ELECTRIC POWER RESEARCH INSTITUTE

Air-Cooled Condenser Design, Specification, and Operation Guidelines

Technical Report

Air-Cooled Condenser Design, Specification, and Operation Guidelines

1007688

Final Report, December 2005

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This report describes research sponsored by the Electric Power Research Institute (EPRI).

The report is a corporate document that should be cited in the literature in the following manner:

Air-Cooled Condenser Design, Specification, and Operation Guidelines. EPRI, Palo Alto, CA: 2005. 1007688.

PRODUCT DESCRIPTION

In contrast to once-through and evaporative cooling systems, use of the air-cooled condenser (ACC) for heat rejection in steam electric power plants has historically been very limited, especially in the United States. However, greater industry focus on water conservation – combined with continued concern over the environmental effects of once-through and evaporative cooling – will almost certainly increase interest in ACC applications. While operating experience and performance data are, to some extent, available from ACC suppliers, consultants, and owner/operators, there is no one repository of such information. This report provides a single resource for ACC application, design, specification, and operation guidelines. It incorporates operating experience from recently commissioned plants to provide insight into issues that confront staff in operating and maintaining ACCs and balancing ACC performance relative to plant performance and output.

Results & Findings

This report provides information to guide the development and specification of ACC design conditions. In doing so, it offers perspectives on economic and operational issues that factor into selecting ACC design points. It also speaks to the dynamics of a typical bid process, with the objective of forging more of a partnership between the supplier and the purchaser. While the report cautions the power plant owner/purchaser relative to performance impacts of wind on the ACC, it recognizes that wind effects are site specific and that more information regarding both impacts and remedial action is required before improved guidance can be provided. Finally, in addition to an examination of startup and commissioning issues, the report provides a test procedure to determine ACC thermal acceptance.

Challenges & Objective(s)

The objective of this specification is to provide engineering and purchasing personnel with information they need to specify, procure, and commission ACCs that have optimum design and performance characteristics for the application at hand. In order to accomplish this, they will need information to assist them in answering the following questions:

What are the primary ACC operating and performance problems?

What information should be provided to bidders in a specification and request for proposal?

How should developers evaluate and compare bids for ACC supply?

What are the most important considerations in conducting performance and acceptance testing?

What are some key ACC commissioning and startup issues?

Applications, Values & Use

This specification provides much-needed industry insights while alerting the purchaser to issues that impact ACC application, specification, and design. Key drivers for increased application of ACCs include the

Scarcity of water and the attendant elimination of evaporative water loss realized by the use of ACCs from both once-through and evaporative cooling systems

Reduction or elimination of thermal pollution and entrainment and impingement issues typically associated with once-through and evaporative cooling

Elimination of the visible plume and drift from the operating cooling system

EPRI Perspective

As a result of the increased interest in water conservation and ACCs in particular, a number of programs are have been commissioned to further examine operational and performance issues. Several such programs – funded by EPRI, the California Energy Commission (CEC), and the U.S. Department of Energy/National Energy Technology Laboratory (U.S. DOE/NETL) – are summarized in the proceedings of the Advanced Cooling Strategies/Technologies Conference, held June 1-2, 2005, in Sacramento, California, and jointly sponsored by EPRI and the CEC. EPRI anticipates that the results of such studies, which include more detailed assessments of wind effects on ACCs and performance enhancement strategies via inlet air cooling, will complement and supplement this specification. Accordingly, aspects of this specification may be considered a work in progress, as ongoing projects will shed light on key areas that will ultimately improve the specification and operational understanding of ACCs.

Approach

Numerous specifications have been developed for ACCs, both internationally and in the United States. In many cases, such specifications do not ensure the optimum economic selection of an ACC for the plant, its environment, and the economic situations in which the plant must compete. Further, in most cases, these specifications have not addressed areas that might be problematic in terms of ACC performance, operation, and maintenance. As a result of site visits and interviews with both plant personnel and suppliers, a number of areas surfaced that deserved additional attention beyond the historical level that they have received. These included wind effects on ACC performance, reliable ACC performance over a range of operating conditions, fouling and cleaning of ACC finned tubes, and inlet air cooling/conditioning systems.

Keywords

Cooling System
Air-Cooled Condenser
Steam Electric Power Plant
Thermal Acceptance Test
Water Conservation
ACC Specification
ACC Performance and Acceptance Testing

ABSTRACT

In contrast to once-through and evaporative cooling systems, use of the air-cooled condenser (ACC) in steam electric power plants has historically been very limited, especially in the United States. However, greater industry focus on water conservation – combined with continued concern over the environmental effects of once-through and evaporative cooling – will almost certainly increase interest in ACC applications. Indeed, in the southwestern United States, this has already occurred.

As a result of limited ACC operating experience and the nature of proprietary and evolving dry-cooling technologies, there is no single repository of performance, operations, and maintenance experience. Recognizing the increased industry interest in ACCs and the aforementioned limitations in available data, the Electric Power Research Institute (EPRI) has commissioned this project to develop ACC procurement guidelines.

This study covers a number of key elements of ACC specifications, including the following:

Assessment of ACC operating and performance issues, including the effects of wind on ACC performance

Development of information that should be included in and solicited via ACC procurement specifications, with emphasis on language that might be incorporated into such specifications

An example procedure for evaluation and comparison of bids

Guidelines for ACC performance and acceptance testing

Issues associated with ACC startup and commissioning

This report's summary of a proposed ACC test guideline is particularly important, as codes for various ACC tests are under development by the American Society of Mechanical Engineers (ASME) and the Cooling Technology Institute (CTI) and are not expected to be published in the foreseeable future.

ACKNOWLEDGMENTS

This effort was aided and the quality of its results improved by the contributions of a number of individuals and organizations.

Dr. John Maulbetsch was a key contributor in the development of performance specification information, and Dr. Detlev Kröger provided project input in the areas of air-cooled condenser (ACC) operational issues, performance testing, and related technical guidance. Additionally, Dr. Ram Chandran contributed in areas of ACC design and performance.

Environmental Systems Corporation contributed significantly to development of the performance testing guidelines. Specific review and comments were provided from representatives of GEA and Marley SPX, ACC suppliers.

These acknowledgements are not meant to imply that the contributors are in complete accord with or endorse all areas of the ACC specification. Where differences in opinion or point of view remain, the specification reflects the views of the author.

LIST OF ACRONYMS

ACC	air-cooled condenser
acfm	actual cubic feet per minute
AISC	American Institute of Steel Construction Inc.
ASD	allowable stress design
ASME	American Society of Mechanical Engineers
AWMA	Air and Waste Management Association
Btu	British thermal unit
CCPP	combined-cycle power plant
CEC	California Energy Commission
CFD	computational fluid dynamics
CRT	condensate receiver tank
CTI	Cooling Technology Institute
dBa	adjusted decibel(s)
ELEP	expansion line end point
FOB	free on board
HRSG	heat recovery steam generator
in. HgA	inch(es) of mercury, atmospheric
ITD	initial temperature difference
J	joule
K	Kelvin
kg/m^3	kilogram(s) per cubic meter
kg/s	kilogram(s)/second
kPa	kilopascal
kW	kilowatt
lbm	pounds mass
m	meter
m/s	meter(s)/second
LMTD	log mean temperature difference
MCC	motor control center
MMBTU/h	million British thermal units/hour
MWe	Megawatt (electric)
OSHA	Occupational Safety & Health Administration
psia	pound(s) per square inch absolute
psig	pound(s) per square inch gage
PTC	Performance Test Code
°R	degree(s) Rankine
RFP	request for proposal
RTD	resistance temperature detector

SRC	single row condenser
TEFC	totally enclosed fan-cooled
UEEP	used energy end point
VFD	variable frequency drive

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INTRODUCTION

1.1 Background

The use of air-cooled condensers (ACCs) for power plant applications is relatively new in the United States. Indeed, the total capacity of power plants with ACCs through the end of the 20th century was less than 2500 MWe. However, by late 2004, that number had nearly tripled, with ACCs serving approximately 7000 MWe in 60 power generation units. This is in contrast to a worldwide installed base of more than 700 systems (Appendix A).

The main drivers for the increased use of ACCs in the United States include the

Scarcity of water and the attendant elimination of evaporative water loss from both once-through and evaporative cooling systems

Reduction or elimination of thermal pollution, entrainment, and impingement issues typically associated with once-through and evaporative cooling

Elimination of visible plume and drift from the operating cooling system (though clearly not the primary driver)

A number of quality studies exist relative to ACCs (references [a-e] at the end of this chapter), though information regarding current designs and operating issues is lacking. Moreover, while operating experience and performance data are, to some extent, available from ACC suppliers, consultants, and owner/operators, there is no single repository of such knowledge. Experience from recently commissioned plants has provided additional insights into operation and maintenance of ACCs and balancing ACC performance relative to plant performance and output. For example, the impact of ambient wind on ACC performance is not well understood by owner/operators or their representatives in the specification and bid evaluation process. This area is highlighted due to the potential for prevailing winds to degrade ACC performance. Beyond additional understanding concerning wind effects, methods for improving ACC performance are under consideration and evaluation; however, no consolidated information on these methods is available to the architect/engineer or owner/operator. This specification provides much-needed industry insights while alerting the purchaser to issues that impact ACC specification and design.

1.1.1 Bid Process Dynamics – Situation Analysis

In general, the bid process for capital equipment in a given utility may effect a set of dynamics that does not necessarily lead to the optimum selection and purchase of equipment for the end user. Factors such as those listed below may influence and characterize this process.

1.1.2 Potential Assumptions and Positions of the Purchaser

The purchaser or their representative assumes that the supplier of the equipment is fully aware of and has incorporated in their bid the features and impacts of the site and process wherein the equipment must perform, and that they have sufficient experience and design expertise to extrapolate from one site or experience to the next.

The purchaser assumes that all suppliers have the same understanding of the technology they are proposing and are essentially offering the same solution or performance, albeit potentially at differenct price points.

The purchaser or their representative may unwittingly discourage technology innovation by requiring a minimum number of installations or years of experience for such offerings.

The purchaser assumes that the equipment proposed to meet their specification will perform in accordance with their expectations under a variety of ambient and plant conditions.

The purchaser concludes that the lowest priced option is the best long-term selection for their plant.

The purchaser attempts to assign all of the risk of performance and operation to the supplier of the equipment, despite the lack of operating experience for such equipment at the purchaser's site and under circumstances that might be present there.

1.1.3 Potential Assumptions and Positions of the Supplier

Based on historical precedent, if the supplier does not submit a low-price offering, they will not be successful in their bidding efforts.

The suppliers/bidders assume that they must meet all requirements of the specification or otherwise be disqualified from the bid process (i.e., make no exceptions to the specification).

They may assume that any features or safety margins added to their bid offering would render them less competitive, even though these features or safety margins may ultimately be in the best interest of the purchaser.

They may be less apt to offer innovative solutions or advancements in the technology for fear of adding costs to their offering or exposing themselves to additional risk in execution or performance.

They must seek any opportunity to reduce risk in an environment where profit margins are typically low and performance penalties may be significant.

While not all of the assumptions and positions listed above prevail in all bid processes, they are typical of the procurement dynamics for major capital equipment purchased in the U.S. power industry. Indeed, it can be argued that the personality of some capital equipment procurement processes is one of "low price" and skepticism regarding innovation versus cooperation and "teaming" between purchaser and prospective supplier. Indeed, if an architect engineer is under a fixed-price contract for the design and supply of a new plant, their motives for selecting a higher cost system, e.g., one with design innovations or additional performance margins, may be diminished. If there is basis for even some of the aforementioned assumptions and positions regarding the prospective bid dynamics, then it is certainly possible that the final selection of supplier and equipment design are not optimal.

At the time that this report was completed, the market for ACCs is supplied by only two major vendors. With the limited number of projects, each of them with large and costly proposal efforts and uncertainties associated with performance testing; the vendors will be more conservative. If the purchasers continue to buy low bid, the number of installations that do not meet owner expectations is likely to increase.

1.2 Objectives

The objective of this specification is to provide engineering and purchasing personnel with information they need to specify, procure, and commission ACCs that feature optimum design and performance characteristics for the application. In order to accomplish this, they will need information to assist in answering the following questions:

What are the primary ACC operating and performance problems?

What information should be provided to bidders in a specification and request for proposal (RFP)?

How should developers evaluate and compare bids for ACC supply?

What are the most important considerations in conducting performance and acceptance testing?

What are some key ACC commissioning and startup issues?

1.3 Process

Numerous specifications have been developed for ACCs, both internationally and in the United States. These specifications, for the most part, cover the design conditions, scope of supply, codes and standards, contract terms, and conditions, etc. However, a review of many of these specifications suggested that the ACC design points were based on prior experience and rules of thumb, but may not reflect optimum design criteria for the economic operating environment anticipated for the plant. Additionally, in many cases, these specifications have not addressed areas that might be problematic in terms of ACC performance, operation, and maintenance. The information contained in these ACC specifications was obtained through a number of site visits. Interviews with plant personnel and suppliers provided a balanced viewpoint on key issues. The following areas surfaced as ones deserving additional attention, beyond the historical level of attention they have received.

Wind Effects – Prevailing winds can be significant at many sites, especially given the typical height of air inlets and fans (e.g., 50–100 ft [15–30 m]) on an ACC. High winds can reduce inlet pressures on ACC upwind fans, leading to decreased airflow rates and cell thermal performance. Prevailing winds can also lead to recirculation of heated exhaust air from the ACC, also reducing ACC performance. This area of wind effects in total represents the major challenge associated with ACC specification, design, and performance.

Range of Operating Conditions – ACCs may be required to operate over ambient temperatures ranging from less than 0°F (-18ºC) to more than 110°F (43ºC). Further, they may be required to undergo "cold starts" (i.e., initial operation without a heat load) and operate successfully over a full range of heat loads. In doing so, particular attention to ACC design and operation is critical to prevent freezing of condensate as well as proper removal of noncondensables.

Fouling of ACC Coils – Many ACCs operate in areas with high ambient dust loadings. This is particularly true in the desert Southwest portion of the United States, where a number of ACCs have recently been commissioned. In some situations, pollen, insects, and other materials can foul heat exchange surfaces. Furthermore, leaky gearboxes lead to carryover of gearbox grease to heat exchange surfaces. It may also be the case that nearby fuel piles – including coal, hog fuel (i.e., wood waste), etc. – can contribute to the inlet air dust loadings to the ACC and resultant fouling. As a result of site visits, it is clear that potential dust loadings, fin-tube cleaning systems, and performance degradation trends warrant additional consideration.

Inlet Air Conditioning – A number of ACC owner/operators have experimented with and/or are using methods for inlet air cooling of their ACCs. The notion of reducing the inlet air dry-bulb temperature, particularly during periods of elevated temperatures, is obviously important when power output requirements are highest. Inlet air cooling typically involves evaporative cooling of the air via filming media or spray systems. In the case of film cooling, additional pressure drop on the inlet air side can be a challenge, since the cooling media is typically installed year-round but operated on occasions of high temperature. In the case of spray cooling, carryover of sprayed droplets can also be problematic. Indeed, spray cooling via atomized sprays has resulted in degradation of finned-tube surfaces at a number of sites. The main reason for this may be improper selection, positioning, and/or orientation of atomizing technologies. Accordingly, anyone considering the use of inlet air cooling via sprays should carefully design away from such conditions.

1.4 Overview of Air-Cooled Condenser Scope of Supply

The actual ACC scope of supply can include a number of areas outside of this specification. However, the following areas are typical of a U.S.-based solicitation.

1.4.1 Finned-Tube Bundle and System

This system is comprised of all finned tubes, including steam distribution manifolds and the condensate collection system.

1.4.2 Structure

Structure refers to all materials and systems required to support and anchor the ACC and its auxiliary equipment. This may also consist of necessary anchors and flanges required for interfacing with site-supplied foundations.

1.4.3 Steam Ducting

Steam ducting, which begins at the turbine exhaust flange and leads to the ACC, typically involves expansion joints, rupture discs, inspection ports, structural steel, necessary vent and drain connections, etc. Also included are blanking plates for each ACC duct necessary for leak testing.

1.4.4 Condensate Receiver Tank

This tank is of sufficient size for condensate collection and a heating/freeze protection system, if required by site conditions.

1.4.5 Air Removal System

The air removal system is comprised of a steam jet air ejector or vacuum pumps and associated skids as well as a basic control system. It also includes the vacuum deaerator system connected to the condensate tanks, vacuum pumps, etc.

1.4.6 Mechanical Equipment

Mechanical equipment includes fans, fan hubs, fan shrouds, drive shafts, gearboxes, and drive motors. It typically includes a protective screen at the inlet to the fan shroud.

1.4.7 Access

Access refers to all stairways, platforms, ladders, manways, etc. to safely access, inspect, maintain, and operate the ACC.

1.4.8 Hoists, Davits, Monorails

Encompassed here are all systems required to remove, convey across the ACC, lower to grade, and replace/maintain all mechanical equipment.

1.4.9 Abatement Systems

Abatement systems may involve noise abatement devices, wind screens, etc.

1.4.10 Instrumentation and Controls

This area refers to all temperature, pressure, flow, and level sensors as well as control devices to properly monitor and control the ACC. Included are thermowells and pressure ports in strategic testing and monitoring locations, and possibly a lightning protection system.

1.4.11 Spare Parts

Spare parts are defined as all backup parts and systems required to reliably maintain the ACC and its control systems. Also included are startup and maintenance lubricants for the gearboxes.

1.4.12 Lightning Protection System

This system, which may be provided by a specialty or electrical contractor or the ACC supplier, encompasses lightning rods, conductive cable, and associated electrical grounding. (Comprehensive lightning protection may not be required on ACCs located near taller structures such as stacks or boilers.)

1.4.13 Cleaning System

The cleaning system includes any vendor-recommended system designed for efficient medium-pressure cleaning of finned tubes.

1.4.14 Factory Testing

Factory testing refers to all shop and key subcontractor tests required to demonstrate that in-factory functionality and quality control requirements are met for each ACC.

1.4.15 Shipping

Shipping takes into account all packing, protection, loading, and transportation modes required to properly transport ACC equipment to the site.

Because the ACC supplier typically is not directly responsible for ACC field erection, construction-related aspects are not part of this specification. More detail on the above components is provided in Chapter 4.

1.5 References

a) D.G. Kröger. *Air-Cooled Heat Exchangers and Cooling Towers*. Thermal Flow Performance Evaluation and Design, Department of Mechanical Engineering, University of Stellenbosch, Private Bag X1, Matieland, 7602, South Africa, 1998.

b) M.W. Larinoff, W.E. Moles, and R. Reichelm. “Design and Specification of Air-Cooled Condensers,” *Chemical Engineering*, May 22, 1978.

c) R. Chandran. “Maximizing Plant Power Output Using Dry Cooling Systems,” presented at the ASME Power Conference, March 30 - April 1, 2004.

d) D.F Sanderlin and R. Chandran. "Operation of Air-Cooled Condensers in Cold Climates," presented at the International Joint Power Generation Conference, 1992.

e) M.S. Sohal and J.E. O'Brien. "Improving Air-Cooled Condenser Performance Using Winglets and Oval Tubes in a Geothermal Power Plant," *Geothermal Resources Transactions*, Vol. 25, Aug. 2001.

