jackson, WY (van der Spek, 2000), at which 15-db reductions in a retrofitted mechanical draft *wet* cooling tower were reported.

Performance Enhancement

It has long been recognized that the major performance and cost issue with dry cooling has been the limitation on performance during the hottest days of the year, where the use of dry cooling can limit plant output and decrease plant efficiency. While it is possible to mitigate this effect by selecting a larger air-cooled unit to begin with, this comes with a considerable increase in the initial cost (see Section 5). Furthermore, the achievable condensing temperature is still limited by the ambient dry bulb no matter what the size. At a design ITD of 20°F (probably an upper practical limit on unit size) in areas where the 1% dry bulb can approach or exceed 120°F, an ITD of 20°F would result in a steam condensing temperature above 140°F and a back pressure in excess of 6 in. Hga. Therefore, the researched enhancement schemes have focused on the use of a limited amount of water during the hottest hours. There are a variety of approaches available:

- Wet helper towers
- Swamp coolers in ACC inlet areas
- Deluge cooling of finned tube surfaces
- Spray injection in several configurations

Analyses performed about 10 years ago (Conradie and D. G. Kröger, 1991) illustrated that substantial performance enhancements could be achieved with a limited use of water, as shown in Figure 7-1 excerpted from Kroeger's text (Kroeger, 1998).

Recently, a spray enhancement project involving laboratory and single-cell testing has been carried out (Maulbetsch and M. N. DiFilippo, 2001). Some full-scale applications exist on small plants such as Chinese Camp (see Appendix C) and Mammoth Lakes Geothermal Plant. System analyses have been conducted recently at the National Renewable Energy Laboratory (NREL) on four approaches to hot-day performance enhancement for a small dry-cooled geothermal plant (C. Kutscher, NREL, personal communication). The four approaches were spray cooling, deluge cooling, wet packing (Munters fill) at the inlet, and a hybrid system combining spray cooling with wet inlet packing. The results suggesting slightly better "rates of return for the spray and the hybrid system" are tentative and unpublished at this time.

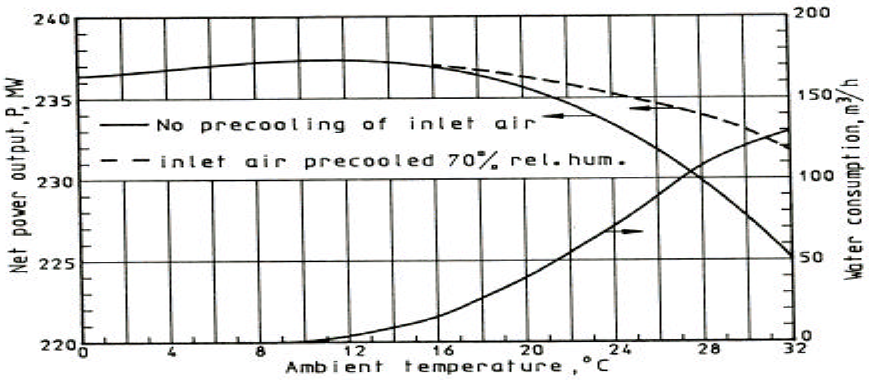

Figure 7-1
Effect of Pre-Cooling of Inlet Air (source: Kroeger 1998)

Deluge cooling has been studied previously and dismissed in a number of publications. The most detailed work, performed as part of the USDOE/EPRI Advanced Concept Test of an ammonia dry cooling system, was reported in Allemann *et al*. (1987) and is described in a little more detail below.

Studies have also been made of a thermal storage, peak-shaving concept in which advantage was taken of the diurnal cycle to provide excess cooling at night to accumulate a reservoir of cold water for use during the hot daytime hours on the following day. This concept has not been pursued commercially at this time.

System Design

A variety of system design configurations have been proposed, studied, and, in some cases, implemented over the past 10 or 20 years.

Direct-Contact Condensers

This design was originally proposed by Heller-Forgo of Hungary and continues to be developed by their successor organization, EGI (Balogh and Z. Takacs, 1998). Often used in conjunction with natural draft towers, it is discussed in Kroeger's text (Kroeger, 1998). No recent advances have been reported.

Ammonia Dry Cooling

This concept was studied extensively by USDOE and EPRI in the 1980s. The system is an indirect type, in which the usual circulating water loop is replaced by a phase-change ammonia loop where the ammonia is evaporated in the tubes of the steam condenser and condensed in an air-cooled condenser. The advantage comes from the elimination of the temperature difference between the ambient dry bulb and the condensing steam temperature normally imposed by the condenser cooling water's temperature rise (range) in conventional indirect systems. The concept was tested and well documented (Allemann, 1986; Allemann, 1981b; Allemann, 1981a;

Allemann and others, 1987), with the participation of several major equipment vendors (Baltimore Air Coil, The Trane Company, Curtiss-Wright, CB&I, and Union Carbide). To date, the concept has not been pursued commercially but may bear reexamination in light of increasing interest in economical dry cooling.

Evaporative Condensers

The use of an evaporative condenser that can be run in a dry mode during colder periods has been used in a number of industrial HVAC and process cooling applications and in at least one power plant (see MassPower, Appendix C). It is discussed at length in Section 8.

NWD

A single reference (Miura and O. Gotoh, 1998) is made to what is described as a "novel wet/dry tower" for plume abatement with no separate finned tube dry section. It works by interrupting the water flow to a section of the fill during cold, plume-forming periods. In that regard, it has some similarities to the evaporative condenser (see above). There is no information given on cost or performance.

Analytical Methods

There is some activity on computational procedures for system optimization and for determining wind effects. This includes work on the following:

- System optimization on a nonlinear constrained problem using a sequential quadratic programming method (Conradie *et al*., 1998)
- Optimization by geometric programming (Ecker and R. D. Wiebking, 1978)
- Computational fluid dynamics (Eldredge, 1995)
- Generalized optimization techniques (Kintner-Meyer and A. F. Emery, 1994)
- Computer-aided optimization (Li and W. Sadiq, 1985)

Progress in these areas will accrue to the benefit of future cooling system designs.

Summary

There is little current research and development work being reported in the open literature on dry or wet/dry cooling systems for power plants. A few important exceptions include improved heat exchanger geometries for finned tube bundles in air-cooled condensers (Bonger, and R. Chandron 1995; Staff report, 1998; Kroeger, 1998); enhancement of air-cooled condenser performance with the use of limited water (Maulbetsch and M. N. DiFilippo, 2001; Balogh and Z. Takacs, 1998); the use of evaporative condensers (Hutton, 1999; Niagara Blower Company); and optimization techniques (Conradie and others, 1998)

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8
EMERGING TECHNOLOGIES

The literature search revealed only one technology that qualifies as an “emerging” technology, i.e., one in an advanced stage of development with the potential to be seriously considered as an alternative water-conserving technology for power plant cooling in the near future, say, within the next 5 years. This technology, based on the evaporative condenser concept, is described in the following section. The cost and performance information is presented in the same format that was used in Section 5 for the wet and dry cooling systems.

Hybrid Cooling Systems: Evaporative Condenser

An alternative cooling technology, widely used in the chemical process and HVAC industries and recently considered for power plant applications, is the evaporative condenser or wet surface air cooler (WSAC). Detailed descriptions of the technology are found in the CTI literature (Hutton, 1999) and on vendor websites and in brochures (Niagara Blower Company). Fundamental treatments of the thermodynamics and heat transfer mechanisms in the process are found in standard texts and handbooks (see references in Hutton, 1999).

A unit by Resorcon (Niagara Blower, Inc.) has been operating at MassPower, a 240-MWe plant in Springfield, MA, since 1993. A description of their operating experience, based on a telephone interview with the plant superintendent, is included in Section 3.

A brief description of the technology follows for convenience of reference. Figure 8-1 provides a schematic view of the equipment. The process fluid, in this case condensing steam, flows through horizontal tubes arranged in bundles. Air flow is induced downward across the tube bundles, turned 180° through mist eliminators and the fan, and then discharged vertically upward. Water is circulated from a collection basin and sprayed onto the top of the tube bundles. It then flows downward, co-current with the air, and returns to the basin. Heat is transferred from the condensing steam through the tube walls to the falling liquid film. It is then rejected to the air stream as both sensible and latent heat, as is the case in a conventional wet cooling tower.

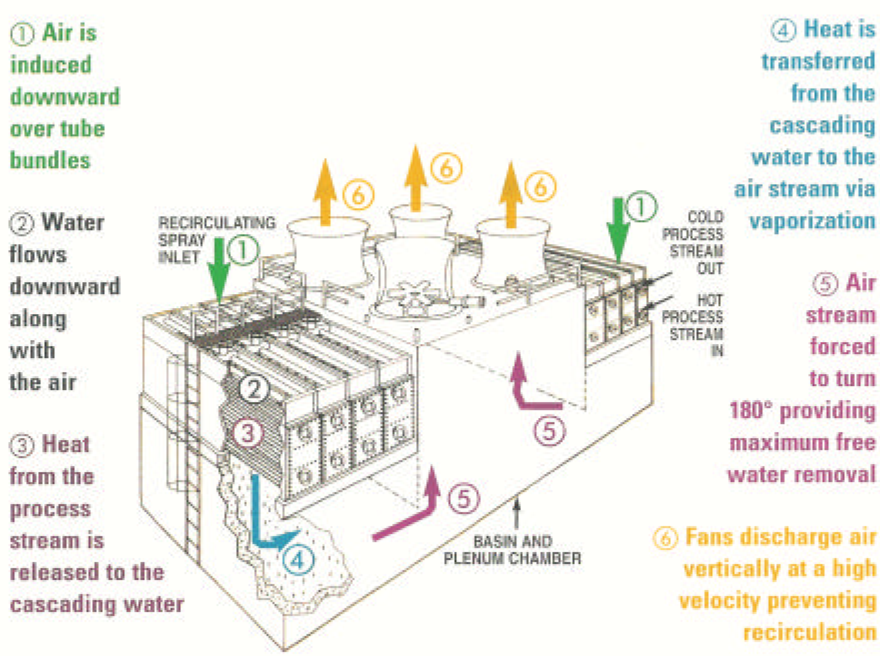

Figure 8-1
Schematic of Wet Surface Air Cooler (source: Niagara Blower Co. brochure)

Comparison to Wet Cooling Tower with Surface Condenser

The primary difference between evaporative condenser technology and a conventional wet cooling tower is that the heat transfer from the condensing steam to the air is accomplished in a single unit rather than in a two-step process. As a result, the temperature difference between the condensation temperature and the atmospheric wet bulb temperature (equal to the sum of the condenser range, the condenser terminal temperature difference, and the cooling tower approach in the conventional case) can be reduced.

In the case of a conventional wet cooling tower, the water enters the tower at an elevated temperature and is cooled by selected range as it falls across the fill. In the case of the evaporative condenser, the water remains at a nearly constant mixed temperature as it is recirculated from the basin and flows across the tube bundles, acquiring heat from the condensing steam at the same time as it rejects heat to the atmosphere by evaporation.

Comparison with Dry Cooling

The primary difference between evaporative condenser technology and dry cooling is that, as in the case of wet cooling, the ultimate heat rejection is through latent heat transfer, which is limited by approach to atmospheric wet bulb temperature, rather than sensible heat transfer limited by the approach to atmospheric dry bulb temperature, which is substantially higher than wet bulb at nearly all times of the year in most locations.

Water Use Considerations

The equipment is normally designed to operate in an “all wet” mode, where water conservation is not a limiting factor and full advantage can be taken of evaporation of the recirculating water to provide maximum cooling.

If water conservation is an objective, the unit can be designed for perhaps 50% water consumption on an annual basis. Full water use would be used during the hotter hours, but water flow could be reduced or turned off to some of the tube bundles during colder periods. Water conservation below 50% consumption is not feasible for this technology since the tubes are not finned, and the performance of a dry bundle is limited. Therefore, extended use in the dry mode would require an uneconomically large unit.

Capital Cost Elements

The system chosen to represent the evaporative condenser technology is the WSAC of the type used at MassPower. A water-conserving version of this technology, known as WSDAC (wet surface/dry air cooler), is also available for water-constrained applications.

The major elements of the capital cost are displayed in Table 8-1.

Table 8-1
Capital Cost Elements for Evaporative Condenser Equipment (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

Element	Comment	Cost
Base unit—evaporative condenser	Fabricated unit including tube bundles, headers, fans, sprays, mist eliminator, piping, and connections	Consistently represents 2/3 (65–70%) of the total installed cost; for full wet operation: ~$70/cell; for wet/dry operation: ~$85/cell
Recirculating spray pumps	Including drive motors	Significant cost element: 4– 6% of total installed cost; lower end of range for wet/dry systems
Concrete (plenum and basin)	Including typical site preparation and installation	Significant cost element: 14–16% (plenum = 5–6%; basin = 8–10%); high end of range for all-wet systems
Steam duct	Includes support structure; varies with site layout	Significant cost element: typically 7–8% of total installed cost
Condensate collection and return	Circulating water system (pumps, piping, valves, etc.)	Typically 2–3% of total installed cost
Electricals and controls	Wiring and hook-up; base price includes instrumentation	Typically 1–2% of total installed cost
Walkways and ladders	—	About 1.5% of total installed cost
Steam ejector	—	Minor; approximately $60k
Installation	Requires crane, anchoring to foundation, piping connections	Typically 1.5–2% of total installed cost
Auxiliary cooling	Separate unit or additional cell on condenser	Typically 5% additional heat load and ~5–6% additional cost; see Table 8-4
Other elements	Typically minor; may be significant at specific sites	Not included in case study estimates or comparisons
- Water supply/intake structure	Location dependent; significant only if water source is far from site or at a much lower elevation	
- Water treatment/blowdown discharge	Usually minor; may be significant if in zero-discharge region	
- Site preparation/access provision	Highly site dependent; likely minor; not likely to be affected significantly by system choice	
- Painting, fire and lightning protection, acceptance testing	Minor costs	
- Delivery and unloading	Entirely site and location dependent; typically minor	

Case Study Results

The effects of ambient meteorological condition and site elevation were addressed through the development of cost estimates for the four selected case study sites. In order to evaluate the tradeoff between initial capital cost and the costs of operating power requirements and capacity/heat rate penalties, two design points were chosen.

For the unconstrained water availability case and the constrained case (50% of unconstrained water use), the units were sized to maintain 2.5 in. Hga back pressure at both average meteorological conditions ($T_{avg.\ dry\ bulb}$ = 67º F; $T_{avg.\ wet\ bulb}$=60ºF) and 1% conditions (the ambient temperature and wet bulb exceeded for only 1% of the time, i.e., 80 hours/year). In dry-mode operation, a turbine back pressure of 2.5 in. Hga cannot be maintained for either the average or the 1% dry bulb temperatures. Therefore, for dry mode operation, the design points were set as 7.0 in. Hga for average temperature and 10.0 in. Hga for 1% temperature.

Tables 8-2 to 8-4 provide capital cost breakdowns by cost element for selected case study examples. Table 8-2 presents cost requirements for the evaporative condenser system with unconstrained water availability for the Central Valley site.

Table 8-2
Capital Cost Breakdown for a 10^6 lb/hr Steam Condenser, Central Valley Site—Unconstrained Water Availability (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

Element	Cost ($1000s)
Base unit	2845
Recirculating spray pumps	266
Concrete works - plenum - basin	 287 439
Steam duct	336
Walkways, ladders	75
Condensate collection and return	150
Electrical and controls	75
Steam ejector	60
Installation	73
Auxiliary cooling @ ~ 5–6% (see Table 8-4 for example breakdown)	257
Total installed cost	**4862**

Table 8-3 provides the same information for a system constrained to 50% of the unconstrained water use on an annual basis.

Table 8-3
Capital Cost Breakdown for a 10^6 lb/hr Steam Condenser, Central Valley Site—Constrained to 50% Water Availability (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

Element	Cost ($1000s)
Base unit	6460
Recirculating spray pumps	477
Concrete works - plenum - basin	 515 795
Steam duct	742
Walkways, ladders	145
Condensate collection and return	150
Electrical and controls	100
Steam ejector	60
Installation	155
Auxiliary cooling @ ~ 5–6% *of unconstrained water case* (see Table 8-4 for example breakdown)	257
Total installed cost	**9855**

Table 8-4 presents the same information for auxiliary cooling with an evaporative water cooler designed to cool 3000 gpm from 125°F to 95°F at average ambient wet bulb temperature (corresponds to ~ 45 x 10^6 Btu/hr or ~ 5-6% of the plant condensing heat load).

Table 8-4
Cost Breakdown for 45 x 10^6 Btu/hour Auxiliary Cooler, Central Valley Site (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

Element	Cost ($1000s)
Base unit	139
Recirculating spray pumps	30
Concrete works - plenum - basin	 34 26
Walkways, ladders	7
Fluid recirculation system	8
Electrical and controls	5
Installation	8
Total installed cost	**257**

Table 8-5 provides capital costs and operating power requirements for the evaporative condenser with unconstrained water availability for the four case study sites.

Table 8-5
Site-to-Site Cost Estimates ($1000s)—Evaporative Condenser, Unlimited Water Availability (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

	Desert Site	Mountain Site	Valley Site	Bay Area Site
Average Meteorology **Design back pressure: 2.5 in. Hga**				
Total Cost	4725	4341	4380	4270
Total BHP (in hp)	2037	1972	2230	1930
- fans (in hp)	1491	1414	1656	1372
- pumps (in hp)	546	558	574	558
1% Meteorology **Design back pressure: 2.5 in. Hga**				
Total Cost	6250	5291	5540	4908
Total BHP (in hp)	3087	2348	2539	2279
- fans (in hp)	2279	1722	1822	1642
- pumps (in hp)	808	626	717	637

Table 8-6 provides capital costs and operating power requirements for the evaporative condenser with constrained water availability for the four case study sites.

Table 8-6
Site-to-Site Cost Estimates ($1000s)—Evaporative Condenser, 50% Water Consumption (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

	Desert Site	Mountain Site	Valley Site	Bay Area Site
Average Meteorology **Wet mode: 2.5 in. Hga** **Dry mode: 7.0 in. Hga**				
Total Cost	9528	9005	9345	8739
Total BHP-wet (in hp)	2002	1726	1935	1756
- fans (in hp)	650	540	617	570
- pumps (in hp)	1352	1246	1318	1186
Total BHP-dry (in hp)	2600	2160	2470	2280
1% Meteorology **Wet mode: 2.5 in. Hga** **Dry mode: 10 in. Hga**				
Total Cost	13,216	11,754	12,485	10,332
Total BHP-wet (in hp)	2772	2464	2610	2156
- fans (in hp)	900	800	850	700
- pumps (in hp)	1872	1664	1768	1456
Total BHP-dry (in hp)	3600	3200	3400	2900

Development of Correlation for Evaporative Condenser Costs

Unconstrained Water Availability

In order to generalize the case study results to other site conditions and design choices, the data were put on a normalized basis of $/kWe and plotted against the difference between the steam condensing temperature (T_{cond}) and the ambient wet bulb temperature (T_{wb}), as is typical for evaporative systems. The result for the unconstrained water case is displayed in Figure 8-2.

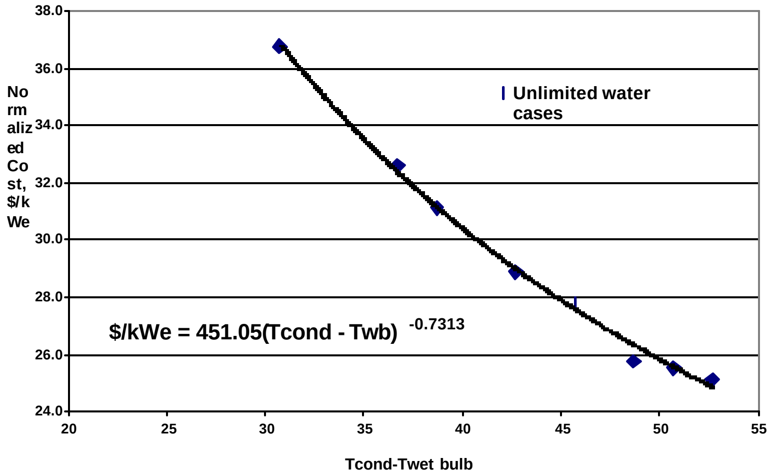

Figure 8-2
Evaporative Condenser Cost vs. Approach (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

A reasonably smooth curve is obtained for temperature differences ranging from 30 to 55°F. The data underlying the correlation are all for a single plant size, condensing approximately 1,000,000 lb/hr of steam. A possible basis for scaling to systems of different size is discussed below under "Comparison with Other Cost Information."

Constrained Water Availability

The generalization of results for units designed for limited water availability is more difficult. Different design criteria were applied to maintain a given back pressure, i.e., 2.5 in. Hga in the wet mode and distinct values for dry operation at average and 1% meteorological conditions. Costs in $/kWe are plotted against two different temperature differences:

- $T_{cond} - T_{wet\ bulb}$ in Figure 8-3, and
- $T_{cond} - T_{dry\ bulb}$ in Figure 8-4.

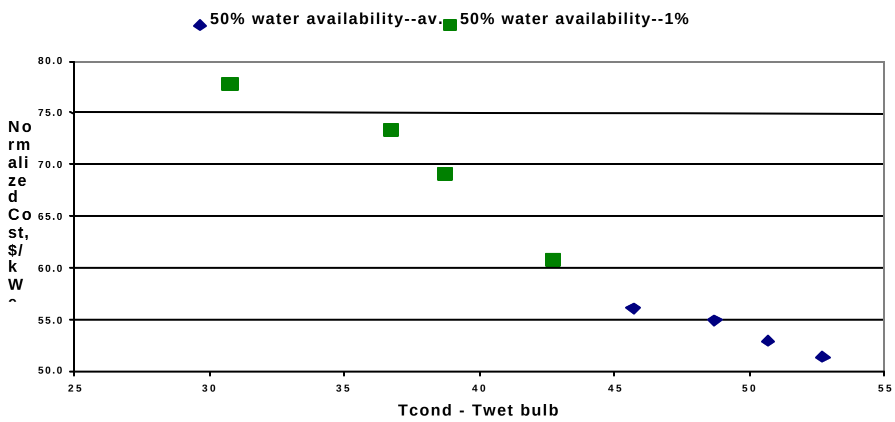

Figure 8-3
Evaporative Condenser Cost vs. Approach—50% Water Availability (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

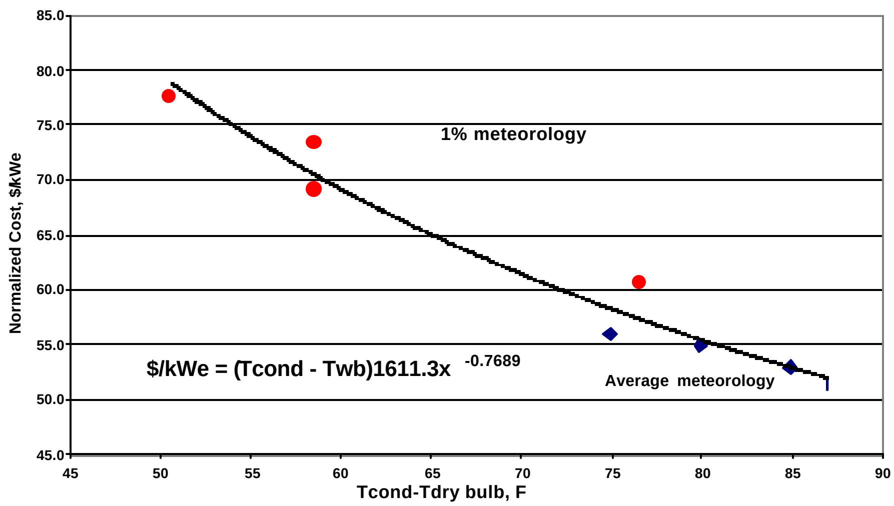

Figure 8-4
Wet/Dry Surface Air Cooler: Normalized Cost vs. Approach (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

The variation in cost cannot be smoothly correlated over the range of conditions for either case. The points aggregate in two groups corresponding to the average and 1% meteorological conditions. This may be in part due to the different design back pressures chosen for the two cases. In addition, since the units are assembled from a discrete number of cells and bundles, discontinuities in the cost may be expected as limits are encountered for different tube types and cell configurations.

An alternative correlation scheme is used based on the cost of the all-wet system as a function of $T_{cond} - T_{wet\ bulb}$ combined with the additional cost imposed by a 50% reduction in water availability. This cost differential in Δ$/kWe is plotted against $T_{cond} - T_{wet\ bulb}$ in Figure 8-5. While the discontinuities remain, a reasonable fit to these points is determined and then added to the wet system cost curve of Figure 8-2. The resultant smoothed cost of the water-limited case is shown in Figure 8-6 and compared to the individual case study costs. The individual differences range from –4.9 to +4.5% with an average absolute difference of 2.3%.

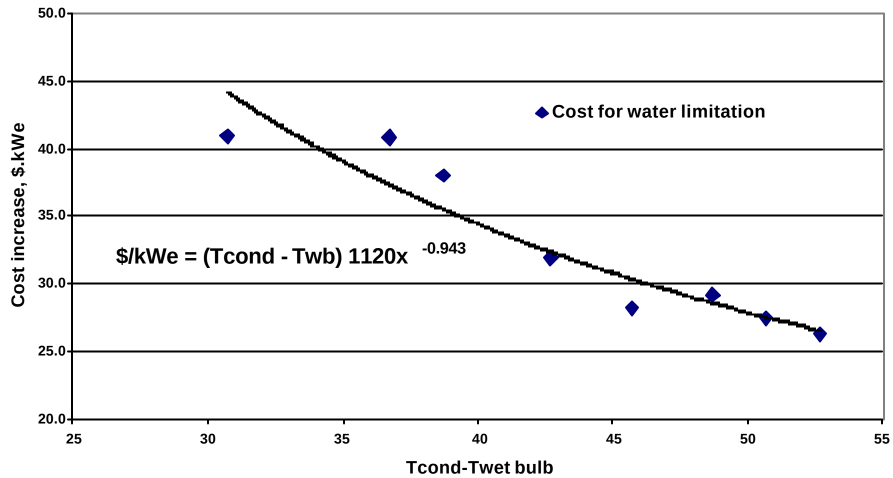

Figure 8-5
Evaporative Condenser—Increase in Cost for Water Conservation (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

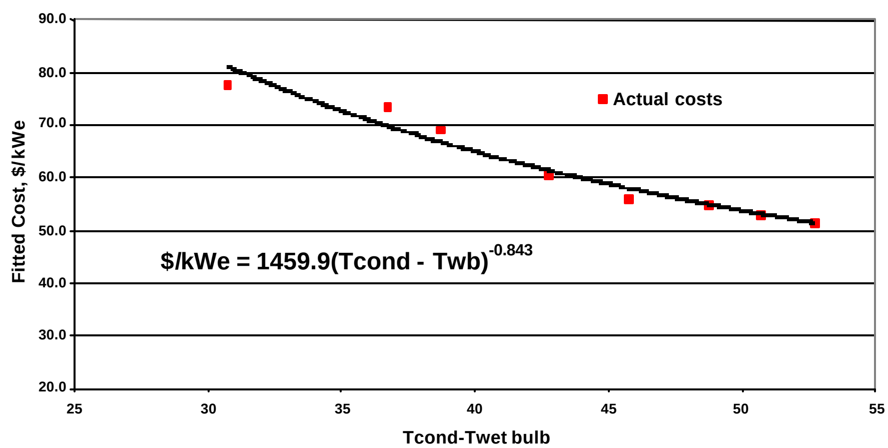

Figure 8-6
Evaporative Condenser Curve-Fitted Cost (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

Power Requirements

Figures 8-7 and 8-8 display the power requirements for both wet- and dry-mode operation, plotted against design approach to wet bulb in the former case and to dry bulb in the latter. Based on the same evaluated cost of power of $3625/kW as was used in the wet cooling tower analysis (see Section 5), the total evaluated costs for the unlimited water case are shown in Figure 8-9. The annual power use in the 50% case depends on the particular operating strategy throughout the year and is not addressed here.