2 CONDENSER COOLING SYSTEMS OVERVIEW – INTRODUCTION AND BACKGROUND

All modern power plants with steam turbines are equipped with a cooling system to condense the steam as it leaves the steam turbine and to maintain a desired turbine exhaust pressure (back pressure). A variety of cooling systems are used at power plants, including once-through cooling, recirculating wet cooling, dry cooling, and hybrid (wet/dry) cooling. Commonly employed cooling systems are described below, while references [a-d] provide information on cooling systems that have seen less use in the United States, at least as of this writing.

2.1 Once-Through Cooling

In once-through cooling, water is withdrawn from a surface water source (e.g., river, lake, ocean, etc), passed through the tubes of a conventional shell-and-tube surface condenser, and returned to that source at an elevated temperature. It is shown schematically in Figure 2-1. Historically, this was the commonly used form of cooling at most plants and still is used on approximately one-half of the nation's generating capacity. It has efficiency advantages related to low auxiliary power requirements (low pumping head and no fans) and lower cooling water temperature given meteorological conditions. However, it is rarely used on new plants due to permitting difficulties related to thermal discharge and intake fish protection regulations.

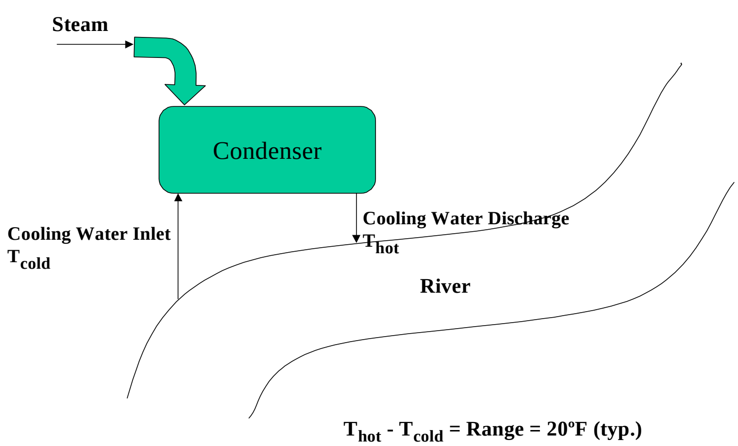

Figure 2-1
Schematic of Once-Through Cooling

2.2 Recirculating Wet Cooling

Recirculating cooling systems are similar to once-through cooling in that the steam is condensed in a water-cooled surface condenser. Recirculating systems differ from once-through cooling systems in that the heated cooling water is not returned to the environment. Rather, this water is sent to a cooling element/device (typically a cooling tower, but in some cases cooling ponds, spray-enhanced cooling ponds, or spray canals), where it is cooled and then recirculated to the condenser. The cooling is substantially accomplished through the evaporation of a small fraction of the circulating water (typically 1–2%), which must be made up from local water sources. Schematics of common wet cooling tower designs, including mechanical- and natural-draft towers in counterflow and crossflow configurations, are shown in Figure 2-2. The most commonly used system in recent years is the counterflow, mechanical draft tower.

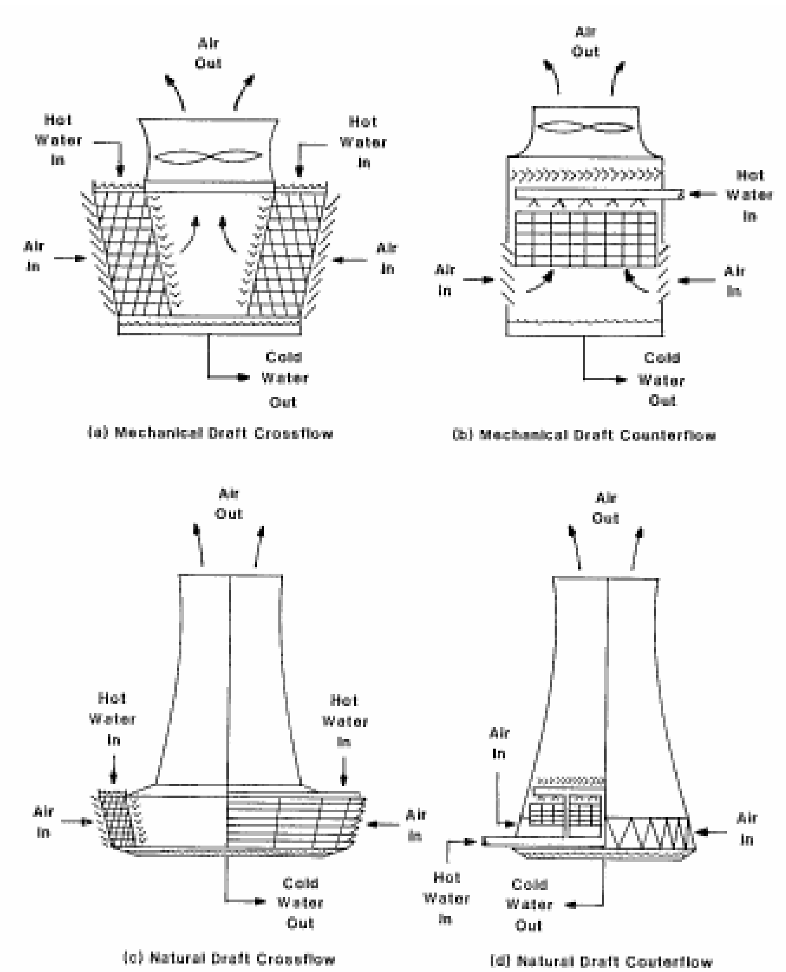

Figure 2-2
Common Wet Cooling Tower Types

2.3 Dry Cooling

Dry cooling refers to the cooling systems in which the ultimate heat rejection is achieved by sensible heating of atmospheric air passed across finned-tube heat exchangers, much like what is accomplished in an automobile radiator. The two general types of dry systems are referred to as direct and indirect dry cooling. Indirect dry cooling systems condense the steam in a surface condenser, as do once-through and recirculating systems, but the heated cooling water is then cooled in an air-cooled heat exchanger. This system, which has not been used in the United States, is shown schematically in 2-3.

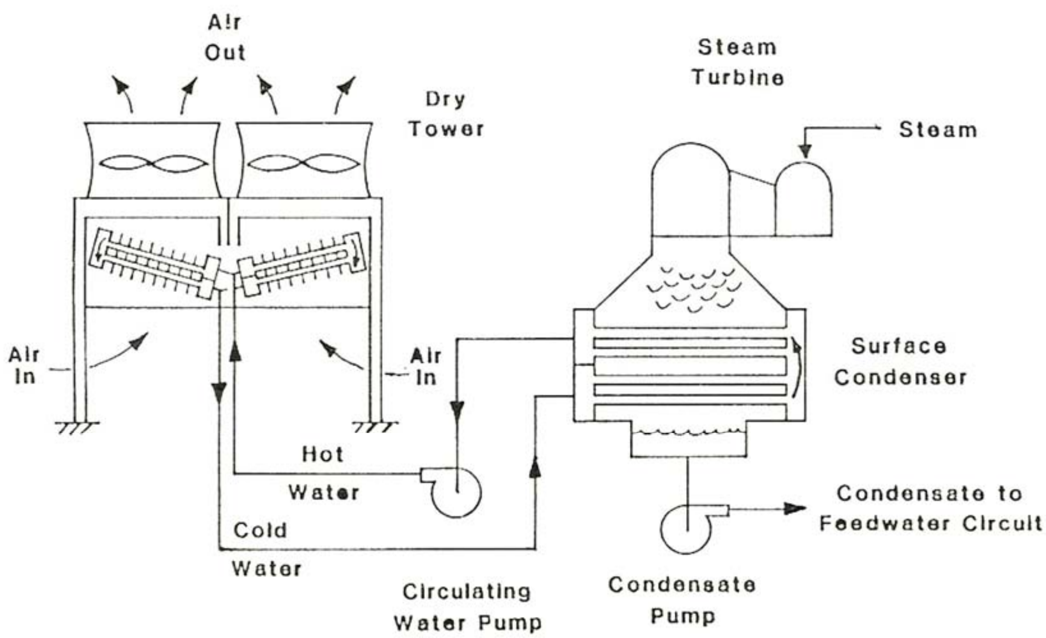

Figure 2-3
Schematic of Indirect Dry Cooling System

In a direct system, the steam is ducted directly to an ACC. Either of these systems can be implemented either as mechanical-draft or natural-draft units.

The direct system with a mechanical-draft ACC has been the system of choice for all dry cooling in the United States and will be the sole focus of this document. The remainder of this chapter identifies and discusses important considerations involved in specifying the design and performance requirements for an ACC for a particular power plant application with given site characteristics.

2.4 Air-Cooled Condensers in the United States

The use of ACCs at power plants in the United States has become more common in recent years. Table 2-1 provides a list of known U.S. installations as of late 2004.

Table 2-1
ACC Installations in the United States

State	Vendor*	Customer	Project/Site	Plant Capacity	Turbine Capacity**		Service
				MWe	MWe	t/h	Date
NY	SPX	Astoria Energy	Astoria			460	2006
MN	GEA	FibroWatt	Fibrominn Biomass		55		2004
AL	GEA	US Army	Ft. Wainwright CHP Plant		3 x 5		2004
CA	GEA	Noresco	32nd St. Naval Station		5		2004
AZ	GEA	Pinnacle West	Snowflake		3		2004
UT	GEA	PacifiCorp	Curant Creek		200		2004
NV	Hamon	Genwest LLC/Silverhawk	Silverhawk	500			2004
NY	Hamon	NYPA/GE/Sargent & Lundy	Poletti	450			2004
NV	Hamon	Reliant/Sargent & Lundy	Big Horn	650			2003
NY	Marley/BDT	Key Span	Ravenswood			278	2002
WA	Marley/BDT	Parsons	Chehalis			490	2001
MA	Hamon	Sithe/Raytheon	Mystic	1600			2002
MA	Hamon	Sithe/Raytheon	Fore River	800			2002
NV	Marley/BDT	Mirant	APEX			697	2001
CA	GEA	Calpine	Otay Mesa		277		2001
MS	GEA	Reliant	Choctaw County		350		2001
NV	GEA	Nevada Power	Moapa		2 x 200		2001
WY	GEA	Black Hill Generation	WyGen I/Unit 3		80		2001
PA	GEA	Reliant	Hunterstown		350		2001
CA	Hamon	Calpine/Bechtel	Sutter	500			2001
CO	GEA	Front Range Power	Front Range		150		2001
MA	Marley/BDT	ABB	Bellingham			2 x 256	2001
WA	GEA	GoldendaleEnergy	Goldendale		110		2000
NY	GEA	PG&E Generating	Atherns		3 x 120		2000
CT	Marley/BDT	ABB	Lake Road			3 x 256	2000
MA	Marley/BDT	ABB	Blackstone			2 x 256	2000
TX	Marley/BDT	ABB	Midlothian			4 x 255	2000
TX	Marley/BDT	ABB	Hays			2 x 256	2000
1990's							
ME	GEA	Rumford Power Associates	Rumford		80		1999
NV	GEA	Sempra Energy / Reliant Energy	El Dorado		150		1999
RI	GEA	Tiverton Power Associates	Tiverton		80		1999
MA	GEA	Energy Mangement Inc.	Dighton		60		1998
CA	Marley/BDT	Bechtel, Crockett	Crockett			275	1996
IL	GEA	Browning Ferris Gas Serv. Inc.	Mallard Lake Landfill		9		1996
MT	Marley/BDT	Billings Generation	Rosebud			210	1995
IA	GEA	Municipal Electric Utility	Cedar Falls		40		1994
MA	GEA	Ogden Martin Sys. of Haverhill	Haverhill Extension		46.9		1994
MI	GEA	Browning Ferris Gas Serv. Inc.	Arbor Hill		9		1994
MN	GEA	Browning Ferris Gas Serv. Inc.	Pine Bend Landfill		6		1994
NY	GEA	MacArther Res. Rec. Agency	Islip		11		1994
NY	GEA	Dutchess County	Dutchess Co. Extension		15		1993
VA	GEA	Mission Energy	Gordonsville		2x50		1993
NY	GEA	Odgen Martin Sys.	Onondaga County		50		1992
NY	GEA	Falcon Seaboard	Saranac		80		1992
WY	GEA	Black Hills Power & Light	Neil Simpson II		80		1992
AK	GEA	University of Alaska	Fairbanks		10		1991
MA	Marley/BDT	CRS Sirrine, Lowell	Lowell			73	1991
NJ	GEA	Odgen Martin Sys.	Union County		50		1991
NY	Hamon	Ogden/Marin/Huntington	Huntington	35			1991
NJ	GEA	Cogen Technologies Inc.	Linden		285		1990
NJ	GEA	Intercontinental Energy	Sayreville		100		1990
NY	Marley/BDT	Indeck Energy	Silver Springs			55	1990
PA	GEA	Falcon Seaboard	Norcon-Welsh		20		1990
WA	GEA	Intercontinental Energy	Bellingham		100		1990
WV	GEA	Energy America Southeast	North Branch		80		1990
1980's							
CT	GEA	Oxford Energy	Exeter		30		1989
WA	GEA	Wheelabrator Environ. Sys.	Spokane		26		1989
NY	GEA	TBG Cogen	Grumman		13		1988
MA	GEA	Ogden Martin Systems	Haverhill		47		1987
PA	GEA	ABB	Hazleton		67.5		1987
IL	GEA	Chicago Northwest	Chicago		1		1986
MA	GEA	American Ref-Fuel	SEMASS		54		1986
ME	GEA	Wheelabrator Sherman Energy	Sherman		20		1985
CA	GEA	Pacific Ultrapower	Chinese Station		22.4		1984
CA	GEA	Pacific Gas & Electric	Gerber		3.7		1981
1970's							
AK	GEA	Chugach Electric	Beluga		65		1979
WY	GEA	Black Hills Power/Pacific Power	Wyodak		330		1977
CA	GEA	Exxon	Benicia		NA		1975

2.4.1 General Description

A schematic of an ACC is shown in Figure 2-6. Figure 2-4 provides a photo of an ACC installation at the Wyodak Station.

In the direct dry cooling system, turbine exhaust steam is ducted from the turbine exit through a series of large horizontal ducts to a lower steam header feeding several vertical risers. Each riser delivers steam to a steam distribution manifold that runs horizontally along the apex of a row of finned-tube, air-cooled heat exchangers arranged in an A-frame (or delta) configuration. A typical full-scale ACC consists of several such rows, sometimes referred to as “streets” or “lanes.” (Three out of five streets are visible in Figure 2-4.)

Each row consists of several cells. Each cell consists of several bundles of finned tubes arranged in parallel, inclined rows in both walls of the A-frame cell, as shown in Figure 2-6. Steam from the steam distribution manifold enters the tubes at the top, condenses on the inner tube walls and flows downward (concurrent with remaining uncondensed steam) to condensate headers at the bottom of the bundles. One cell in each row – typically one of the center cells out of five or six along the row – is a “reflux” or “dephlegmator” cell. This cell functions to remove noncondensable gases from the condenser. Uncondensed steam from the other cells in the row along with entrained noncondensables flow along the condensate header to the bottom of the reflux-cell tube bundles. An air-removal system (vacuum pumps or steam ejector) removes the noncondensables through the top of the reflux cell bundles. Additional condensation takes place in this cell and the condensate runs down (flowing countercurrent to the entering steam) into the condensate header. The condensate flows by gravity to a condensate receiver tank from which it is pumped back to the boiler or heat recovery steam generator (HRSG).

The finned tubes, typically about 30–40 ft (9–12 m) long, are clustered in bundles typically 8 ft (2.5 m) across. For typical cell plan dimensions of 40 ft x 40 ft (12 x 12 m), there are five bundles on each face, or 10 bundles per cell. Finned-tube geometries have evolved from circular tubes with wrapped, round fins to elliptical tubes with plate fins, usually arranged in two to four rows in a staggered array. Most recent ACC designs have used elongated, nearly rectangular flow passages separated by plate fins, referred to as a single row condenser (SRC). Figure 2-7 shows pictures of several finned-tube geometries, while Figure 2-8 depicts the SRC finned tube.

Large axial flow fans – typically 28–34 ft (8.5–10 m) in diameter – are located in the floor of the cells, providing forced-draft air cooling to the finned-tube heat exchangers. These typically low-speed fans feature two-speed (100/50 rpm) drives and five to eight blades. Designs vary considerably depending on allowable noise levels at the site.

A 500-MW combined-cycle plant – condensing approximately 1.0–1.2 million pounds of steam per hour (125–150 kg/s) – might typically have 30–40 cells, each having the aforementioned geometry, arranged in a 5 x 6 or 8 x 5 or two 4 x 5 layouts. The ACC footprint is typically 200 ft x 250 ft (60 m x 75 m). Vertical steel columns and extensive bracing support the cells and fans. The fan deck is often 60–80 ft (20–25 m) above grade, with the steam duct at the top of the cells rising to 100–120 ft (30–35 m) or more above grade.

The cells are typically surrounded by a windwall to reduce the possibility of hot air recirculation and wind effects. This windwall is not shown in the schematic (Figure 2-6), but is visible in the Figure 2-5 photograph as a yellow wall surrounding the A-frame tube bundles.

Figure 2-4
Air-Cooled Condenser at El Dorado Generating Station – Depicting 3 of 5 Streets

Figure 2-5
Wyodak Air-Cooled Condenser

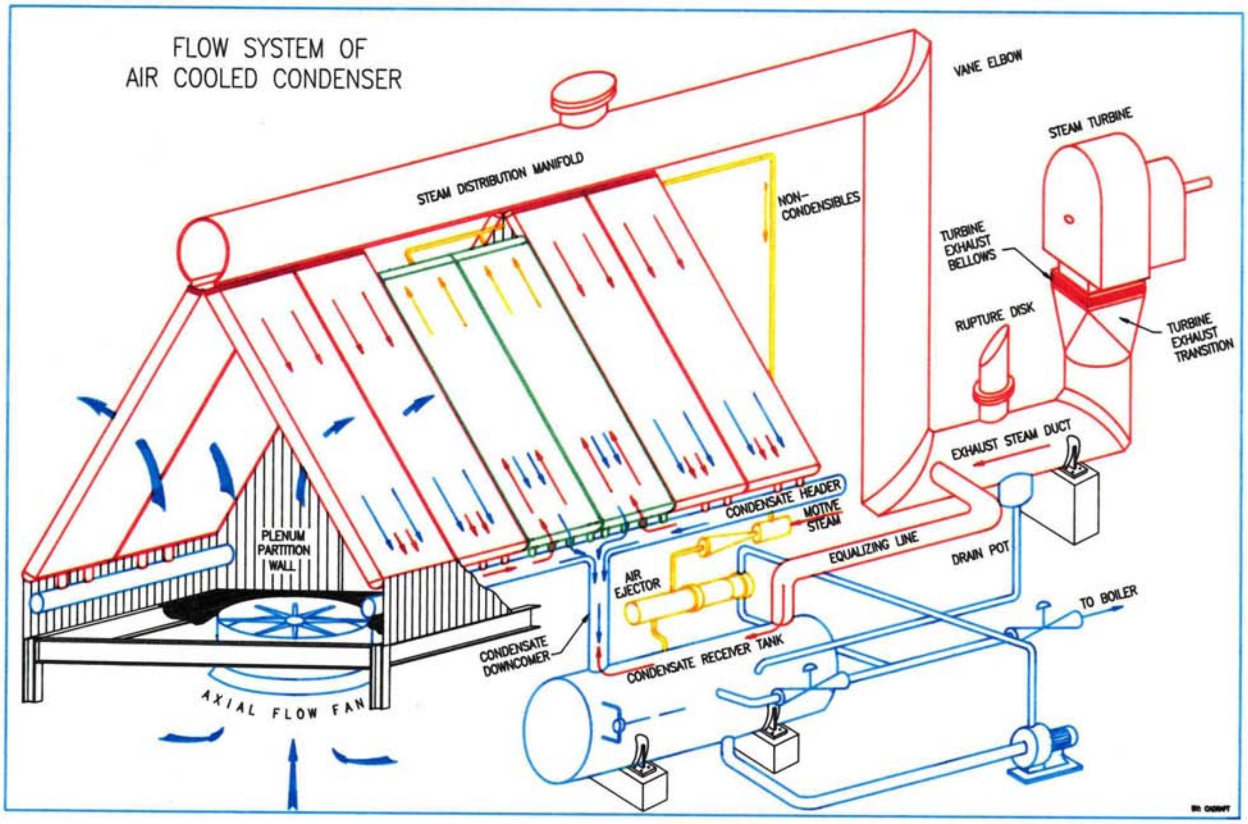

Figure 2-6
Schematic of an Air-Cooled Condenser (Courtesy of Marley Cooling Tower Company)

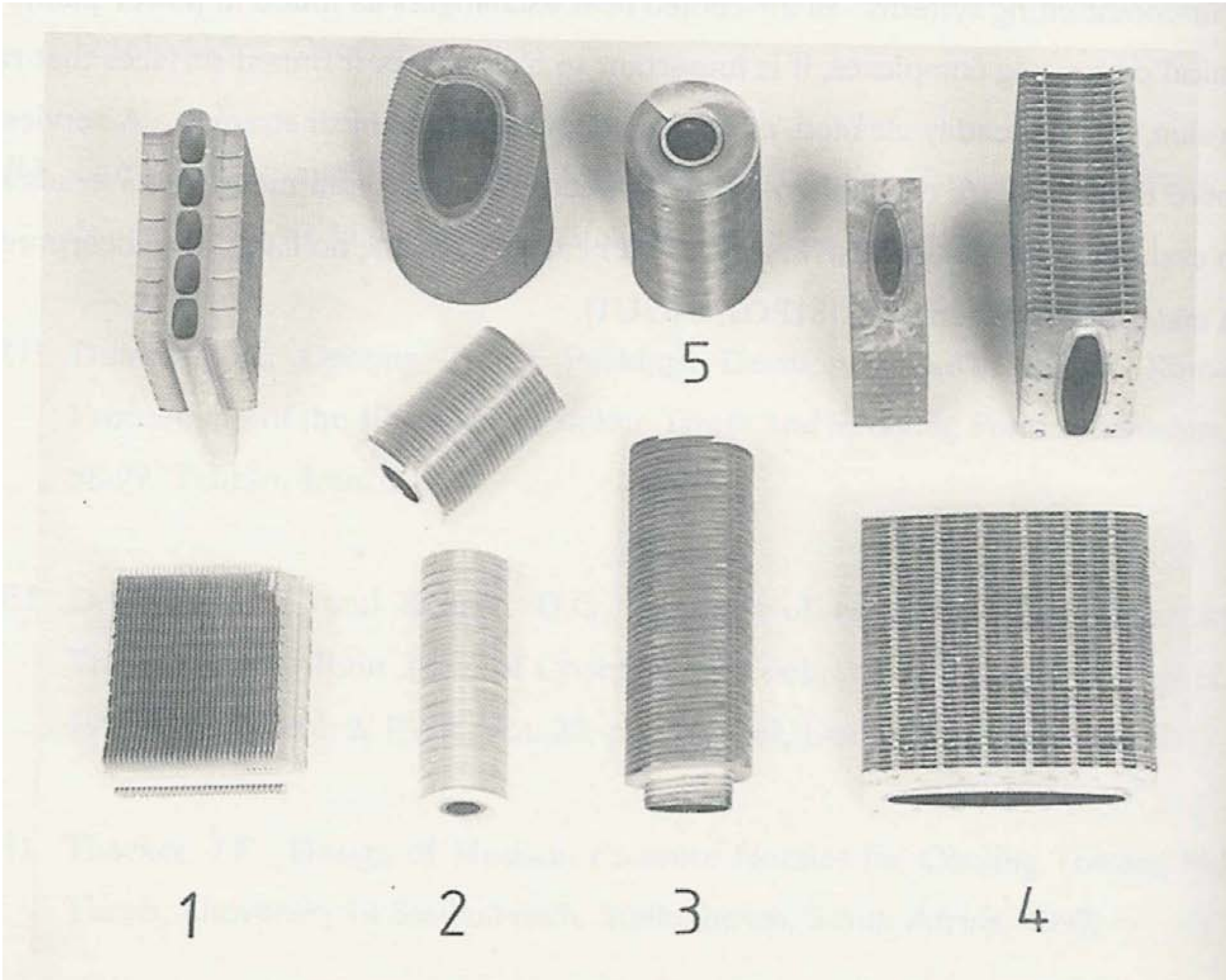

Figure 2-7
Photographs of Several Finned-Tube Geometries [e]

Figure 2-8
Photograph of a Single Row Condenser Finned Tube [e]

2.4.2 Air-Cooled Condenser Performance

The previous section described general ACC circuitry as well as the flow paths of the steam, condensate, and cooling air. The condensation of steam requires the removal and transfer of large quantities of heat (~1000 Btu/lb steam [2.3 x 10^6 J/kg]) to the atmosphere. The heat of condensation is transferred to the cooling air stream by the temperature difference between the condensing steam and the cooling air.

The condensing side temperature is nearly isothermal, since a saturated two-phase mixture of water and steam is present. The condensing temperature is related to the condensing pressure, which is equal to the turbine exit pressure, or back pressure, minus any pressure drop in the steam lines between the turbine exhaust flange and the heat exchanger bundle inlet. Figure 2-9 gives the relationship between the condensing temperature and condensing pressure over the range of pressures typically relevant to cooling system design.

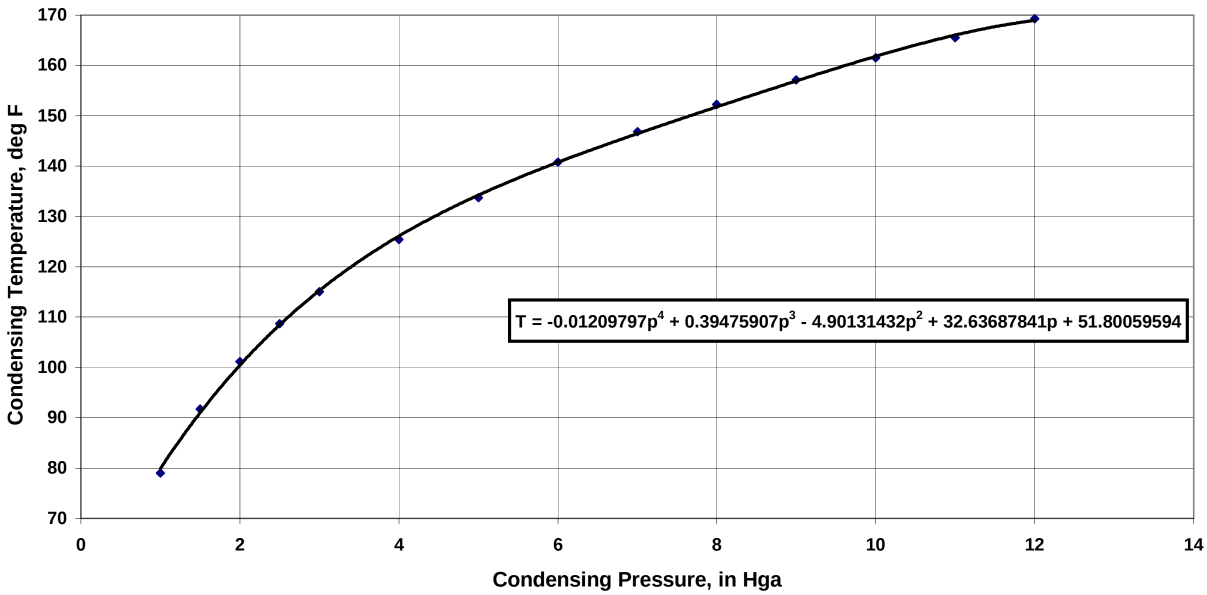

Figure 2-9
Condensing Temperature vs. Condensing Pressure

The cooling air enters at approximately the ambient dry-bulb temperature and is heated, typically by 25–30°F (~14–17°C), as it passes across the heat exchanger bundles. Under some conditions, a small amount of the heated air leaving the ACC may be entrained (recirculated) into the inlet air stream, resulting in an increased inlet temperature to some cells. There is a discussion of selection of the ACC design point in Section 3.2.

2.4.2.1 ACC Performance Characteristics

The ACC design point is frequently characterized by the difference between the condensing temperature (T_{cond}) and the entering air temperature ($T_{a\ inlet}$), known as the initial temperature difference (ITD).

$$ITD = T_{cond} - T_{a\ inlet} \qquad \textbf{Equation 2-1}$$

For a given ACC, the heat load Q [Btu/hr (W/s)] is related to the ITD by

$$Q/ITD = \text{Constant} \qquad \textbf{Equation 2-2}$$

Alternatively, *for a given heat load*, the size (number of cells, heat transfer surface) is inversely related to the ITD as

$$\text{ACC "Size"}\ \alpha\ 1/ITD^{\beta} \qquad \textbf{Equation 2-3}$$

where a low ITD corresponds to a large ACC.

Figure 2-10 shows ACC size (as number of cells) vs. ITD for a heat load of about 1.0 billion Btu/hr (0.3 x 10^9 W) (approximately 1.0 million lb steam/hr [125 kg/s]). It is important to note that the number of cells does not vary continuously, but is always a number that can be factored into a reasonable rectangular or square array. Typical values are 20 (4 x 5 or 5 x 4), 25 (5 x 5), 30 (5 x 6 or 6 x 5), 36 (6 x 6), 40 (5 x 8, two 5 x 4), etc.

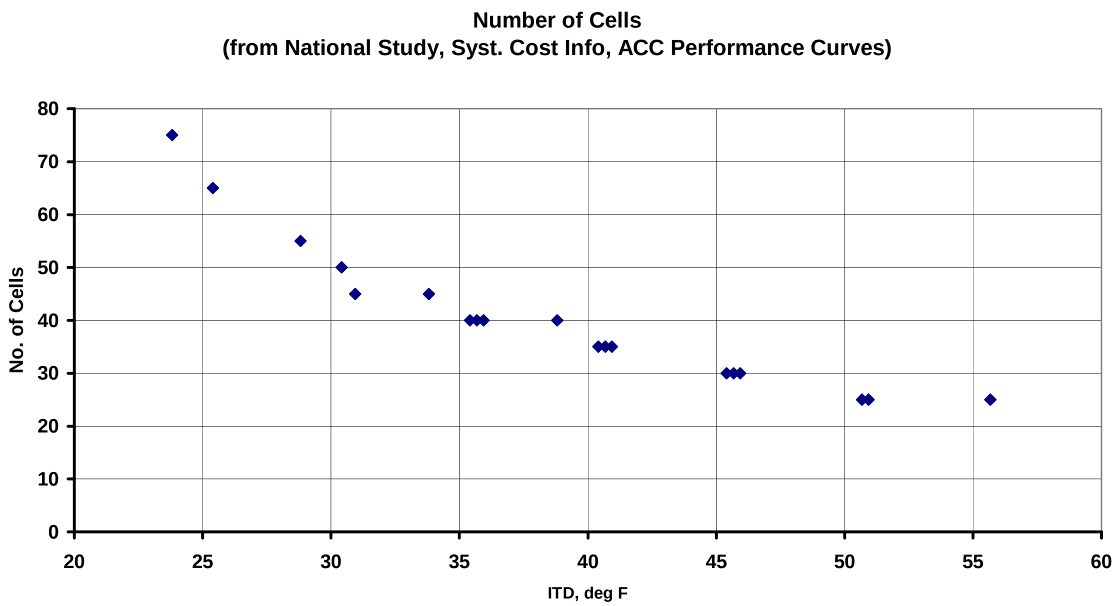

Figure 2-10
Air-Cooled Condenser Size vs. Initial Temperature Difference

(For a condensing duty of ~1.0 million lb steam/hour)

ACC fans for recent designs are typically rated at 200 hp (150 kW), as this is a common motor size, and the input power may be 480-V supply. With ITD increases at constant heat load, the design fan power decreases along with the ACC size and number of cells. As noted earlier, the number of cells does not change continuously. However, in order to maintain a reasonably square array, the number may shift from 25 to 30 to 36, with ITD variations from the mid-50s to the upper 30s (as, for example, in Figure 2-11). Over this range, the performance shifts are modulated by varying the design fan horsepower over the range of ITDs for a given number of cells. Additional variability can also be obtained by changing tube length, fin spacing, and other design values. Figure 2-11 shows a typical range of design fan power for an ACC sized for 1.0 million lb steam/hour (125 kg/s) over a wide range of ITDs corresponding to various ACC sizes.

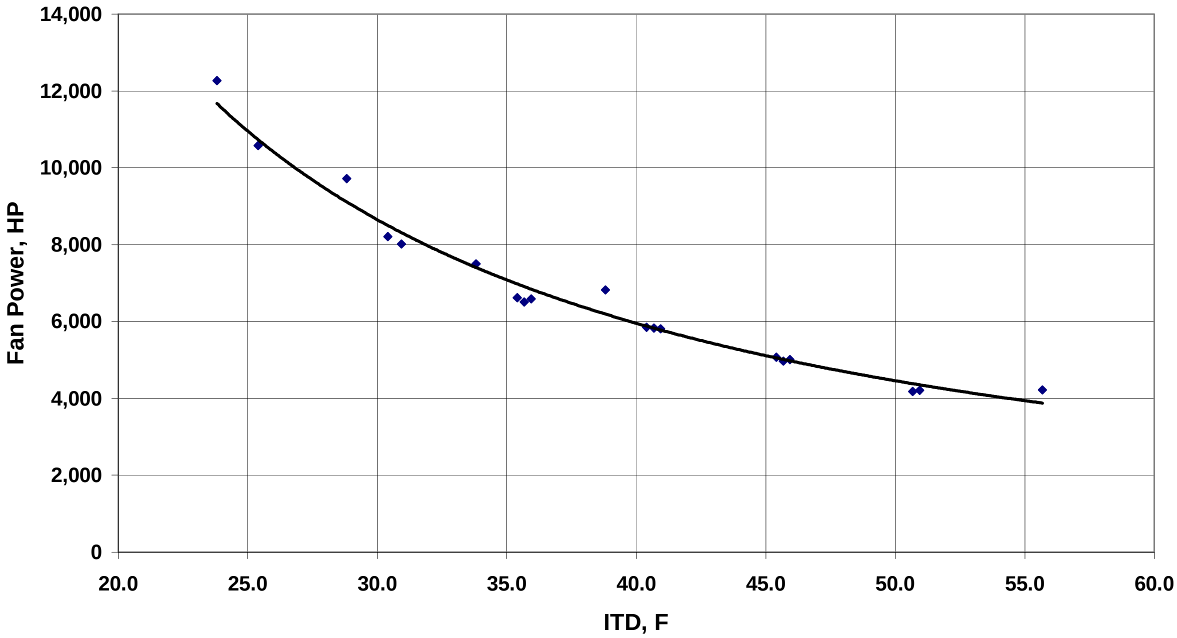

Figure 2-11
Design Fan Power vs. Initial Temperature Difference

(For a heat duty of 1.0 million lb steam/hour [125 kg/s])

2.5 References

a) J.S. Maulbetsch and K.D. Zammit. "Cost/Performance Comparisons of Alternative Cooling Systems," presented at the California Energy Commission (CEC)/EPRI Advanced Cooling Strategies/Technologies Conference, Sacramento, California, June 1-2, 2005.

b) Balogh and S. Zolton. "The Advanced Heller System: Technical Features and Characteristics," presented at the CEC/EPRI Advanced Cooling Strategies/Technologies Conference, Sacramento, California, June 1-2, 2005.

c) P. Demakos. "Closed-Loop Wet Surface Evaporative Cooling Systems for Steam Condensing and Auxiliary Loop Cooling in Water Limited Power Plants," presented at the CEC/EPRI Advanced Cooling Strategies/Technologies Conference, Sacramento, California, June 1-2, 2005.

d) Benefiel, J.S. Maulbetsch, and M. DiFilippo. "Water Conservation Options for Wet-Cooled Power Plants," presented at the CEC/EPRI Advanced Cooling Strategies/Technologies Conference, Sacramento, California, June 1-2, 2005.

e) D.G. Kröger. *Air-Cooled Heat Exchangers and Cooling Towers*. Thermal Flow Performance Evaluation and Design, Department of Mechanical Engineering, University of Stellenbosch, Private Bag X1, Matieland, 7602, South Africa, 1998.

3
AIR-COOLED CONDENSER SPECIFICATION

Specification of an ACC requires consideration of two key issues. The first is the approach to choosing an ACC to meet a particular level of performance at specified operating conditions for a given plant at a given site. The second, and more complex question, is the method of determining the *preferred* or *optimum* design point for that plant at that site.

The following discussion will be organized into two sections:

ACC sizing for a given operating point

Determination of the optimum design point

The initial discussion will illustrate the approaches for the simplest set of design considerations. In addition, there are a number of plant and site characteristics and business factors that influence the choice of the optimum ACC. These will be identified and discussed later in this section. The reader should also review references [a-g] at the end of this section along with proceedings of the CEC/EPRI Advanced Cooling Strategies/Technologies Conference, held June 1-2, 2005, in Sacramento, California.

3.1 Sizing an Air-Cooled Condenser

The minimum amount of information required to establish the simplest ACC design point is as follows:

Steam flow, W, lb/hr (kg/s)

Turbine exhaust steam quality, x (lb dry steam/lb turbine exhaust flow)

Turbine back pressure, p_b, inch(es) of mercury, atmospheric (in. HgA) (kPa)

Ambient dry-bulb temperature, T_{amb}, ºF (ºC)

Site elevation, ft (m) above sea level

“Steam flow” refers to the total flow passing through the steam turbine exhaust flange and consists of both dry steam and entrained liquid water droplets.

“Steam quality” refers to the fraction of the steam flow that is dry steam and is expressed as a decimal fraction or a percent. All dry steam at saturation conditions has a quality of 100% (x = 1.0). An equivalent description sometimes used is “steam moisture” (ξ), defined as the percent of liquid water in the “steam flow.” Therefore,

$$\xi = 1.0 - x \qquad \textbf{Equation 3-1}$$

These properties are used, along with the thermodynamic properties of steam and water (including the latent heat of vaporization, h_{fg} [Btu/lb (J/kg)], at the design condensing pressure) to determine the heat load, Q [Btu/hr (W)], which must be handled by the ACC. The heat load is determined by the total steam flow, and the difference between the enthalpy of the inlet steam, $h_{steam\ inlet}$ [Btu/lb (J/kg)] and the enthalpy of the leaving condensate, h_{cond} [Btu/lb (J/kg)], according to the following equation:

$$Q[\text{Btu/hr (W)}] = W[\text{lb/hr (kg/s)}] * x[\text{lb/lb (kg/kg)}] * h_{fg}[\text{Btu/lb (J/kg)}] \qquad \textbf{Equation 3-2}$$

The turbine steam flow and quality at the plant design load are obtained from information provided by the turbine vendor.