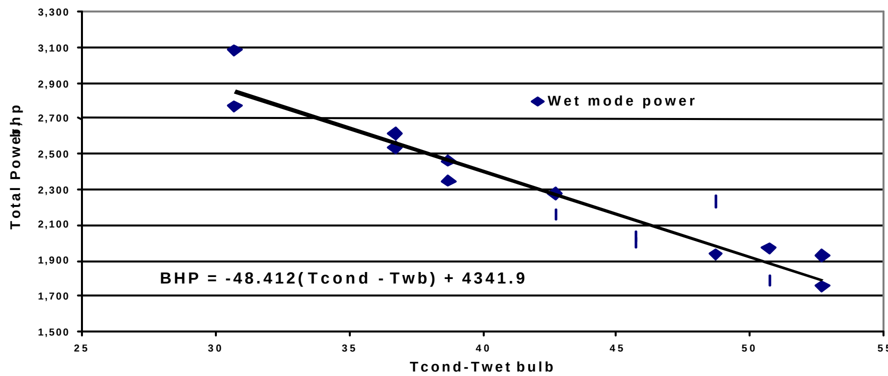

Figure 8-7
Evaporative Condenser—Fan and Pump Power vs. Approach (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

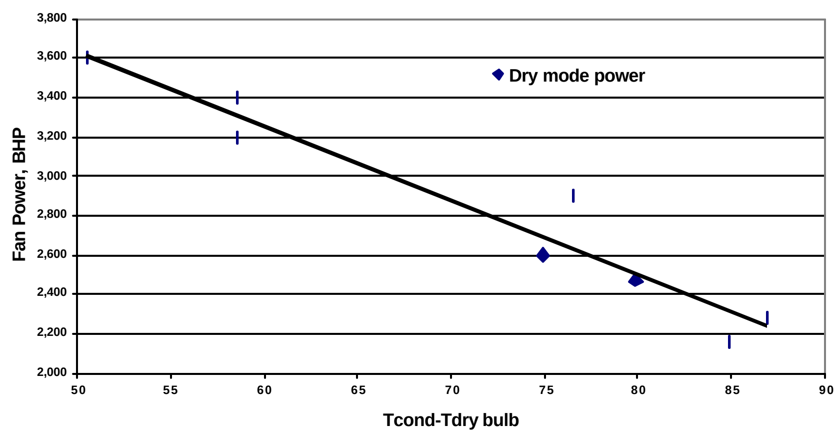

Figure 8-8
Evaporative Condenser Power Requirements in Dry-Mode Operation (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

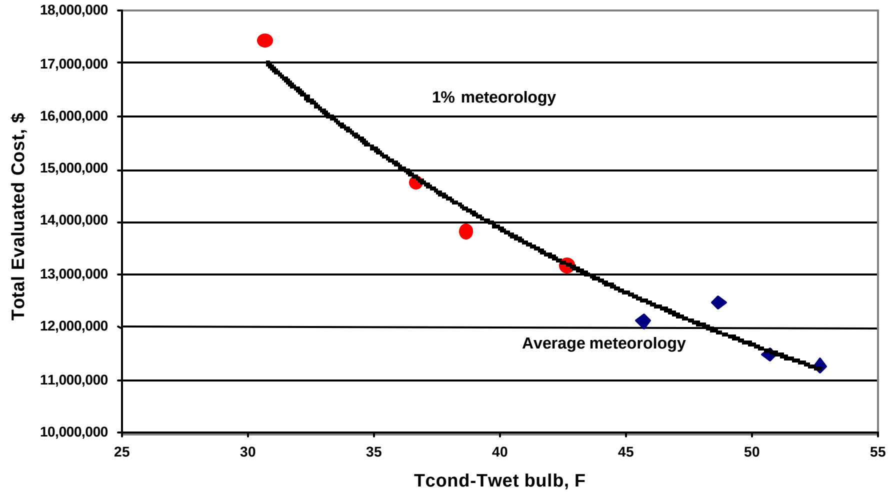

Figure 8-9
Evaporative Condenser Total Evaluated Cost vs. Approach (for New 500-MWe Combined-Cycle Plants with 170-MWe Steam Cycle)

Comparison with Other Cost Information

An independent check of the calculated evaporative condenser costs can be performed by a comparison with costs presented by Hutton (1999). These costs were developed for three condensing loads (100,000, 250,000, and 400,000 lb steam/hr) for two condensing pressures (3 and 4 in. Hga) at an ambient wet bulb temperature of 78º F.

Figure 8-10 and 8-11 display curves constructed from these data in $/kWe vs. steam condensation load at two values of $T_{cond} – T_{wet\ bulb.}$ (37ºF and 47 ºF). Corresponding points for the case study results at 1,000,000 lb steam/hr were read from Figures 8-2 and 8-7 and plotted on Figures 8-10 and 8-11. They are seen to fall on a reasonable extrapolation of the Hutton (1999) results.

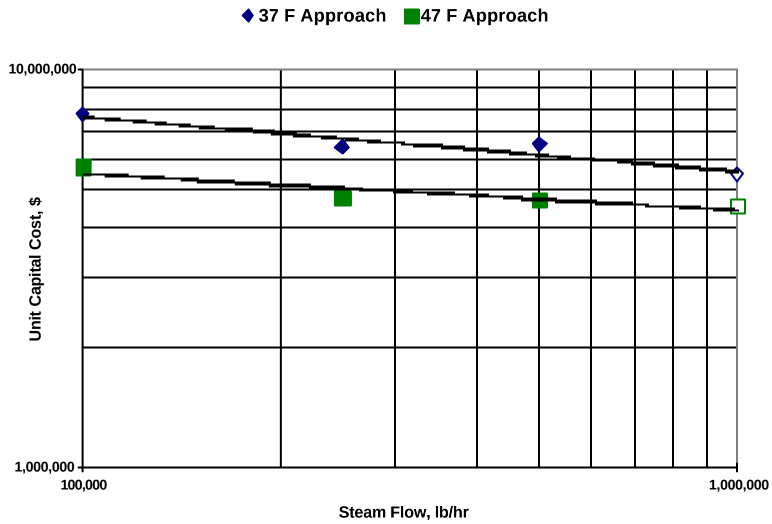

Figure 8-10
Evaporative Condenser Cost Comparison

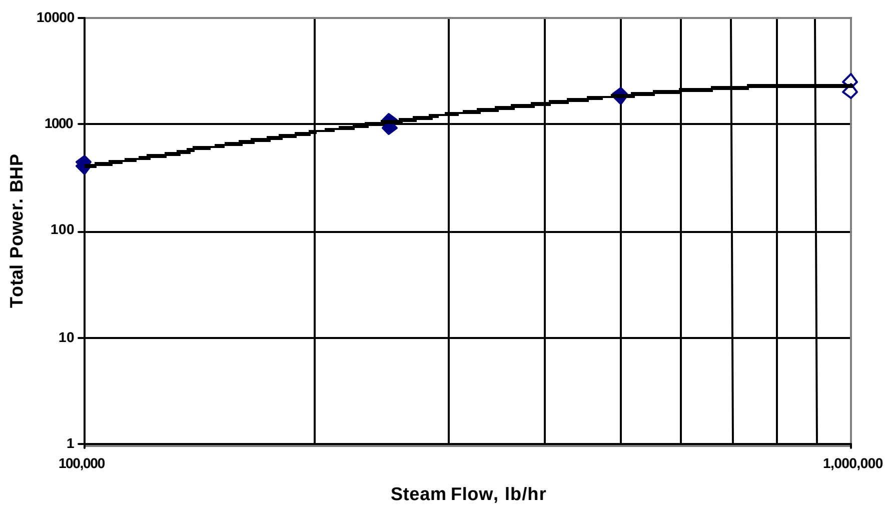

Figure 8-11
Evaporative Condenser Power Use Comparison

Summary

The evaporative condenser provides a cooling system alternative with capital cost and auxiliary power requirements about halfway between the wet and dry systems evaluated in Section 5.

Evaporative condenser designs can be developed both for conditions of unlimited water availability and for conditions where some water use constraints exist.

The system is not an economical choice for conditions where water availability is severely limited. Maximum achievable water conservation is about one-half that of a wet cooling system.

For unlimited water availability, where the evaporative condenser would compete with wet cooling towers, the estimated costs for the four case study sites range from 1.5 to 2 times those for wet cooling tower surface condensers presented in Section 5. In addition, evaporative condensers have higher auxiliary power requirements.

Evaporative condenser designs provide some ancillary benefits, however, including the ability to operate in a plume abatement mode during colder periods, the ability to operate with lower-quality water without scaling or corrosion, and improved characteristics in freezing conditions. These qualities led to its choice in the MassPower application described in Section 3.

If water conservation (up to 50%) is desired, the cost of the unit approximately doubles and the power requirements increase by approximately 20 to 30%. The unit also incurs substantial efficiency and capacity penalties while operating in the dry mode. Economical designs, in these estimates, are sized for 7 in. Hga back pressure at average meteorological conditions and 10 in. Hga back pressure if sized for 1% meteorology (temperature level exceeded for only 80 hours per year). The costs and power requirements are well below those for an all-dry system, but only 50% of the annual water savings can be realized and, during periods of wet operation, the withdrawal rates are the same as for wet cooling systems.

References

1. Hutton, Dave. Improved Power Plant Performance with Evaporative Steam Condensing. 1999 Feb; CTI TP99-08.

2. Niagara Blower Company. Available at: http://www.niagarablower.com/wsac.htm.

9
SUMMARY AND CONCLUSIONS

This report documents the results of a study of alternative cooling systems for electric power generating plants. The people and businesses of California require increasing amounts of both energy and water. On occasion, the use of water for power plant cooling can conflict with the water needs of the residential, commercial, industrial, and agricultural sectors and with environmental requirements to maintain adequate in-stream flows. This has resulted in increasing interest in California and elsewhere in the use of dry or hybrid wet/dry cooling systems.

This study examined alternative cooling systems and compared them in several dimensions:

- Water consumption
- Plant performance
- Cost
- Operations and maintenance (O&M)
- Environmental effects

In addition, current R&D and technology development efforts were surveyed to assess the prospects for significant changes in the comparative attributes of the alternative systems in the near term.

Results of the study indicate that while dry and wet/dry systems undeniably reduce water requirements at power plants, they do so at some cost—both capital costs for the cooling system components themselves and associated "penalty" costs resulting from higher auxiliary energy requirements to operate the systems, reduced plant efficiency, occasionally limited plant output, and perhaps additional operating and maintenance requirements. Additionally, while the dry systems reduce or eliminate many of the environmental effects of wet cooling, some effects do exist, and others have been postulated.

The following paragraphs summarize the results and conclusions of this study in specific areas.

Water Consumption

Dry cooling reduces the amount of water used in a power plant by eliminating the consumption of water through evaporation in a wet cooling tower. In order to understand the significance of the reduction, it is necessary to distinguish between cooling system water consumption and total plant water consumption, which includes many uses that exist regardless of which cooling system is employed (see Table 1-1 and the accompanying text).

For plants equipped with wet recirculating cooling systems (wet towers) for steam condensation, the evaporation of water in the cooling tower is the largest single water use in the plant, accounting for approximately two-thirds of total water consumption at a gas-fired combined-cycle plant (and perhaps 95% at a stand-alone thermal steam plant). The following comparisons are based on combined-cycle plants, with two-thirds of plant output from the combustion turbines and one-third from the steam turbine.

For the cooling system alone (including condensation of steam turbine exhaust plus auxiliary cooling estimated at 5% of the condensing heat load), dry cooling affords a 95% reduction in water use from, typically, 250 gallons per MWh of plant output to perhaps 10 gallons/MWh.

For the entire plant, additional water use—primarily for emissions control and turbine inlet cooling where used—is estimated at approximately 100 gallons per MWh (plant output), representing the mid-range of values surveyed and reported by the California Energy Commission (CEC, 2001a). This results in plant water use of 350 gallons per MWh with wet cooling and 110 gallons per MWh with dry cooling, an overall reduction of approximately 70%.

Plant Performance

In essentially all situations, the use of water as the cooling medium is the cheapest way to provide cooling at power plants. Furthermore, wet cooling always results in higher annual plant output and in more efficient plant performance during most of the year. The relevant questions are *how much* more expensive and *how great* is the effect on plant output and efficiency. The performance issues fall in three categories.

Operating Power Requirements

The energy required to operate cooling system fans and pumps is energy that must be generated but that cannot be exported or sold by the plant. The cost of this reduced output is incurred for the life of the plant. It is normally expressed as the present worth of a kilowatt that could produce energy at a projected price per MWh for the assumed plant life. This value can be traded off against the capital cost of the cooling system (which can normally be reduced through the expenditure of higher amounts of fan and pump power).

For wet systems in this analysis, the present worth of the power was approximately equal to the capital cost of the cooling system for an optimized system and approximately 160% of the capital cost for a “low first cost” system.

For dry systems, the operating power for a given heat load is much greater—by a factor of 4 to 6—than that for an optimized wet system.

Efficiency Penalties

The ability of a cooling system, wet or dry, to reject heat to the environment is affected by ambient conditions—ambient wet bulb temperature for wet cooling towers and ambient dry bulb temperature for dry systems. As these temperatures rise, the turbine back pressure also rises, with a resultant reduction in turbine efficiency. Plant output can sometimes be maintained by increasing the steam flow to the turbine if the steam supply system has additional capacity. Alternatively, in the case of a fixed steam supply, the plant output will decrease.

Capacity Penalties

If ambient temperatures rise sufficiently high, the achievable back pressure may exceed the maximum allowable back pressure specified for the turbine. In this case, steam flow must be reduced to protect the turbine, and plant output will be limited at that level.

Cost

The performance and cost issues are inextricably related. The initial choice of a larger, higher capacity and, hence, more expensive cooling system will result in higher plant capacity and more efficient operation for the life of the plant. This is true for all cooling systems, wet as well as dry.

In general, a proper cost comparison among alternative cooling systems must be made between optimized selections at a particular site. An optimized system would normally be defined as one that minimizes the sum of all costs---initial capital cost, operating energy cost, efficiency penalty cost, and capacity penalty cost---for the life of the plant. The magnitudes of these cost elements are dependent on many variables. Some of the most important include the operating characteristics of the plant generating components, the meteorology at the plant site, the present and projected costs of fuel, the present and projected price of power, and the projected demand profiles for the plant. The choice of the optimum or preferred design also depends on the relative importance assigned to present vs. future costs, which depends strongly on the economic objectives and business plans of the plant owner. Therefore, the “optimum system” or the “system of choice” might range from a system of lowest first cost to one of lowest total lifetime cost projected for a 30-year or longer lifetime.

To illustrate the methodology for determining the various cost elements and for selecting an “optimum” system based on a range of criteria, case studies were conducted at four sites typical of conditions in California. In each case the cooling system was sized to condense the turbine exhaust steam from the steam portion of a nominal 500-MW gas-fired combined-cycle plant typical of what is currently being proposed and built in California.

Initial capital costs for wet and dry cooling systems that could provide suitable cooling for the four case study sites vary greatly, ranging from $2.7 to $4.1 million for wet systems and from $18 to $47 million for dry systems. Figure 9-1 displays the initial capital costs for wet and dry cooling systems of both “lowest first cost” and “optimized” designs for each of the four sites.

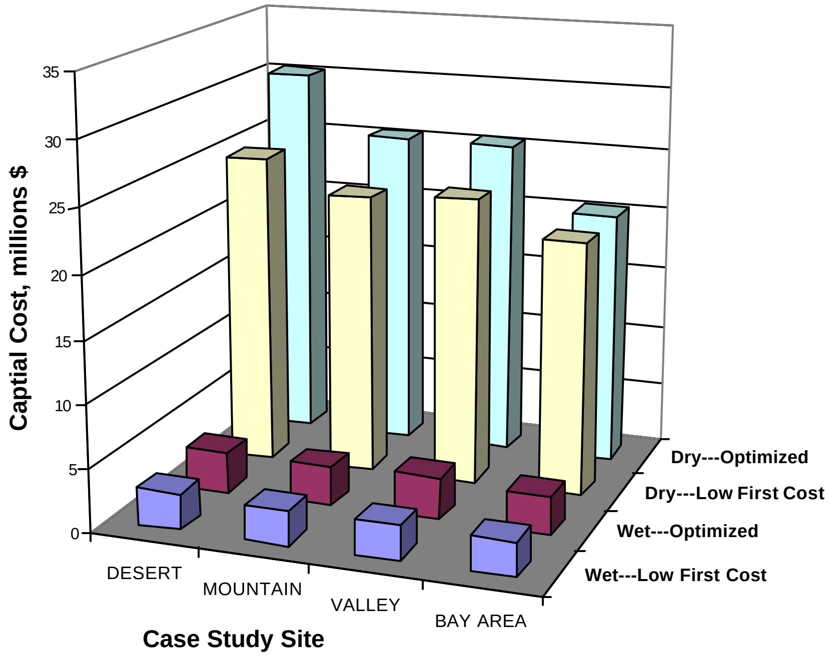

Figure 9-1
Case Study Cost Summary

Wet Systems

In the case of wet systems, the optimized system is defined by the tradeoff between capital cost and operating energy costs as determined in Section 5. The heat rate and capacity penalty costs incurred by the plant because of cooling system capability limitations on hot, humid days are not an important influence in the determination of the optimum for wet cooling in any of the cases studied.

Dry Systems

In the case of dry systems, the optimum is determined by the tradeoff between total evaluated cost of the cooling system (including both the capital cost and the operating energy cost) vs. the heat rate and capacity penalties incurred by the plant because of cooling system capability limitations during the hottest days of the year.

As discussed in detail throughout the report, conclusions about the relative cost of wet vs. dry cooling are difficult to generalize, and they depend on many site-specific considerations. Clearly, for the cases studied, the cost of dry cooling for either a “low first cost” design or an “optimized” design is substantially greater than the cost of a comparable wet system. This is most frequently the case. Therefore, as noted in Section 3, one must conclude that in those instances where dry cooling is selected—which has occurred with increasing frequency in the U.S and worldwide in recent years—the choice is driven by other considerations such as severe water use limitations at otherwise preferred sites, environmental pressures, and the avoidance of licensing delays.

Operations and Maintenance

Comparisons of O&M requirements were based not on detailed analyses or surveys but rather on information obtained during interviews and discussions with staff at operating plants equipped with dry or hybrid systems, all of whom had had previous experience at other plants using wet cooling towers. In each case, the water-conserving system was described as basically trouble free and easy to operate. No additional staff was required, nor was any unusual amount of staff time allocated to the cooling system beyond occasional scheduled cleaning of the finned tube surfaces—and this could be performed with the unit on line.

Therefore, no additional costs were assigned to dry or wet/dry systems for extra O&M requirements.

Environmental Effects

The comparison of environmental impacts associated with the alternative cooling systems examined nine areas, including the effect on water resources addressed above. The other eight areas are

- Waste management, including water discharge;
- Hazardous materials;
- Air emissions;
- Noise;
- Visual resources;
- Public health;
- Biological resources; and
- Agriculture and soils.

In many of these areas, the effects are proportional to the water use and the associated withdrawal, evaporation, blowdown, drift, and plume production. This is the case for impacts on waste management, water discharge, hazardous materials, public health, and agriculture and soils. In these cases, the use of dry cooling eliminates the effects entirely, and hybrid systems reduce them in proportion to the reduction in water use, which is a function of cooling system operating profile.

The other areas are discussed further below.

Air Emissions

Dry systems eliminate effects associated with drift and volatile components stripped from the water in wet towers. Hybrid systems do so in proportion to their water use.

However, the use of dry systems imposes penalties on plant efficiency and capacity, which requires that additional fuel be burned at the plant or elsewhere on the system to produce the same net power generation. This leads to an increase in the emissions associated with the combustion process. While this is a second-order effect on a state- or system-wide basis, it be a measurable effect in the vicinity of the plant and should be considered on a case-by-case basis.

Noise

The noise from wet systems comes from both the water falling through the tower fill and the fans and air motion. In the case of dry systems, the water noise is obviously eliminated, but the fan and air noise may be increased since the quantity of air moved through the system is greater.

The importance of this effect is site specific, but, in locations where noise abatement is necessary, the use of special low-noise fans may be required. For so-called "ultralow- noise fans" the increase in cost is estimated at approximately 10%. Similar noise abatement may be required for the wet towers in some cases.

Visual Resources

Dry and wet/dry systems effectively eliminate the occurrence of visual plumes. However, they require physically larger, taller structures than wet cooling and, as such, can be a more obvious element at a plant site. The importance of this feature is also site specific.

Biological Impacts

The primary focus in this area is on entrainment and impingement losses regulated under Section 316(b) of the Clean Water Act. Dry and wet/dry systems obviously reduce this impact in proportion to the reduction in total plant water use, estimated above as approximately 70%.

A postulated effect of dry cooling is so-called "atmospheric entrainment," where insects and even small birds may be entrained with the inlet air stream. To date, no research or information is available on the importance of this issue.

Current R&D

There is little current R&D work being reported in the open literature on dry or wet/dry cooling systems for power plants. A few important exceptions include improved heat exchanger geometries for finned tube bundles in air-cooled condensers (Bonger, 1995; Staff report, 1998; Kroeger, 1998); enhancement of air-cooled condenser performance with the use of limited water (Maulbetsch, 2001; Balogh, 1998); the use of evaporative condensers (Hutton, 1999; Niagara Blower Company); and optimization techniques (Conradie, 1998).

Emerging Technologies

The evaporative condenser provides a cooling system alternative with capital cost and auxiliary power requirements about halfway between the wet and dry systems evaluated in Section 5. Design choices are available both for conditions of unlimited water availability and for conditions where some water use constraints exist.

The system is not an economical choice for conditions where water availability is severely limited; maximum achievable water conservation is about one-half that of a wet cooling system. In water-conserving applications, the cost of the unit approximately doubles, and the power requirements increase by approximately 20 to 30%. The unit also incurs substantial efficiency and capacity penalties while operating in the dry mode. The costs and power requirements are well below those for an all-dry system but only 50% of the annual water savings can be realized and, during periods of wet operation, the withdrawal rates are the same as for wet cooling systems.

For cases of unlimited water availability, where the evaporative condenser would compete with wet cooling towers, the estimated cost for the four case study sites ranges from 1.5 to 2 times those for wet cooling tower surface condenser presented in Section 5. Evaporative condensers also have higher auxiliary power requirements. They provide some ancillary benefits, however, including the ability to operate in a plume abatement mode during colder periods, the ability to operate with lower-quality water without scaling or corrosion, and improved characteristics in freezing conditions. These qualities led to its choice in the MassPower application as described in Section 3.

References

California Energy Commission (CEC, 2001a), Environmental Performance Report of California's Electric Generation Facilities, P700-01-001, July, 2001

Bonger, R. and R. Chandron. New Developments in Air-cooled Steam CondensingPalo Alto, CA: EPRI; 1995: Paper 18.

Staff report. Single row condensers build on success. 1998 Jul: 43.

Kroeger, D. G. Air-cooled Heat Exchangers and Cooling Towers. New York: Begell House; 1998.

Maulbetsch, J. S. and M. N. DiFilippo. Spray Cooling Enhacement of Air-Cooled Condensers, Madadnia, J and H. Koosha. Proceedings of XIIth International Conference on Cooling Towers; Sydney, Australia. International Association of Hydraulic Research; 2001.

Balogh, Andras and Z. Takacs. Developing Indirect Dry Cooling Systems for Modern Power Plants [Web Page]. 1998. Available at: http://www.nemesis.at/publication/gpi_98_2/articles/33.html .

Hutton, David. Improved Power Plant Performance with Evaporative Steam Condensing. 1999 Feb; CTI TP99-08.

Niagara Blower Company. Available at: http://www.niagarablower.com/wsac.htm.

Conradie, A. E.; J. D. Buys, and D. G. Kröger. Performance Optimization of Dry-Cooling Systems for Power Plants through SQP Methods. Applied Thermal Engineering. 1998; 18(1-2):25-45.

Appendix A
ANNOTATED BIBLIOGRAPHY OF COOLING SYSTEM LITERATURE

Adams, A.P. and B. G. Lewis. Bacterial Aerosols Generated by Cooling Towers of Electrical Generating Plants. Cooling Tower Institute (1978).
Notes: ***One of the earliest studies of the possibility of infectious air-borne discharge from cooling towers using contaminated make-up water. A listing of pathogenic and opportunistic microbes existing in sewage is given along with some test results of aerosolized bacteria (particles per cubic meter) as a function of the number of bacteria in the circulating water. No firm conclusions were drawn other than to suggest the use of good disinfection methods in cooling tower water treatment with particular attention to chlorine dioxide. 16 references.***

Adams, S. and J. Stevens. Strategies for Improved Cooling Tower Economy. Cooling Tower Institute (1991).
Notes: ***Provides an analysis, for mechanical draft wet cooling towers of (1) how the choice of design point (maximum wet-bulb temperature at which cold water temperature can be delivered) and (2) effect of variable fan operation on annual tower energy consumption. The point is demonstrated with simple, largely qualitative examples with no recommendations about how to trade off lost capacity on hot days against capital cost savings from smaller towers. No references are given***.

Allemann, R. T. *et al.* Wet-Dry Cooling Demonstration: Test Results. 86. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Summary report on over 10 years work (EPRI and USDOE) on an Advanced Dry Cooling System using a phase-change ammonia transport loop between the plant condenser and an air-cooled tower with both supplementary deluge cooling and a capacative system for shaving heat rejection peaks during daily hot periods. Extensive performance data is provided as well as comparison with (calculated) conventional dry cooling system performance. No cost data are provided. Extensive references.***

Allemann, R. T. Advanced Concepts Test (ACT) Facility Summary Safety Report. 81. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Safety analysis and operational safety report for the EPRI/USDOE Advanced Concept Test facility. The emphasis was on the risks associated with the use and handling of ammonia as the heat transport fluid between the surface condenser and the air-cooled condenser. The analysis concluded that all applicable safety codes and standards for the operation of the system were met and that it should be reliable for a power generation application.***

Allemann, R. T. Development of an Advanced Concept of Dry/Wet Cooling of Power Generating Plants. 81. Palo Alto, CA, Electric Power Research Institute.

Notes: ***Description of the tests conducted at the EPRI/USDOE Advanced Concept Test facility. This report was one of a series of reports to document the ammonia phase change dry cooling system and to make the knowledge available to utility planners. This report presents the overview of the facility design and the various technical support studies including component testing and material selection. It also outlines the operational acceptance tests and operator training that was planned prior to initiating the test work. 30 references.***

Allemann, R. T., B. M. Johnson, and E. V. Werry. Wet-Dry Cooling Demonstration: A Transfer of Technology. 87. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Final report on the four year project to develop and demonstrate the ammonia phase-change dry cooling system at Pacific Gas and Electric Company's Kern station. Test results are presented on the performance of the system in three operating modes: two combining dry and evaporative cooling and one combining dry and capacitive cooling. The test program included both passive and active performance tests, system response to normal system transients and system recovery from unusual system conditions. 42 references.***

Air-Cooled Heat Exchangers: A Guide to Performance Evaluation. Unknown. American Institute of Chemical Engineers.
Notes: ***Report of the Air-Cooled heat Exchanger Subcommittee of the American Institute of Chemical Engineers. The document is intended to specify test methods for conducting and interpreting field performance tests of air-cooled heat exchangers with ambient air on the outside tube surface and process stream on the inside. It does not set performance specifications or discuss deviations from manufacturer's predicted performance. 26 references.***

Air Cooled Heat Exchangers. 91. American Society of Mechanical Engineers.

Armstrong, C. H. and R. S. Schermerhorn. Economics of Dry Cooling Towers Applied to Combined-Cycle Power Plants. 73. American Society of Mechanical Engineers.
Notes: ***One of the first treatments of the cost of dry cooling when used on combined cycle plants. A case study for an 85MWe unfired combined cycle plant was presented. The costs were shown to be highly dependent on the relationship of unit capability vs. ambient temperature. The possibility that the adoption of dry cooling would significantly accelerate the project schedule due to a reduction in the time required to complete the environmental review and to shorter construction times enabled by shop construction of the tower was noted as an important economic advantage. No references.***

Aschner, F.S. Planning Fundamentals of Thermal Power Plants. John Wiley & Sons, New York, NY (Unknown).
Notes: ***Review of design fundamentals for condensers on condensing power plants. This material is one part of a full textbook on the planning and design of steam power plants. This section covers both surface condensers and barometric condensers (of the Forgo-Heller type) and air cooled condensers. No cost information is provided. No references.***

Baker, D. Cooling Tower Performance. Chemical Publishing Company, New York, NY, (1984).
Notes: ***Textbook on cooling towers. (Only the Table of Contents has been looked at.)***

Coverage appears to be restricted to wet cooling towers. It is unclear whether cost information is included although there is a Section on Specifications and Bid Analysis. A complete treatment of the theory and calculational methods is provided.