Specification of the five quantities above is sufficient to obtain a "budget" estimate from ACC vendors. The following example illustrates the considerations in selecting an appropriate design point.

An ACC for installation at a 500-MW (nominal), gas-fired, combined-cycle plant located in an arid desert region might select the following design values:

Steam flow, W, lb/hr (kg/s):	1.1×10^6 (137.5)
Quality, x, lb/lb (kg/kg)	0.95 (0.95)
Back pressure, p_b, in. HgA, (kPa)	4.0 (13.5)
Ambient temperature, $T_{amb,}$ F (C)	80 (26.7)
Site elevation – Sea level	[p_{amb} = 29.92 in. HgA (101.3 kPa)]

The values were selected as follows:

3.1.1 Steam Flow

Figure 3-1 displays the design steam flow for a number of modern plants plotted against *steam turbine* output. Although some differences exist among plants of comparable size, a reasonable correlation is shown and given by

$$\mathbf{W(lb/hr) = 17{,}459 * (MW_{steam})^{0.8132}} \qquad \textbf{Equation 3-3}$$

For a nominal 500-MW, 2 x 1 combined-cycle plant, the steam-side capacity is approximately one-third of the plant total, or about 170 MW, with a corresponding steam flow of approximately 1.1×10^6 lb/hr (137.5 kg/s).

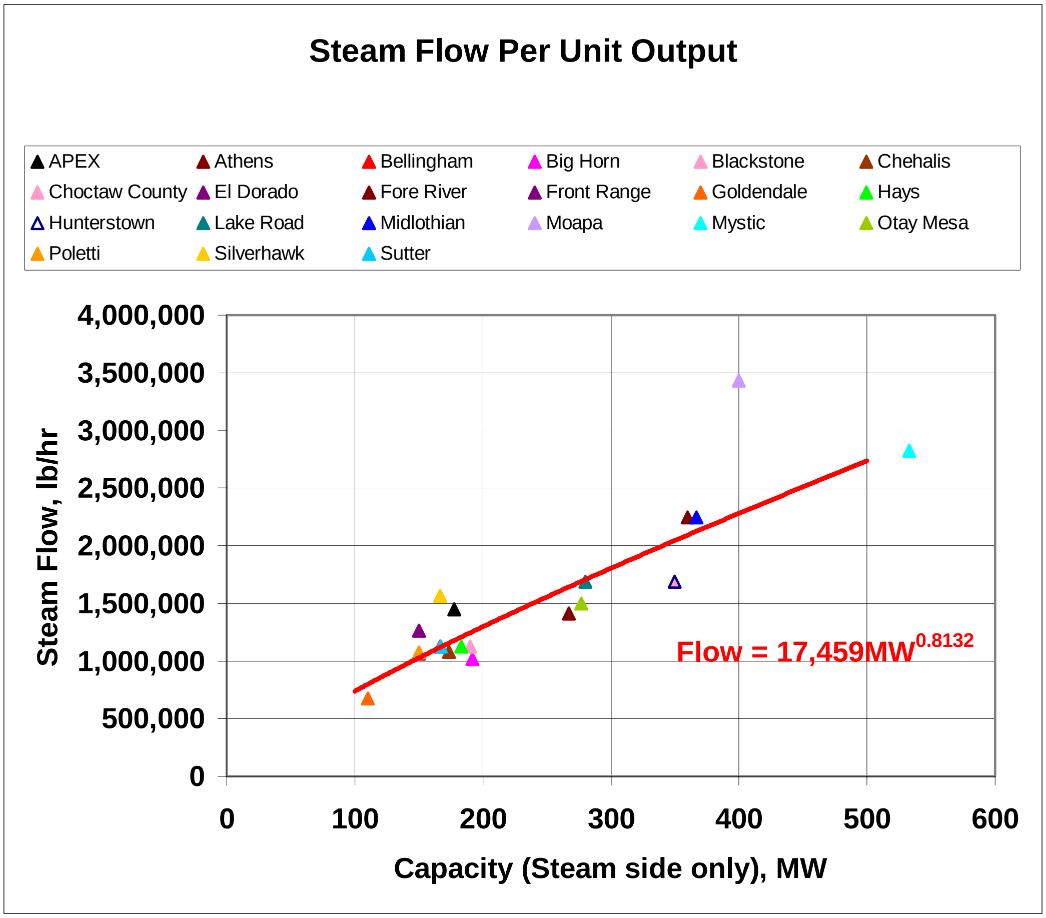

Figure 3-1
Steam Flow vs. Turbine Output

3.1.2 Steam Quality

Turbine steam exit quality (or enthalpy) must be obtained from the specific turbine design information or be determined from full-scale turbine tests. Typical values range from 0.92–0.98. For estimating purposes, a quality of 0.95 (5% moisture) represents a reasonable value.

3.1.3 Turbine Back Pressure and Ambient Temperature

For a given heat load, the combination of turbine back pressure and ambient temperature at the design point essentially determines ACC size, fan power, cost, and off-design performance.

Back Pressure – Over the normal operating range, the turbine efficiency improves (heat rate decreases) as the back pressure is lowered. Figure 3-2 displays a typical load correction vs. back pressure curve for a turbine selected for use on a combined-cycle plant with an ACC. Below about 2.0–2.5 in. HgA (6.8–8.5 kPa), no further reduction in heat rate is achieved and, in some instances, a slight increase occurs. Most turbines are restricted to operating at back pressures below 8.0 in. HgA. Typical guidelines are: "alarm" @ 7.0 in. HgA (23.7 kPa) and "trip" @ 8.0 in. HgA (27.1 kPa). For this example, the back pressure was set at an intermediate value of 4.0 in. HgA (13.5 kPa).

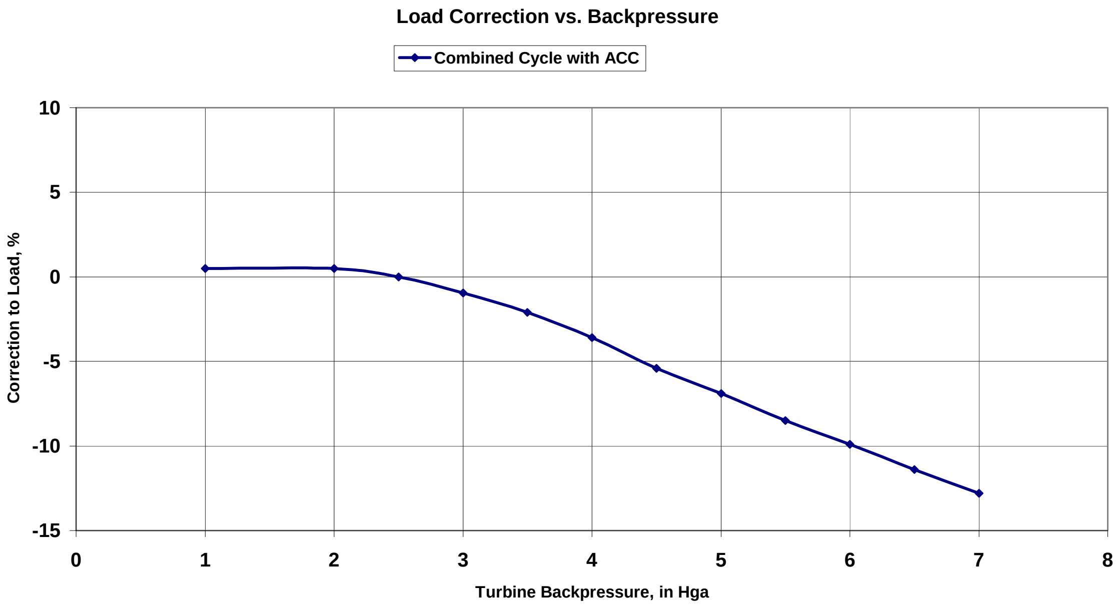

Figure 3-2
Steam Turbine Performance vs. Back Pressure

Ambient Temperature – At the desert site chosen for this example, the ambient temperature varies widely during the year. Figure 3-3 shows a temperature duration curve based on 30-year average data from El Paso, Texas. Other southwestern sites are comparable. A summer average temperature of 80 ºF is reasonably consistent with the choice of an intermediate back pressure for an example operating point.

A back pressure of 4.0 in. HgA (13.5 kPa) corresponds to a condensing temperature of 126.1ºF (52.3ºC), giving an ITD (from equation 2-1) of 46.1ºF (25.6ºC). From Figures 2-10 and 2-11, this would require an ACC of about 30 cells and a fan power requirement of about 5000 hp (3750 kW).

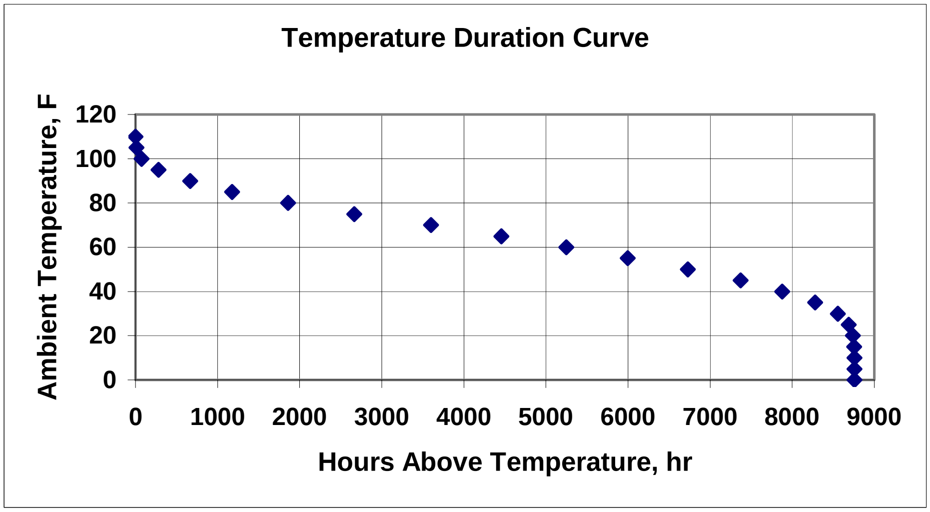

Figure 3-3
Example Temperature Duration Curve for an Arid Southwest Site

3.2 Selecting the "Optimum" Design Point

As noted above, the choices of back pressure and ambient temperature essentially fix the ACC design size for a given heat load. However, there is no assurance that the choices above result in a preferred design for the particular plant, site, and business conditions. This design optimization requires that selection of the ACC design point be based on an informed tradeoff between initial capital cost of the installed equipment and the operating and penalty costs to be incurred during the operating life of the plant.

A large ACC with a low ITD entails high capital costs and high fan power consumption. However, it achieves lower turbine back pressures at any given ambient temperature with correspondingly higher plant efficiency and output throughout the year. In addition, a large ACC may defer or avoid the need to reduce load on the highest temperature days during the summer. These tradeoffs are illustrated schematically in Figure 3-4.

Figure 3-5 [g] provides a trend of ACC design choices for modern ACCs. Over the past 20 years, experience and market forces have led to the selection of larger ACCs (lower ITDs). Such ACCs have resulted in higher capital cost but improved performance, greater plant efficiency throughout the year, and increased plant capacity during the hotter periods of the year. The trend has moved toward ITDs in the mid-40 ºF range.

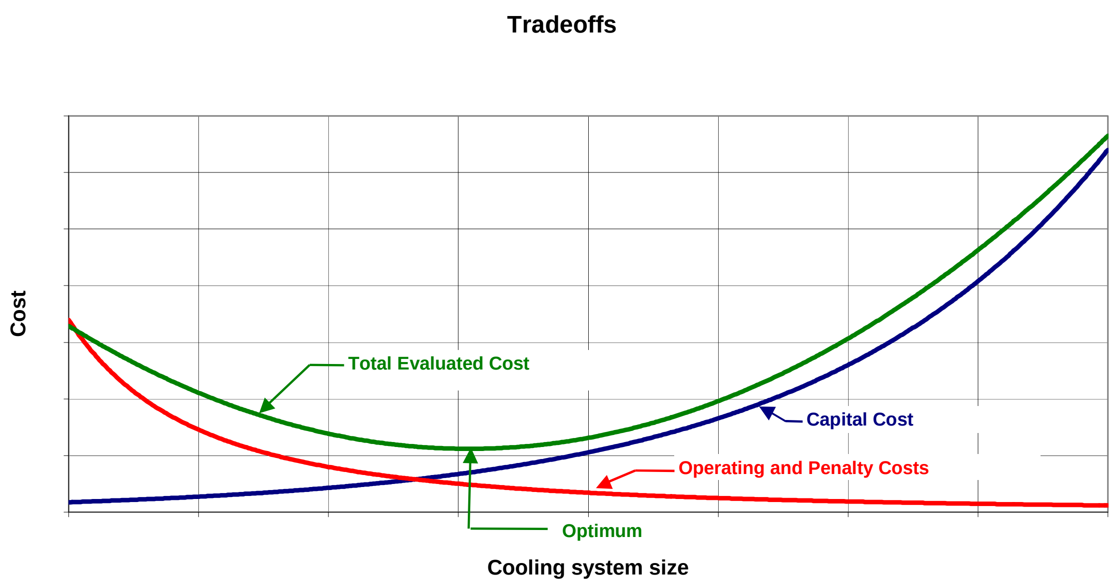

Figure 3-4
Schematic of Air-Cooled Condenser Cost Tradeoffs

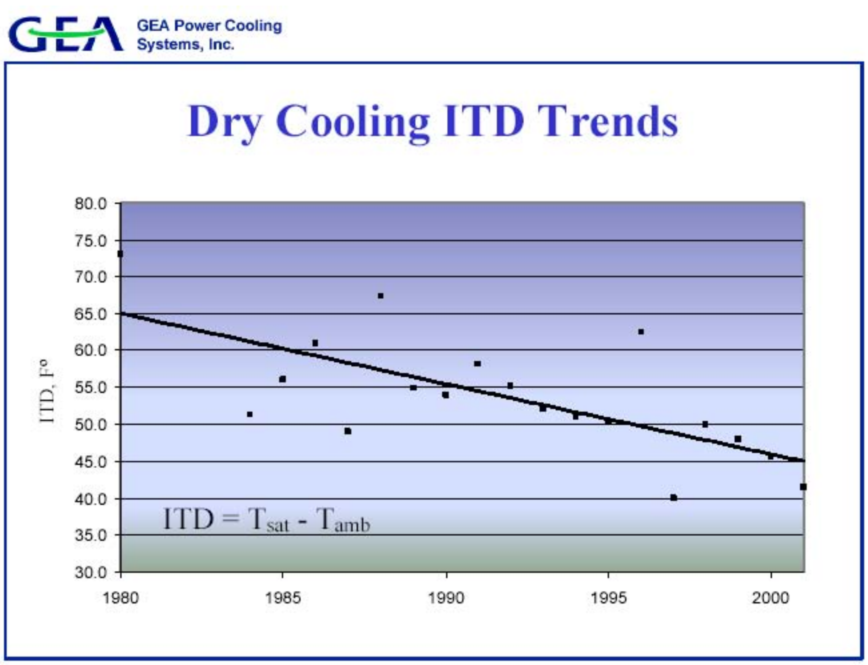

Figure 3-5
Dry Cooling Initial Temperature Difference Trends

In general, there are two approaches to selecting the optimized design point. The first, and most common, is to base the choice on a few performance goals at selected ambient conditions during the year. The second is to perform a complete optimization analysis to determine which ACC specification would correspond to the minimum annual (or lifetime) total evaluated cost.

3.2.1 Selected Performance Goals

The choice of desired performance at a few conditions depends on a knowledge of plant operating characteristics (such as heat rate and capacity as a function of ambient temperature and steam turbine back pressure) along with the owner's business strategy and financial performance goals.

For example, the purchaser's primary concern might be to maintain the rated plant output and heat rate at average annual site conditions. Alternatively, summertime performance, especially performance at periods of peak system demand, may be a particularly important consideration.

One set of performance goals covering these situations might be the following:

Maintain rated plant performance at a steam turbine back pressure of 2.5 in. HgA (8.5 kPa) at the annual average temperature.

Maintain an acceptable heat rate at average summertime conditions and a maximum output reduction from rated design output of 2%, corresponding to a back pressure of 4.0 in. HgA (13.5 kPa).

Avoid turbine alarm conditions at a back pressure of 7.0 in. HgA (23.5 kPa) at a temperature reached no more than 50 hours per year.

These conditions for a site in the arid U.S. Southwest (Figure 3-3) are shown in Table 3-1.

Table 3-1
Selected Performance Goals for an Arid Southwest Site

	Ambient Temperatures, ºF/(ºC)		
	Annual Average	**Summer Average**	**50 Hottest Hours**
$T_{ambient}$, ºF/(ºC)	65/(18)	80/(27)	110/(43)
Back Pressure, in. HgA/(kPa)	2.5/(8.5)	4.0/(13.5)	7.0/(23.5)
Condensing Temperature, ºF/(ºC)	108.7/(42.6)	120.2/(49)	146.9/(63.8)
ITD, ºF/(ºC)	43.7/(24.2)	40.2/(22.3)	36.9/(20.5)

This rough comparison of the ITDs at each condition suggests that the most demanding condition, requiring the largest ACC, occurs during the peak summer hours. Similarly, the smallest and least expensive ACC will satisfy the annual average requirement, but will result in higher back pressure and, hence, lower efficiency and lower output at the summer average and peak temperature conditions. It must be emphasized that this comparison is very approximate, but is intended to illustrate the point that knowledge of plant characteristics and performance goals over the range of expected site conditions can be used to bracket the options for specifying ACC performance.

This is a common approach to specifying the ACC design point, but to be done properly, it must include consideration of a number of factors in much more detail than the simple assumption of desired back pressure. Some of these factors include plant design characteristics, expected power price as a function of ambient conditions, severity and duration of peak load periods, the contractual status of the plant within the larger power system, and the effect of other site conditions such as prevailing wind patterns. All such factors must be considered when assessing the relative importance of seasonal performance goals in terms of ACC capability and cost. These issues are reviewed in greater detail in succeeding sections.

3.2.2 Complete Optimization

While consideration of a few chosen performance goals, as described above, is a common approach to ACC specification, it does not assure that the selected ACC is the economically optimum choice for the plant. To determine the optimum design, a more complete analysis is required to balance the initial capital costs against the continuing operating, maintenance, and penalty costs. The full set of costs to be considered includes:

ACC capital cost, including equipment and installation

Cooling system operating and maintenance costs

Operating cost, primarily in the form of ACC fan power consumption

Maintenance cost, centered on routine inspection, cleaning, motor and gearbox upkeep, etc.

Penalty costs

Heat rate penalty cost, as determined from the influence of ACC performance on plant efficiency and output reduction throughout the year

Capacity penalty cost, as determined from the limitation on plant output required to avoid last-stage turbine over pressure during the hottest hours

Each of these costs must be evaluated for a range of ACC sizes, using ACC cost and fan power correlations vs. design ITD, site temperature duration curves, and steam turbine heat rate (or output reduction) curves.

The capital cost and the future operating and penalty costs must then be expressed on a common basis. This can be done in one of two ways.

3.2.3 Capital Cost Annualization

In this method, the operating, maintenance, and penalty costs are calculated for the first year of operation using current cost information. The initial capital cost is then converted to an equivalent annual cost as a function of expected plant life, n, and an expected discount rate, i, using an amortization factor given by:

$$F = \{i \times (1 + i)^n\}/\{(1 + i)^n - 1\} \qquad \text{Equation 3-4}$$

Figure 3-6 displays this conversion.