Balogh, Andras and Z. Takacs. Developing Indirect Dry Cooling Systems for Modern Power Plants. 98.
Notes: ***Description of the Heller-type indirect dry cooling systems placed on the Web by EGI Contracting/Engineering Co. Ltd., a Hungarian company now a division of GEA, which builds and installs these systems. Reasonably detailed but qualitative information is provided on the design and operation of the system and comparison to conventional wet tower and air-cooled condensers. No cost information is provided. No references.***

Bartz, J. A. Dry cooling of power plants: a mature technology? Power Engineering . 88.
Notes: ***Review and brief description of some existing dry cooling tower installations in South Africa and other foreign countries and a qualitative projection of future U.S. requirements for the technology. Gives a short description of the ammonia phase change system which was being developed by EPRI and USDOE at that time and a short discussion of performance issues with conventional wet cooling. No references.***

Beck, A. and M. Schaal. Water Requirements of an Inland Nuclear Power Station: Engineering and Economic Aspects. Desalination **40**, 19-24 (1982).
Notes: ***Cost comparison of alternative cooling systems for a 950 MWe nuclear power plant in Israel. Costs were estimated for four cases: wet cooling with fresh water; wet cooling with sea water make-up; wet/dry cooling and dry cooling. It was concluded that it was more economical to desalt sea water for make-up to a wet tower than to use dry cooling. The most economical alternative was to design a wet tower with the capability of accepting sea water as make-up. 4 references.***

Bonger, R. and R. Chandron. New Developments in Air-cooled Steam Condensing. EPRI TR-104867, Paper 18. 95. Palo Alto, CA, EPRI.
Notes: ***Review of dry cooling systems and a more detailed treatment of natural draft air-cooled condensers and the single row condenser (SRC) using a elliptical tube which was just being introduced at that time. Economic comparison criteria were simplified but clearly stated and detailed designs for the compared cases were presented. Direct dry cooling systems were found to be substantially cheaper than indirect systems and natural draft systems cheaper than mechanical draft for the direct cases. The elliptical tube, single row condenser was shown to have significant performance advantages. No references.***

Bukowski, Joe. Taking the Heat off Industrial Processes. Consulting-Specifying Engineer , 46-50. 95.
Notes: ***An elementary, introductory article of heat rejection alternatives, primarily for industrial applications. Identifies a few rules-of-thumb and general guidelines for initial choices of preferred systems for particular applications and environments. No quantitative information for comparing systems.***

Burger, Robert. Cooling Towers, the Overlooked Energy Conservation Profit Center. EPRI TR-104867. 94.
Notes: ***Addresses the question of how much improved cooling system performance is worth. Three case studies, two for power generation plants, are presented. The cost savings associated with a given temperature reduction in the cold water return temperature are given. Methods for improving the performance of existing towers are given including a review of an advanced fill to replace conventional wood packing. 7 references***.

Burns, J. M. *et al*. The Impacts of Retrofitting Cooling Towers at a Large Power Station. EPRI TR-104867 . 94.
Notes: ***Presents a cost evaluation of retrofitting the PSE&G's Salem Station from a once-through cooling system to recirculating, wet cooling towers. A dry cooling alternative is considered briefly but rejected on qualitative conclusions about cost and lack of experience. Cost estimates are provided but with very little information about the source of the information. A good summary of the several cost categories that must be considered is given. 3 references.***

Buys, J.D. and D. G. Kroeger. Cost-Optimal Design of Dry Cooling Towers Through Mathematical Programming Techniques. ASME Trans. **111**, 322-327 (1989).
Notes: ***Presents a highly mathematical treatment of the cost optimization of a natural-draft dry-cooling tower. Casts the problem as a generalized, non-linear constrained mathematical minimization problem and uses published numerical methods to accomplish the solution. Worked example is instructive. The cost functions are introduced (without derivation) but an extensive reference list may provide the basis for the algorithms. Useful only if computational packages are to be developed. 17 references.***

Buys, J.D. and D. G. Kroeger. Dimensioning Heat Exchangers for Existing Dry Cooling Towers. Energy Conversion and Management **29**, 63-71 (1989).
Notes: ***Develops a method for determining the optimum design of a finned tube bundle to be installed in an existing tower for which heat exchanger retrofit is desired. Example is provided for a natural draft tower and a detailed numerical case is worked through. The objective function to be minimized is the cost at a given cooling load. Sensitivity studies to illustrate the effect of varying certain independent parameters are given. 13 references.***

Carroll, George, D. F. Dyer, and G. Maples. Comparison of Phased Cooling Systems with Conventional Cooling Systems---Performance and Economics. 77.
Notes: ***Investigated the effect of including cold water storage with conventional cooling system. Little relevance to the evaluation of wet vs. dry systems. No copy available.***

Choi, M. and L. R. Glicksman. Computer Optimization of Dry and Wet/Dry Cooling Tower Systems for Large Fossil and Nuclear Power Plants . 79. Cambridge, MA, Massachusetts Institute of Technology.
Notes: ***Program for optimization of dry cooling for power plants. The objective function to be minimized is the total evaluated cost of the cooling system. Cases were run for both fossil and nuclear plants. Lost capacity was replaced with gas turbines. The use of both conventional and high back pressure turbines was considered. The result was found to be most sensitive to***

the choice made for replacing lost capacity. The use of a separate wet tower to handle some of the cooling duty on the hottest days was also examined and found to be a big economic advantage even if only a small amount of water was available. 37 references.

Christopher, P.J. and V. T. Forster. Rugeley Dry Cooling Tower System . Proc. Instn. Mech. Engrs. **184, Pt. 1**, 197-221 (1969).
Notes: ***A detailed description of one of the first large scale dry cooling systems---a natural draft, barometric condenser (Heller system) at the Rugeley Power Station in the U.K. Full descriptions of the design, materials selection, and operating data are provided. Operation and maintenance experience is reviewed with particular attention to corrosion issues. An extensive discussion and authors' replies are attached expanding somewhat on the question of whether the tower was properly optimized. (It appears that the tower was perhaps oversized to economically optimized.) Very little cost information is included. 2 references.***

Conradie, A.E. and D. G. Kröger. Performance Evaluation of Dry-Cooling Systems for Power Plant Applications. Applied Thermal Engineering **16**, 219-232 (1996).
Notes: ***A performance model for dry cooling systems. A detailed model is formulated of the relevant mass, momentum and energy balance equations and engineering design relations for dry cooling systems in power plant applications. Worked examples for both natural and forced draft cases are given. The model gives both capital and operating costs over the life of the plant. It can be used to conduct sensitivity analyses to determine savings associated with small changes in many design choices. 31 references.***

Conradie, A.E., J. D. Buys, and D. G. Kröger. Performance Optimization of Dry-Cooling Systems for Power Plants through SQP Methods. Applied Thermal Engineering **18**, 25-45 (1998).
Notes: ***A dry cooling system optimization method (see also Ecker and Wiebking, 1978) formulated as a non-linear constrained problem using the Sequential Quadratic Programming (SQP) method. The objective function to be minimized is the ratio of total system cost to plant output. A detailed cost estimating methodology is included. Examples for both a natural and forced draft system are given. The emphasis is on the development and description of the methodology rather than the results so there is little quantitative discussion of costs. 36 references.***

Conradie, T. A. and D. G. Kröger. Enhanced Performance of a Dry-Cooled Power Plant through Air Precooling. ASME Paper No. 91-JPGC-Pwr-6 Presented at the International Power Generation Conference. 91.
Notes: ***Study of the use of spray enhancement cooling to reduce capacity loss at dry cooled plants on hot days. Thermodynamic calculations of the effect of adiabatic pre-cooling of tower inlet air by water injection into the air stream is presented. Po0lynomial curve fits are given to relate plant output to tower cold water return temperature. A worked example is given for a power pant and annual water consumption and annual power output is calculated for a range of ambient temperatures. No attention is given to the design of the spray enhancement system or to potential operating problems for the tower. It is assumed throughout that the finned tube surfaces remain dry. 5 references.***

Conradie, T. A. and D. G. Kröger. Enhanced Performance of a Dry-Cooled Power Plant through Air Precooling. 91. International Power Generation Conference, San Diego, CA.
Notes: ***Study of the use of spray enhancement cooling to reduce capacity loss at dry cooled plants on hot days. Thermodynamic calculations of the effect of adiabatic pre-cooling of tower inlet air by water injection into the air stream is presented. Po0lynomial curve fits are given to relate plant output to tower cold water return temperature. A worked example is given for a power pant and annual water consumption and annual power output is calculated for a range of ambient temperatures. No attention is given to the design of the spray enhancement system or to potential operating problems for the tower. It is assumed throughout that the finned tube surfaces remain dry. 5 references.***

Cooper, George P. and J. W. Cooper, Jr. Watts Bar Nuclear Unit 1 Cooling Tower Thermal Performance Upgrade: A Value ADD Engineering Approach" . EPRI TR-104867. 94.
Notes: ***Summary of a TVA upgrade of a natural draft wet cooling tower at the Watts Bar nuclear power plant. Modifications to the tower's fill system and the spray nozzles resulted an improvement in tower performance from 88% (12% shortfall) to 106% of design capability and an effective increase of 6 MWe in the capability of the plant. The cost of the upgrade of about $1.5 million as recovered in one year of operation. 4 references.***

Cuchens, J. W. and R. J. VanSickle. Crossflow Cooling Tower Performance Upgrade. EPRI TR-104867. 94.
Notes: ***Review of several upgrade possibilities to improve the performance of mechanical draft, cross-flow wet cooling towers. The several possibilities considered ranged from the installation of auxiliary towers, adding cells to the existing tower, refurbishing towers with new fill and converting from cross-flow to counter-flow. The benefit to cost ratio was found to be significantly better (~ x4.) for the auxiliary tower option than for any of the others (~ x1. To x2.). 2 references.***

Curcio, John *et al.* Advanced Dry Cooling Tower Concept. 75. Cambridge, MA, Energy Laboratory, Massachusetts Institute of Technology.
Notes: ***No information available.***

Ecker, J.G. and R. D. Wiebking. Optimal Design of a Dry-Type Natural-Draft Cooling Tower by Geometric Programming. Journal of Optimization Theory and Applications **26**, 305-323 (1978).
Notes: ***An optimization program for designing natural draft dry towers. The emphasis is on the mathematical formulation of a constrained optimizations problem. The objective function to be minimized is the total (operating plus fixed) annual cost with specified performance constraints. The design goes to the level of finned tube bank details (tube size and spacing, fin height, spacing and thickness). There are two worked examples. The cost numbers are chosen without derivation or justification but merely to enable an example calculation. 18 references.***

Eldredge, T. V. Cooling Tower Modeling with Computational Fluid Dynamics. EPRI TR-104867. 95. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Presents a computational fluid dynamics (CFD) approach to calculating cooling tower performance and how it is affected by meteorological conditions, tower geometry, water flow rate, flow maldistribution, fill plugging and other three dimensional effects. The results are***

compared to measurements of cold water temperature on two natural draft cooling towers. The effect on tower performance of injecting scrubbed flue gas into the tower chimney is also investigated. 15 references.

EPRI. Proceedings: Cooling Tower And Advanced Cooling Systems Conference. EPRI TR-104867. 95. Palo Alto, CA, Electric Power Research Institute.

EPRI. Proceedings: International Cooling-Tower and Spray Pond Symposium. EPRI GS-6976. 90. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Papers from the 1990 IAHR/EPRI meeting on cooling towers and spray ponds. The volume contains 60 papers, mostly on wet cooling systems but 7 on dry and hybrid cooling towers. (Summaries of important papers related to dry cooling are provided under the entries for the individual papers: Tesche; Bodas (2); Alt; Leitz; Bergmann and Csaba; Mozes)***

Dinelli, G. and B. Bellagamba. Proceedings: International Cooling-Tower Conference. EPRI GS-6317. 89. Palo Alto, CA, Electric Power Research Institute .
Notes: ***Contains 28 papers on several aspects of cooling tower technology: modeling of cooling system performance and design; operating experience; new cooling concepts; optimization criteria and methods; environmental impact; and summary of a round table discussion on optimization methods, design and acceptance testing and materials selection and maintenance. Primary emphasis of the meeting was on wet cooling. Eight papers on dry or wet/dry systems. (Summaries of the important papers related to dry systems given under the entries for the individual papers.) {du Preez and Kroger; Bodas; Szentgyorgyi and Salamon; Olsha and West; Palfalvi; Szabo and Szentgyorgyi; Fischer and Sommer; Fleury and Bellot)***

EPRI. TAG™ Technical Assessment Guide. 91. Palo Alto, CA, Electric Power Research Institute.
Notes: .***A set of up-to-date economic factors and cost assessment methodologies. The methods provide a means for comparing capital investment evaluations in a consistent manner. A number of example cases are worked out. No specific cases include cooling system comparisons but the general information on utility economics is useful in developing appropriate energy replacement penalty costs. No references except in footnotes.***

Fay, H. P. and T. R. Litton. Save Water, Stop Plume with Parallel Condensing. EPRI TR-104867. 94. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Reviews the installation of the first two wet/dry parallel condensing systems in commercial operation---one for water conservation, the other for plume abatement. Full system descriptions and performance information vs. ambient dry-bulb temperature are given. No cost information is provided. No references.***

Fricke, H. D. and A. M. Czikk. Performance of a Steam Condenser for Dry Cooling. EPRI CS-4016. 85. Palo alto, CA, Electric Power Research Institute.
Notes: ***{See above citation}***.

Fricke, H. D. *et al.* Advanced Dry-Cooling Demonstration: Summary. 86. Palo Alto, CA, ElectricPower Research Institute.
Notes: ***{See above citation}***.

Fricke, H. D., K. McElroy, and D. J. Webster. Heat Transfer Characteristics of a Dry and Wet/Dry Advanced Condenser of Cooling Towers". 82. Palo Alto, CA, Electric Power Research Institute.
Notes: ***One of three (along with the next two citations) documenting various aspects of the design, operation and test results of the ammonia phase-change advanced dry cooing system developed by EPRI and USDOE in the late 1970's and early 1980's. summary information is provided in {REFERNCE}***.

Fryer, B. C. A Review and Assessment of Engineering Economic Studies of Dry Cooled Electrical Generating Plants. 76. Richland, WA, Battelle Pacific Northwest Laboratories.
Notes: .***Review of the several engineering economic studies of the comparative costs of dry and wet cooling conducted in the early 1970's. At the time, there was no widely agreed upon approach to conducting proper cost comparisons. Also it was not clear that the dry tower designs had been optimized. Finally, since the energy and capacity replacement penalties which were calculated under the methods used at that time accounted for as much as two-thirds of the cost of the systems, the need for a consistent means of evaluating these penalties was emphasized. It is not clear that the methods used during that era of a fully regulated power business are useful in a deregulated era. 31 references.***

Ortega, F. M. Layton. 95.
Notes: ***Provides comparative cost estimates of an all-dry air-cooled condenser and a parallel wet/dry system designed for water conservation (nominally 50% of the water consumption of a wet cooling tower). Estimates are for "typical" U.S. site and include installation costs. An indication of cost vs. size for air-cooled condensers is given by comparing three different turbine exhaust pressures (4., 6. And 8. In Hg) for a given design dry bulb temperature (90F). Costs ranged from $13.2 to 8. Million (including installation) for a 600MM Btu/hr heat duty. No references.***

Goldschagg, H. B. *et al.* Air-Cooled Steam Condenser Performance in the Presence of Crosswinds. EPRI TR-104867. 95. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Analysis and correction of wind-related performance problems at ESKOM's Matimba power station. A computational procedure for determining the flow field and its effect on fan performance during windy conditions is presented. The results of a proposed solution, based on the analysis, in preventing turbine trips and providing significant performance improvement is given. 16 references.***

Goldschagg, Hein. Winds of Change at Eskom's Matimba Plant. Modern Power Systems , 43-45. 99.
Notes: ***Review of dry cooling at ESKOM's (South Africa) Matimba Plant (6 x 665 Mwe on air-cooled condensers). Site conditions and physical descriptions of the towers are given as are the operating procedures. Discussion focuses on wind-related problems. Certain wind directions unexpectedly resulted in serious performance losses and frequent turbine trips for high back***

pressure. Analysis and eventual solution of the problem through modifications to the configuration of the turbine hall and the ACC windwalls is reviewed. Highlights a need for flow modeling in advance of construction. No references provided.

Guyer, E. C. and D. L. Brownell. Wet/Dry Cooling for Cycling Steam-Electric Power Plants. 80.
Notes: ***A computational procedure for incorporating utility capacity supply and energy production economics into the design criteria and operational strategy of wet/dry cooling systems. Conclusions were that there was little difference between the optimum designs for base-load and cycling plants, particularly if relatively little water was available for cooling (< 250 acre-feet per year). 16 references.***

Guyer, E. C. and J. A. Bartz. Dry cooling moves into the mainstream. Power Engineering . 91.
Notes: ***A brief review of the state-of-the-art of dry cooling. At the time (~1990) the use of dry cooling was increasing in the U.S. A list of recent installations is provided. Some of the installations are described and a summary of the basic types of dry cooling systems is given. 8 references.***

Guyer, E. C. Dry Cooling: Perspectives on Future Needs. 91.
Notes: ***Survey of needs for and utility attitudes toward dry cooling. A review of the environmental regulations and the then current expectations for water supply and potential shortages is given. The status of existing dry cooling systems in use at the time is provided. An historical survey of installations in the U.S. showed a significant increase in the late 1980's up to the date of the report. An extensive review of the literature at the time is given 72 references.***

Hamilton, Thomas H. Developing the Worth of Colder Water in a Steam Turbine Generating Station. 2000. Cooling Tower Institute.
Notes: ***Presents a systematic calculation procedure for evaluating the lost energy penalties associated with reduced tower performance. Examples are for wet cooling towers at steam-electric plants but the methodology applies to dry systems. Extension to combined-cycle plants is non-trivial but straight-forward. The computational scheme is detailed and laborious in this age of computerized computational tools but serves to illustrate the elements of the comparison procedure well. A useful starting point for an approach to the wet-dry comparisons. No references.***

Hauser, L. G. *et al.* An Advanced Optimization Technique for Turbine, Condenser, Cooling System Combinations. Proc. of American Power Conference. Vol. 33, p. 427-448. 71.
Notes: ***Thorough description of the methodology for selecting the optimum cooling system and comparing optimized designs of the different classes of cooling system. Typical curves of present worth evaluation vs. cooling system performance are provided. The effect of various turbine heat rate characteristics are presented and explained. 4 references.***

Hendrickson, Paul L. An Overview of Economic, Legal and Water Availability Factors Affecting the Demand for Dry and Wet/Dry Cooling for Thermal Power Plants. 77. Battelle Northwest Laboratories.
Notes: ***No copy available.***

Hirschfelder, G. Der Trockenkühlturm des 300-MW-THTR-Kernkraftwerkes, Schmehausen-Uentrop. 73. VGB Krasftwekstechnik.
Notes: ***Review of the design, operation and economics of the natural draft dry cooling tower at Schmehausen in Germany. This was a unique design which was never replicated. The tower no longer exists but is an interesting case study of an innovative design. The tower structure consisted of a central column from which three rings were suspended. A cable network was attached to the rings and aluminum plates were attached to the cable network to provide the tower shell. No references (In German.)***

Hoffmann, J. E. and D. G. Kroeger . The Response of a Large Natural Draft Dry-Cooling Tower to Ambient Temperature Stratification. EPRI TR-104867. 94. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Detailed analysis and computational scheme for performance of natural draft dry cooling systems in the presence of a thermal inversion. Comparisons with data are given. Relevant only for natural draft systems---not for combined cycle plants in California. 20 references.***

Horsak, Randy D. Heat Rejection from Geothermal Power Plants. 79. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Cost and performance comparisons of wet and dry systems for geothermal plant applications. Little relevance to the combined cycle plant case.***

Hu, M. C. Engineering and Economic Evaluation of Wet/Dry Cooling Towers for Water Conservation". November, 1976. Washington, DC, U. S. Department of Energy.
Notes: ***No copy available.***

Hutton, David. Improved Power Plant Performance with Evaporative Steam Condensing. 99.
Notes: ***Presentation of the use of evaporative condensers (units where the evaporative cooling takes place directly on the outer surface of the condenser tubes). The system has been applied in a few instances in small installations and may be applicable to 5 to 100 MWe units. Cost savings of around 10% in comparison to condenser/cooling tower systems are claimed. 6 references.***

Hutton, David and C. W. Carlson. Fiberglass Closed-Circuit Cooling Towers: Design Considerations for Power Industry Applications. 94.
Notes: ***Discussion of the use of fiberglass for structural component of wet cooling towers. A brief review of the types of power plant cooling systems is given. The benefits of the choice of fiberglass for tower construction along with the implications of this choice for tower structural design are presented. Some practical observations based on experience with industrial process applications is given. Benefits are in low cost, low maintenance and ease of construction. No references.***

Proceedings of the 9th IAHR Cooling Tower and Spraying Pond Symposium. 94. International Association for Hydraulic Research.
Notes: ***Sessions were held on effects on local environment; cooling tower operation; cooling tower modeling; components; design and performance. The primary emphasis of the meeting***

is on wet cooling towers and spray systems. Eight papers deal with dry or wet-dry systems. Summaries of the more relevant papers are given in the individual citations (Leitz; Tesche; Tesche et al.; Cinski; van der Spek; Kroger; Nemeth; Ludvig; Bouton and Monjoie; Nagel; Schrey) **(37 papers)**

Iovino, G *et al.* Optimal Sizing of Natural Draft, Dry Cooling Towers for ENEL Combined Cycle Power Plants.
Notes: ***An optimization method for natural draft dry cooling towers of combined cycle power plants (See also Ecker and Wiebking, 1978 and Conradie, Buys and Kröger, 1998). The case study is for an indirect, dry system and the objective function to be minimized is the cost of energy. Sensitivity studies on the effect of differing meteorological conditions and cost assumptions are given. A good treatment of the effect of turbine heat rate characteristics and tower approach is given. 2 references.***

Junge, Erik. Innovations in Cooling Tower Design. 2000.
Notes: ***Marketing article to promote a proposed design and new tower configuration developed by TowerTech. The article highlights three problems with wet towers: high maintenance problems with fans; failure of water distribution system and the buildup of sludges (sometimes hazardous) in the tower basin. The proposed design includes a rotary spray nozzle to obtain better coverage of the fill and the replacement of the basin with perimeter troughs, and the placement of the fans at the bottom of the tower for easier access and a drier environment. The description is extremely brief and, in all likelihood, is not a feasible approach. No references***.

Kast, G. A. and S. D. Adams. Kakkonda geothermal plant uses hybrid cooling. Modern Power Systems . 95.
Notes: ***Description of a wet/dry tower designed for plume abatement purposes at a 30MWe geothermal plant in Kakkonda, Japan. Heat rejection is 100% to the wet tower in the summer and 20%/80% dry wet in winter months. 14 ft. high fan stacks were used to promote plume mixing and plume rise for environmental reasons. Apparently a barometric, direct contact condenser was used, since it is claimed that the circulating water is "partially geothermal condensate." No cost information is provided. No references.***

Kintner-Meyer, Michael and A. F. Emery. Cost-Optimal Analysis of Cooling Towers. ASHRAE Annual Meeting. pp. 92 -- 101. 94. ASHRAE.
Notes: ***Optimization procedure for cooling tower design is presented. The objective function to be optimized is the combined capital and operating cost. The application is for refrigeration systems so the operating penalties, which include the effect on chiller coefficient of performance are qualitatively different from power generation applications. The paper gives a good example of optimization methods for cooling tower costs against range and approach but no insight into source of cost information or into the effect of actual tower or fill design variables. Reference has limited value for the wet vs. dry cost comparisons of this report. 12 references.***

Kooy, R. J., R. J. Laverman, and J. L. Seale. Performance of a Capacative Cooling System for Dry Cooling. 86. Palo Alto, CA, Electric Power Research Institute.

Notes: ***Description of design and performance of the thermal storage components of the advanced phase-change ammonia dry cooling system developed and tested by EPRI and USDOE. No copy currently available.***

Kosten, H. and R. Wyndrum. Wet, Dry and Hybrid Systems--A Comparison of Thermal Performance. EPRI TR-104867. 95. Palo Alto,CA, Electric Power Research Institute.
Notes: ***Describes the choices available for condensing and cooling systems for the condensation of turbine exhaust steam. The paper describes the major types of system and their operating characteristics, operating and maintenance costs and other selection considerations. Life cycle cost analyses are discussed. Problem check lists for wet, dry and wet/dry systems are given. Simplified cost comparisons are included. An excellent introductory survey of the subject. No references.***

Kroeger, D.G. Air-cooled Heat Exchangers and Cooling Towers. Begell House, New York (1998).
Notes: ***Comprehensive text on the design and performance of dry cooling systems. The text is over 600 pages long. The 10 Sections and 3 appendices give a detailed treatment of the fluid flow, heat transfer and thermodynamic theory of cooling systems complete with detailed worked examples. Descriptions of operating plants using dry cooling, the effects of different meteorological conditions and methods for selection of optimum systems are presented. The several hundred references catalog the up-to-date literature of the field as of 1998. (Information to obtain the book is available at the Begell House, Inc. website; http://wwwbegellhouse.com/heat_exchangers/aircooled.html)***

LaRonge, Thomas M. Bugs and Bugaboos of Cooling System Components. 2000. Cooling Technology Institute.
Notes: ***Review of cooling tower problems associated with water quality control. Emphasis is on microbial induced corrosion problems and methods for ameliorating and correcting such problems. A brief review of mechanical failures and their cause is presented. Largely qualitative discussion based on author's considerable experience in cooling tower water treatment and maintenance. 1 reference.***

LeFevre, M.R. New Technology and Cooling Tower Design Practices. Combustion 28-32 (1977).
Notes: ***General discussion of various types of cooling towers. Discussion is limited to wet systems and includes mechanical and natural draft towers, round mechanical draft towers and fan-assisted, natural-draft towers. A brief review of where in the U.S. the various designs choices are preferred (as of 1977). No references.***

Li, K. W. and W. Sadiq. Computer-Aided Optimization of Cooling Systems. Computers in Engineering Conference No. 3. pp. 165-171. 85. ASME International.
Notes: ***A relatively simplified optimization method for power plant cooling systems. Presents a case study for a mechanical draft wet cooling tower on a 670 MWe plant. The effect of varying approach and condenser terminal temperature difference is illustrated. No detailed optimization of tower design variables is attempted. No cost information is included. 4 references.***

Lindahl, P. and R. W. Jameson. Plume Abatement and Water Conservation with the Wet/Dry Cooling Tower. CTI Journal **14**, (1993).
Notes: ***One of the classic references on hybrid wet/dry cooling towers for both plume abatement and water conservation. Excellent descriptions of the configuration and thermodynamic operating principles of the several tower types are given. Attention is given to the selection of the design point for real applications. No cost information is included. No references.***

March, F. and F. Schulenberg. GEA-Information: Air cooled condenser for a 160 MW steam power station, planning of the plant and experience gained in two years of operation. 70. Bochum, Germany, GEA-Kühlturmbau und Luftkondensation, GmbH.
Notes: ***Review of two years operating experience on the air-cooled condenser at the 160 MWe power plant in Utrillas, Spain. The performance data confirmed the design calculations and demonstrated a reserve capacity of about 8%. No serious operating problems were noted. Annual cleaning with compressed air blowing during operation was sufficient to maintain performance. No efficiency loss was observed as a result of windy conditions. No cost information is provided. No references.***

Mathews, R.T. Air cooling in chemical plants. Chemical Engineering Progress **55**, 68-72 (1959).
Notes: ***An old reference which analyzes the use of air-cooled heat exchangers in the chemical process industry. Contains a detailed set of considerations and criteria for deciding when air cooling is potentially a preferred choice. Some discussion of design temperature selection based on site meteorology. Suggested design temperature is the "2% temperature". (The temperature that is exceeded for 175 hours per year.) Many of the considerations are appropriate for power plant applications, although the economic evaluation discussion is quite limited. No references.***

Maze, R.W. Air vs. Water Cooling. The Oil and Gas Journal 74-78 (1974).
Notes: ***Comparison of air and water cooled heat exchangers for petrochemical applications. Typical values for temperature differences and comparative costs are provided. Attention is given to problems related to cold weather operation. Cost information would be difficult to translate for power plant applications. 3 references.***

McHale, C.E.*e.al*. New Developments in Dry Cooling of Power Plants. Combustion 28-36 (1990).
Notes: ***Description of the advanced phase-change dry cooling system developed and tested by EPRI and the USDOE in the 1980's. The system was an indirect dry cooling system with a surface condenser and an ammonia loop between the condenser and the dry tower, thus eliminating the condenser range from the temperature difference between the ambient dry bulb and the steam condensing temperature. Comparisons are provided with other cooling system designs and the costs and incentives for the use of dry cooling are reviewed. 18 references.***

Merchant Power on the Go Power Engineering (July, 2000).

Michell, F.L. and D. H. Drew. On-Line Performance Monitoring of the 1300 MW Natural Draft Cooling Towers on American Electric Power's General James M. Gavin Plant. CTI Journal (1997).
Notes: ***Discussion of the methods for monitoring very large, natural draft cooling towers and the problems with the equipment for making the measurements. Measurements of ambient temperatures and wet bulbs, circulating water flow rates and data averaging procedures are discussed. The paper is of limited value for the issues of cooling system selection for combined cycle plants in California but does emphasize the difficulties of determining actual cooling performance being realized from an operating cooling tower. 2 references***.

Miliaris, E.S. Power Plants with Air-Cooled Condensing Systems. The MIT Press, Cambridge, Massachusetts (1974).
Notes: ***Textbook consisting of 14 Sections and 4 appendices 234 pages long. It is organized by cooling system type and includes direct and indirect systems, barometric and surface condensers, detailed descriptions of heat transfer and fluid flow computational procedures for finned tube heat exchangers, and extensive treatments of the thermodynamic optimization methods for application to power plants. Additional Sections deal with the system planning aspects of system selection by utility companies. Detailed information on plants in existence at the time is provided. A treatment of determining the loss of plant capability and increased cooling system load at increased back pressure included. Extensive references are given in each Section; total for the book is several hundred.***

Mirsky, G. and J. Bauthier. Cooling Towers: New Developments for New Requirements. EPRI TR-104867. 94. Palo Alto. CA, Electric Power Research Institute.
Notes: ***Addresses the design of cooling towers for combined cycle power plants. The discussion is organized around three sets of requirements: water use and pollution, noise and visibility. Particular attention is given to determining the cooling tower requirements for the special case of a combined cycle power plant where the steam part of the cycle accounts for only about 1/3 of the plant output but for which the cycle efficiency may be fairly poor compared to stand-alone steam plants. Order of magnitude cost comparisons are provided based on the authors' corporate experience. No sources are provided. 4 references.***

Mirsky, G., K. Bryant, and J-P. Libert. The Latest Worldwide Technology in Environmentally Designed Cooling Towers. CTI Journal **13**, (1992).
Notes: ***Reviews some advanced approaches to improving the environmental characteristics for cooling towers at power plants. Specific topics include low noise designs; injection of scrubbed flue gas into tower air stream; low drift designs; new EPA testing methods; thermal performance; asbestos removal and replacement; water conservation; zero discharge towers; and plume abatement or elimination. Order of magnitude cost estimates are given for some of these approaches.***

Missimer, J.R. and D. Wheeler. Characterization of Drift Rates and Drift Droplet Distribution for Mechanical Draft Cooling Towers. (1997).
Notes: ***Discusses drift measurement methods and presents data on drift rates and drift droplet size distribution obtained from 57 drift tests made over the previous 23 years. The results are grouped by tower type and by drift eliminator. Results from different sampling methods***

(isokinetic sampling and sensitive paper sampling) are compared. A distinction is made between mechanically generated drift and droplets condensed in the tower exhaust stream. 14 references.

Mitchell, R. D. Survey of Water-Conserving Heat Rejection Systems. 89. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Summary of all economic comparison studies of wet, dry and wet/dry systems available at the time of the report. A clear comparison of the important assumptions made in each study and the effect that the choice of assumptions has on the cost comparisons is provided. Very concise descriptions of the appropriate methodology for determining all the important cost elements and in obtaining optimized systems for purposes of comparison are given. This is the most comprehensive treatment of the comparison methodology that has appeared in the open literature to date. 63 references.***

Miura, T. and O. Gotoh. The New Wet/Dry Cooling Tower without Finned Tube Dry Section (NWD). (1998).
Notes: ***Presents a new design concept (called "NWD" for "novel wet/dry tower) for plume abatement with no separate finned tube dry section. The approach uses a combined wet/dry fill with a section to which the water flow can be interrupted. The wet/dry passages consist of 15 to 25% of the total fill area. Performance test results on a single mechanical draft, cross-flow cell are reported. Satisfactory plume abatement was documented, but no information on cost or on relative cooling capability when operated in the plume abatement mode is available. No references***

Wet/dry cooling tower eliminates plume at Teesside. Modern Power Systems , 57-60. 95.
Notes: ***Presents a description of the retrofit of a fan-assisted hyperbolic cooling tower in the U.K. with a parallel path wet/dry tower for plume abatement. The plant is a combined-cycle 1725 MWe plant operated in conjunction with a chemical complex to which it supplies steam. A full description of the supplementary cooling tower is given including some overall cost estimates. The unit was not yet in operation at the time of the paper so no performance or O&M information is available. No references.***

Monjoie, M.a.J.-P.L. Testing Procedures for Wet/Dry Plume Abatement Cooling Towers. CTI Journal (1994).
Notes: ***Procedures for testing of wet/dry plume abatement towers are presented. Full discussion of the measurements to be made, the equipment to be used, calibration techniques are given. The thermodynamics and fluid flow of plume abatement are reviewed, and the calculations required to verify performance are laid out in detail. Criteria for fulfilling the plume performance guarantee are clearly stated: thermal guarantee is fulfilled; the average maximum humidity is lower than expected; no local air state exceeds the maximum humidity guarantee value by more than 7%. No references.***

Mukherjee, R. Effectively Design Air-Cooled Heat Exchangers. Chemical Engineering Progress 26-46 (1997).
Notes: ***Detailed discussion of design methods for air-cooled heat exchangers. Applications are for the chemical process industry but much of the material applies equally well. No***

optimization is considered, but each of the design decisions (tubes, headers, fins, fans, etc) are discussed in detail. No cost information is provided. 17 references plus 7 suggestions for "Further Reading."

Murphy, D. Cooling towers used for free cooling. ASHRAE Journal 16 (June, 1995).
Notes: ***Deals with chilled water systems which reject heat to a cooling tower. It is recognized that in some climates, there are days when the cooling tower alone can provide cold water at sufficiently low temperature to meet the chilled water needs of the building or process. Configurations are suggested to implement this approach as an energy saving method. No cost or performance data is given. No relevance to the comparison of wet vs. dry systems***.

Naegelen, R. J., J. L. Seale, and M. Husain. Detailed Design for Incorporating Chicago Bridge & Iron (CBI) Capacitive Cooling System in the ACT Facility in Bakersfield, California---Vol. 1: Executive Summary. 82. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Description of the thermal storage component of the advanced concept test facility(ACT) built and operated by EPRI and USDOE to test an indirect dry cooling system using phase-change ammonia as the intermediate heat transport loop. This first volume reviews the concept of thermal storage in this application and estimates the performance and cost benefits to be obtained.No copy available.***

Naegelen, R. J., J. L. Seale, and M. Husain. Detailed Design for Incorporating Chicago Bridge & Iron (CBI) Capacitive Cooling System in the ACT Facility in Bakersfield, California---Vol. 2. 82. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Description of the thermal storage component of the advanced concept test facility(ACT) built and operated by EPRI and USDOE to test an indirect dry cooling system using phase-change ammonia as the intermediate heat transport loop. This second volume presents the description of the equipment, its operation at the test facility and the test results. No copy available.***

Norton, C.H. and K. W. Rowland. Design, Operation and Water Treatment of a Wet Finned Tube Exchanger Cooling Tower. CTI Journal (1989).
Notes: ***Presents a description of a "wet" finned tube heat exchanger cooling tower for compressed gas cooling by Southern California Gas Company. The units consists of fiberglass fill over which recirculated water is passed and through which inlet air can be directed during hot periods. The precooling of the air increases the capacity of the finned tube dry section. Annual water consumption is reduced significantly (55% to 86%) relative to conventional wet cooling. Water treatment procedures for scaling control are discussed. No references***.

O'Boyle, Paul. Be cool, be flexible. Modern Power Systems , 53. 99.
Notes: ***Describes a business arrangement where cooling water supply is "outsourced" to a company (Global Water Technologies) who install or upgrade cooling towers at their cost and contract to deliver a certain amount of water at a specified temperature for the contract term. No performance or cost information is provided. No references.***

Oosthuizen, P. C. Performance Characteristics of Hybrid Cooling Towers. ME Thesis. 95. University of Stellenbosch, Dept. of Mechanical Engineering.

Notes: ***Theoretical and experimental study of the performance of wet/dry cooling towers in hybrid and separate configurations. A computer program is developed which predicts the thermal performance of both parallel and series path water flow rectangular hybrid towers. Example calculations are provided. Experimental data on one cooling tower fill was obtained and the computational procedure for generating the heat transfer and pressure drop characteristics from the experimental data is given. Particular attention is given to the thermodynamics of plume prediction models and the psychometrics governing the formation of visible plumes are discussed. Calculations of inlet recirculation and a method for estimating the extent of ineffective fill area is provided. 97 references.***

Special Report: Cooling Towers. Power Magazine . 73.
Notes: ***Excellent comprehensive special report on cooling towers; major sections on natural and mechanical draft wet towers and on dry towers. Good systems descriptions are provided along with discussions of design, operation and maintenance issues. Some typical performance data is given but no detailed design or cost information. Although the report in 27 years old, much of the information is still relevant although there are big differences in materials, water treatment methods and the economics which dictate the cost of alternate systems. For example, the emphasis on natural draft hyperbolic towers is no longer justified and the dry system designs have progressed considerably since the writing of the report. No references.***

Quigley, K. and Karl Wilber. Development of Cooling Tower Performance Impacts of Utility and Process Plants. CTI Journal (1991).
Notes: ***Presents methods for determining the performance penalties and their contribution to total evaluated costs for cooling towers. Five cases, three of which are for power plants, are presented. Results are presented in the form $/°F for estimating the cost of tower performance shortfall. 9 references.***

Rose, J. C., D. J. Wilson, and G. H. Cowan. Air-Cooled Heat Exchanger Performance Specification. 73. Harwell, UK, Atomic Energy Research Establishment.
Notes: ***Detailed specifications for design, specification and location of air-cooled heat exchangers. The report covers considerations of unit location, specification of ambient design temperatures, fan and plenum configuration, fan and drive selection, thermal performance guarantees, noise level guarantees and air-flow measurement for performance tests. No references.***

Smith, E. C. and M. W. Larinoff. Power Plant Siting, Performance and Economics with Dry Cooling Tower Systems. Proc. of American Power Conference. 32, p. 544-578. 70.
Notes: ***Excellent summary of performance and economics of dry cooling tower systems. A detailed methodology is set forth for accounting for all aspects of the cost of dry cooling. The optimization studies suggested that lowest total owning and operating costs with dry cooling tower systems occur with turbine exhaust temperatures in the range of 8” Hg and above. The authors recommended that turbine manufacturers could contribute to the field by offering high back pressure turbines which were not available at that time. 16 references.***

Smith, James O. and M. Muder. Commonwealth Edison (COMED) Byron Generating Station Unit 1 Crossflow to Counterflow Conversion. EPRI TR-104867. 94. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Presents the results of a cross-flow to counter flow conversion of two hyperbolic cooling towers at Commonwealth Edison's Byron Nuclear plant. The conversion was motivated by a deterioration in tower performance due to ice damage from winter operations. The paper is of limited value for the issue of cooling system choice for combined-cycle plants in California but does contain some information of the cost of reduced performance. No references.***

Spilko, J. Dry cooling enhances Syria's Teshrin plant operation. Power Engineering , 40-41. 94.
Notes: ***Description of the design, performance and operation of the 200 MWe gas-fired plant at Teshrin, Syria. The system includes delugeable fin-fan coolers to assist in making peak load on the hottest days. A natural draft design was chosen to reduce plant auxiliary power requirements and to improve the plant net heat rate. No cost or quantitative performance information is given. No maintenance problems were reported. Annual cleaning with a water-wash system maintains performance and is done by the normal plant maintenance crew.***

Staff report. Parallel combined cycle at Altback/Deizisau HKW 2. Modern Power Systems . 95.
Notes: ***Brief description of a hybrid 20%dry/80% wet cooling tower chosen primarily for plume abatement. No cost or performance information is provided. No references.***

Staff report. Single row condensers build on success. Modern Power Systems , 43. 98.
Notes: ***Brief discussion of the use of single row condensers in air-cooled condenser applications. Some specific application (Sutton Bridge combined cycle plant in the UK (780MWe); the Ogden-Huntington waste-to-energy plant in the US; the Herdersbrug combined cycle plant in Bruges, Belgium) are identified. The extension of the use of SRC to natural draft towers is proposed and reference is made to a Hamon paper at the Power Gen meeting in Milan, Italy in 1998. 1 reference.***

Streng, A. Circular Hybrid Cooling Towers. CTI Journal (2000).
Notes: ***Describes circular hybrid towers as a preferred alternative to conventional (rectangular) cell type towers primarily because the problem of plume recirculation is reduced. Field data on recirculation and its effect on inlet wet-bulb temperature is provided. Performance comparisons between circular and cell-type towers over a year's range of ambient conditions is given. References to field tests and operating reports (primarily German) are given. Little relevance to wet vs. dry comparisons.***

Streng, A. Combined Wet/Dry Cooling Towers of Cell-Type Construction. Journal of Energy Engineering **124**, 104 (1998).
Notes: ***State-of-the-art of technical developments in wet/dry cooling towers. The emphasis is on plume abatement and not water conservation. Detailed treatments of the tower design and the thermodynamics of plume abatement are provided. Some attention is given to selection of materials of construction. No cost or thermal performance information is provided. 9 references.***

Surface, M.O. System designs for dry cooling towers. Power Engineering 42-50 (1977).
Notes: ***General review of dry cooling tower designs and their applications. The paper provides a good overview of the several types of dry cooling system---direct, indirect more or less conventional design along with some concepts that were being proposed in the late 1070's---plastic surfaces, rotating disk surfaces, and the ammonia phase change system. The tradeoffs with the use of high back-pressure turbines are discussed. A cost analysis is provided of the range of options from all-dry systems with and without high-back pressure turbines, wet/dry systems with make-up water requirements ranging from 1% to 40% of wet tower, and a mechanical draft wet cooling tower. No references***.

Swanekamp, R. Profit from latest experience with air-cooled condensers. POWER 78 (1994).
Notes: ***Excellent review of operating issues with air-cooled condensers at several power plants. Anecdotal descriptions of loss of capacity during hot weather periods, freeze-up problems in winter, recirculation problems during windy conditions and requirements for tube bank cleaning are given. Identifies operating plants in California (Camarillo Plant in Ventura county), Virginia (Doswell plant), Montana (Rosebud plant in Colstrip, MT) and New Jersey (Sayreville co-gen plant) and Hawaii (Maalaea Unit 15). No references***.

Thiel, F. and D. Clute. A landmark for private power in Mexico. 27-29 (1998).
Notes: ***Description of a 700 MWe combined cycle plant (Samalayuca II) in Mexico. This is the first large-scale private power project in the country. This reference is included only because it is mentioned that an air-cooled condenser was chosen for water conservation reasons. No information about the cooling system is provided. No references.***

Trage, B., A. J. Ham, and Th. C. Vicary. The Natural Draught, Indirect Dry Cooling System for the 6 x 686 MWe Kendal Power Station, RSA. Jt. ASME/IEEE Power Generation Conference. 90-JPGC/Pwr-25. 90.
Notes: ***Detailed description of the unit is provided. It is noted that the choice of an indirect system, while not expected to be economically preferred, was made in order to provide large-scale comparative performance and operating data to the 6 x 665 MWe plant that had just been constructed with a direct dry cooling system at Matimba. Information on the net output of the unit during the year at different ambient temperatures is given. Some attention is given to the effect of wind on tower performance with comparisons to towers at other sites including a tower in Armenia (Razdan Power Station) and in Germany (Schmehausen). No economic information is provided. No references.***

Tsatsaronis, G. and A. Valero. Thermodynamics meets economics. Mechanical Engineering 84-86 (1989).
Notes: ***An exergy analysis of power conversion systems to identify sources of thermodynamic inefficiencies. Simple optimization methods are given. No specific mention is made of power plant cooling systems although the methodology is applicable. No quantitative cost information or any worked examples are provided. No references.***

Tsou, John J. Jenco and K. Zammit eds. Proceedings: Cooling Tower Technology Conference. EPRI TR-108483. 97.
Notes: ***Contains 22 papers on several aspects of cooling tower technology: performance***

improvement; design innovations; fouling and corrosion prevention; and operation and maintenance. Primary emphasis of the meeting was on wet cooling with only three papers on dry or wet/dry systems. (Summaries of the important papers related to dry systems given under the entries for the individual papers.) {Goldschagg et al., Hoffmann and Kroeger, Mirsky and Bauthier}.

van der Spek, H. F. and P. J. M. Nelissen. Advanced Low Noise Cooling Fans. EPRI TR-104867. 95. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Describes a research program to develop and test low noise fans for cooling tower application. Good description of testing procedures. Analyses and data are presented relating noise reduction to fan power reduction. Good description is provided of the sources of noise and the general approach used to reduce it. 2 references***.

Varley, J. Eskom's Majuba: at the peak of its career. Modern Power Systems . 1999 .
Notes: ***Description of the Majuba Power Station in South Africa, a 4100 MWe plant with six units, three of which use air-cooled condensers. The difference between the units on wet cooling towers and those on air-cooled condensers is an increase in capacity from 660 MWe to 715 MWe. Complete qualitative description of the plant is given but no performance or cost information is provided. No references.***

Von Cleve, H-H. Comparison of Different Combinations of Wet and Dry Cooling Towers. ASME Winter Annual Meeting. ASME Paper No. 75-WA/Pwr-10. 75. New York, NY, ASME, International.
Notes: ***Review of the use of combined or hybrid wet and dry systems for either water conservation or plume abatement. The combination of wet cooling towers with direct air cooled condensers is the usual system of choice for meeting water consumption or maximum turbine back pressure constraints. For plume abatement a combined wet/dry tower is required. The effect of design objectives, site characteristics and water availability on the system choice is discussed. No references.***

Wheeler, K. R. *et al.* Deposition and Corrosion Phenomena on Aluminum Surfaces Under Deluged Dry Cooling Tower Conditions. 81. Palo Alto, CA, Electric Power Research Institute.
Notes: ***Description of the deluge enhancement feature of the EPRI/USDOE Advanced Concept Test facility. The design of the equipment and details of the tests are given along with data from the tests. No copy is currently available.***

Willa, J.L. Evolution of the Water Cooling Tower. CTI Journal (1991).
Notes: ***History of wet cooling towers and their evolution to power plant size units. Particular attention is paid to the changes in materials of construction over time. Comparison of the advantages and disadvantages of cross-flow vs. counter-flow units is given. No references.***

Willa, James L. How to Improve the Thermal Performance of Cooling Towers. CTI-EPRI Cooling Tower Conference. 2000. Palo Alto, CA, Electric Power Research Institute.
Notes: ***An excellent survey of common operating and maintenance problem found in wet cooling towers and methods for addressing them. Estimates of performance improvement to be gained from these fixes are given. No quantitative cost information is given. 2 references***.

Zaloudek, F. R., L. J. Brown, and R. T. Allemann. Advanced Concepts Test Facility---Measurements and Suggested Test Plan. 80. Palo Alto, CA, Electric Power Research Institute. Notes: ***Description of the tests conducted at the EPRI/USDOE Advanced Concept Test facility. This report was one of a series of reports to document the ammonia phase change dry cooling system and to make the knowledge available to utility planners. This report presents the test strategy, the determination of the required measurement accuracy, the specification of the instrumentation and the statistical treatment of the results. 5 references.***

BDT Engineering.

GEA.

Hamon Cooling Systems.

Marley Cooling Tower.

Niagara Blower Company.

Aull, Rich and Tim Krell. Design Features and Their Effect on High Performance Fill. 2000. Houston, TX, Cooling Technology Institute.

Hobson, E., T. H. Massey, and P. Lindahl. NPF Cooling Tower Fill--Its Development and Demonstration. 95. Houston, TX, Cooling Technology Institute.

Richardson, J. and M. G Trulear. Recent Advances in High Alkaline Cooling Water Treatment. 2000. Houston, TX, Cooling Technology Institute.

Howarth, J. and C. Nalepa. First Field Trials of Single-Feed, Liquid Bromie Biocide for Cooling Towrs. 2000. Houston, TX, Cooling Technology Institute.

Gill, J. S., J. R. Parson, and R. C. Gordon. A New Treatment for Calcium Carbonoate Control in Alkaline Conditions. 97. Houston, TX, Cooling Technology Institute.

Suptic, D. M. A Non-Metallic Air Cooled Heat Exchanger for Cooling Tower Plume Reduction. 99. Houston, TX, Cooling Technology Institute.

Missimer, J. R., D. E. Wheeler, and K. W. Hennon. The Relationship Between SP and HGBIK Drift Measurement Results--New Data Creates a Need for a Second Look. 98. Houston, TX, Cooling Technology Institute.

Randall, J. D. *et al.* Cooling Tower Plume Abatement at Chicago's O'Hare Airport. 98. Houston, TX, Cooling Technology Institute.

Feltzin, A. E. and D. Benton. A More Nearly Exact Representation of Cooling Tower Theory. 91. Houston, TX, Cooling Technology Institute.

DesJardins, R. J. Using the EPRI Test Data to Verify a More Accurate Method of Predicting Cooling Tower Performance. 92. Houston, TX, Cooling Technology Institute.

LeFevre, M. R. Eliminating the Merkel Theory Approximations--Caan It Replace the Empirical Temperature Correction Factor. 84. Houston, TX, Cooling Technology Institute.

LeFevre, M. R. Influence of Air and Water Tepeature on Fill Characteristics Curve. 85. Houston, TX, Cooling Technology Institute.

Fulkerson, R. D. A Comparison of Crossflow Cooling Tower Splash-Type Fills. 99. Houston, TX, Cooling Technology Institute.

Fulkerson, R. D. Comparative Evaluation of of Counterflow Cooing Tower Fills. 88. Houston, TX, Cooling Technology Institute.

Mirsky. G. R. and M. Monjoie. Film Fill: Recent Research and Application Data. 90. Houston, TX, Cooling Technology Institute.

Schulenberg, F. Der Luftkondensator für den 356-MW-Block in Wyoming/USA. 77. Sammelband VGB-Konferenz.

Van der Walt *et al.* The Design and Operation of a Dry Cooling Tower system for a 200 MW Turbo-genrator at Grootvlei Power Station, South Africa. Proceedings of the 9th World Energy Conference. 74.

Kosten, G. J. *et al.* Operating Experience and Peformance Testing of the World's Largest Air-cooled Condenser. American Power Conference. 81.

Von Cleve, H.-H. Die Luftgekühlte Kondensationsanlage des 4000 MW-Kraftwerks Matimba/Südafrika. VGB Kraftwerkstechnik **64**, (1984).

Scherf, O. Die luftgekühlte Kondesationsanlage im 150-MW-Block des Preussag-Kraftwerkes in Ibbenbüren, Energie und Technik. 260-264 (1969).

Simpson, N. Air-Cooled Condenser Fits Steam Plant to Arid Site. Electrical World (June 8, 1970). 70.

van der Spek, H. F. The Mechanical Retrofit of a Large Cooling Tower. Cooling Tower Conference. 2000.

Maulbetsch, J. S. and M. N. DiFilippo. Spray Cooling Enhacement of Air-Cooled Condensers. Madadnia, J and H. Koosha. Proceedings of XIIth International Conference on Cooling Towers. 2001. International Association of Hydraulic Research.

Appendix B
EXISTING AND PLANNED DRY AND HYBRID COOLING SYSTEMS

Major Manufacturers and Contact Information

- **BDT Engineering,** Balcke-Dürr, Inc., 405 N. Reo Street, Tampa, FL 33609, 813-289-1516, Contact: Mr. Ralph Wyndrum
- **Marley Cooling Tower,** 7401 W. 129th Street, Overland Park, KS 66213, 913-664-7588, Contact: Mr. Paul Lindahl
- **GEA Power Cooling Systems, Inc.,** 5355 Mira Sorrento Place, Suite 600, San Diego, California 92121, 619-457-0086, Contact: Mr. Jamie Clark
- **Hamon Cooling Towers**, 58-72 E. Main Street, Somerville, NJ 08876; Contact: Dr. Ram Chandran

Installations

The remainder of this section contains information provided by the manufacturers listed above.

BDT
ENGINEERING
Balcke-Dürr, Inc.