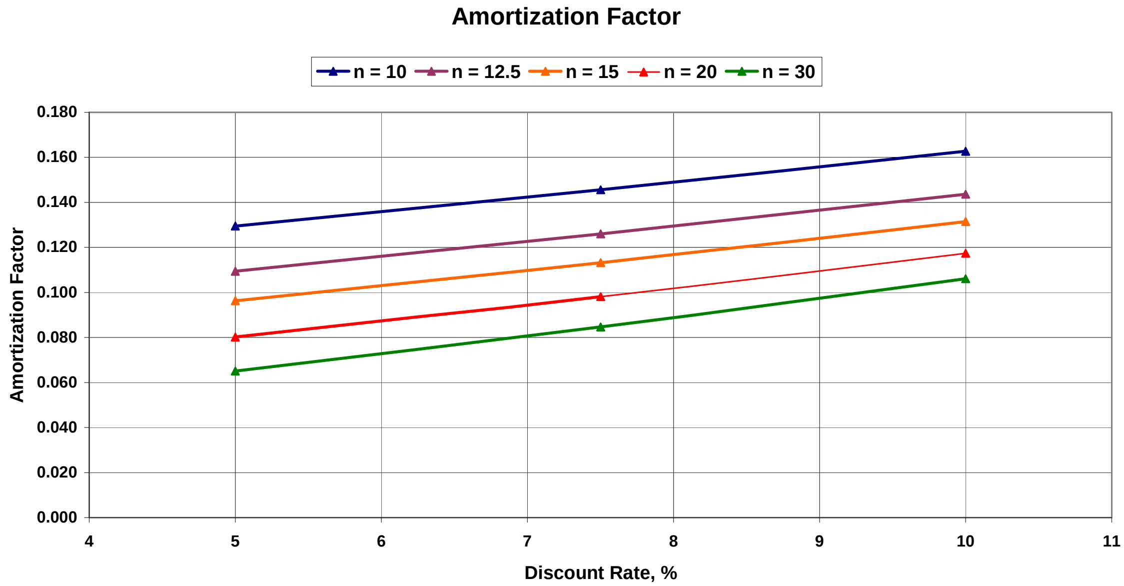

Figure 3-6
Variation in Amortization Factor

3.2.4 Lifetime Total Evaluated Cost

In this method, the annual operating, maintenance, and penalty costs are projected year-by-year into the future for the expected life of the plant. This requires assumed future costs and prices for labor, fuel, and power. Each yearly cost is then discounted to a present value using expected long-term discount rates. The yearly costs are then summed and added to the initial capital cost.

The results of the two methods vary to the extent that future increases in the prices of labor, fuel, and electricity differ from one another and cannot be properly represented by a single amortization factor, as in the “capital cost annualization” method.

Describing and providing the data, correlations, and procedures for the complete optimization methodology are beyond the scope of this document. However, a complete treatment is developed and presented in a recent EPRI report [c].

That EPRI study featured a case study for a combined-cycle plant at a hot, arid site with the ambient temperature duration curve shown in Figure 3-3. The results are shown in Figure 3-7. The minimum annualized cost is seen to occur at an ITD of approximately 45ºF. Two points are noteworthy.

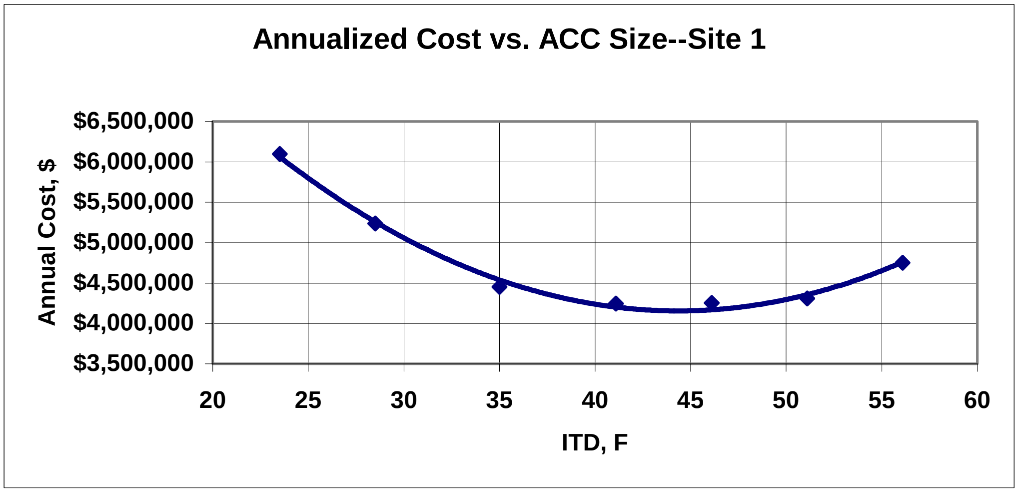

Figure 3-7
Annualized Cost vs. Air-Cooled Condenser Size

This result is reasonably consistent with the overall trend of ACC selections shown in Figure 3-5.

The result is at the high ITD (small ACC size and cost) end of the example choices shown above for the “Selected Performance Goals” approach in Table 3-1. This has no generalized significance but suggests that for the assumptions used in that particular example, the sacrifice of hot weather performance in exchange for a lower cost ACC resulted in a lower annualized cost.

It would be expected that assumptions of higher expected value of electric power (either throughout the year or especially at peak demand periods in the summer) might alter the result. Figures 3-8 and 3-9 display the results of the same example for a range of annual average (Figure 3-8) and peak load (Figure 3-9) power prices. As shown, increases in either one drive the optimum ACC size to lower ITDs and thus larger ACCs.

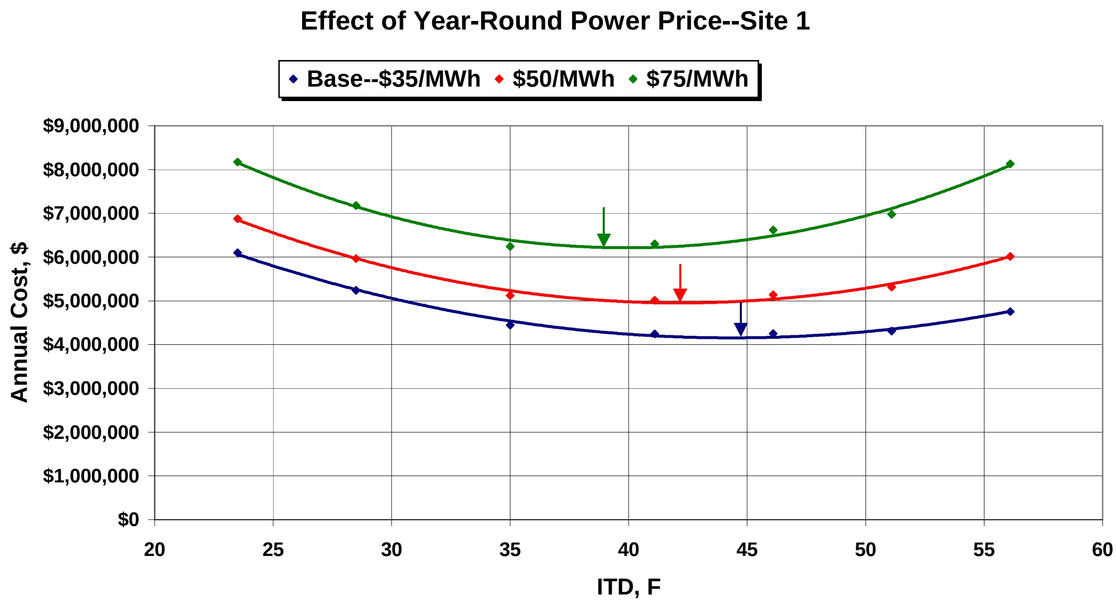

Figure 3-8
Effect of Annual Average Power Price

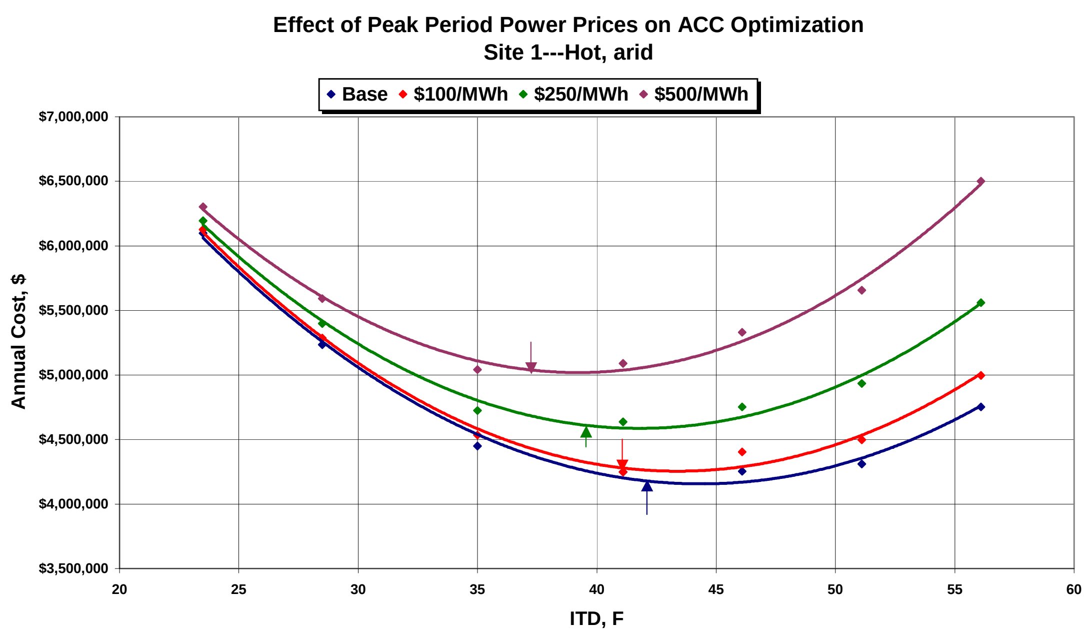

Figure 3-9
Effect of Peak Power Price

3.3 Basic Design Specifications

In addition to these basic quantities, the ACC design (and cost) will be affected by a number of plant and site characteristics, which are listed here but will be discussed in the following sections.

Site topography and features

Site elevation

Site meteorology

Annual temperature duration curves

Prevailing wind speeds and directions

Extreme conditions (hottest day, freezing conditions)

Topography and obstructions

Nearby hills, valleys, etc.

Nearby structures, coal piles, etc.

Nearby heat sources, including auxiliary coolers, plant vents, etc.

Other heat sources or interferences

Noise limitations at the ACC itself or at some specified distance, as set by neighboring communities or open space sanctuaries

Maximum height restrictions

"Footprint" constraints (length, width)

Location restrictions, particularly distance from turbine exhaust

Seismic loads, requirements, and zones, as developed on a site-specific basis, with pertinent data obtained from U.S. Geological Survey databases as part of their National Seismic Hazard Mapping Project

3.3.1 Site Elevation

The site in the earlier examples was assumed to be at sea level with a normal barometric pressure of 29.92 in. HgA. The effect of site elevation on ACC design is minor but not negligible.

An analysis of the effect of elevation on ACC size, fan power, and cost over an elevation range of sea level to 5000 ft (1500 m) indicates no change in the basic size (number of cells) or fan horsepower, but a moderate cost increase of 5–10%. This is explained in the following way.

At 5000 ft (1500 m), the ambient air density is only 83% of that at sea level. Using conventional fan laws, a 6% increase in fan speed would result in the same fan power but a reduction in the mass flow of air of about 12%. For the same heat load, the air temperature rise will therefore be 12% greater with a corresponding decrease in the log mean temperature difference (LMTD), which is the driving force for heat transfer across the finned-tube bundles. To compensate for this reduced heat transfer driving force, modest increases in the finned-tube surface area are made. Similar modifications of some structural dimensions (such as tube length to maintain comparable air-side velocities, turning losses, mixing coefficients, and other flow characteristics) would likely be required and appear reasonably consistent with a modest (~5–10%) increase in cost.

3.3.2 Plant Type

The choice of the optimum ACC is affected by the performance characteristics of the type of plant at which it is to be used. The comparison of most interest is between simple-cycle steam plants (normally coal-fired) and gas-fired combined-cycle power plants (CCPP). While similar considerations would apply to nuclear steam plants, there is little experience with the use of dry cooling at nuclear plants.

One important difference is simply size. Most CCPPs of recent design have been (nominally) 500-MW plants, with a steam turbine providing approximately one-third of the plant output at design, or about 170 MW. Coal-fired steam plants, on the other hand, range from 350–500 MW or larger and the entire output is generated by the steam turbine. Therefore, even neglecting differences in steam turbine heat rate, the heat load to be rejected through the ACC is typically two to three times greater at a steam plant than at a CCPP. While the equipment cost for an ACC is essentially linear with heat load, a significantly larger unit may have higher costs for extended steam supply ducting and a higher structure. This increase in height results from the need to elevate the fan deck more for a larger cluster of cells in order to provide free flow of air to the interior cells.

Another and more important distinction is the difference in steam turbine performance characteristics between the two plant types. Because of the higher steam turbine inlet pressure and temperature in steam plants, the turbines typically have lower heat rates, lower steam flow per unit output, and shallower output correction curves (vs. back pressure) than turbines designed for CCPPs. The comparison between the output correction curves is shown in Figure 3-10.

As a result of the lower reduction in output at elevated back pressure, the steam plant suffers less performance penalty during the hotter periods. The ACC therefore optimizes at a smaller size, higher ITD, and lower cost per unit heat load. A recent study [c] indicates that ACCs at steam plants optimize at ITDs in the low- to mid-50s (degrees F), while those at CCPPs optimize in the mid-40s (degrees F). Having said this, a number of recent ACC procurements have resulted in selections with ITDs in the 28–40 ºF range. In some merchant plant situations, the ability to competitively deliver power during summer can drive the selection to lower ITDs.

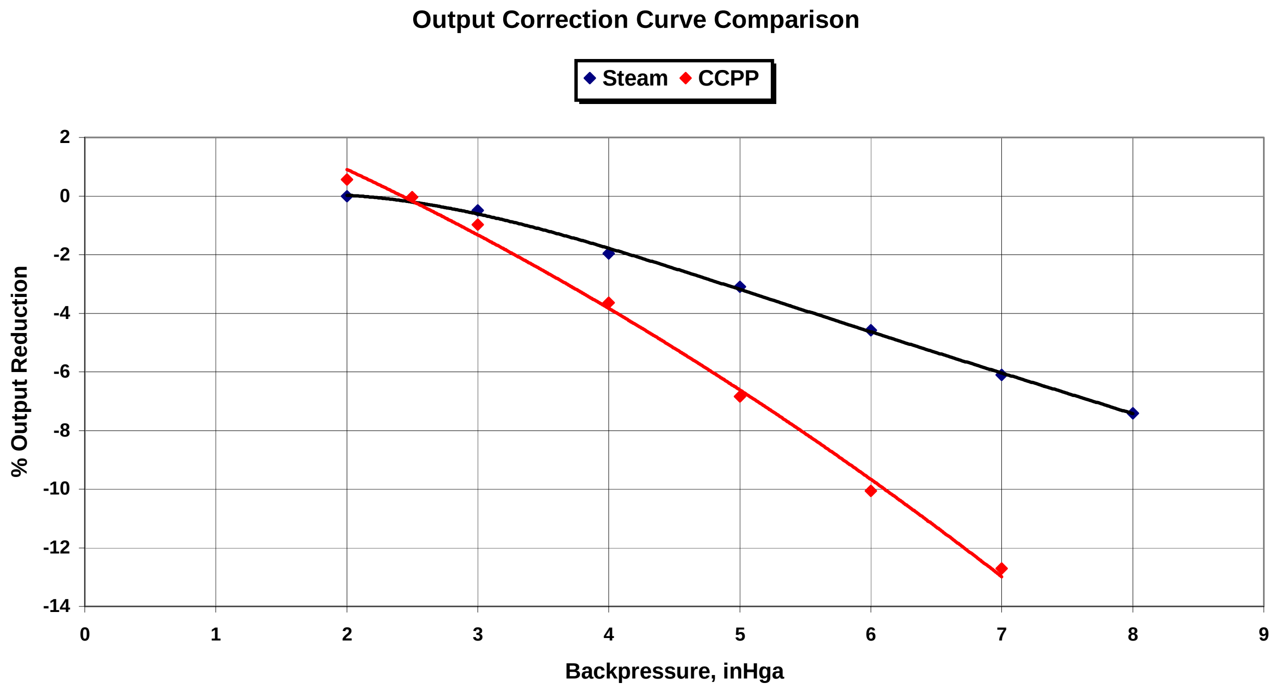

Figure 3-10
Turbine Output Corrections

3.3.3 Plant Design and Operating Strategy

Plant design and planned operating strategy also have an important influence on the choice of the optimum design point for the ACC. The crucial question is: What are the purchaser's expectations for plant performance during the hottest hours of the year? During the high-temperature periods, the steam turbine back pressure will increase, resulting in increased turbine heat rate. In this situation, the plant must accept a reduction in plant output, in which case the cost penalty is valued as lost revenue from reduced output. If the plant was designed and built to do so, it can increase its firing rate, send a higher flow rate of steam to the turbine, and hold the power output at the design level. The penalty is valued as increased fuel cost. It must be recognized in this latter case, that the ACC must be sized to handle a *higher* steam flow at the *highest* ambient temperatures, resulting in a larger, costlier turbine and ACC than would be chosen for the former (constant firing rate) assumption.

For a combined-cycle plant, the situation is more complicated. As the ambient temperature increases, not only is the steam cycle portion of the plant affected by the higher steam turbine back pressure, but also the combustion turbine portion of the plant is affected as well. Combustion turbines are essentially constant volume flow machines. With a rise in ambient temperature, the mass flow of air to the combustion turbines decreases as the inlet air density decreases. Consequently, the power output of the combustion turbines decreases as well as the hot gas flow to the heat recovery steam generator (HRSG) and the steam flow to the steam turbine. As a result, in areas where high summertime temperatures are expected, nearly all

CCPPs are equipped with both combustion turbine inlet air coolers and duct burners. The inlet air coolers, which are either chillers or evaporative coolers of the spray or matrix type, help maintain the mass flow of air to the combustion turbines closer to the design flow. Duct burners supplement the input to the HRSG as the combustion turbine exhaust gas flow decreases. In some cases, where the plant owner has specified that design plant output must be maintained throughout the year, the steam turbine may be sized to deliver nearly one-half the plant output (nominally 250 MW) during the hot periods, instead of the nominal one-third (~170 MW). Obviously, this requires an ACC sized for the higher load at the highest ambient temperature.

3.3.4 Site Meteorology

The ambient dry-bulb temperature and its variation throughout the year are the most important site characteristics influencing the optimum design point of an ACC. Figure 3-11 displays the temperature duration curves for two sites with very different meteorology. They are based on 30-year average data from a hot, arid site in the Southwest and a cold, arid site in the Northern Plains.