(June 2000)

BDT Engineering: Air-Cooled Turbine Exhaust Steam Condensers, Worldwide Project Experience

Client	Location	Steam Flow lb / h	Back Pressure	Installed
Asea Brown Boveri (ABB)	Lake Road, CT	3 x 520,000	2.50"HgA	Engineering
ABB	Hays, TX	2 x 500,000	2.40"HgA	Engineering
ABB	Blackstone, MA	2 x 540,000	2.20"HgA	Construction
Electricite de France (EDF)	Rio Bravo, Mexico	1,100,000	3.0"HgA	Construction
ABB	Midlothian, TX	4 x 500,000	2.40"HgA	Construction
Mitsubishi Heavy Industries	Chihuahua, Mexico	970,000	2.76"HgA	Construction
ABB	Monterrey, Mexico	2 x 545,000	2.24"HgA	2000
ABB	Enfield, England	804,400	2.1 "HgA	1999
Thomassen Power Systems Doga / Mission Energy	Esenyurt, Turkey	390,000	7.5 "HgA	1998
EPA Taiwan/ Chung-Hsin Electric & Machinery	Hsinchu, Taiwan	205,955	4.43 "HgA	1998
EPA Taiwan/ Chung-Hsin Electric & Machinery	Pali, Taiwan	308,577	4.43 "HgA	2000
ESP Geko / HKW Feldberg	Feldberg,Germany	44,100	5.9 "HgA	1997
ESP Geko / HKW	Dresden, Germany	63,900	35.5 "HgA	1997
ML Ratingen / MHKW Pirmasens (Single Row)	Germany	117,200	3.3 "HgA	1997
ABB Enertech AG / KVA Niederurnen	Switzerland	35,000	2.9 "HgA	1996
D.B. Anlagen / VERA Hamburg (Single Row)	Germany	33,000	5.9 "HgA	1996
Bechtel	Crockett, CA	608,000	2.0 "HgA	1996
Caliqua Basel / KVA Gamsen	Switzerland	38,800	2.9 "HgA	1996
Statwerke Kiel / MVA Kiel	Germany	45,200	103 "HgA	1996
Siemens KWU / AEZ Kreis Wesel	Germany	165,300	2.9 "HgA	1996
Siemens KWU / SBA Furth (Single Row)	Germany	104,100	4.1 "HgA	1996
AVI Twente, Hengelo / Twente	Netherlands	194,400	2.5 "HgA	1996
Billings Generation	Billings, MT	463,696	7.5"HgA	1995
Stork Ketels / Wapenveld	Netherlands	103,200	2.9 "HgA	1995
NEMA Netzschkau / Izmit (Single Row)	Turkey	43,000	2.3 "HgA	1995
Blohm & Voss / SAVA Brunsbuttel (Single Row)	Germany	30,900	3.5 "HgA	1995
ML Ratingen / MVA Offenbach (Single Row)	Germany	75,000	3.5 "HgA	1995

Client	Location	Steam Flow lb / h	Back Pressure	Installed
ESP Heinzwerke / Sulzbach-Rosenberg	Germany	41,400	5.9 "HgA	1994
Caliqua Basel / KVA Thurgau	Switzerland	130,100	14.7 "HgA	1994
Bechtel	Rochester, MA	220,250	3.5"HgA	1993
PowerGen/Siemens	United Kingdom	1,877,900	2.7"HgA	1993
Krupp Stahl / Bochum	Germany	36,400	38 "HgA	1993
MAN GHH / GSB Ebenhausen	Germany	70,500	6.2 "HgA	1993
ABB Nurnberg / AVA Augsburg	Germany	122,700	3.5 "HgA	1993
Blom & Voss Batam	Indonesia	57,500	13.3 "HgA	1992
CRS Sirrine	Lowell, MA	160,000	3.25"HgA	1991
CNF Constructors	Fitchburg, MA	127,000	3.5"HgA	1991
Indeck Energy	Silver Springs, NY	120,000	2.5"HgA	1990
Rutgerwerke	W. Germany	88,000	5.0"HgA	1990
Lurgi MSW Bazenheid	Switzerland	3,100	3.5"HgA	1989
Siemens/MWS Cogen. Weissenhorn	W. Germany	83,000	4.5"HgA	1989
Chemische Fabrik Budenheim	W. Germany	6,000	1.8"HgA	1989
Blohm and Voss MSW,Beselich	W. Germany	13,200	3.0"HgA	1988
Blohm and Voss MSW Pinneberg	W. Germany	68,000	6.0"HgA	1987
ABB Baden, Kabul	Afghanistan	243,000	3.5"HgA	1987
SERT MSW Harelbeke	Belgium	44,000	1.5"HgA	1985
Stadtwerke Frankfurt for MSW Frankfurt	Germany	55,000	15.0"HgA	1985
BBC Mannheim (ABB), Touss Unit 4 150 MW Power Station	Iran	792,000	8.0"HgA	1984
BBC Mannheim, Touss Unit 3 150 MW Power Station	Iran	792,000	8.0"HgA	1984
BBC Mannheim for MWS/Geiselbullach	Germany	72,600	4.0"HgA	1984
BBC Mannheim MSW Neustadt	W. Germany	57,200	3.6"HgA	1984
Kringlen MSW Linthgebiet	Switzerland	58,700	4.0"HgA	1983
BBC Mannheim, Touss Unit 2 150 MW Power Station	Iran	792,000	8.0"HgA	1983
BBC Mannheim, Touss Unit 1 150 MW Power Station	Iran	792,000	8.0"HgA	1983
Standard Messo, MSW Stapelfeld	West Germany	17,600	2.7"HgA	1982
Techn. Werke Ludwigshafen	West Germany	39,600	3.0"HgA	1982
Babcock Krauss Maffei Imperial, MSW Burgau	West Germany	26,400	6.0"HgA	1982
Widmer + Ernst	West Germany	57,900	3.7"HgA	1982

MSW Ingolstadt

Client	Location	Steam Flow lb / h	Back Pressure	Installed
B C Berlin, MSW Krefeld	West Germany	130,500	5.5"HgA	1981
Stork Boilers	Netherlands	90,200	3"HgA	1981
Goepfert + Reimer, Iserlon	West Germany	110,000	15"HgA	1980
G H, Hattingen	West Germany	71,500	5.5"Hg	1980
Cabot	West Germany	29,900	6"HgA	1979
Mura Biel	Switzerland	24,200	19.5"HgA	1978
Didier	Netherlands	4,600	10.5"HgA	1977
Widmer + Ernst, Hamburg	West Germany	178,200	3.6"HgA	1976
SSK v. Schaewen	West Germany	17,800	30"HgA	1976
City of Frankfurt	West Germany	52,800	15"HgA	1976
B A S F, Antwerpen	Belgium	19,100	27"HgA	1976
DuPont	West Germany	4,400	30"HgA	1976
Borsig, Ruhrgas	West Germany	118,800	6.6"HgA	1975
Stadt Bremerhaven	West Germany	176,000	14"HgA	1975
Krupp	Poland	44,000	24"HgA	1974
DuPont	West Germany	7,300	30"HgA	1974
V K W, Goppingen	West Germany	92,400	4.5"HgA	1974
DuPont	West Germany	6,400	30"HgA	1972
AEG-Kanis Turbines	West Germany	110,000	4"HgA	1972
Hamburg				
G H, Rottka	West Germany	44,000	3"HgA	1971
K H D, Koln	West Germany	7,000	12"HgA	1971
Bechtel/Canada	Australia	79,000	6"HgA	1969
Kwinana				
Stadtwerke	West Germany	39,000	4.5"HgA	1969
Solingen				
Glanzstoff	West Germany	28,600	30"HgA	1968
Koln				
Wirus Werke	West Germany	4,600	30"HgA	1968
Saline	West Germany	700	30"HgA	1967
Ludwigshafen				
AEG-Kanis, Cabot	West Germany	55,000	23"HgA	1966
KEW/Werhohl	West Germany	22,000	33"HgA	1961

Marley Cooling Tower
A United Dominion Company

Marley: Parallel Path Wet Dry Cooling Towers (Plume Abatement Towers)

Project	Design	Fill Type	Model	Order Date	Order No
Atlantic Richfield Company Pasadena, TX	22,000 gpm 115 °-90 °-80°		65B-4-03 PPWD	1971	12-61-71
Northeast Utilities Middletown, CT	92,000 gpm 106.6 °-85 °-75 °		674-4-77 PPWD	1971	12-37-71
Eastman Kodak Windsor, Co	28,000 gpm 100.2 °-72.2 °-65.0 °		664-5-02 PPWD	1973	12-316-73
Maruzen Petrochemical Ltd. Licensee Kobe, Japan (Sinko-Pfaudler Licensee)	Licensee		642M-72-07 PPWD	1973	
Olympic Park Montreal, Quebec, Canada	8,200 gpm 103 °-85 °-76.5 °		667-3-2 OPWD	1975	MC-53013-75
J.R. Simplot Pocatello, ID	3,300 gpm 110 °-75 °-63 °		659-01 OPWD	1975	12-334-75
Eastman Kodak Rochester, NY	19,500 gpm 120 °-85 °-78 °		6615-4-02 PPWD	1975	12-331-75
P.S. New Mexico Waterflow, NM (Water Conservation Tower)	220,000 gpm 112.4 °-100 °-66 °		2@ 664-4-05(5)	1975	12-333-75
Texaco Montreal, Quebec, Canada	33,760 gpm 111.1 °-85 °-75 °		6710-0-04 OPWD	1978	X-54342-78
Dome Petroleum Sarnia, Ontario, Canada	9,000 gpm 93 °-80 °-73 °		659-0-02 OPWD	1980	X-54605-80
Dupont (Special design to condition dilution air for a water scrubber tower)	1,200 gpm 91.3 °-81.5 °-70 °		666-4-01 OPWD	1983	28-52403-83
Philadelphia Museum of Art Philadelphia, PA	42,000 gpm 95 °-85 °-78 °		2@MS1-1882 PPWD	1988	12-826-88
Selkirk II Cogeneration Selkirk, NY	90,000 gpm 114.3 °-90 °-74 °	MX75 Fill	12236-28-09	1993	035082
Chicago O'Hare Airport Chicago, IL	24,000 gpm 95 °-85 °-79 °		83029-6.0-4 PPWD	1994	04444

Project	Design	Fill Type	Model	Order Date	Order No
Formosa Heavy Industries	70,450 gpm	MC67 Fill	499-5.0-4	1994	046494
(Steam Coils) Taipei, Taiwan	111.2 ^{o} -91.4 ^{o} -84.2 ^{o}		PPWD		
Teesside Power Plant Teesside, England	124,856 gpm 105 ^{o} -75.0 ^{o} -41 ^{o}	MC75 Fill	486-4.0-11 PPWD	1994	066313
Foxwoods Casino Mashantucket, CT	42,000 gpm 95.2 ^{o} -83 ^{o} -78 ^{o}	MC75 Fill	84545-6.0-4 PPWD	1996	102078
Archer Daniels Midland Decatur, IL	10,000 GPM 110 ^{o} -85 ^{o} -78 ^{o}		1 Cell Retrofit ClearFlow	11/97	121895
ICI Billingham United Kingdom	205M^3/HR 38 ^{o} C-23 ^{o} C-17.2 ^{o} C		EW433-3.0-01 ClearFlow	11/97	37-00001
CIBA Speciality Chemicals United Kingdom	750M/HR 38 ^{o} C-23 ^{o} C-17.2 ^{o} C		EW433-3.0-01 ClearFlow	11/97	37-00001
Archer Daniels Midland Decatur, IL	90,000 GPM 110 ^{o} -85 ^{o} -78 ^{o}		9 Cell Retofit ClearFlow	01/98	129667
Archer Daniels Midland United Kingdom	205M^3/HR 35 ^{o} C-21 ^{o} C-15.6 ^{o} C		EW454-5.0-01 ClearFlow	01/98	37-00002
Formosa Plastics Point Comfort, TX	80,000 GPM 105 ^{o} -93 ^{o} -83 ^{o}		488-5.4-04 ClearFlow	3/99	144177
BASF Monaca, PA	17,000 GPM 99 ^{o} -85 ^{o} -75 ^{o}		F445-4.0-03 ClearFlow	5/99	147494

Thermal and Energy
Technology Division

GEA Power Cooling Systems, Inc.

610 West Ash Street, 17th Floor
San Diego, CA 92101
Tel.: (619) 232-7200
Fax: (619) 232-7177

GEA: Direct Air Cooled Condenser Installations

STATION OWNER (A/E)	SIZE [Mw(e), Steam-Side Only]	STEAM FLOW [Lb/Hr]	TURBINE BP [IN HgA]	DESIGN TEMP. [deg. F]	YEAR	REMARKS
Neil Simpson I Station **Black Hills Power & Light Co.** Gillette, WY (Stearns Roger)	20	167,550	4.5	75	1968	Coal Fired Plant
Norton P. Potter Gen. Station **Braintree Electric Light Dept.** Braintree, MA (R. W. Beck)	20	190,000	3.5	50	1975	Combined Cycle
Benecia Refinery **Exxon Company, U.S.A.** Benecia, CA	NA	48,950	9.5	100	1975	
Wyodak Station **Black Hills Power & Light Co.** **and Pacific Power & Light Co.** Gillette, WY (Stone & Webster)	330	1,884,800	6.0	66	1977	Coal Fired Plant
Beluga Unit No. 8 **Chugach Electric Assoc., Inc.** Beluga, AK (Burns & Roe)	65	478,400	5.6	35	1979	Combined Cycle
Gerber Cogeneration Plant **Pacific Gas & Electric** Gerber, CA (Mechanical Technology Inc.)	3.7	52,030	2.03	48	1981	Combined Cycle Cogeneration
NAS North Island Cogen Plant **Sithe Energies, Inc.** Coronado, CA	4.0	65,000	5.0	70	1984	Combined Cycle Cogeneration (Supplied & Erected)
NTC Cogen Plant **Sithe Energies, Inc.** San Diego, CA	2.6	40,000	5.0	70	1984	Combined Cycle Cogeneration (Supplied & Erected)
Chinese Station **Pacific Ultrapower** China Camp, CA (Ultrasystems Eng. & Const.)	22.4	181,880	6.0	97	1984	Waste Wood
Dutchess County RRF Poughkeepsie, NY (Pennsylvannia Engineering)	7.5	50,340	4.0	79	1985	WTE

Sherman Station **Wheelabrator Sherman Energy Co.** Sherman Station, ME (Atlantic Gulf)	20	125,450	2.0	43	1985	Waste Wood
Olmsted County WTE Facility Rochester, MN (HDR Techserv)	1	42,000	5.5	80	1985	WTE
Chicago Northwest WTE Facility **City of Chicago** Chicago, IL	1	42,000	15 PSIG	90	1986	WTE
SEMASS WTE Facility **American Ref-Fuel** Rochester, MA (Bechtel, Inc.)	54	407,500	3.5	59	1986	WTE (Converted to PAC System™ in 1999)
Haverhill Resource Rec. Facility **Ogden Martin Sys. of Haverhill** Haverhill, MA (Stone & Webster)	46.9	351,830	5.0	85	1987	WTE
Hazelton Cogeneration Facility **Continental Energy Associates** Hazelton, PA (Brown Boveri Energy Systems)	67.5	420,000	3.7	47	1987	Combined Cycle Cogeneration (Supplied & Erected)
Grumman **TBG Cogen** Bethpage, NY (General Electric)	13	105,700	5.4	59	1988	Combined Cycle Cogeneration (Converted to PAC System™ in 1997)
Cochrane Station **Northland Power** Cochrane, Ontario, Canada (Volcano, Inc.)	10.5	90,000	3.0	60	1988	Combined Cycle Cogeneration
North Branch Power Station **Energy America Southeast** North Branch, WV (Fru-Con Construction Corp.)	80	622,000	7.0	90	1989	Coal Fired Plant
Sayreville Cogen Project **Intercontinental Energy Co.** Sayreville, NJ (Westinghouse Electric Corp.)	100	714,900	3.0	59	1989	Combined Cycle Cogeneration
Bellingham Cogen Project **Intercontinental Energy Co.** Bellingham, MA (Westinghouse Electric Corp.)	100	714,900	3.0	59	1989	Combined Cycle Cogeneration
Spokane Resource Rec. Facility **Wheelabrator Spokane Inc.** Spokane, WA (Clark-Kenith Inc.)	26	153,950	2.0	47	1989	WTE (Supplied & Erected)
Exeter Energy L. P. Project **Oxford Energy** Sterling, CT	30	196,000	2.9	75	1989	PAC System™
Peel Energy From Waste **Peel Resources Recovery, Inc.** Brampton, Ontario, Canada (SNC Services, Ltd.)	10	88,750	4.5	68	1990	WTE

Nipigon Power Plant **Transcanada Pipelines** Nipigon, Ontario, Canada (SNC Services, Ltd.)	15	169,000	3.0	59	1990	Combined Cycle Cogeneration
Linden Cogeneration Project **Cogen Technologies, Inc.** Linden, NJ (Ebasco Constructors, Inc.)	285	1,911,000	2.44	54	1990	Combined Cycle Cogeneration
Maalaea Unit #15 **Maui Electric Company, Ltd.** Maui, Hawaii (Stone & Webster)	20	158,250	6.0	95	1990	Combined Cycle
Norcon - Welsh Plant **Falcon Seaboard** North East, PA (Zurn/Nepco, Inc.)	20	150,000	2.5	55	1990	Combined Cycle Cogeneration
University of Alaska, Fairbanks Fairbanks, AK	10	46,000	6.0	82	1991	Combined Cycle Cogeneration
Union County RRF **Ogden Martins Sys. of Union County** Union, NJ (Stone & Webster)	50	357,000	8.0	94	1991	WTE (Supplied & Erected)
Saranac Energy Plant **Falcon Seaboard** Saranac, NY (Zurn/Nepco, Inc.)	80	736,800	5.0	90	1992	Combined Cycle Cogeneration
Onondaga County RRF **Ogden Martins Sys. of Onondaga Co.** Onondaga, NY (Stone & Webster)	50	258,000	3.0	70	1992	WTE (Supplied & Erected)
Neil Simpson II Station **Black Hills Power & Light Co.** Gillette, WY (Black & Veatch)	80	548,200	6.0	66	1992	Coal Fired Plant (Supplied & Erected)
Gordonsville Plant **Mission Energy** Gordonsville, VA (Ebasco Constructors Inc.)	2 x 50	2 x 349,150	6.0	90	1993	Combined Cycle
Dutchess County RRF Expansion Poughkeepsie, NY (Westinghouse Electric / RESD)	15	+ 49,660	5.0	79	1993	WTE
Samalayuca II Power Station **Comision Federal de Electricidad** Samalayuca, Mexico (Bechtel Corporation)	210	1,296,900	7.0	99	1993	Combined Cycle
Potter Station **Potter Station Power Limited** Potter, Ontario (Monenco/Bluebird)	20	181,880	3.8	66	1993	Combined Cycle

Streeter Generating Station **Municipal Electric Utility** **City of Cedar Falls, Iowa** Cedar Falls, Iowa (Stanley Consultants)	40	246,000	3.5	50	1993	PAC System™ (Supplied & Erected)
MacArthur Resource Rec. Facility **Islip Resource Recovery Agency** Ronkonkoma, New York (Montenay Islip Inc.)	11	40,000	4.8	79	1993	WTE (Supplied & Erected)
North Bay Plant **Transcanada Pipelines** North Bay, Ontario, Canada	30	245,000	2.0	53.6	1994	Combined Cycle
Kapuskasing Plant **Transcanada Pipelines** Kapuskasing, Ontario, Canada	30	245,000	2.0	53.6	1994	Combined Cycle
Haverhill RRF Expansion **Ogden Martin Sys. of Haverhill** Haverhill, MA	46.9	+44,500	5.0	85	1994	WTE
Arbor Hills Landfill Gas Facility **Browning-Ferris Gas Services Inc.** Northville, MI (European Gas Turbines Inc.)	9	87,390	3.0	50	1994	Combined Cycle
Pine Bend Landfill Gas Facility **Browning-Ferris Gas Services Inc.** Eden Prairie, MN (European Gas Turbines Inc.)	6	58,260	3.0	50	1994	Combined Cycle
Pine Creek Power Station **Energy Developments Ltd.** Pine Creek, Northern Territory, Australia (Davy John Brown Pty. Ltd.)	10	95,300	3.63	77	1994	Combined Cycle
Cabo Negro Plant **Methanex Chile Limited** Punta Arenas, Chile (John Brown)	6	74,540	4.0	63	1995	Methanol Plant
Esmeraldas Refinery **Petro Industrial** Esmeraldas, Ecuador (Tecnicas Reunidas, S. A.)	15	123,215	4.5	87.3	1995	Combined Cycle
Mallard Lake Landfill Gas Facility **Browning-Ferris Gas Services Inc.** Hanover Park, IL (Bibb & Associates Inc.)	9	101,400	3.0	49	1996	Combined Cycle
Riyadh Power Plant #9 **SCECO** Riyadh, Saudi Arabia (Raytheon Engrs. & Const., Inc.)	4 x 107	4 x 966,750	16.5	122	1996	Combined Cycle (1200 MW Total)

Barry CHP Project **AES Electric Ltd.** Barry, South Wales, UK (TBV Power Ltd.)	100	596,900	3.0	50	1996	Combined Cycle
Zorlu Enerji Project **KORTEKS** Bursa, Turkey (Stewart & Stevenson International)	10	83,775	3.5	59	1997	Combined Cycle
Tucuman Power Station **Pluspetrol Energy, S.A.** El Bracho, Tucuman, Argentina (Black & Veatch International)	150	1,150,000	5.0	99	1997	PAC System™
Dighton Power Project **Dighton Power Associates, Ltd.** Dighton , MA (Parsons Power Group, Inc.)	60	442,141	5.5	90	1997	Combined Cycle
El Dorado Energy **El Dorado LLC** Boulder, NV (Kiewit/Sargent & Lundy)	150	1,065,429	2.5	67	1998	Combined Cycle
Tiverton Power Project **Tiverton Power Associates, Ltd.** Tiverton, RI (Stone & Webster Engineering Corp.)	80	549,999	5.0	90	1998	Combined Cycle
Coryton Energy Project **Intergen** Corringham, England (Bechtel Power Corporation)	250	1,637,312	2.5	50	1998	Combined Cycle
Rumford Power Project ***Rumford Power Associates, Ltd.*** Rumford, ME (Stone & Webster Engineering Corp)	80	545,800	5.0	90	1998	Combined Cycle
Millmerran Power Project **Intergen / Shell Coal** *Toowoomba, Queensland, Australia* (Bechtel International)	2 x 420	2 x 2,050,000	5.43	88	1999	Coal Fired
Bajio Power Project **Intergen** *Quertetaro, Guananjuato, Mexico* (Bechtel International)	150	1,307,000	3.54	71.4	1999	Combined Cycle
University of Alberta **University of Alberta** *Edmonton, Alberta, Canada* *(Sandwell)*	25	277,780	9.15	59	1999	Gas Fired Cogeneration
Monterrey Cogeneration Project **Enron Energra Industrial de Mexico** Monterrey, Mexico (Kawasaki Heavy Industries)	80	671,970	5.8	102	2000	Combined Cycle Cogeneration

Gelugor Power Station **Tenaga Nasional Berhad (TNB)** *Penang, Malaysia* (Kawasaki Heavy Industry)	120	946,600	6.8	89.6	2000	Combine Cycle Cogeneration
Front Range Power Project *Fountain, Colorado* (TIC/UE Front Range JV)	150	1,266,477	3.57	80	2000	Combine Cycle
Goldendale Energy Project **Goldendale Energy Inc.** *Goldendale, Washington* (NEPCO)	110	678,000	4.5	90	2000	Combine Cycle PAC System
Athens Power Station **PG & E Generating** *Athens, New York* (Bechtel Power)	3 x 120	3 x 749,183	5	90	2000	Combined Cycle
Moapa Energy Facility **Duke Energy Moapa, LLC** *Clark County, Nevada* (Duke/Fluor Daniel)	2 x 200	2 x 1,718,790	6.25	103	2001	Combined Cycle (1200 MW Total)
Wygen 1, Unit 3 Power Project **Black Hills Generation, Inc.** Gillette, Wyoming (Babcock & Wilcox)	80	548,200	6.0	66	2001	Coal Fired Plant
Hunterstown Power Project **Reliant Energy** *Hunterstown, Pennsylvania* (Black & Veatch)	350	1,690,000	4.6	90	2001	Combined Cycle (890 MW Total) Low Noise (51 dBA @ 400 ft)
Choctaw County Power Project **Reliant Energy** French Camp, Mississippi (Black & Veatch)	350	1,690,000	4.6	90	2001	Combined Cycle (890 MW Total)
Otay Mesa Energy Center **Calpine** San Diego, California (Duke/Fluor Daniel)	277	1,501,332	3.47	74	2001	Combined Cycle

Hamon: Direct Air Cooled Condenser Installations

RECENT MAJOR REFERENCES POWER & INDUSTRY
AIR COOLED STEAM CONDENSER (DRY COOLING)

320 ACC > 10000 MW SINCE 1962
45 SRC > 9000 MW SINCE 1991
28 COUNTRIES - AIR T° -62°C UP TO +50°C

PLANT Mwe	CLIENT		TYPE OF PLANT	Condenser MWth	START-UP	TUBE TYPE
450	NYPA / GE / SARGENT & LUNDY / POLETTI	US	COMB-CYCLE	316	2004	SRC
350	TRANSALTA / DELTA HUDSON / CHIHUAHUA III	US	COMB-CYCLE	205	2004	SRC
650	RELIANT / SARGENT & LUNDY / ARROW CANYON (1)	US	COMB-CYCLE	300	2003	SRC
650	RELIANT / SARGENT & LUNDY / BIG HORN (1)	US	COMB-CYCLE	300	2003	SRC
360	YALOVA / AK ENERJI / SKODA (2)	TR	COMB-CYCLE	2 X 120	2002	SRC
1600	SITHE / RAYTHEON / MYSTIC	US	COMB-CYCLE	830	2002	SRC
800	SITHE / RAYTHEON / FORE RIVER	US	COMB-CYCLE	415	2002	SRC
600	BECHTEL / HSIN TAO	TW	COMB-CYCLE	394	2001	SRC
470	CALPINE / BECHTEL / SUTTER	US	COMB-CYCLE	331	2001	SRC
350	ELECTRABEL / ALSTOM / ESCH-S-ALZETTE	LUX	COMB-CYCLE	220	2001	SRC
220	EDF / SALTILLO	MEX	COMB-CYCLE	147	2001	SRC
160	HYUNDAI E.C. / BARIA	VN	COMB-CYCLE	141	2001	SRC
130	EDISON / JESI	IT	COMB-CYCLE	115	2001	SRC
15	THUMAIDE / CNIM IPALLE	BE	MUN. SOLID WASTE	42	2001	SRC
780	ENTERGY / MITSUBISHI / DAMHEAD CREEK	GB	COMB-CYCLE	453	2000	SRC
55	ZORLU ENERJI / BURSA	TR	COMB-CYCLE	35	2000	SRC
12	EPR SCOTLAND / ABENGOA / WESTFIELD	UK	MUN. SOLID WASTE	25	2000	SRC
780	ENRON / STONE&WEBSTER / SUTTON BRIDGE	GB	COMB-CYCLE	443	1999	SRC
75	ANACONDA / ABB POWER / MURRIN MURRIN	AU	COMB-CYCLE	2 X 59	1998	SRC
350	ELECTRABEL / GEC ALSTHOM / BAUDOUR	BE	COMB-CYCLE	214	1998	SRC
132	SONDEL / CELANO	IT	COMB-CYCLE	115	1998	SRC
14	LENTJES ENERGIETECHNIK / WÜRZBURG	DE	MUN. SOLID WASTE	32	1998	SRC
460	ELECTRABEL-SPE / TBL / BRUGGE	BE	COMB-CYCLE	348	1997	SRC
350	ELECTRABEL-SPE / TBL / GENT	BE	COMB-CYCLE	222	1997	SRC
330	MITSUBISHI TAKASAGO	JP	COMB-CYCLE	210	1997	SRC
150	CENTRO ENERGIA / FWI / TEVEROLA	IT	COMB-CYCLE	122	1997	SRC
130	FIAT AVIO / COASTAL HABIBULLAH / QUETTA	PK	COMB-CYCLE	88	1997	SRC
105	KEPCO / HALIM	KR	COMB-CYCLE	82	1997	SRC
32	LINDE / NOVI URENGOY	RU	CHEMICAL	73	1997	SRC
31	LINDE / NOVI URENGOY	RU	CHEMICAL	70	1997	SRC
25	SEGHERS / INDAVER	BE	MUN. SOLID WASTE	56	1997	SRC
20	TUNTEX	TW	MUN. SOLID WASTE	55	1997	SRC
45	ABB TURBINEN / FRANKFURT-ODER	DE	CCPP-URB. HEATING	30	1997	SRC
8	LINDE / BASF LUDWIGSHAFEN	DE	CHEMICAL	11	1997	SRC
	EST-GEKO / HKW MEUSELWITZ	DE	URBAN HEATING	11	1997	SRC
360	SIEMENS / EAST. ELECT. / KING'S LYNN	UK	COMB-CYCLE	204	1996	SRC
150	CENTRO ENERGIA / FWI / COMUNANZA	IT	COMB-CYCLE	122	1996	SRC
54	ML-GAVI WIJSTER	NL	MUN. SOLID WASTE	100	1996	SRC
8	RMZ / HOUTHALEN	BE	MUN. SOLID WASTE	16	1996	SRC
3	SCHWÖRERHAUS / HOHENSTEIN	DE	URBAN HEATING	8	1996	SRC
1	NSC / IISUKA CITY	JP	MUN. SOLID WASTE	6	1996	SRC
670	MITSUBISHI / JANDAR	SY	COMB-CYCLE	500	1995	SRC
130	EDISON / SAN QUIRICO	IT	COMB-CYCLE	112	1995	SRC
70	ABB STAL / GAS EDON ERICA	NL	COMB-CYCLE	52	1995	SRC
70	ABB STAL / GAS EDON KLAZIENAVEEN	NL	COMB-CYCLE	52	1995	SRC
75	ABB TURBINE / FICHTNER / GERA-NORD	DE	URBAN HEATING	30	1995	SRC
30	ABB STAL / PGEM / BORCULO	NL	COMB-CYCLE	24	1995	SRC
10	ABB KESSELANLAGEN / KEZO / HINWIL	CH	MUN. SOLID WASTE	20	1995	SRC
20	BLOHM & VOSS / SCHWERIN	DE	URBAN HEATING	47	1994	SRC
130	ABB / PPC / CHANIA	GR	COMB-CYCLE	100	1993	SRC
35	OGDEN / MARIN / HUNTINGTON	USA	MUN. SOLID WASTE	64	1991	SRC
465	VARIOUS		27 POWER PLANTS	575	1962/94	ROUND/SRC

SRC = single row condenser
(1) letter of intent - order June 2001
(2) letter of intent - order before September 2001

Appendix C
MATERIALS FROM DRY COOLING SYSTEM OWNER/OPERATORS

Interview Meetings

- **Crockett Co-Generation Plant**, ESOCO Crockett, Inc., 550 Loring Avenue, Crockett, CA, 94525, (510) 787-4100 (main).
- **El Dorado Energy,** El Dorado Energy LLC, 701 Eldorado Valley Drive, P.O. Box 62470, Boulder City, NV 89006-2470, 702-568-8206 (main), 702-568-8213 (fax),
- **Calpine**, 620 Coolidge Drive, Suite 200, Folsom, CA 95630, 916-608-3800 (main), 916-985-5655 (fax)

Crockett Co-Generation Plant

Site Visit Report (June 12, 2000)

Personnel: Peter H. So (host), Plant Engineer, 510-787-4105, 510-787-4150 (FAX), peteso@crockettcogen.com

Owners: NRG Energy (owns 138.4 of 240Mw) and others. Subsidiary of Northern States Power Company, 1221 Nicollet Mall, Suite 700, Minneapolis, MN, 55403-2445, 612-373-5300 (main), 800-241-4674 (toll free), 612-373-5312 (fax).