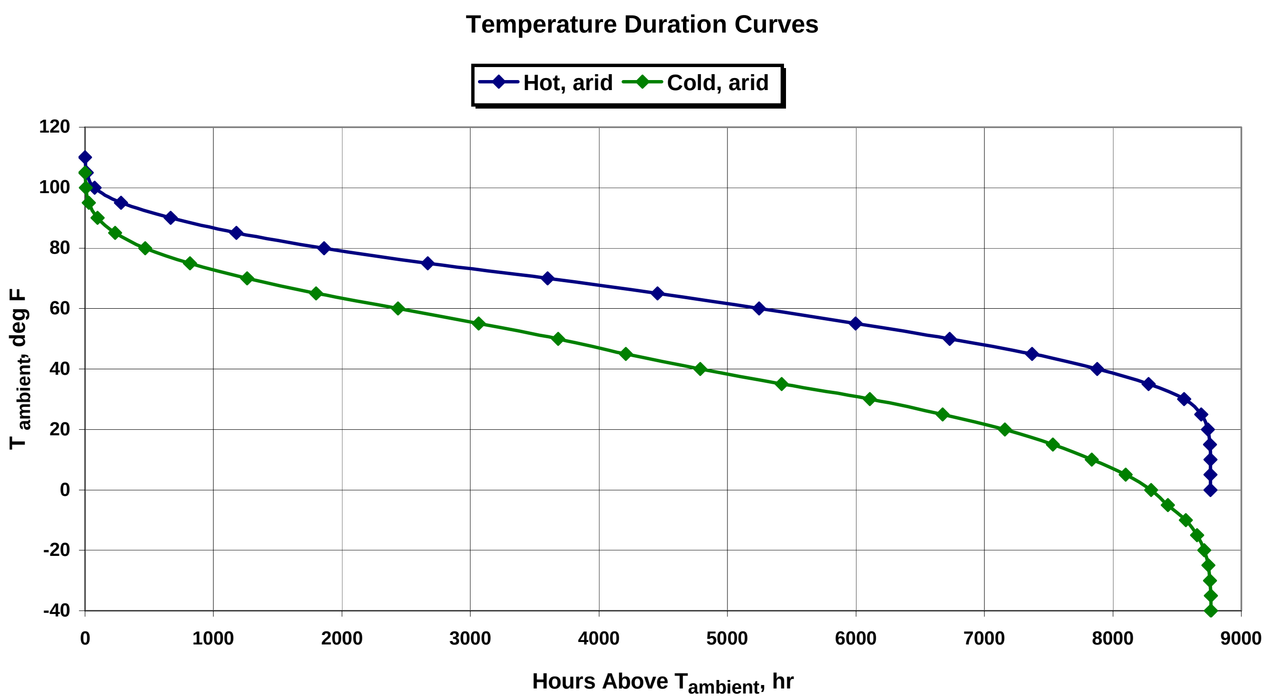

Figure 3-11
Sample Site Temperature Duration Curves

The characteristics of the temperature profiles at the two sites are given in Table 3-2.

Table 3-2
Comparative Characteristics of Site Meteorology

	Southwest Desert	Northern Plains
Ambient Temperatures, ºF (ºC)		
Annual average	65 (18)	42 (5.5)
Summer average	80 (27)	65 (18)
Extreme high (median)	105 (41)	100 (38)
Extreme low (median)	15 (-9)	-28 (-33)
Durations, hours		
Hours above 100 ºF	~ 60	~ 6
Hours below freezing	~340	~3000

Although the highest temperature reached during the year is nearly the same at both sites, the duration of sustained hot weather is much shorter at the plains site. For most of the year, the plains temperatures are from 10–30 ºF (5–15 ºC) lower. Freezing conditions are a rare occurrence in the desert, but the potential for such conditions exists for nearly one-third of the year in the Northern Plains.

As a result, the annual average heat penalty is much lower and the high ambient temperature capacity penalty is incurred for a much shorter time at the plains site compared to the desert site. These lower penalty costs drive the optimum system design to a much smaller ACC with a much lower capital cost for the plains site.

3.3.5 Economic Factors

In addition to plant and site characteristics, certain elements of economic expectations and business strategy have an important influence on the selection of the optimum design point. Noteworthy among these are

Status of generation operation (regulated vs. unregulated)

Expected duration of plant ownership

Future economic projections (fuel and electricity price, inflation, etc.)

Variability of electricity price with ambient temperature

Differing circumstances and assumptions regarding these factors can greatly affect the size of the cost penalties associated with ACC performance limitations at high temperatures. Such factors will also impact the relative importance of initial capital cost vs. future operating costs and performance penalties. These considerations are highly specific to individual companies and their economic expectations. Therefore, a detailed analysis will not be provided here. However, a broad indication of the qualitative effect of these factors on ACC optimization will be given.

3.3.6 Electricity Price vs. Ambient Temperature

In unregulated markets, the price of electricity exhibits high volatility with availability and demand and can strongly depend on location within the power grid. Table 3-3 shows behavior of wholesale power prices at a Southern California location during the months of July for the seven years of 1997 through 2003. It shows that the price during the hottest three days of the month was an average of nearly 50% above the monthly average and in 2000 was nearly double the average. For the next hottest three days, the increases were lower but still substantial.

Table 3-3
Electricity Price Variability With Ambient Temperature

Price for Palo Verde (Southern California) (for month of July)		
Year	**Hottest Three Days**	**Next Three Days**
	% above average	% above average
1997	40	25
1998	45	25
1999	61	35
2000	96	64
2001	43	34
2002	50	29
2003	16	9
Average (excluding 2000 and 2003)	**48**	**48**

At sites with long periods of very hot weather, the capacity penalty can be an important element of the annual cost. If the ACC limits output at precisely the time of year when electricity prices are at the highest levels, the potential revenue loss can be large. Therefore, an optimization computation that accounts correctly for an expected large increase in power price on hot days will drive the optimum design point toward a low ITD (large ACC).

3.3.7 Multiple Guarantee Points

As noted previously, the specification of heat load, back pressure, and ambient temperature essentially sets the ACC design. However, given the importance of wind effects on the ACC's performance, prevailing wind conditions, especially during warmer months, should also be considered. In addition to wind effects, the purchaser may have more than one set of conditions that must be met, and it may not be obvious to the non-specialist which of these conditions is the most demanding. Examples include:

A maximum allowable back pressure on the hottest expected day

A minimum allowable plant efficiency at the annual average operating conditions

A lower, but still acceptable, efficiency under average summer conditions

A required performance level for several different steam flows and at different ambient temperatures. This may be particularly important for

- Combined-cycle plants where duct burning may be employed during hot periods, which shifts a higher load onto the steam cycle
- Cogeneration plants at which the host site may be unable or unwilling to accept steam under certain conditions, imposing a larger steam flow change on the ACC

Table 3-4 lists a set of 10 operating conditions considered important to one combined-cycle, cogeneration plant equipped with an ACC. Four of these conditions were designated as "guarantee points."

Table 3-4
Multiple Operating and "Guarantee" Points

Item	Units	Case 1	Case 2	Case 3	Case 4	Case 5	Case 6	Case 7	Case 8	Case 9	Case 10
Guarantee Point		X			X				X		X
Specified Points											
Ambient Temperature	F	85	65	65	65	65	65	85	96.00	33	65
Steam Flow	lb/hr	605,000	510,000	241,300	320,000	373,000	152,700	531,500	567,000	622,000	608,000
Steam Quality		0.90	0.91	0.93	0.92	0.92	0.96	0.90	0.91	0.90	0.90
Backpressure	inHga	3.2	1.65	1	1	1.15	1	2.7	3.80	1.2	2.1
Calculated Values											
Heat Duty	Btu/hr	550,500,000	474,100,000	233,100,000	305,700,000	353,600,000	152,400,000	486,900,000	519,400,000	581,000,000	557,200,000
ITD	F	58.8	54.6	36.1	36.1	41.0	36.1	52.6	54.4	74.6	63.6

Of the several "multiple guarantee" points, it is always the case that one of the points is the most demanding or the "limiting guarantee point." If the performance at this "limiting" point is achieved, then all other points will be met or exceeded. This, therefore, is the *design* point. In the actual design process, each of the points must be carefully analyzed to determine which is the limiting case. However, a first-cut at ranking the points can be made using equation 2-2. For a given ACC, the heat load divided by the ITD is reasonably constant over a wide range of operating points. Therefore, if the quantity [Q/ITD] is calculated for each case, the highest value will be the limiting case. Table 3-5 displays the results for such a ranking for the 10 cases in Table 3-4. In this instance, the "hot day guarantee" point (Case 8) is the limiting case and therefore the design point.

Table 3-5
Ranking of Operating Points to Establish Limiting Case

Case No	Guarantee Point	Ambient Temperature	Steam Flow	Backpressure	ACC Const (Q/ITD)
		F	lb/hr	in Hga	Btu/hr-F
8	*	96	567,000	7.7	9,549,395
1	*	85	605,000	6.5	9,368,680
7		85	531,500	5.5	9,251,930
10	*	65	608,000	4.3	8,767,392
2		65	510,000	3.4	8,690,946
5		65	373,600	2.3	8,614,087
4	*	65	320,000	2.0	8,478,840
9		33	622,000	2.4	7,787,497
3		65	241,300	2.0	6,465,220
6		65	152,700	2.0	4,226,939

3.3.8 Site Wind Conditions

It is well known that the influence of wind is to reduce ACC cooling capability. This is the result of both hot air recirculation and degraded fan performance under windy conditions.

Recirculation – An operating ACC discharges a large volume of air heated typically to about 30 ºF (17 ºC) above the inlet air temperature. Under quiescent ambient conditions, the plume rises essentially vertically above the unit and does not interfere or mix with the airflow entering the unit through the open sidewall areas below the fan deck. (See Figure 2-6).

As previously explained, the A-frame cells are typically protected with a windwall, which is erected completely around the unit and usually extends from the fan deck to the top of the A-frames. While helping to reduce recirculation, this wind screen also prevents crosswinds from impinging on the outer surfaces of the cell walls, thus opposing the flow of cooling air through the bundles. However, at higher wind velocities, two additional impacts occur. First, the plume of heated air is bent over in the downwind direction, bringing it closer to the inlet areas below the fan deck. Second, the bulk of the ACC cells and their windwall act as a bluff body and produce a low-pressure wake region and associated vorticity that can draw a portion of the plume down below the top of the windwall and the fan deck. Under these circumstances, the heated air can become entrained in the inlet stream of ambient air, resulting in an ACC inlet temperature that is slightly higher than the surrounding ambient air. This phenomenon, referred to as recirculation, is also well known with wet cooling towers.

Some studies recommend the use of a "recirculation allowance" (a common practice with wet towers) in which the design *inlet* temperature is assumed to be 2–3 ºF (1–1.5 ºC) higher that the design *ambient* temperature.

Degraded Fan Performance – As noted earlier, the fans on an ACC are large, low-speed, axial-flow fans and have a static pressure rise (at design conditions with uniform, parallel axial flow at the inlet) of perhaps 0.3–0.5 in. H_2O (80–125 Pa). In the presence of a significant crosswind shearing across the fan inlet, turning losses at the inlet can be a significant factor on performance. In addition, the distortion of the velocity profile (or velocity triangle) on the leading edge of the fan blades can result in a stall condition over all or a portion of the fan. If this occurs, there can be a large and sudden reduction in the airflow to the affected cell. In contrast to recirculation, which primarily affects downwind cells, the fan performance degradation is most apparent in the upwind cells.

It is generally believed that the effect on fan performance is the more important of the two effects. However, this can vary with details of the site topography, the presence of nearby obstructions, ACC orientation relative to the prevailing winds, and any other factors influencing wind speed, direction, turbulence, and gustiness at the ACC.

From the viewpoint of the purchaser, several points are noteworthy.

The effect of wind on ACC performance can be significant. Figure 3-12 shows data at an operating coal-fired steam plant, indicating an 8–14% reduction in turbine output for wind speeds ranging from 7.5–20 mph (3.4–8.5 m/s), for a wind direction where a large building is directly upwind of the ACC.

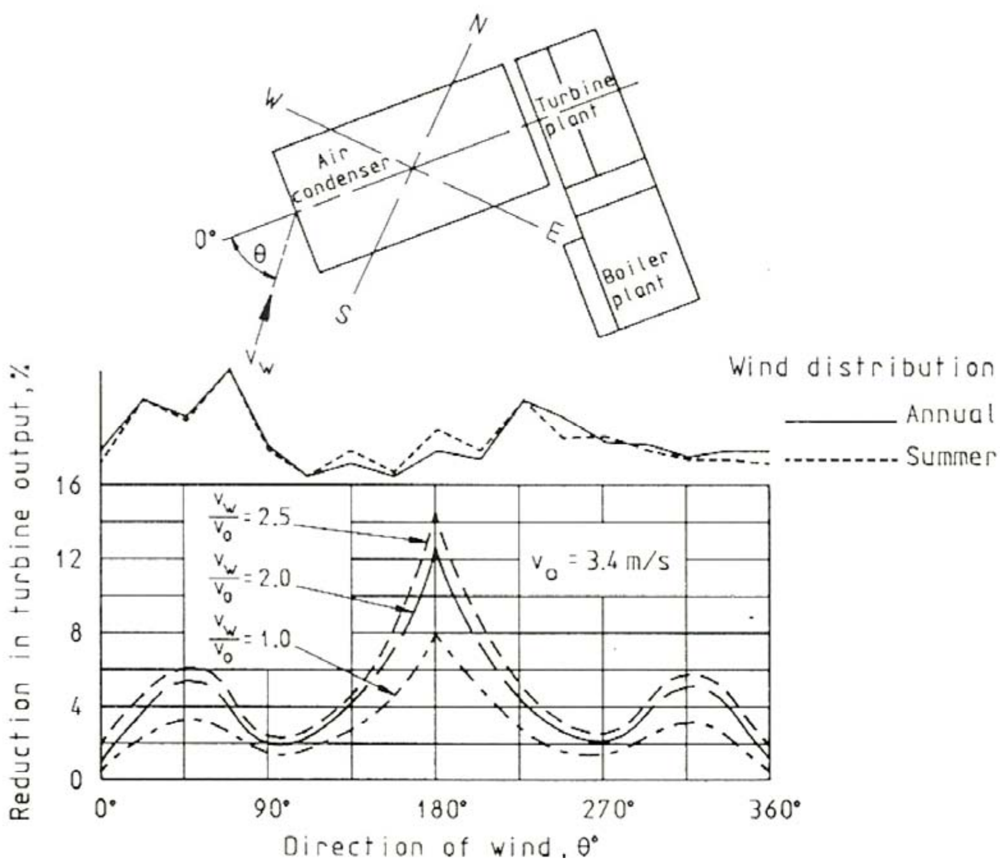

Figure 3-12
Effect of Wind on Air-Cooled Condenser Performance at Wyodak

It is very difficult to accurately predict the effect of wind on performance without detailed site- and design-specific computational fluid dynamics (CFD) or physical modeling.

If meteorological conditions at the site are such that high winds coincide with high summertime temperatures, the situation is exacerbated. During hot periods, the plant is likely to be operating at high back pressures close to the alarm/trip points, even under still air conditions. At that time, a sudden or sustained period of high wind may result in a turbine “trip” (i.e., shutdown based on exceedance of a preset pressure limit). Such high winds may require a voluntary shedding of load to avoid the plant trip condition at just the time when demand and the power prices are at their highest points of the year.

All existing test codes as well as those currently under development stipulate that ACC testing will be conducted at low wind speeds, typically less than 3–5 m/s. Therefore, a successful acceptance test does not ensure that the performance will not degrade significantly under higher wind conditions.

Designing an ACC to be immune to wind effects would be difficult. Indeed, suppliers may be reluctant to bid and guarantee such a design. Even if they were willing to do so, the cost would likely be very high. It should also be recognized that it could be difficult to conduct a rigorous acceptance test of a high-wind design since the high-wind, high-temperature conditions of interest may occur only rarely and intermittently.

If the choice is made, therefore, to forego high-wind performance guarantees in order to reduce capital costs, the supplier should be asked to provide some estimates of the expected reduction in performance with increasing wind speed, so that the purchaser understands the consequences of the decision and has the information to evaluate the tradeoffs intelligently. This reduction prediction may include both recirculation (i.e., increased inlet air temperature) as well as projections of reduced airflow and ACC performance.

There is little data available in the open literature to assist purchasers in estimating the probable effect of wind, and even if it were available, site-specific differences may dominate performance and render the data inapplicable to the purchaser’s site. The safest approach under the current state-of-the art is to request a model of on-site test results for the design as part of the bid package. From there, a more informed decision can be made relative to the resultant balance of increased capital (and operating) costs against power sales revenues and associated margins.

If, as a result of anticipated wind effects, a purchaser desires to inject additional performance margin in the specification, it may best be done in terms of artificially higher heat loads or a recirculation allowance on inlet temperature. The solicitation of “additional heat transfer surface” or “lower tube cleanliness” may result in different proposals and different actual performance levels from competing ACC suppliers.

3.3.9 Site Noise Limitations

Noise limitations can be an important consideration at some sites, with significant effect on ACC performance and cost. In all cases, Occupational Safety & Health Administration (OSHA) regulations [f] limit the noise at ground level for the ACC to about 85 adjusted decibels (dBa). In the absence of particular far-field limitations, the usual design with standard fans produces a noise level at 400 ft (140 m) from an ACC boundary of about 65 dBa.

However, some sites have special requirements. Plants located near residential neighborhoods; facilities such as schools, hospitals, and houses of worship; critical habitat areas; or national parks and other sensitive recreational areas may require significant noise reductions at specified distances.

Figure 3-13 shows a rough estimate of the effect of noise reduction on ACC capital cost [g]. Progressively greater levels of noise reduction are achieved in various ways. For modest reductions of 5 dB or less, fans with more blades can be run at lower speeds requiring about the same power. Reductions of up to about 10 dB can be achieved with reduced airflow, compensated for by increased heat transfer surface, but not requiring any additional cells. For reductions above 10 dB, a combination of more surface per cell, additional cells, and low noise fans will be required. When the limit of what can be achieved with fan selection and ACC design modification is reached, somewhere in the range of 15–17 dBa, external noise barriers will be required.

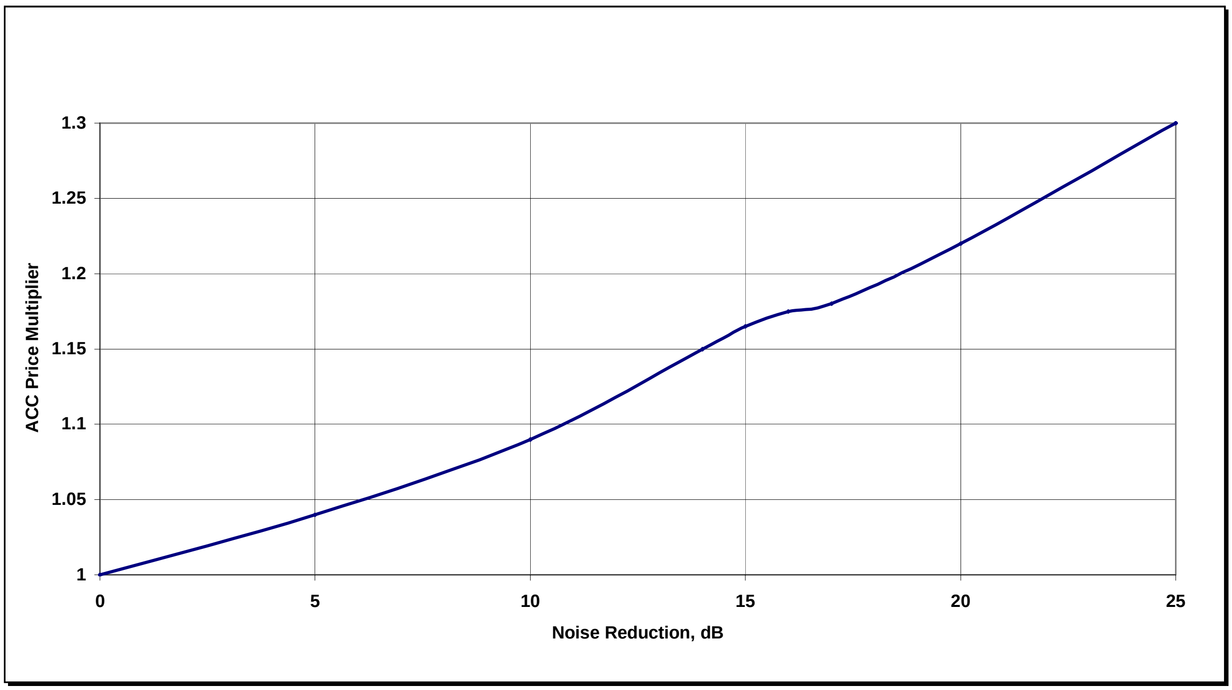

Figure 3-13
Air-Cooled Condenser Cost Multiplier vs. Noise Reduction

Low noise fans come in several categories, characterized in some descriptions as “standard,” “low noise,” “very low noise,” and “super low noise.” Tables 3-6 and 3-7 show the comparative performance and cost information for these four categories.