Constructor: Bechtel Power Company

Plant description:

- Gas-fired co-generation units
- 240 Mwe (~2/3 gas turbines; 1/3 steam turbine)
- Turbine Generator: General Electric (Frame 7H turbines)
- Heat Recovery Steam Generator: Vogt
- Plant went commercial in 1996.
- Provides steam to C&H Sugar on neighboring site:
- 450 psi steam to 3 auxiliary boilers (50 – 350 kph; nominal 250 kph)

- Approx. 2/3 of condensate returned; discarded if it contains detectable TOC; otherwise, polished and reused
- Electric to PG&E
- Gas/steam turbines on single shaft.
- Can augment HRSG with 20 Mw duct burner
- Steam augmentation to gas turbines of ~20 Mw.

Air-cooled condenser: (See attached sketch and spec. sheet end of section)

Balcke-Duerr, 405 N. Reo Street, Suite 300, Tampa, FL 33609, 813-289-1516 (main). Contact: Ralph Wyndrum, 813-342-4916 (direct), 813-342-7916 (fax).

ACC Model No.: 120-9310-8810-2---4-TR

Dry cooling was chosen for two primary reasons:

1. space considerations (plant on 2 ½ acre site)
2. regulatory schedule (estimated 5 years to get permit to use Bay water)

A general description of the system includes:

- 15 cells; 12 for steam condesaton/3 for plant water cooling (lube oil/bearings/etc)
- Cells in three parallel banks of 5 cells (4 for condenser) each. One cell in each bank in reflux configuration for non-condensable removal. (See sketch 1.)
- All heat exchanger, fan and other descriptive information in attached spec. sheet.

Operating issues include**:**

- Unit runs well
- No problems with wind effects on performance
- Freeze protection has not been a problem but the cold weather operating controls sometimes act strangely; i.e., fans will turn off and operators don't know why. Apparent conflict between Balcke-Duerr operating system and the plant Foxboro I&C package---not a serious problem.
- No air in-leakage problems; vacuum systems/de-aerators work well.
- Water chemistry not difficult to control (HRSG at pH = 9 with amine treatment).
- High make-up rates because of condensate not being returned by C&H or having to be discarded.
- Results in complete turnover of cycle water every 2 or 3 days.

Air-side cleaning issues include:

- Frequency—manufacturer's recommendation "at least annually"; last cleaning interval was 18 months.
- We got somewhat conflicting stories about the need for and effect of cleaning. The main contaminant is sugar dust from loading/unloading activities at C&H. It coats the fins and biological activity creates a tough black layer.
- So felt that the fouling was not obvious and there was no particular change in appearance as a result of cleaning. Others said an adherent black layer came off leaving the tubes a light gray.
- So felt there was no particular improvement in performance as a result of the cleaning. Others reported that the automatic controls had cut some of the fans back from full to half speed at similar conditions after cleaning.
- Wash water is discharged to straits.

There is no systematic monitoring of the ACC performance. Since the gas turbines and the steam turbines are on a single shaft, there is no way to know how the two parts contribute to the output. However, Crockett does keep data records of ambient temperature, steam flow and cold water temperature leaving the ACC. It would seem to be possible to extract at least average performance information over some period of time and generate cleanliness factors for the unit.

The general estimate was that very hot days cost the company 3 to 5 Mw on the steam side at "7 or 8" on the gas turbine side.

Noise is not much of a problem. Limits are < 50db at the nearest residence and at 300 yards into the Bay. There is apparently no trouble in maintaining these levels. The noise on the fan deck was not excessive—comfortable to walk around.

Corrosion issues include:

- No known corrosion problems on the tubes or fins, at the fin-tube contacts points or at the tube header connections. The use de-mineralized water to wash and do not use external sprays for hot weather performance enhancement.
- Crockett is considering installing sprays (talking with MeeSpray Humidification Division (See "Other Contacts") and recognize that there might be corrosion/scaling problems if not operated properly. There are examples of spray enhanced units locally (Shell Martinez and TOSCO) See "Other Contacts."

Worker safety issues include:

- Some problems with knowing when particular fans are locked out or just "off".
- Gear box oil is awkward to change, requires workers to hang through the fan blades.
- Can get too hot on discharge side of exchangers—must limit working time to approximately 15 minutes.

Costs issues include:

- No information on capital costs.
- Operating costs (15 fans each 150 Hp; draws about 2 Mw, biggest power requirement at the plant).
- Maintenance costs are not separately accounted for by units. They have a sponsorship program where individual technicians/operator "adopt" a particular component and become the "plant expert" on it.

The plant is run with 26 employees. Each shift has 2 operators and 1 supervisor per shift (3 x 4 = 12). The day shift has 2 mechanical and 3 I/C technicians (2 + 3 = 5).

Other contacts include:

Jan Polawachek,Foster-Wheeler, TOSCO (air-cooled process cooler with spray enhancement).

Janet Opio,air-cooled unit.

David J. Ayres, Manger, Utilities Department, Martinez Refining Co., Division of Equilon Enterprises, LLC, P.O. Box 711, Martinez, CA 94553-0071, 925-313-3378 (direct), 925-313-3059 (fax), djayres@equilon.com.

Tae H. Lee, Sales, Mee Industries (spray enhancement for gas turbine inlets and air-cooled equipment), 204 West Pomona Avenue, Monrovia, CA 91016, 626-359-4550 (direct), 626-359-4660 (fax), lee3873@aol.com.

Humidification Division, 1651 Katy Lane, Fort Mill, SC 29715, 803-547-2380 (main), 803-547-2379 (fax), www.meefog.thomasregister.com.

System Description: Crockett Co-generation Project, Air-Cooled Condenser

1.0 The Air-Cooled Condenser System

1.1 Function of an Air-Cooled Condenser

The Air-Cooled Condenser (ACC) serves to condense steam from the steam turbine exhaust and/or the steam turbine bypass system. As the name implies, ambient air is used as the cooling medium. The exhaust steam is condensed within the finned bundles by the heat transfer with the air passing over the external surface of the finned tube bundles. The air absorbs the steam's latent heat of condensation and is heated in the process. The steam is condensed and is then returned to the ACC condensate tank to be re-used as boiler feedwater .

1.2 General Description

Turbine exhaust or bypass steam is supplied to the condensing elements through dual 96"0 exhaust steam duct and three(3) 75"0 distribution manifolds. Each 96"0 steam duct and the 96"0 riser distribution manifold includes a drain pot to remove any condensate developed in the ducting during start-up and normal operation.

The finned tube elements are arranged in a triple-roof configuration. Each roof consists of four(4) ACC modules. Each module contains one(l) fan and ten(10) finned-tube bundles. The complete ACC unit consists of 120 (12 x 10) finned-tube bundles. Each finned-tube bundle consists of200, single pass, CS-HDG oval finned tubes. The tubes are arranged in three parallel rows between tube sheets/headers.

The finned-tube bundles are arranged in two roofs and assembled on a 341-6" high steel support structure. The ACC is located on the roof of the turbine building. The turbine exhaust steam duct exits the turbine from dual side exhausts, is routed to the ACC riser manifold and divides into three(3) duct risers outside of the turbine building.

The steam distribution manifolds at the apex of the tube bundles directs the steam to finned-tube bundles in the nine(9) condenser modules. The nine(9) condenser modules function as parallel flow condensers. The steam and condensate flow downward, in the same direction.

The remaining steam flows from the bottom of the condenser modules to the three(3) reflux modules where it is condensed in a counter-flow mode. Counter-flow, or Reflux, condensation consists of upward flowing steam and downward flowing condensate. The counter-flow mechanism permits the warm steam to be in contact with the condensate, thus minimizing sub-cooling and the risk of freezing. Figure 1 on the following page illustrates the arrangement of the parallel flow and counter flow modules.

Condensate from the ACC is collected in the 18"0 condensate collection headers and is gravity drained to the ACC condensate hotwell tank.

Non-condensibles are removed from the ACC at the top of the reflux modules and are evacuated by the Steam Jet Air Ejector System (SJAE). The evacuation system consists of one single-stage hogging (start-up) ejector and 2 x 100% two-stage steam holding (operating) ejectors. Motive steam is utilized to operate the SJAE system.

Cooling air is delivered through the finned tube bundles by ultra-low noise, forced draft axial fans located at the base of the ACC bundle delta. The fans are driven by two-speed electric motors and parallel shaft gearboxes.

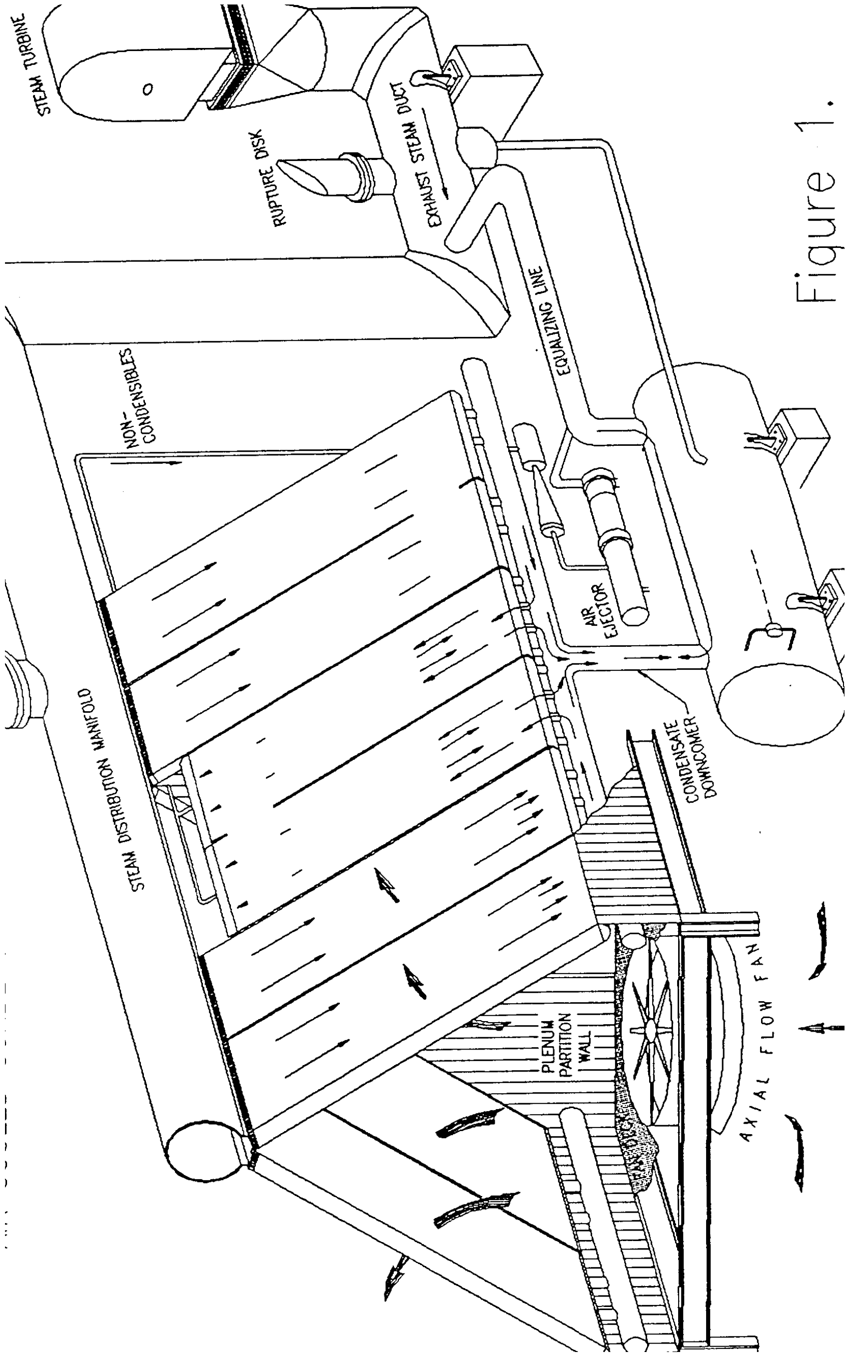

Figure C-1
Air-Cooled Condenser, Crockett Co-generation Project

1.3 Air-Cooled Condenser Design Data

1.3.1 Site Meteorological Data:

Site Elevation above MSL	10

	Ambient Air Temperature, ºF	105 to 35
1.3.2	ACC Design Data	**(Design Case Cl)**
	Turbine Exhaust Steam flow, lb./hr.	320,000
	Turbine Exhaust Steam Pressure, in.HgA	2.04
	Exhaust Steam Moisture, %	92.2
	Net Heat Load, MM Btu/hr	305.7
	Design Temperature, ºF	250
	Design Pressure, psig	FV to 14.9
1.3.3	ACC Guarantee Data	
	Turbine Exhaust Steam flow, lb./hr.	320,700
	Turbine Exhaust Steam Pressure, in.HgA	2.04
	Turbine Exhaust Steam Enthalpy, BTU/lb.	1,025
Air Temperature, ºF		Inlet Ambient 65
	Fan Power @ Motor Terminals, kW	1,290

BECHTEL P.O. № 22311-M-004
BALCKE-DÜRR, INC. JOB № ACC-3007

1.4 Air-Cooled Condenser System Component Data

The following technical information is included for the ACC and the furnished equipment. Additional component details can be found elsewhere in this manual.

1.4.1 General

1.4.1.1	Manufacturer :	BALCKE-DÜRR, INC 405 N. Reo St. Suite 300 Tampa, FL 33609 813-289-1516
1.4.1.2	BDI Contract № :	ACC - 3007
1.4.1.3	ACC Model № :	120 - 9310/8810 - 200 - 4 - TR

1.4.2 Mechanical Equipment

1.4.2.1 **FANS**

A.	Manufacturer :	Alpina Equipmentos
B.	Diameter, ft. :	29
C.	Fan Speed, rpm :	63 / 32.5
D.	Number of blades :	5
E.	Tip speed, fpm :	5,740
F.	Axial Thrust, lb. :	C Fans = 1,649 R Fans = 1,595
G.	Blade material :	Fiberglass/Epoxy
H.	Fan horsepower, BHP :	C Fans = 124 R Fans = 113
I.	Total static pressure, in. H_20 :	C Fans = 0.347 R Fans = 0.347
L.	Air delivery per fan, ACFM	C Fans = 1,060,883 R Fans = 1,003,887
M.	Blade pitch angle, ° :	C Fans = 30.0 R Fans = 28.0

1.4.2.2 **GEARBOXES**

A.	Type :	Helical, Two-Stage, Parallel Shaft
B.	Manufacturer:	Flender Gear Corp.
C.	Ratio :	18.6 : 1
D.	Input Horsepower :	150
E.	Service Factor :	2.0
F.	Gear Efficiency, %:	≈ 98

6

ENERGY NATIONAL INC. CROCKETT COGENERATION PROJECT
AIR-COOLED CONDENSER
BECHTEL P.O. Nº 22311-M-004
BALCKE-DÜRR, INC. JOB Nº ACC-3007

1.4.2.3 **MOTORS**

A.	Type:	447TC - TEF(
B.	Manufacturer:	Siemens
C.	Horsepower:	150 / 37
D.	Volts/Ph/Hz:	460 / 3 / 60
E.	Full load, rpm (nominal):	1,175 / 575
F.	Service factor:	1.15
G.	Motor weight, lb.:	2,000
H.	Wr^2, lb./ft^2.:	550
I.	Maximum number of successive starts:	2 cold and 1 ho per hour

1.4.2.4 **SJAE SYSTEM**

Hogging Ejector (start-up)

A.	Evacuated Volume, $ft.^3$:	42,100
B.	Evacuation period, min.:	30
C.	Motive steam	
	Pressure, psig:	120
	Temperature,°F:	350
D.	Motive steam consumption, lb./hr:	6,350

Holding Ejector (operation)

A.	Number and capacity:	2 x 100%
B.	Number of stages:	2
C.	Air suction flow, lb./hr.:	56
D.	Total suction flow, lb./hr.:	124
E.	Motive steam	
	Pressure, psig:	120
	Temperature,°F:	350
F.	Motive Steam Consumption (per ejector train), lb/hr.:	606 + 450
G.	Inter/aftercondenser:	1 x 100%
H.	Design Conditions : I/A Condenser	
	Shell side, psig/°F:	FV&150 /300
	Tube side psig/°F:	150 / 300

ENERGY NATIONAL INC. CROCKETT COGENERATION PROJECT
AIR-COOLED CONDENSER
BECHTEL P.O. Nº 22311-M-004
BALCKE-DÜRR, INC. JOB Nº ACC-3007

1.4.2.5 **FINNED TUBE BUNDLES**

A.	Manufacturer:	Deutsche Babcock (SA)
B.	Tube Material:	Carbon steel, BS/1449 CR37/23
C.	Fin Material:	Carbon Steel, DIN/624 ST4 GBK
D.	Protective Coating:	HDG - 50 Microns
E.	Fin pitch, fins/inch:	Row 1: 7.25 Row 2-4: 8.5

1.5 **Air-Cooled Condenser Performance Curves**

The ACC performance curves are included such that the operation of the ACC can be predicted for various operating conditions. The following curves are included:

Backpressure Range, in.HgA:	2.0 to 10.0
Released Heat Range, % of Design:	50% to 190%
Inlet Air Temperature Range, °F:	30 to 100

Fan State	Nº Fans Fast	Nº Fans Slow	Nº Fans Off
25	12	0	0
23	10	2	0
21	8	4	0
19	6	6	0
17	4	8	0
15	2	10	0
13	0	12	0
11	0	10	2
9	0	8	4
7	0	6	6
5	0	4	8
3	0	2	10

The above fan states coincide with those presented in control logic diagram L2.

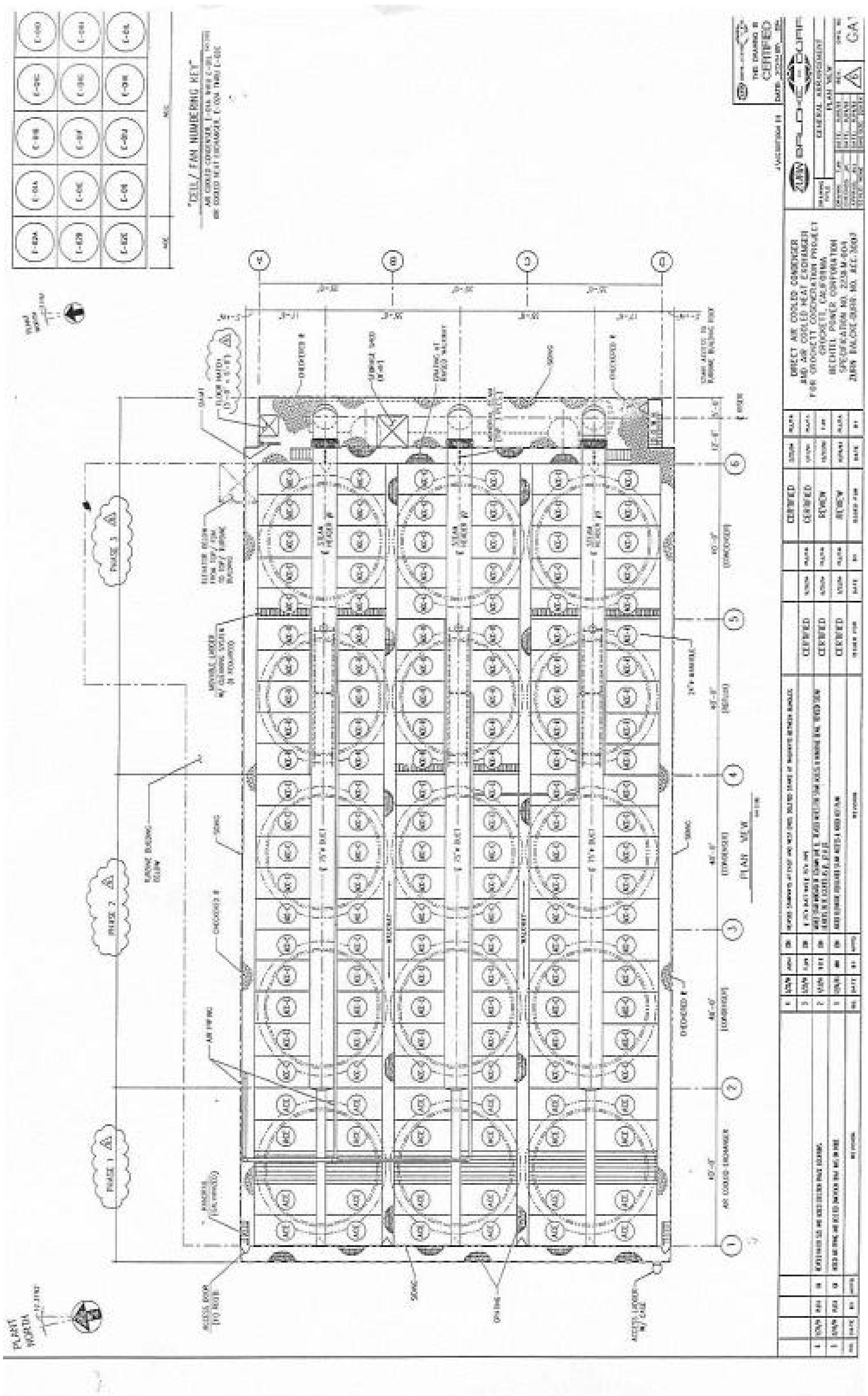

Figure C- 2
ACC Schematic (Crockett Cogeneration Project)

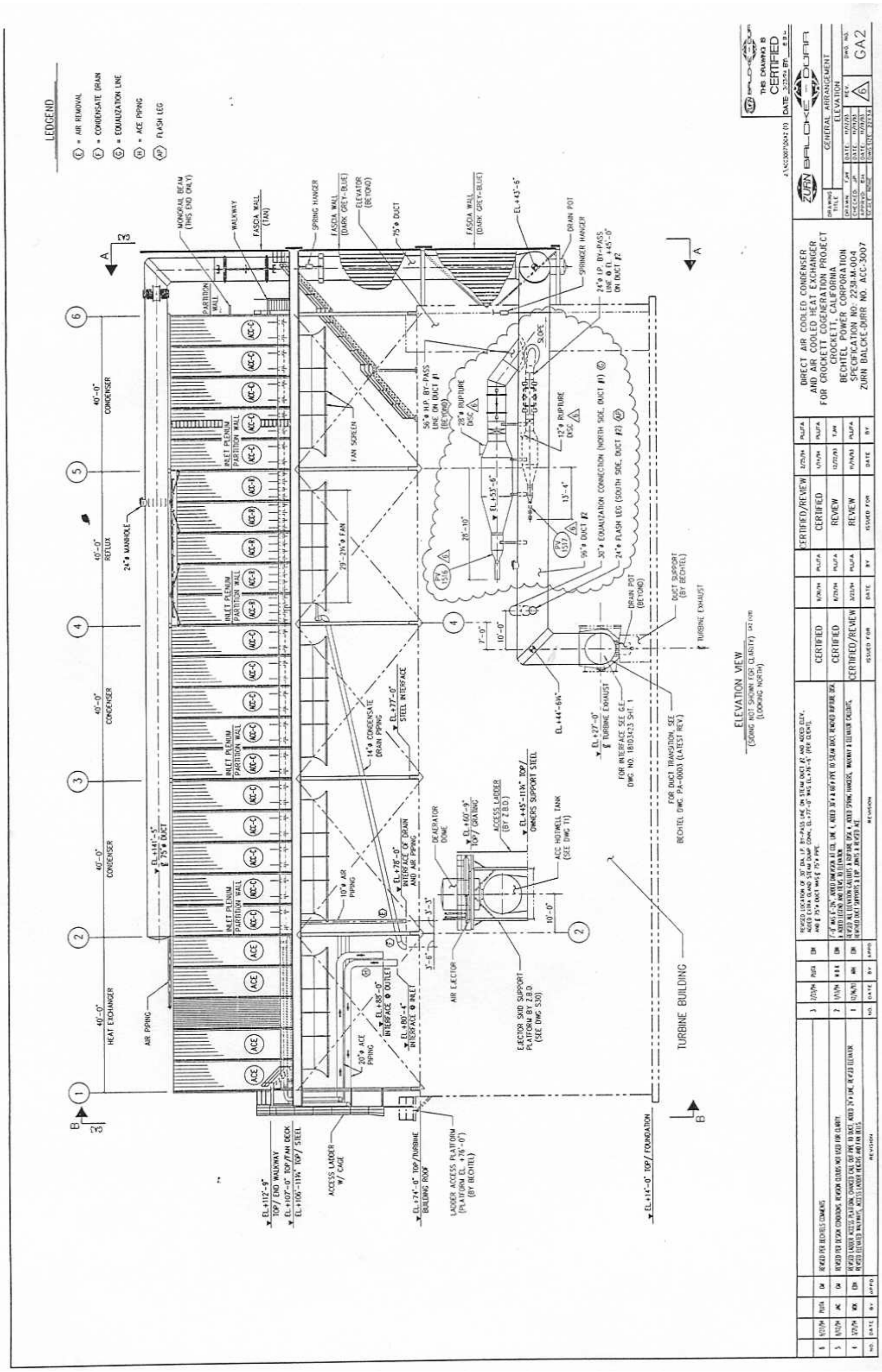

Figure C- 3
ACC Schematic (Continued)

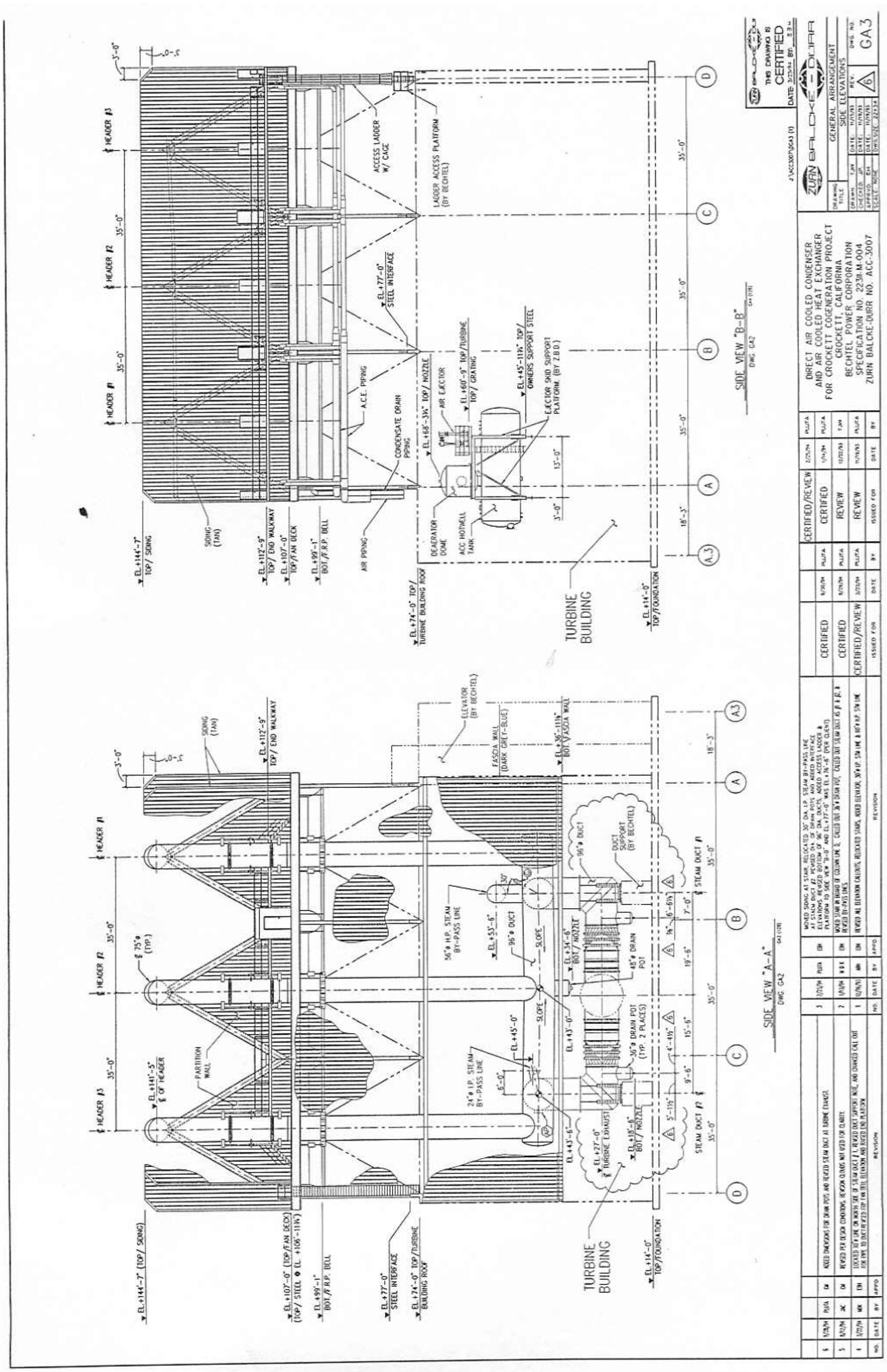

Figure C- 4
ACC Schematic (Continued)