The correspondence between an approximate 3.5-fold increase in fan cost for a 16-dB noise reduction and an ACC cost multiplier of ~x 1.17 for a similar noise reduction implies a fan cost of about 7% of the ACC cost for standard fans, which appears reasonable.

Table 3-6
Low Noise Fan Performance [g]

ACC Fans---34 ft. diameter 1,468,200 ACFM; 0.375 " WG				
	Standard	**Low Noise**	**Very Low Noise**	**Super Low Noise**
Blades	4	6	6	6
RPM	116.9	83.9	65.4	65.4
Tip Speed	12,481	8,957	6,988	6,988
HP	137.4	139.9	158.2	161.4
Sound Power	104.8	99.1	92.6	89.9
Sound Pressure (@ 400 ft)	56.2	48.6	42.8	40.1

Table 3-7
Low Noise Fan Cost Comparisons [g]

ACC Fans---Relative Cost Factors (in %)				
	Standard	**Low Noise**	**Very Low Noise**	**Super Low Noise**
Blades	5	5	5	5
Weight	100	115	145	325
Fan Costs	100	140	225	450
Gearbox Costs	100	100	125	125
Transport	100	100	125	300

3.3.10 Use of Limited Water Supply

The preceding discussions make clear that "hot day" performance of ACCs and the associated heat rate and capacity penalties have a significant influence on the optimum design point. High penalty costs drive the optimization toward larger and more costly ACCs.

An alternative approach to the design of large ACCs can be considered if a limited amount of water is available at the site for use as supplementary wet cooling during those limited periods of hot weather. Two systems are commonly considered.

3.3.11 Hybrid Systems

Hybrid systems, in this context, refer to those with a conventional, shell-and-tube surface condenser and a wet cooling tower installed in parallel with an ACC. The system is shown schematically in Figure 3-14.

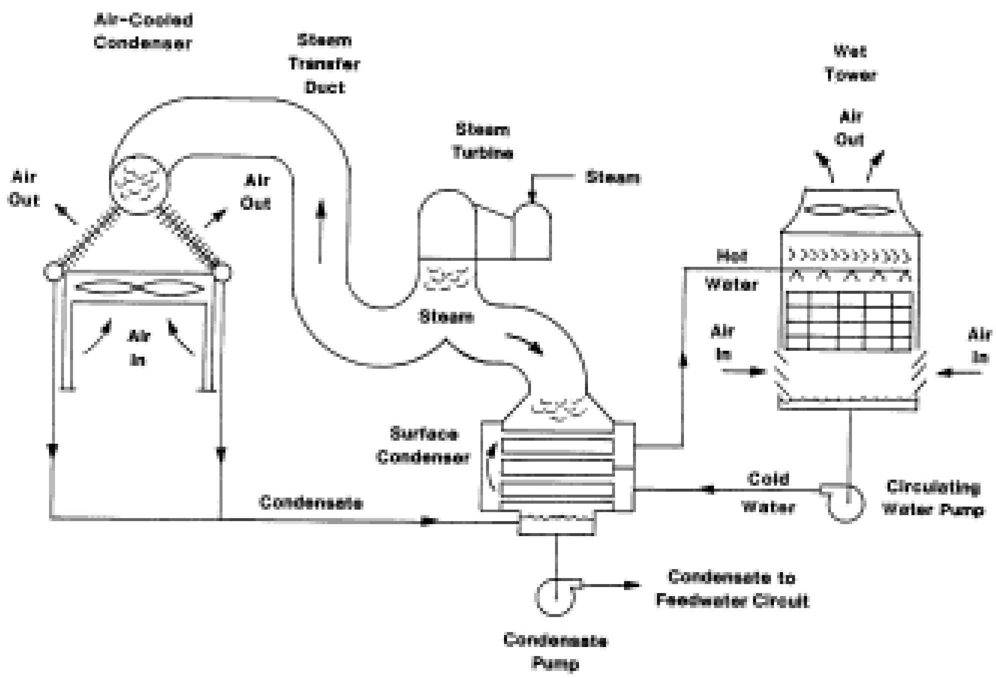

Figure 3-14
Hybrid (Dry/Wet) Cooling System

During peak load hot periods, cooling water from the wet tower is circulated through the surface condenser, which then draws steam away from the ACC. The system is self-balancing. The steam flow will divide to establish an operating point in which the condensing pressures in the ACC and surface condenser are the same. The heat load on the ACC is thus reduced and the turbine back pressure is lower than it would have been for an ACC operating alone.

The system permits the use of a smaller and therefore less expensive ACC than would have been required in an all-dry design. On the other hand, the system incurs the costs of a small wet-cooling system, which, though small in size compared to what would be required for an all-wet system, requires the full complement of equipment. This includes the shell-and-tube condenser, the cooling tower, circulating water pumps and piping, intake and discharge lines, and structures and associated water treatment capability.

In-depth discussion and analysis of the tradeoffs are beyond the scope of this document. A more detailed discussion is available in a recent EPRI report [c]. General guidelines have been presented [d] that suggest for annual water availability, ranging from 15–85% of the water required for all-wet cooling, the capital cost of the hybrid system is less than that for an optimized all-dry ACC system.

3.3.12 Spray Enhancement

Another approach, called spray enhancement, is shown schematically in Figure 3-15. It involves the spraying of water into the inlet air stream of an ACC. The water evaporates and cools the air before it enters the finned-tube heat exchanger bundles. ACC performance is improved to a level consistent with an artificially lowered ambient temperature. Cooling effects of 5–10 ºF (2.5–5 ºC) are readily achieved, which is normally more than sufficient to avoid the need to reduce load.

This approach has been tested at several installations on an ad hoc basis, where sprays were retrofitted to an existing ACC that had lower-than-desired performance on hot days. The performance enhancement is achieved with a relatively low-cost retrofit, which has resulted in a satisfactory rating. The approach, if not carefully designed and operated, runs some risk of scaling or corrosion damage to the finned-tube bundles from unevaporated spray droplets impacting the surfaces. Current research and development work will provide well-documented cost, performance, and operating procedure information for spray enhancement, which will assist in formulating reliable design guidelines.

To date, no new ACC designs have incorporated spray enhancement into the original design. Some European installations have included deluge cooling in the Heller systems, which use indirect dry cooling, typically with natural draft cooling towers. Detailed discussion of spray enhancement or deluge cooling options is beyond the scope of this report. Recent information on spray enhancement and the deluge system was reported at the CEC/EPRI Advanced Cooling Strategies/Technologies Conference, June 1-2, 2005, and is available from EPRI.

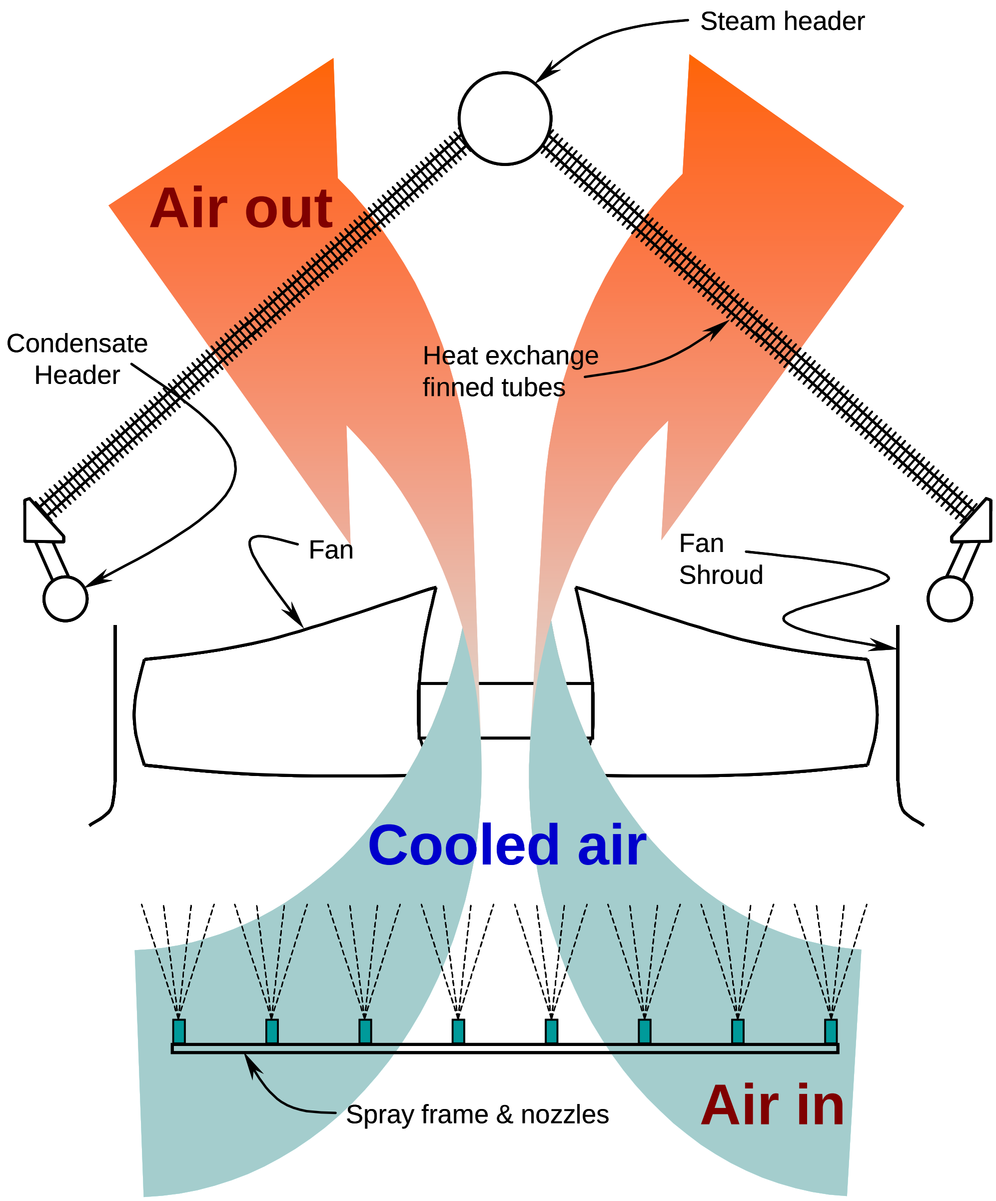

Figure 3-15
Schematic of Spray Enhancement Arrangement

3.4 References

a) Balogh and Z. Takacs. *Developing Indirect Dry Cooling Systems for Modern Power Plants*. 98.

b) J.M. Burns and W.C. Micheletti. *Comparison of Wet and Dry Cooling Systems for Combined Cycle Power Plants*. Burns Engineering Services, Inc. and Wayne C. Micheletti, Inc.: November 4, 2000. Final Report (Version 2.1).

c) *Comparison of Alternate Cooling Technologies for U.S. Power Plants*: *Economic, Environmental, and Other Tradeoffs*. Electric Power Research Institute, Palo Alto, CA: August 2004. Final Report 1005358.

d) J.S. Maulbetsch, M. DiFilippo, K.D. Zammit, M. Layton, and J. O'Hagan. "Spray Cooling – An Approach to Performance Enhancement of Air-Cooled Condensers," presented at EPRI's *Cooling Tower Technology Conference*. Electric Power Research Institute, Palo Alto, CA: August 2003. Proceedings Report 1004119.

e) D.G. Kröger. *Air-Cooled Heat Exchangers and Cooling Towers*. Thermal Flow Performance Evaluation and Design, Department of Mechanical Engineering, University of Stellenbosch, Private Bag X1, Matieland, 7602, South Africa, 1998.

f) OSHA. *Occupational Noise Exposure*/1910.95.

g) D. Sanderlin and F. Ortega. *Dry Cooling History in North American Power Plants. Dry Cooling for Power Plants – Is This the Future?* Air and Waste Management Association (AWMA): 2002.

4
AIR-COOLED CONDENSER COMPONENT SPECIFICATION

4.1 General Requirements

The ACC and associated equipment shall be designed to meet the performance requirements developed as a result of the process outlined in Chapter 3. The fabrication and testing of equipment will be in accordance with recognized codes and standards. A performance testing procedure was developed and is included in Chapter 6 of this specification. A design life of 25 years or more is likely achievable under typical operating conditions, assuming that the ACC system is not operated in a corrosive environment or one where extremes in temperatures and operating conditions preclude such expectations.

4.2 Specific Components

4.2.1 Finned-Tube Bundle and System

In general, finned-tube bundle systems, shown in Figure 4-1, consist of the heat exchanger finned tubes and associated headers. The finned-tube systems may consist of multiple rows of tubes but are more commonly single-row tube systems characteristic of today's low-capital-cost scope of supply.

The finned tubes are normally arranged in a sloping A-frame type installation, from the main steam header to the condensate collection system. Connections are of welded type. The total finned-tube system needs to accommodate expansion and contraction with changes in thermal loads.

Materials of construction may be galvanized carbon steel with a minimum thickness of 0.06 in. (1.5 mm) prior to galvanizing. Aluminum-clad tubes are also acceptable with aluminum finned heat exchange surfaces brazed to the tubes.

As to strength and arrangement of fins and connections, the fins shall be capable of withstanding spray cleaning pressures of up to 1200 psi at a distance of one foot or greater. To facilitate cleaning, the number of fins should not exceed more than 10–15 fins per inch. This spacing limitation may vary as a function of the nature of the ambient and airborne contaminants that might be entrained with the inlet air.

Figure 4-1
Example of Steam Ducting and Finned-Tube Bundles

4.2.2 Structure

Structural steel and associated steel work shall be in accordance with the latest edition of the American Institute of Steel Construction Inc. (AISC) *ASD Manual of Steel Construction* [a]. Associated welding shall conform to the American Welding Society *Standard Code for Welding in Building Construction* [b]. All connections for the structural steel members shall be designed and detailed for bearing-type connections. If the ACC is expected to operate in a snow and ice environment, it shall be designed for local snow and ice loadings absent any type of heat tracing. The structure shall be designed for the required seismic rating of the site.

4.2.3 Steam Ducting

The steam ducting begins at the turbine exhaust flange and leads to the ACC. The ducting typically includes expansion joints, rupture discs, inspection ports, structural steel, necessary vent and drain connections, etc. Blanking plates for each ACC duct permit leak testing of the entire ACC. Figure 4-2 below depicts steam ducting on a combined-cycle plant in Nevada.

Figure 4-2
Example Steam Ducting and “Pant Leg” Distribution

The steam ducting also includes a sufficiently sized tank for condensate collection and, depending upon site conditions, may require a heating/freeze protection system. An example of this is provided in Figure 4-3, again, from a combined-cycle plant in the southwestern United States.

Figure 4-3
Example Condensate Tank, Piping, and Access

4.2.4 Air Removal System

Air removal systems are normally of the steam-jet air ejector type. Those having surface-type condensers are typically designed in accordance with the Heat Exchange Institute's *Standards for Steam Jet Vacuum Systems* [c].

The air removal system shall be designed for accommodating both noncondensable and air in-leakage that might occur in the operation of the system over a range of heat loads. Further, it shall be equipped with isolation valves for the steam inlet, condensable inlet, and discharge of the ejectors. Two 100% air removal systems are typically required.

The air removal systems shall be configured and supplied as stand-alone units, including all piping, drains, pressure relief valves, gauges, etc. Common accessories required include:

Volumetric flow and temperature indicators

Steam strainers

Bypass line with valves affording determination of flow rates

4.2.5 Mechanical Equipment

4.2.5.1 Fans

This equipment includes the complete fan assemblies, comprised of blades, hub, and seal disks to provide optimum efficiencies over the expected range of ambient temperatures and fan speed modulation. Fan drives may be variable speed or multi-speed, including two-speeds forward. The supplier shall verify that the fans and the ACC will perform in minimum and maximum density inlet air environments, including those associated with warm weather conditions, plus a recirculation allowance. Fan blades shall be fiberglass-reinforced epoxy and fan hubs shall be galvanized. (Smaller fan diameters may warrant consideration of aluminum or other similar materials of construction). Fan blade and hub assemblies shall be designed to facilitate adjustment of blade pitch following installation and operation in the ACC.

Fan blades, complete with hubs, shall be assembled and statically balanced before shipment. Obviously, a record system of fan blade and hub assemblies must be maintained and communicated with the shipment of separate fan blades and hubs. Replacement fan blades shall be manufactured in such a fashion as to be interchangeable, without adverse impacts to static balancing.

Fan blade systems and operations will be designed so that there are no natural frequencies set up between the intended operations of the fans and the ACC structure itself. Fan systems shall further be designed so as not to exceed 2 mils (50 microns) maximum vibration amplitude under any operating condition for the bearings and bearing pedestals. Fans shall be capable of operating at 110% of their design operating speeds.

Fan systems shall be equipped with inlet bell rings to improve the entering airflow characteristics upstream of the fan. The inlet fan rings shall be fabricated from fiberglass or polypropylene. The inlets to each fan shall be protected with a screen capable of preventing objects such as local birds, entrained paper, plastic bags, and the like from being carried into the rotating fan assemblies. The fan system installation, including inlet bell, shall result in fan tip clearances not to exceed the performance and installation guidelines set forth by the fan manufacturer.

4.2.5.2 Gearboxes

Gearboxes can be of the hypoid, helical, or spiral bevel type designed for continuous service. They shall be designed to meet the Cooling Technology Institute's Standard 111, *Gear Speed Reducers* [d], and to operate for 100,000 hours before major repair or replacement. They shall also be designed to operate in both directions and accept design thrust values. An American Gear Manufacturer's Association rating of 2.0 shall apply to the design and operating criteria.

Gearboxes shall be equipped with a means of simple access for filling and servicing of lubricating oil. They shall also be equipped with a vent line for ease of filling. In addition, each gearbox shall have a magnetic drain plug system for pick up and retention of metal particles in the gearbox oil. Gearbox internal lubrication will be forced type.

The gearbox shaft shall be equipped with a flexible coupling that will accommodate typical angular misalignments in the gearbox /fan shaft drive shaft system. The “flex-coupling” will also be designed to fail under manufacturer-recommended thrust levels.

Figure 4-4 below, shows a motor and offset gearbox on an ACC in the southwestern United States.