CUSTOMER : CROCKETT COGENERATION	EQUIPMENT NAME : AIR COOLED CONDENSER
PROJECT : CROCKETT COGENERATION PROJECT	EQUIPMENT NO. : E-01A THRU L (Letter corresponding to # of cells)
LOCATION : CROCKETT, CA	
SERVICE OF UNIT: TURBINE STEAM CONDENSING	MANUFACTURER: BALCKE - DÜRR, INC
SIZE: 105' W x 160' L TYPE: ROOF TYPE	(INDUCED/FORCED) DRAFT FORCED NO. OF BAYS 12
SURFACE PER UNIT - FINNED TUBE 3,350,385	BARE TUBE 289,840 FT²
HEAT EXCHANGED 305.7 x10⁶ BTU/HR	MTD. EFF 15.1 °F NO.OF UNITS REQ'D ONE
TRANSFER RATE - FINNED TUBE 6.04 BARE TUBE, SERVICE 69.85	CLEAN BTU/HR,FT²·°F

PERFORMANCE DATA TUBE SIDE

FLUID CIRCULATED: TURBINE EXHAUST STEAM — LETHAL SERVICE ? YES / NO

TOTAL FLUID ENTERING: See Sheet 3 — 320 000 LB/HR

	IN		OUT				IN LIQUID	IN VAPOR	OUT LIQUID	OUT VAPOR
TURBINE PRESSURE			2.04	IN. HGA	DENSITY	LB/FT³				
TURBINE TEMPERATURE			101.7	°F	VISCOSITY	CP				
STEAM LB/HR	295,040				THERM.CON. BTU/HR.FT·°F					
WATER LB/HR	24,690		320,000		SPECIFIC HEAT BTU/LB·°F					
NONCOND. LB/HR,MW					SURFACE TEN. DYNE/CM					
TEMPERATURE					POUR / FREEZE POINT	°F	BUBBLE POINT			
FOULING RESIST.	0.0005			HR.FT²·°F/BTU	LATENT HEAT	BTU/LB	CRITICAL TEMPERATURE			
PRESSURE DROP ALLOW./CAL	/			PSI	INLET PRESSURE	PSIA	CRITICAL PRESSURE			PSI

PERFORMANCE DATA - AIR SIDE

AIR QUANTITY,TOTAL 57.71 x10⁶ LB/HR / ~~SCFM~~	AIR QUANTITY/FAN 1,060,767 ACFM	ACTUAL STATIC PRESSURE 0.35 IN. H₂O
ALTITUDE ABOVE SEA LEVEL 70 FT	TEMPERATURE IN (DESIGN DRY BULB) 65 °F	TEMPERATURE OUT 88.5
FACE VELOCITY 470 STD.FT/MIN	MASS VEL (NET FREE AREA) LB/HR.FT²	MIN. DESIGN AMBIENT 35

MAINTENANCE CONSIDERATIONS

DESCRIBE TYPE OF FOULING:	RESTRICTION (IF ANY)	VELOCITY: FT/S TUBE OD: IN
METHOD OF CLEANING:	FREQUENCY OF CLEANING: ANNUAL	

MECHANICAL DESIGN CONSIDERATIONS

MAX. EXPECTED OPERATING PRESSURE 5 PSIG — MAX. EXPECTED OPERATING TEMPERATURE 225

WILL UNIT BE SUBJECTED TO FREQUENT THERMAL CYCLING? YES / NO — DESCRIBE: Daily start - Full Load - Stop

WILL UNIT BE SUBJECTED TO FREQUENT PRESSURE CYCLING? YES / NO — DESCRIBE: Daily Range approx 1" HgA to 15" HgA

DESCRIBE UPSET, START-UP, OR FUTURE OPERATING CONDITIONS, AND MINIMUM AND MAXIMUM FLOWRATE (IF APPLICABLE):

DESIGN - MATERIALS - CONSTRUCTION

DESIGN PRESSURE FV TO 14.9 PSIG	TEST PRESSURE 22 PSIG	DESIGN TEMP. MAX./~~MIN.~~ 250 / 35
TUBE BUNDLE	HEADER TYPE STEAM DUCT	TUBE MATERIAL CS - HDG
SIZE 31' L x 8' W	MATERIAL C.S.	~~SEAMLESS~~/ WELDED ELLIPTICAL
NO./BAY 10 NO. TUBE ROWS 4	NO. PASSES 1 SLOPE 60° ~~IN./FT~~	OD ~ IN MIN. THICKNESS 0.06
ARRANGEMENT	PLUG MATERIAL N.A.	NO./BUNDLE 200 LENGTH 30.5 FT
BUNDLES 120 IN PARALLEL IN SERIES	GASKET MATERIAL N.A.	PITCH 1.77 IN
BAYS 12 IN PARALLEL IN SERIES	CORROSION ALLOWANCE 1/16 IN	FIN. TYPE TENSION - ELLIPTICAL
BUNDLE FRAME CS - HDG	NO., SIZE INLET NOZZLE N.A. IN	MATERIAL CS - HDG
MISCELLANEOUS	NO., SIZE OUTLET NOZZLE N.A. IN	OD — IN STOCK THICKNESS 0.014
STRUCT.MOUNT., ~~GRADE/ PIPERACK~~ ROOF	SPECIAL NOZZLES N.A. IN	NO./IN 8.5 FIN DESIGN TEMP. 250
SURFACE PREPARATION	RATING & FACING N.A.	CODE -ASME VIII, DIV 1 STAMP ~~YES~~ / NO
~~LOUVERS~~ AUTO MANUAL	TI PI	SPECS.:
VIBRATION SWITCHES 1 PER FAN	CHEMICAL CLEANING N.A.	

MECHANICAL EQUIPMENT

FAN. MFR. & MODEL ALPINA - 8EM2K5	DRIVER. TYPE TWO SPEED - ELECTRIC	SPEED REDUCER. TYPE GEARBOX
NO./BAY 1 REV/MIN 63	MFR. SIEMENS	MFR. & MODEL FLENDER
DIAMETER 29 FT NO. BLADES 5	NO./BAY 1 HP/DRIVER 150	NO./BAY 1
PITCH ADJ. ~~AUTO~~ ANGLE 30	REV/MIN 1200 / 600	AGMA RATING LATER HP RATIO 18.
MATERIAL BLADE FRP HUB CS	ENCLOSURE TEFC	SUPPORT. STRUCTURAL / ~~PEDESTAL~~
HP/FAN.DES. 122 MIN. AMB.	VOLT, PHASE, CYCLE 460 / 3 / 60	

CONTROL ACTION ON AIR FAILURE - FAN PITCH, MINIMUM / MAXIMUM / LOCK UP — LOUVERS OPEN / CLOSE / LOCK UP

DEGREE CONTROL OF OUTLET PROCESS TEMPERATURE / MAXIMUM COOLING / + OR - °F

RECIRCULATION NONE / INTERNAL / EXTERNAL OVER SIDE / EXTERNAL OVER END — STEAM COIL YES / NO

NOTE: *GIVE TUBE COUNT OF EACH PASS WHEN IRREGULAR.

REV.	DESCRIPTION	ORIG. BY	DATE	CHECK BY	DATE	PROC APPR.	DATE	PROJ APPR.	DATE
0	Issued for Purchase	BCL	11/29/93	GJS	11/29/93				

CROCKETT COGENERATION PROJECT

AIR COOLED CONDENSER PROCESS DATA SHEET

Job No. : 22311

Specification No.: M-004 — REV. 0

Appendix A

SHEET 2 OF 4

(RWW3 12/8/93)

Figure C- 5
ACC Process Data Sheet

CROCKETT COGENERATION PROJECT
PURCHASE ORDER 22311-M-004, REV 0
APPENDIX A
ACC DATA SHEET, SHEET 3 OF 4
AIR COOLED CONDENSER PERFORMANCE SUMMARY

Item	Units	Case A	Case B	Case C	Case C1 Guar Pt	Case D	Case E	Case F	Case G Guar Pt	Case H	Case J1 Guar Pt
Ambient	°F	85	65	65	65	65	65	85	96	33	65
RH	%	30	60	60	60	60	60	30	23	80	60
ST Exhaust Flow	lb/hr	605,000	510,000	241,300	320,000	373,600	152,700	531,500	567,000	622,000	608,000
ST Exhaust Enthalpy	Btu/lb	1022	1016	1036	1025	1022	1067	1022	1035	1009	1012
ST Exhaust Pressure	psia	3.2	* 1.65	* 1.00	1.0	* 1.15	* 1.00	* 2.70	3.80	* 1.20	2.10
ACC Vendor Guaranteed ST Exhaust Pressure Measured at Turbine Exhaust Flange	psia	3.2			1.0				3.80		2.10
Released Heat	MMBtu/hr	* 550.5	* 474.1	* 233.1	* 305.7	* 353.6	* 152.4	* 486.9	* 519.4	* 581.6	* 557.2
Makeup Flow to Receiver	lb/hr	212,000	74,000	357,000	276,000	72,300	356,000	211,000	210,000	later	113,600
Makeup Water Temp	°F	125	138	141	142	138	141	123	125	later	132
Far Field Noise Guarantee	See	Attach	Spec	22311-	G-003						
Guaranteed Total ACC and ACE Electrical Load Measured at the Motor Terminals	kW	ACC: 1240 ACE: 154			ACC: 1290 ACE: 160				ACC: 1219 ACE: 150		ACC: 1290 ACE: 160
Guaranteed Residual Oxygen Leaving Cond Receiver	ppb	less	than	20 ppb	for	all	Cases				

* to be provided by Bidder (RWW3 12/16/93)

November 29, 1993
file: wp51\crockett\accperf

Figure C- 6
ACC Process Data Sheet

	CUSTOMER : CROCKETT COGENERATION PROJECT : CROCKETT COGENERATION PROJECT LOCATION : CROCKETT, CA	EQUIPMENT NAME : CLOSED COOLING WATER HEAT EXCHANGER (ACE) EQUIPMENT NO. : E-02A thru E-02 C (Corresponding to the # of cells) 3 x 50% BAYS - 2 OPERATING - 1 SPARE
	SERVICE OF UNIT: COOLING OF PLANT AUXILIARY COOLING WATER	MANUFACTURER: BALCKE-DURR, INC.
	SIZE: 105'W x 40'L TYPE: ROOF TYPE	(INDUCED/FORCED) DRAFT FORCED NO. OF BAYS 3
	SURFACE PER UNIT - FINNED TUBE 223 518 FT²	BARE TUBE 22503 FT² (DATA FOR 2 BAYS)
2	HEAT EXCHANGED: 26,000,000 BTU/HR	MTD. EFF 16.8 °F · NO.OF UNITS REQ'D ONE
2	TRANSFER RATE - FINNED TUBE — — 6.98	BARE TUBE, SERVICE 68.8 · CLEAN BTU/HR.FT²·F

PERFORMANCE DATA TUBE SIDE

	FLUID CIRCULATED: 10% ETHYLENE GLYCOL AND WATER (VOL)			LETHAL SERVICE ? YES / NO				
	TOTAL FLUID ENTERING: 2 555 536 LB/HR				IN		OUT	
		IN	OUT		LIQUID	VAPOR	LIQUID	VAPOR
1	VAPOR LB/HR, MW			DENSITY LB/FT³	62.5		62.6	
2	LIQUID LB/HR	2555536	2555536	VISCOSITY CP	0.69		0.67	
1	STEAM LB/HR			THERM.CON. BTU/HR.FT·F	0.34		0.34	
1	WATER LB/HR			SPECIFIC HEAT BTU/LB·F	0.97		0.96	
	NONCOND. LB/HR,MW			SURFACE TEN. DYNE/CM				
2	TEMPERATURE °F	124.2	115.3	POUR / FREEZE POINT °F	BUBBLE POINT °F			
	FOULING RESIST.	0.002	HR.FT²·F/BTU	LATENT HEAT BTU/LB	CRITICAL TEMPERATURE °F			
1	PRESSURE DROP ALLOW./CAL	—	/ 3 PSI	INLET PRESSURE 25 PSIA	CRITICAL PRESSURE PSIA			

PERFORMANCE DATA - AIR SIDE (DATA FOR 2 x 50% BAYS)

1	AIR QUANTITY,TOTAL 8.14 x 10⁶ LB/HR / ~~SCFM~~	AIR QUANTITY/FAN 963948 ACFM	ACTUAL STATIC PRESSURE 0.247 IN. H₂O
1	ALTITUDE ABOVE SEA LEVEL 70 FT	TEMPERATURE IN (DESIGN DRY BULB) 96 °F	TEMPERATURE OUT 109.1 °F
1	FACE VELOCITY 563 ~~STD.~~FT/MIN	MASS VEL (NET FREE AREA) LB/HR.FT²	MIN. DESIGN AMBIENT 35 °F

MAINTENANCE CONSIDERATIONS

	DESCRIBE TYPE OF FOULING:	RESTRICTION (IF ANY)	VELOCITY: FT/S · TUBE OD: IN.
	METHOD OF CLEANING:	FREQUENCY OF CLEANING: ANNUAL	

MECHANICAL DESIGN CONSIDERATIONS

	MAX. EXPECTED OPERATING PRESSURE PSIG	MAX. EXPECTED OPERATING TEMPERATURE °F
	WILL UNIT BE SUBJECTED TO FREQUENT THERMAL CYCLING? YES / NO	DESCRIBE
	WILL UNIT BE SUBJECTED TO FREQUENT PRESSURE CYCLING? YES / NO	DESCRIBE
	DESCRIBE UPSET, START-UP, OR FUTURE OPERATING CONDITIONS, AND MINIMUM AND MAXIMUM FLOWRATE (IF APPLICABLE):	

DESIGN - MATERIALS - CONSTRUCTION

1	DESIGN PRESSURE 40 PSIG	TEST PRESSURE 50 PSIG	DESIGN TEMP. MAX./MIN. 173.6 / 35 °F
	TUBE BUNDLE	**HEADER.** TYPE "D"	**TUBE.** MATERIAL CS - HDG
	SIZE 29'L x 8'W	MATERIAL CS	~~SEAMLESS~~ / WELDED ELLIPTICAL
	NO./BAY 8 · NO. TUBE ROWS 3	NO. PASSES 2 · SLOPE 60° ~~IN / FT~~	OD ~ IN · MIN. THICKNESS 0.06 IN
	ARRANGEMENT	PLUG MATERIAL NA	NO./BUNDLE 182 · LENGTH 28.6 FT
	BUNDLES 24 IN PARALLEL IN SERIES	GASKET MATERIAL NA	PITCH 1.5 IN
	BAYS 3 IN PARALLEL IN SERIES	CORROSION ALLOWANCE 1/16 IN	**FIN.** TYPE TENSION - ELLIPTICAL
	BUNDLE FRAME CS - HDG	NO., SIZE INLET NOZZLE 2 @ 6" IN	MATERIAL CS - HDG
	MISCELLANEOUS	NO., SIZE OUTLET NOZZLE 2 @ 6" IN	OD ~ IN · STOCK THICKNESS 0.014 IN
	STRUCT.MOUNT., ~~GRADE / PIPERACK~~ ROOF	SPECIAL NOZZLES — IN	NO./IN 8.5 · FIN DESIGN TEMP. 250 °F
	SURFACE PREPARATION	RATING & FACING WELD ENDS	CODE -ASME VIII, DIV 1 · STAMP ~~YES~~ / NO
	~~LOUVERS~~ AUTO MANUAL	TI PI	**SPECS.**:
	VIBRATION SWITCHES 1 PER FAN	CHEMICAL CLEANING	

MECHANICAL EQUIPMENT

	FAN. MFR. & MODEL ALPINA 8EM2K4	**DRIVER.** TYPE ONE SPEED ELECTRIC	**SPEED REDUCER.** TYPE GEARBOX
2	NO./BAY 1 · REV/MIN 63	MFR. SIEMENS	MFR. & MODEL FLENDER
	DIAMETER 29 FT · NO. BLADES 4	NO./BAY 1 · HP/DRIVER 125	NO./BAY 1
	PITCH ADJ. ~~AUTO~~ · ANGLE 28	REV/MIN 1200	AGMA RATING 2 x HP · RATIO 18.6
	MATERIAL, BLADE FRP · HUB CS	ENCLOSURE TEFC	SUPPORT, STRUCTURAL / ~~PEDESTAL~~
1	HP/FAN,DES. 87 · MIN. AMB.	VOLT, PHASE, CYCLE 460/3/60	
	CONTROL ACTION ON AIR FAILURE - FAN PITCH, MINIMUM / MAXIMUM / LOCK UP		LOUVERS OPEN / CLOSE / LOCK UP
	DEGREE CONTROL OF OUTLET PROCESS TEMPERATURE / MAXIMUM COOLING / + OR - °F		
	RECIRCULATION NONE / INTERNAL / EXTERNAL OVER SIDE / EXTERNAL OVER END		STEAM COIL YES / NO
	NOTE: *GIVE TUBE COUNT OF EACH PASS WHEN IRREGULAR.		

REV.	DESCRIPTION	ORIG. BY	DATE	CHECK BY	DATE	PROC APPR.	DATE	PROJ APPR.	DATE
2	REVISED FLOW PER BECHTEL	RWW3	5/5/94						
1	REVISED PER BECHTEL REQUIREMENT	RWW3	1/27/94						
0	Issued for Purchase	DCL	11/29/93	[illegible]	11/29/93				

BECHTEL	**CROCKETT COGENERATION PROJECT** **AIR COOLED HEAT EXCHANGER PROCESS DATA SHEET**	Job No.: 22311 Specification No.: M-004 · REV. Appendix A · 0 SHEET 4 OF 4

(RWW3 5/5/94)

Figure C-7
ACC Process Data Sheet

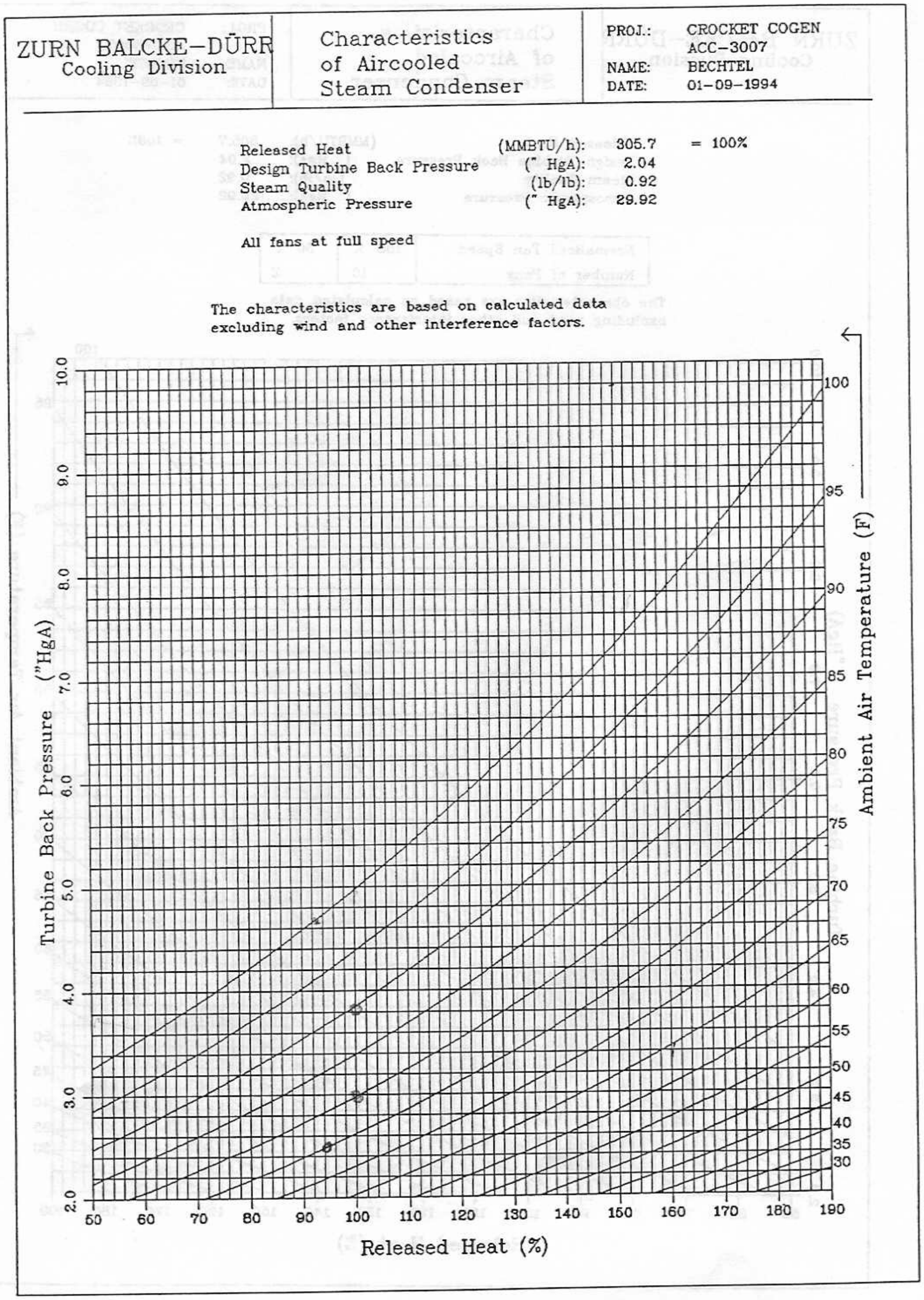

Figure C- 7
Characteristics of ACC (Crockett Cogeneration Project)

El Dorado Energy

Technical Memorandum

October 28, 2000

Subject: Meeting Notes - El Dorado Visit - ACC

To: John Maulbetsch

Matt Layton

Joe O'Hagan

Kent Zammit

From: Mike DiFilippo

I met with George Tatar of El Dorado Energy at the Boulder City Power Plant. George is the Facility Manager and can be reached as follows:

George Tatar 702-568-8206 (office)

El Dorado Energy, LLC 702-568-8213 (fax)

701 Eldorado Valley Drive george-tatar@eldoradoenergy.com

PO Box 62470

Boulder City, NV 89006-2470

We took a brief plant tour in the pouring rain. The tour was shortened because George had another meeting. Also, I did not bring my camera (mea culpa, mea culpa).

The following was discussed:

The Boulder City plant is rated at 540 MW - two GTs (each rated at 180 MW) and the one ST (rated at 180 MW). Start up occurred in May, 2000. George noted that the plant has peaked at 480-490 MW. Sargent Lundy (Chicago) was the design engineer. Construction is complete (the appearance of the plant was neat and impressively laid out).

Dry cooling was selected because:

The politics of using fresh water in the water-short Las Vegas area.

There was no available municipal reclaimed water for cooling towers.

The property boundaries of the plant will not accommodate evaporation ponds for cooling tower blowdown. [Note the evaporation rate is equivalent to the Blythe area.]

El Dorado looked at cooling towers with VCE (brine concentrators) versus ACC, and based on their analysis, dry cooling was more economic. Also, the ACC support structure was designed for Zone 2 seismic making it less costly.

The Westinghouse steam turbine was a first of its kind:

It can operate at back pressures of 2 to 8.5 inches.

Initially, the turbine had problems with its thrust bearing (I hate when that happens), but the problem has been repaired. The problem was not related to the ACC.

There are now seven of these turbines on order, in manufacture or installed.

The following information was gathered on the ACC:

GEA supplied the ACC.

The ACC is only used for steam condensation. There is a separate closed-loop fin-fan cooler (supplied by Balke Durr) used only for auxiliary cooling.

The ACC has 5 banks with six fan bays.

The fans are huge - 34 feet diameter (maybe 36 feet - George was not sure). Each multiple-speed fan requires 200 HP for a total of 6,000 HP (4.5 MW). When I visited the plant it was 60-65 F and all fans were in service and spinning slowly.

The plant has been on line for 6 months and there are no signs of corrosion. The fins are probably carbon steel (George was not sure). Like almost all power plants, El Dorado is using a specialty chemical provider to monitor and control steam chemistry.

El Dorado did not buy the cleaning apparatus (we saw at Crockett). They have not had problems with sand/dust in the fin crevices (as yet) even though they experience notable gusting in their area.

There is a distinct loss of cooling during periods with gusting winds. Evidently, during these periods, some fans are starved of air flow (that's GEA analysis of the situation).

He had no idea of the cost of the ACC.

George is actually thinking of ways to pre-cool air to the ACC to enhance overall power plant performance during the hottest days of the years. When ambient air temperature exceed 110 F, plant performance drops dramatically. We discussed the following:

One idea George is thinking of is to install evaporative coolers around the open areas of the ACC. The coolers would cool some of the air drawn into the fans.

He has looked at Mee Fog and another fogging system (I didn't get the name), but thought the approach was too expensive. I'm not sure how he evaluated it.

George, really wants to look at pre-cooling alternatives. As I was leaving, he said would be interested in a collaborative effort to evaluate pre-cooling. Let's plan discuss this early next week. I think he would try to sell the idea of some sort of shared expense to his management. Also, this ties into one of the TC's we've been discussing.

Description: El Dorado Energy

(Boulder City, NV) - El Dorado Energy, a 480-megawatt power plant jointly owned by Reliant Energy and Sempra Energy, has begun commercial operation, selling electricity into the wholesale power markets of Nevada, California and the southwestern United States (photos).

The $280 million natural gas-fired power generating plant, located near Boulder City, Nev., about 40 miles southeast of Las Vegas, is designed to provide reliable, safe and cost-effective power to support the growing economies of the region.

"We're pleased to begin commercial operations," said George Tatar, El Dorado facility manager. "Testing and initial start-up operations were conducted for the last several weeks to scrutinize and fine-tune all operational and environmental systems for safety, reliability and commerciality."

The new plant, under construction since April, 1998, is capable of generating enough energy to serve nearly a half-million households.

"One of the nation's first large-scale merchant power plants, El Dorado Energy is perfectly positioned to sell competitively priced energy into three of the nation's fastest growing markets, southern Nevada, Arizona and Southern California," said Darcel Hulse, president of Sempra Energy Resources, the power generation unit of Sempra Energy. "In addition, this generation will help support Sempra Energy's retail-focused growth strategy."

"This is an important step for Reliant Energy in our strategy to compete in the major power markets in the western United States and across the country by building an asset-backed, energy trading and marketing organization," said Joe Bob Perkins, president and chief operating officer of Reliant Energy's Wholesale Group.

The state-of-the-art plant complex employs advanced, combined-cycle gas turbine and photovoltaic technology, making it one of the most efficient and environmentally friendly power generating facilities in the United States. The plant features a water-saving, air-cooled condenser in lieu of a traditional, "wet-cooled" water system. Also, the facility is equipped with a 100-kilowatt solar field, comprised of 256 panels that track the sun's path through the Nevada sky. With its photovoltaic technology, El Dorado Energy is well-positioned to become a prototype for other such facilities.

Boulder City benefits from profit-sharing incentives, a 20-year land lease that generates $800,000 in revenue per year and a power-purchase agreement. According to Boulder City Mayor Bob Ferraro, the revenue the city receives will help ensure the preservation of the unique quality of life that the historic town has maintained since Hoover Dam was built.

“El Dorado Energy’s partners – Sempra Energy and Reliant Energy – have lived up to their promises to be good corporate citizens," Ferraro said. "The land-lease agreement has produced substantial revenue for Boulder City, which has been particularly beneficial given our previous budgetary situation. The companies have fulfilled their commitment to this community by funding an aggressive tree-planting effort, providing funds to pave Boulder City alleys and generously supporting a multitude of Boulder City community organizations and events.

"Reliant Energy (NYSE: REI), based in Houston, Texas, is an international energy services and energy delivery company with $15.3 billion in annual revenue and assets totaling more than $26 billion. The company has a wholesale energy trading and marketing business that ranks among the top five in the U.S. in combined electricity and natural gas volumes and has a presence in most of the major power regions of the U.S. It also has power generation and wholesale trading and marketing operations in western Europe.

Sempra Energy Resources acquires and develops power plants for the competitive market, as well as natural gas storage, production and transportation assets. Sempra Energy Resources is a subsidiary of Sempra Energy (NYSE: SRE), a San Diego-based Fortune 500 energy services holding company, with 12,000 employees, revenues of nearly $5.5 billion and more than 9 million customers in the United States, Europe, Canada, Mexico and South America. (Sempra Energy Resources and El Dorado Energy are not the same companies as the utilities, SDG&E/SoCalGas, and are not regulated by the California Public Utilities Commission.)

Figure C- 8
El Dorado Energy Center

Figure C- 9
ACC at El Dorado

Calpine

September 18, 2000.

(Monday, 2:30 PM)

Kim Stucki
Calpine
620 Coolidge Drive
Folsom, CA
916-608-3839

Topics for discussion

General:

- What are the factors which influence the choice of cooling system for your power projects?

- **Assume**: Gas-fired, combined cycle (~ 500MWe) power plant at an inland California location.

Capital cost comparisons:---Cost of system from turbine flange to ultimate discharge----

- How is the comparison done?
- Is there a systematic optimization to select the design points for comparison?
- If so, how are the "penalty" costs evaluated
    - capacity shortfall on hottest periods
    - efficiency penalties during rest of year
    - lost revenues or contractual penalties for capacity replacement (or failure to deliver)
- If no optimization, how are comparative design points selected?
    - Average temperature over year?
    - Design back-pressure at 1 or 2 ½% exceeded dry bulb (wet-bulb)?

Approaches to mitigating hot day penalties:

- What is the effect of ambient dry bulb on plant output and efficiency?
    - On gas turbines
    - On steam side
- Use of high back-pressure turbines
    - Any experience
    - Cost/reliability
    - Efficiency penalties during rest of year
- Use of inlet sprays
    - On gas turbines
    - On air cooled condensers
- Other wet/dry systems
    - Indirect dry cooling
    - Heller system

Other considerations:

- Licensing issues
    - Reduced licensing time (what is it worth to get on-line "six months sooner".

    - Drift/plume
    - Noise
    - Site placement/footprint

Use of recycled/reclaimed or lower quality water

- Water quality criteria
- Treatment costs
- Cost of water
- Other licensing issues
    - Drift/plume issues
    - Worker health/safety issues
    - Reliability of supply

Operating experience:

- With dry cooling
    - Cycle chemistry problems
    - Tower (finned tube heat exchanger maintenance)
    - Fouling/corrosion
    - Cleaning frequency
    - Use in regions other than hot/dry (Northeast---Massachusetts, Long Island, etc)

- With recycled water (Delta and Pittsburg/Los Medanos)
    - Supply problems
    - Maintenance issues (any plants currently operating?)

Figure C-10
ACC at Sutter Energy Center

Chinese Station: Contact Information

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Appendix D
CASE STUDY SITE CHARACTERISTICS

Site characteristics are presented for Desert, Mountain, Valley, and Bay Area sites. State bulb and dry bulb data are presented in Table D-1.

Table D- 1
California Wet Bulb and Dry Bulb Data

California Wet Bulb & Dry Bulb Data

Source: Evaluated Weather Data for Cooling Equipment Design, Addendum 1, Winter and Summer Data
Fluor Products Company Inc., 1964

	Elevation	Dry Bulb, F			Wet Bulb, F			Ratio - Wet Bulb/Dry Bulb		
Location	Feet	1%	2.5%	5%	1%	2.5%	5%	1%	2.5%	5%
	-30	111	109	106	81	80	79	0.730	0.734	0.745
	3	85	81	77	65	63	62	0.765	0.778	0.805
	3	89	85	81	71	68	66	0.798	0.800	0.815
	8	83	79	75	65	63	62	0.783	0.797	0.827
	11	111	109	106	80	79	78	0.721	0.725	0.736
	16	94	92	89	70	69	68	0.745	0.750	0.764
	17	100	97	94	72	70	69	0.720	0.722	0.734
	19	86	83	80	71	70	68	0.826	0.843	0.850
	27	94	90	87	70	68	67	0.745	0.756	0.770
	28	101	98	96	72	70	69	0.713	0.714	0.719
	30	84	81	78	69	68	67	0.821	0.840	0.859
	34	87	84	81	72	70	69	0.828	0.833	0.852
	35	83	80	77	69	68	67	0.831	0.850	0.870
	38	81	78	75	64	63	61	0.790	0.808	0.813
	39	85	80	76	68	67	65	0.800	0.838	0.855
	43	84	80	78	70	69	67	0.833	0.863	0.859
	50	72	65	65	61	60	59	0.847	0.923	0.908
	52	80	77	73	64	62	61	0.800	0.805	0.836
Richmond	55	85	81	77	66	64	63	0.776	0.790	0.818
	57	80	77	74	69	68	67	0.863	0.883	0.905
	70	90	88	85	70	69	67	0.778	0.784	0.788
	70	102	100	97	71	70	69	0.696	0.700	0.711
	72	98	94	90	71	69	67	0.724	0.734	0.744
	74	87	85	82	64	62	61	0.736	0.729	0.744
	86	102	99	96	71	70	69	0.696	0.707	0.719
	91	101	98	96	73	71	70	0.723	0.724	0.729
	92	101	98	95	72	70	69	0.713	0.714	0.726
	99	86	83	80	69	68	67	0.802	0.819	0.838
	100	87	84	81	67	66	65	0.770	0.786	0.802
	115	92	89	86	72	71	70	0.783	0.798	0.814
	116	93	94	87	72	71	70	0.774	0.755	0.805
	125	87	84	80	66	65	63	0.759	0.774	0.788
	167	95	93	90	70	68	67	0.737	0.731	0.744
	178	102	99	96	73	72	70	0.716	0.727	0.729
	195	96	92	88	69	67	66	0.719	0.728	0.750
	203	67	65	63	60	59	58	0.896	0.908	0.921
	205	102	100	97	71	70	69	0.696	0.700	0.711
	238	85	82	79	65	64	63	0.765	0.780	0.797
	263	91	89	86	72	71	70	0.791	0.798	0.814
	312	94	90	87	72	70	69	0.766	0.778	0.793
	315	89	85	82	65	64	63	0.730	0.753	0.768
	326	101	99	97	73	72	71	0.723	0.727	0.732
	354	102	100	97	72	71	70	0.706	0.710	0.722
Blythe	390	111	109	106	78	77	76	0.703	0.706	0.717
	411	110	108	105	79	78	77	0.718	0.722	0.733
Redding	495	103	101	98	70	69	67	0.680	0.683	0.684
Bakersfield	495	103	101	99	72	71	70	0.699	0.703	0.707
El Cajon	525	98	99	92	74	73	72	0.755	0.737	0.783
Livermore	545	99	97	94	70	69	68	0.707	0.711	0.723
	552	82	79	76	65	63	61	0.793	0.797	0.803
	575	100	92	94	73	72	71	0.730	0.783	0.755
	620	98	96	93	71	70	68	0.724	0.729	0.731
	660	95	92	89	73	72	71	0.768	0.783	0.798
	676	101	99	97	72	71	70	0.713	0.717	0.722
Burbank	699	97	94	91	72	70	69	0.742	0.745	0.758
Pasadena	864	96	93	90	72	70	69	0.750	0.753	0.767
Needles	913	112	110	107	76	75	74	0.679	0.682	0.692
San Fernando	977	100	97	94	73	72	71	0.730	0.742	0.755
Ontario	995	100	97	94	72	71	70	0.720	0.732	0.745
San Bernadino	1125	101	98	96	75	73	71	0.743	0.745	0.740
Auburn	1297	98	96	93	70	68	67	0.714	0.708	0.720
	1318	99	96	93	72	71	70	0.727	0.740	0.753
	1511	99	96	94	72	71	69	0.727	0.740	0.734
Barstow	2142	104	102	99	73	72	71	0.702	0.706	0.717
Banning	2349	101	98	95	73	72	71	0.723	0.735	0.747
Palmdale	2517	103	101	98	70	68	67	0.680	0.673	0.684
Yreka	2625	96	94	91	68	66	65	0.708	0.702	0.714

California Wet Bulb/Dry Bulb Ratio vs. Dry Bulb is presented in Figure D-1.

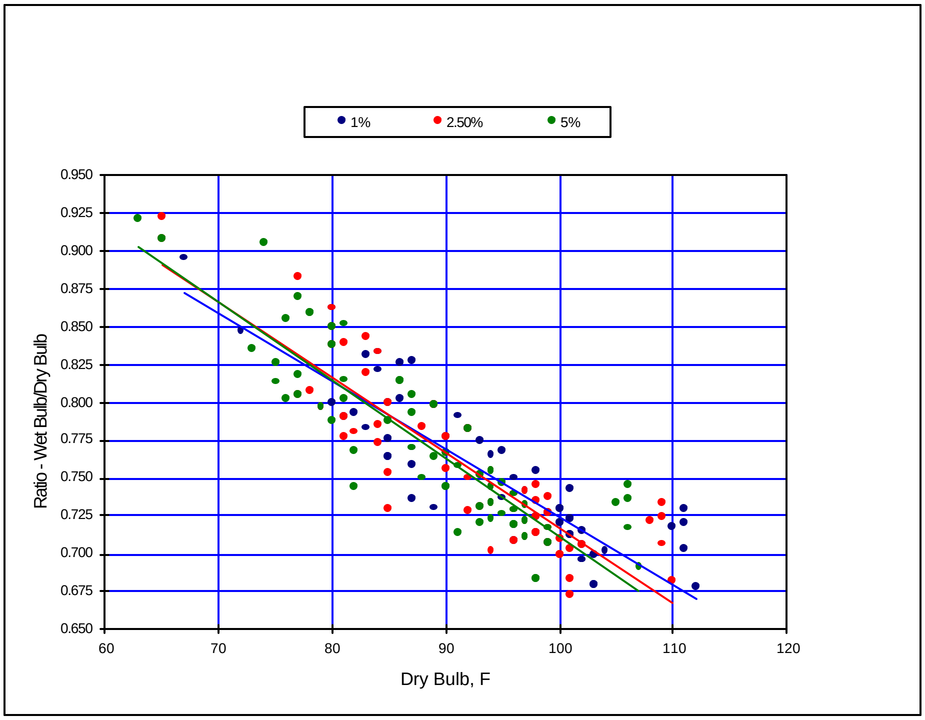

Figure D-1
California Wet Bulb/Dry Bulb Ratio vs. Dry Bulb (Elevation to 1500 ft)

Desert Site

The desert site is similar to the location of the Blythe project (elevation: ~400 ft.). Meteorology data was taken from Blythe, California.

Figure D-2
Desert Site Location

Table D- 2
Records and Average Temperatures, Blythe, California

Month	Avg. High	Avg. Low	Avg. Precip.	Rec. High	Rec. Low
January	66.7° F	41.6° F	.4 in	89° F (01/25/1951)	20° F (01/08/1971)
February	72.6° F	45.7° F	.3 in	93° F (02/28/1986)	22° F (02/16/1990)
March	78.4° F	50.1° F	.3 in	100° F (03/27/1986)	30° F (03/13/1956)
April	86.5° F	56.2° F	.2 in	107° F (04/29/1992)	38° F (04/10/1975)
May	95.2° F	64° F	0 in	114° F (05/29/1983)	43° F (05/29/1971)
June	105° F	72.7° F	0 in	123° F (06/28/1994)	46° F (06/01/1980)
July	109° F	81.3° F	.3 in	123° F (07/28/1995)	62° F (07/19/1987)
August	106.9° F	80.2° F	.6 in	120° F (08/01/1972)	62° F (08/31/1957)
September	100.8° F	72.6° F	.4 in	121° F (09/01/1950)	53° F (09/26/1971)
October	89.7° F	60.9° F	.3 in	111° F (10/02/1980)	27° F (10/30/1971)
November	75.6° F	48.9° F	.3 in	95° F (11/01/1997)	27° F (11/20/1994)
December	66.4° F	41.4° F	.5 in	87° F (12/29/1980)	24° F (12/15/1971)

Table D- 3
Highest Ambient Temperatures, Blythe, California

Percent Exceeded	1990 Temp. ºF	1991 Temp. ºF	1992 Temp. ºF	1993 Temp. ºF
2.5	104.4	102.5	103.5	103.9
2.0	105.3	102.8	104.4	104.6
1.5	106.6	103.6	105.3	105.5
1.0	107.9	104.4	106.4	106.6
.50	109.9	105.5	108.9	108.2
0	117.2	111.1	112.9	116.3

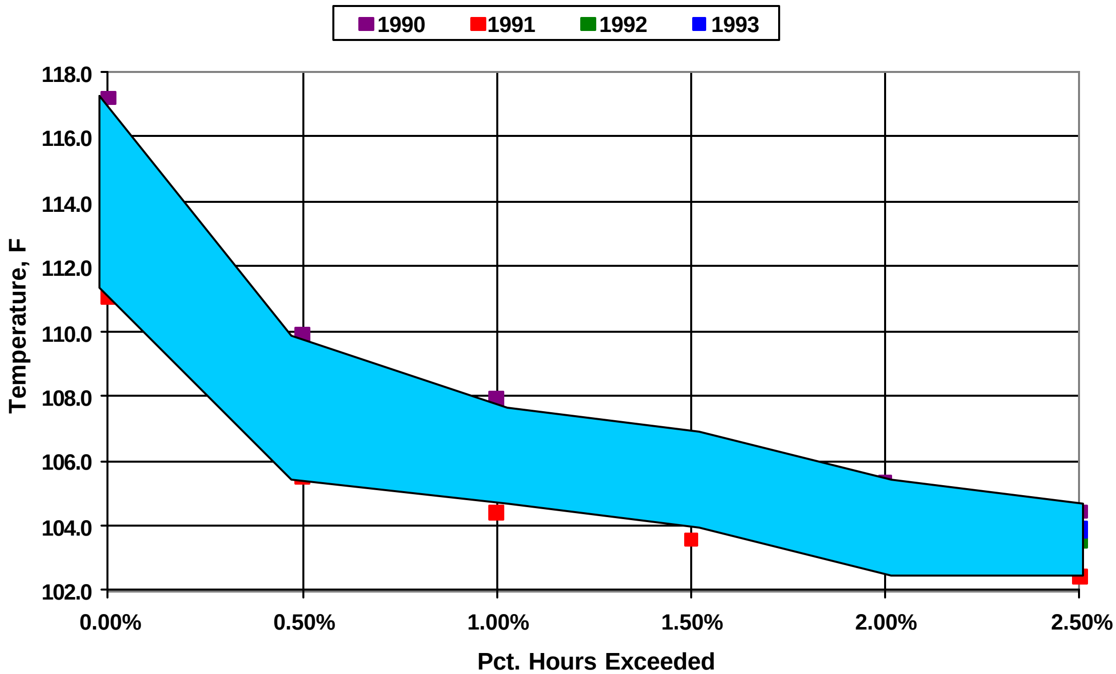

Figure D-3
Percent Hours Exceeded Highest Temperature

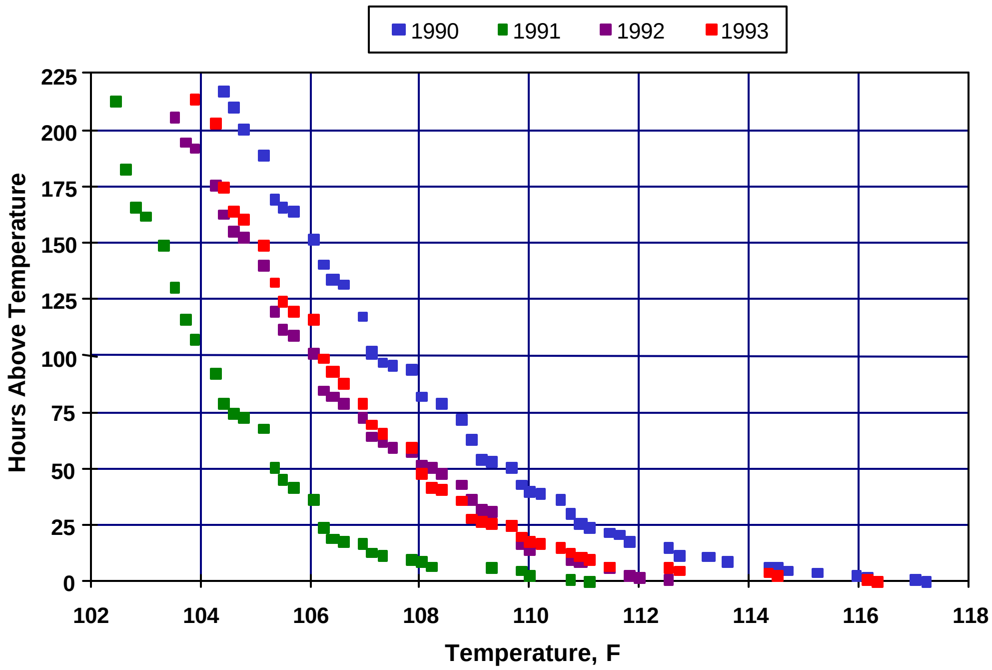

Figure D-4
Temperature Occurences—Highest Temperatures, Blythe

Mountain Site

The mountain site is similar to the location of Three Mountain project (elevation: ~1000 ft). Meteorology data was taken from Redding, California and adjusted for elevation.

Figure D-5
Mountain Site Location

Table D- 4
Records and Average Temperatures, Redding, California

Month	Avg. High	Avg. Low	Avg. Precip.	Rec. High	Rec. Low
January	55.3° F	35.7° F	6.1 in	77° F (01/17/1994)	19° F (01/14/1997)
February	61.3° F	40° F	4.5 in	83° F (02/25/1992)	21° F (02/05/1989)
March	62.5° F	41.7° F	4.4 in	85° F (03/26/1988)	28° F (03/05/1997)
April	69.9° F	46° F	2.1 in	94° F (04/09/1989)	31° F (04/01/1999)
May	80.5° F	52.3° F	1.3 in	104° F (05/06/1987)	36° F (05/04/1999)
June	90.4° F	61.8° F	.6 in	111° F (06/26/1987)	42° F (06/01/1990)
July	98.3° F	64.7° F	.2 in	118° F (07/20/1988)	54° F (07/21/1999)
August	95.7° F	63.1° F	.5 in	115° F (08/06/1990)	51° F (08/28/1995)
September	89.3° F	58.8° F	.9 in	116° F (09/03/1988)	46° F (09/24/1993)
October	77.6° F	49.2° F	2.2 in	105° F (10/11/1991)	33° F (10/31/1989)
November	62.1° F	41.4° F	5.2 in	88° F (11/13/1995)	23° F (11/23/1993)
December	54.7° F	35.2° F	5.5 in	78° F (12/16/1998)	17° F (12/21/1990)

Valley Site

The valley site is similar to the location of the La Paloma project (elevation: ~300 ft). Meteorology data was taken from Bakersfield, California.

Figure D-6
Valley Site Location

Table D- 5
Records and Average Temperatures, Bakersfield

Month	Avg. High	Avg. Low	Avg. Precip.	Rec. High	Rec. Low
January	56.9° F	38.6° F	.9 in	82° F (01/31/1984)	20° F (01/13/1963)
February	63.9° F	42.6° F	1.1 in	87° F (02/22/1989)	25° F (02/15/1990)
March	68.9° F	45.8° F	1 in	92° F (03/29/1969)	31° F (03/03/1966)
April	75.9° F	50.1° F	.6 in	101° F (04/30/1981)	34° F (04/27/1984)
May	84.6° F	57.3° F	.2 in	107° F (05/24/1982)	37° F (05/01/1988)
June	92.4° F	64° F	.1 in	114° F (06/28/1976)	44° F (06/23/1943)
July	98.5° F	69.6° F	0 in	115° F (07/01/1950)	52° F (07/18/1987)
August	96.6° F	68.5° F	.1 in	112° F (08/09/1981)	52° F (08/26/1942)
September	90.1° F	63.5° F	.2 in	112° F (09/03/1955)	45° F (09/26/1948)
October	80.7° F	54.8° F	.3 in	103° F (10/05/1980)	29° F (10/30/1971)
November	66.8° F	44.7° F	.7 in	91° F (11/06/1949)	28° F (11/20/1994)
December	56.5° F	38.3° F	.6 in	83° F (12/03/1979)	19° F (12/24/1990)

Table D- 6
Highest Ambient Temperatures, LaPaloma

Percent Exceeded	1993 Temp. °F	1994 Temp. °F	1995 Temp. °F
2.50	95.1	97.1	96.3
2.00	95.7	98.2	97.4
1.50	96.4	99.1	98.5
1.00	97.5	100.4	99.9
0.500	99.3	101.5	101.9
0.00	105.9	103.5	105.7

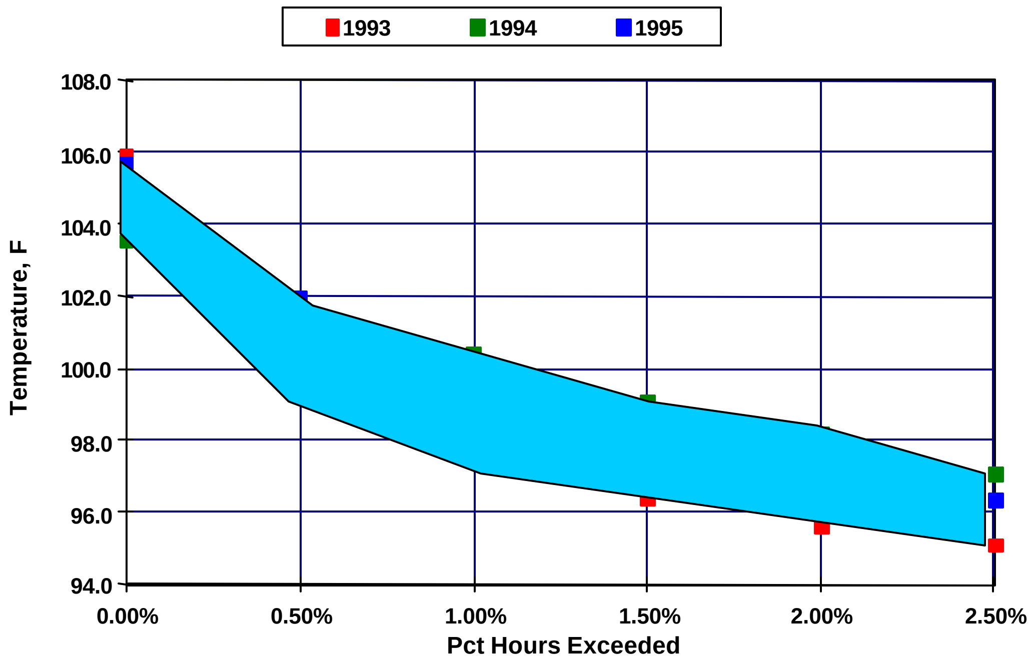

Figure D-7
Percent Hours Exceeded Highest Temperature, LaPaloma

Bay Area Site

(The Bay Area Site is similar to the location of the Contra Costa project (elevation: ~10 ft). Meteorology data taken from Pittsburg, California

Figure D-8
Bay Area Site Location

Table D- 7
Records and Average Temperatures, Pittsburg

Month	Avg. High	Avg. Low	Avg. Precip.	Rec. High	Rec. Low
January	53° F	35.9° F	2.6 in	72° F (01/31/1976)	20° F (01/24/1962)
February	60.4° F	40° F	2.1 in	76° F (02/28/1985)	25° F (02/06/1989)
March	65.1° F	42.8° F	2 in	88° F (03/27/1988)	27° F (03/04/1966)
April	71.4° F	45.6° F	.9 in	94° F (04/30/1981)	28° F (04/19/1961)
May	78.9° F	50.4° F	.3 in	103° F (05/29/1984)	35° F (05/05/1975)
June	85.7° F	55.4° F	.1 in	117° F (06/17/1961)	35° F (06/04/1982)
July	90.8° F	56.8° F	0 in	110° F (07/15/1972)	41° F (07/05/1961)
August	89.6° F	56.3° F	.1 in	109° F (08/05/1998)	43° F (08/06/1958)
September	85.7° F	54.4° F	.2 in	109° F (09/04/1955)	41° F (09/14/1980)
October	77.5° F	49.2° F	.9 in	102° F (10/05/1964)	28° F (10/30/1971)
November	63.9° F	42.7° F	1.9 in	85° F (11/02/1967)	24° F (11/17/1958)
December	53.6° F	36.6° F	1.9 in	75° F (12/01/1977)	18° F (12/12/1972)

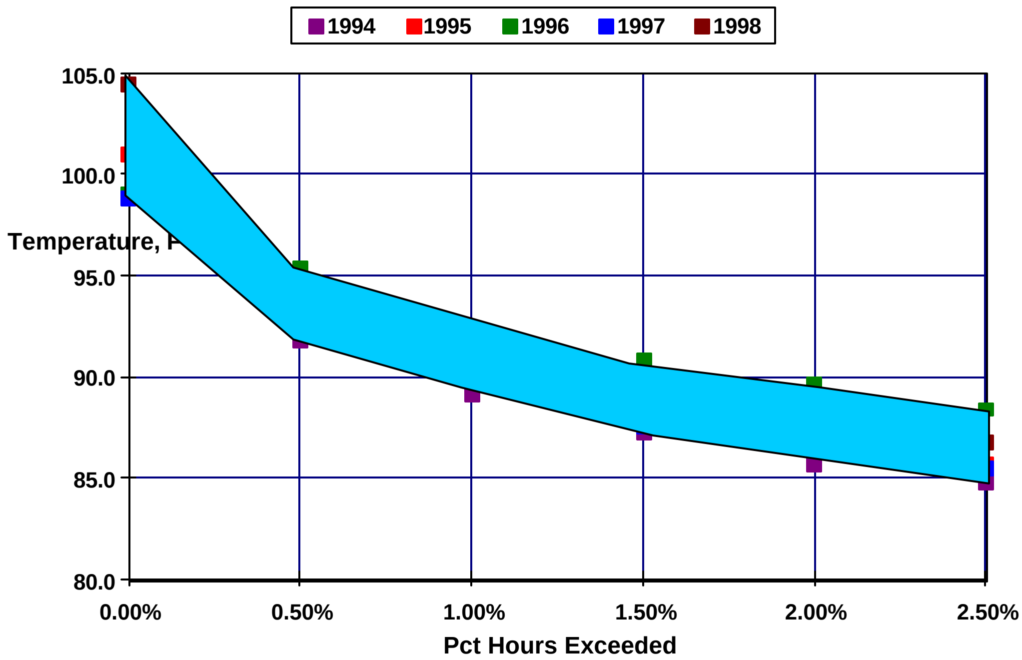

Figure D-9
Percent Hours Exceeded Highest Temperatures, Contra Costa, Unit 8