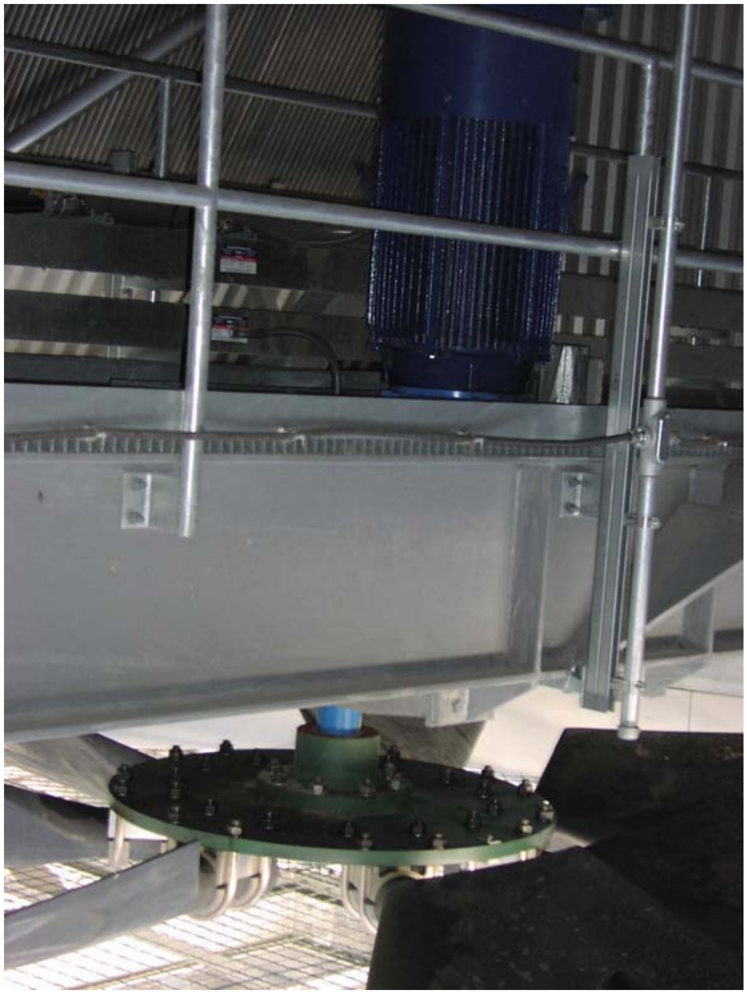

Figure 4-4
Example Motor, Gear, and Fan Hub Assembly

4.2.6 Access

General – Access refers to all stairways, platforms, ladders, manways, etc. to safely access, inspect, maintain, and operate the ACC.

Recommended Access Type and Locations:

- Single stairway on one end of the ACC
- Two caged ladders on opposing end of the ACC from stairway access
- Grating platforms connecting the ACC cells and accessing the mechanical equipment
- Walkway around the ACC perimeter at the tube-bundle condensate collection level
- Access to steam duct rupture disc and any valving that may require manual operation
- Access to instrumentation and sensor locations, including all permanent and temporary test wells or ports
- Hinged doorways with automatic closure and full seals for access to each end of a street and between each cell within a street
- Optional rolling staircase for access to upper surfaces of tube bundles – may also include cleaning spray nozzles

Galvanizing – All access platforms, gratings, stairways, ladders, etc. shall be hot-dip galvanized steel materials.

Dimensioning – All walkways and stairways shall be a minimum of 3.3 ft (1 m) wide and shall have no obstructions. Clearance above walkways shall be a minimum of 7 ft (2.2 m). All catwalks shall be a minimum of 1.6 ft (0.5 m) wide.

Appurtenances – All stairways, catwalks, and platforms will be equipped with kickplates and handrails as well as intermediate piping and baseplates.

Figure 4-5 below shows a walkway and lighting on a typical ACC.

Figure 4-5
External Walkway and Caged Ladder on a Typical Air-Cooled Condenser

4.2.7 Hoists, Davits, Monorails

General – All systems required to remove, convey across the ACC, lower to grade, and replace/maintain mechanical equipment fall into the general category of hoists, davits, and monorails. Such equipment will be permanently mounted, with removable panels at intermediate and end walls to convey equipment through and out of the ACC.

Monorail Beams – This includes a full run of monorail along the length of any entire "street," i.e., along each condensing run. It shall be designed to suspend and convey the weight of a fan blade bundle, fan hub, fan motor, or gearbox. The beams will have sufficient overhang on one end of the ACC to raise or lower removed or replacement assemblies to and from grade to the fan elevation level.

Monorail Trolley and Hoist – The monorail trolley and hoist, as shown in Figure 4-6, include a movable roller-type assembly and electric hoist with movable operating panel and cable to run the length of a condensate run or street.

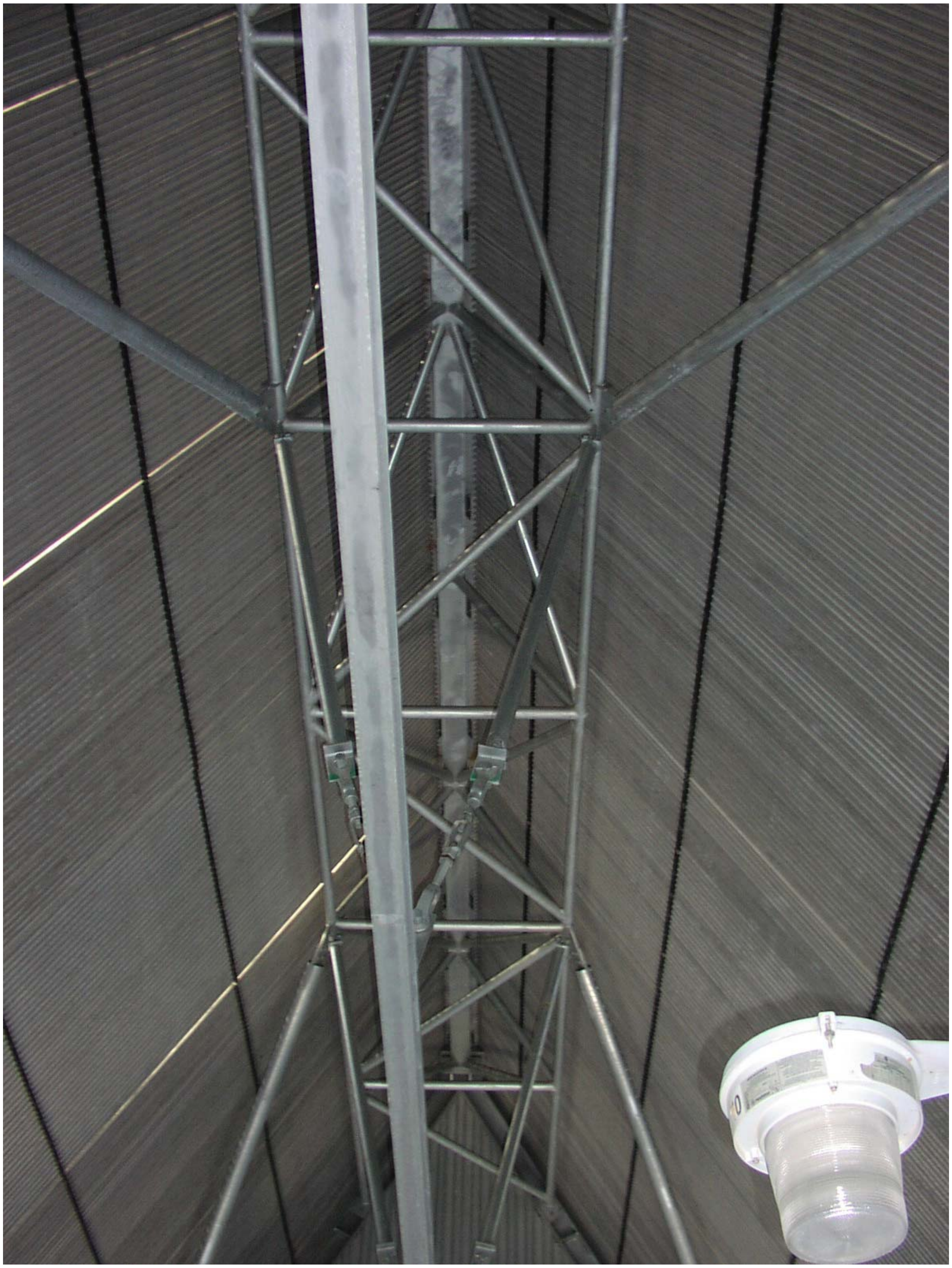

Figure 4-6
Example Monorail System and Truss Work for Conveying Motor and Gear Assemblies for Service and Repair

4.2.8 Abatement Systems

Abatement systems may include noise abatement devices, wind screens, etc. These systems tend to be site-specific, especially those having to do with wind abatement. Some general guidelines for abatement systems are provided below.

Noise Abatement – Sound-pressure levels can be reduced via sound walls and similar external sound absorption systems; however, these are not typically required for ACCs and would obviously result in additional capital cost. The primary noise abatement vehicle is the fan and fan design. The degree to which either noise absorption devices or low noise fans are employed depends upon the sound-pressure levels required by the site. Most plant sites in the United States are sufficiently distant from residential or population areas to preclude the use of low noise fans or noise abatement devices. The Crockett Cogeneration Plant in Crockett, California, uses low noise, low velocity fan designs.

Wind Screens – A variety of wind screen designs have been employed at sites in the United States. In some cases, the screens serve multiple purposes, namely, reduction of entrained debris and reduction of wind effects. In cases where debris entrainment is an issue, the screens may be deployed in the area upwind of the ACC. An example of this type of situation is shown in Figure 4-7.

Figure 4-7
Upwind Screen for Reduction of Wind-Entrained Debris and Wind Effects (Chinese Camp Cogen)

When winds alone are of concern, wind screens may be deployed under the ACC in a variety of arrangements. As these are considered experimental at this time, and are under consideration by research groups such as EPRI and the CEC [e], no additional guidance is provided in this document.

4.2.9 Instrumentation and Controls

General – Instrumentation and controls refer to all temperature, pressure, flow, level sensors, and devices to properly monitor and control the ACC. It also includes thermowells, pressure ports, etc. in strategic testing and monitoring locations. A lightning protection system may be included, depending on the apportionment of work between the electrical and instrumentation contractor and the ACC supplier.

Optimization of Operation – Instrumentation and controls may be provided to modulate operation of the ACC fans in order to optimize operation of the turbine generator, minimize subcooling, and eliminate risks of the system freezing during low loads and low ambient temperatures.

Specific Control and Monitoring Features (and their locations):

- Fan vibration, alarm, and cutout switches – remote
- Gearbox lube oil pressure and level – local
- Steam duct temperature sensors – remote
- Steam duct pressure sensors – remote. Tubing runs shall be sloped downward to prevent pressure head from the condensate. Pressure sensors shall be designed to prevent impacts of velocity pressure.
- Condensate return temperature sensors – remote
- Condensate level sensors – remote
- Vacuum pump skid hogging flow rate
- Wind speed and direction sensor – remote. Location of such sensors should be done with objectives clearly in mind. For instance, if wind speed and direction information is for monitoring and research only, multiple locations, including at the fan level and upwind of the ACC, should be considered. If these data are for performance testing, placement guidance is provided in Chapter 6.

All sensor systems shall be designed and installed such that they can be safely removed while the ACC is in operation, without impacting ACC or plant operation.

Additional guidance on potential instrumentation and sensor locations is provided below, based on input to the American Society of Mechanical Engineers Performance Test Code 30 [f] Committee (ASME PTC 30 Committee) in progress at the time of this writing.

Turbine Exhaust Pressure – At least four pressure taps with basket tips, symmetrically disposed, in the steam duct near the connection to the turbine exhaust flange

Turbine Exhaust Temperature – A minimum of one thermowell, in the steam duct, near the connection to the turbine exhaust flange

Local Wind Speed and Direction – Anemometer and wind vane at least 3 m above the wind walls, on the corner of the ACC facing the prevailing wind

Condensate Flow Rate – The flow element should be installed at a point at least 10 diameters of straight pipe in the condensate line downstream of the condensate pump, with a removable flanged spool that can be used to install an in-line flow transducer

Condensate Temperature – At least two thermowells in the condensate tank

Isolation – Isolation valves on all drain inlets to the condensate tank

Fan Power Measurements – Accessible wattmeter taps where the fan power cables exit the motor control center (MCC) or variable frequency drive (VFD) cabinets

4.2.10 Spare Parts

Spare parts include all backup parts and systems required to reliably maintain the ACC and its control systems. This also refers to startup and maintenance lubricants for the gearboxes. Recommended spare parts include:

One spare fan and hub assembly

One spare gearbox

One each pressure and temperature transducer used for routine monitoring and control

4.2.11 Lightning Protection System

This system includes lighting rods, conductive cable, and appropriate grounding. It may best be provided by the electrical or grounding contractor for the plant in total, including integration into the plan for other high-elevation structures such as combustion stacks.

4.2.12 Cleaning System

The cleaning system includes a vendor-recommended cleaning system designed for efficient medium-pressure cleaning of finned tubes. Figures 4-8 and 4-9 below show a popular “library ladder” type cleaning system, which can be used to periodically clean the finned tubes with a high-pressure (i.e., 500–1500 psig) supply system. The movable ladder affords coverage of the total exit plane of the finned-tube bundles. It is important to note that flushing of the tubes is recommended countercurrent from the direction of airflow in order to achieve maximum cleaning of collected contaminants.

Figure 4-8
Movable Cleaning Ladder System Above Tube Bundles

Figure 4-9
Close Up of Spray Cleaner Valving on Movable Cleaning System

4.2.13 Factory Testing

Factory testing includes all shop and key subcontractor tests required to demonstrate that in-factory functionality and quality control requirements are met.

4.2.14 Shipping

Shipping refers to all packing, protection, loading, and transportation required to properly transport ACC components from the point(s) of manufacture to the job site or nearby staging area.

4.2.15 Additional Options

There are options that are available, and should be considered. One example is a system for continuous performance monitoring. GEA offers a system that can track ACC performance at their headquarters and alert the plant if something is wrong (heavy fouling, excessive air in-leakage, etc.).

Automatic fan controls are another important consideration. The few units that don’t have such controls rely on the operators to pay attention and know when to put fans on full or half speed or off. Variable speed fans are also getting more attention and some installations considering retrofitting them.

4.3 References

a) American Institute of Steel Construction Inc. *ASD Manual of Steel Construction.* 9th Edition, Volumes 1-2. 170-M021-00. Chicago, Illinois: AISC Inc., 2003.

b) American Welding Society. *Standard Code for Welding in Building Construction*. AWS D 1.1/D 1M. Miami, Florida: American Welding Society, 2004.

c) Heat Exchange Institute. *Standards for Steam Jet Vacuum Systems*. 5th Edition. Chicago, Illinois: Heat Exchange Institute, 2000.

d) Cooling Technology Institute. *Gear Speed Reducers – STD-111*. Houston, Texas: CTI, Revised May 1998.

e) J.S. Maulbetsch. Panel Discussion and Presentation at the CEC/EPRI Advanced Cooling Strategies/Technologies Conference, Sacramento, California, June 1-2, 2005.

f) American Society of Mechanical Engineers. *PTC 30 – 1991 Air Cooled Heat Exchangers.* New York, New York: ASME, 1991.

5
AIR-COOLED CONDENSER BID EVALUATION

General verification of ACC performance can be conducted by solicitation and evaluation of some of the following information.

5.1 General Requirements Overview

5.1.1 Initial Temperature Difference (ITD)

The ITD will typically be in the range of 25–60ºF (14–33.3ºC). Note that ITDs approaching the low end of this range will result in equipment sizing that may not be economical for a specific plant, notwithstanding the obvious benefits to turbine efficiency. On the other hand, high ITDs, especially in the event of wind-induced performance deficiencies, may well result in derating of the power generation unit or a steam turbine trip.

5.1.2 Steam Quality

Steam quality is the weight fraction of steam or percentage of steam at the turbine exhaust. It is typical to have some moisture in the exhaust steam. Usual values of steam quality are 90–95%, but may be lower depending on operating conditions of the system. If steam quality were to exceed 100%, it would suggest superheated steam still exists at the turbine exhaust.

5.1.3 Steam Turbine Exhaust Pressure

Steam turbine exhaust pressure, commonly referred to as “back pressure,” will typically be in the range of 2.5–7.5 in. HgA. Pressures above this level will typically exceed steam turbine manufacturers’ warranties. Accordingly, this high level may be set as a “trip point” (i.e., automatic shutdown) for the unit.

5.1.4 Verification of Supplier Performance Requirements for the Air-Cooled Condenser

This section focuses on the single row condenser (SRC) design as it is the most widely offered in response to current ACC bid solicitations.

Number of Cells – The number of cells (also referred to as modules) is clearly an important part of the supplier data. Obviously, the number of cells dictates the amount of mechanical equipment (i.e., fans, motors, gearboxes) required by the ACC. Furthermore, many current large-scale SRC designs use components whose dimensions are optimized for shipping and erection. For example, an ACC cell may use 33-ft (~10-m) diameter fans, individual tube bundle sections of approximately 36 ft (~11 m) in length and 8 ft (~2.5 m) in width, and 5 bundles per cell per side, for a plan area of 36 ft by 40 ft (~11 m by 12 m) per cell. As a result, the total number of cells often dictates a number of ACC features, including the mechanical equipment as well as the total amount of heat transfer surface.

The total number of cells or modules is the sum of the primary and secondary modules. The primary modules are responsible for the majority of heat transfer and condensing processes, while the secondary modules are responsible for residual heat transfer and noncondensables collection and evacuation.

Number of Primary Modules – The number of primary modules is typically about 80% of the total number of modules.

Length of Primary Modules – The length of the primary modules is typically on the order of 33–40 ft (10–13 m) for an SRC type system.

Number of Secondary Modules – The number of secondary modules is typically about 20% of the total number of modules and there is typically one module per row (or street).

Length of the Secondary Modules – These modules are typically shorter than the primaries by about 3–5 ft (~1–1.5 m).

Primary Module Dimensions – (Width) – Obviously, the width of the primary modules must be greater than the fan diameter and typically run on the order of 15–25% larger than the fan diameter.

Fan Characteristics – Fan diameters for ACCs used in most recent power plant applications are typically 30–37 ft (10–12 m). The number of blades per fan will minimally be 5 but may be as many as 8–10, depending on the fan supplier and the performance requirements.

Motor Characteristics – Fan motor power must be equal to that required by the fan shaft power divided by the motor and gearbox efficiencies. It is typically the case that a margin of 5–10% is provided, in addition to service factor margins.

5.1.5 Additional Vendor-Supplied Data

Beyond the guidance provided in Chapter 3, a bid specification may solicit the following information:

Overall heat transfer coefficient, **U**, (based on air-side surface area)

Total air-side surface area, **A**

Total mass flow rate of air at each design condition, $\mathbf{m`}_{\mathbf{air}}$

Fan static pressure **(p_{static})** or total system pressure drop

Log Mean Temperature Difference (LMTD)

Steam duct pressure drop

Heat exchanger bundle pressure drop (steam side)

5.1.6 Important Items for Verification

Thermal Duty – It is important to verify that the thermal duty solicited (i.e., the amount of heat to be rejected) is matched or exceeded by the supplier's offering.

$$Q_{required\ -} = m\grave{}_{steam} \text{ x } (h_{\ steam,\ (turbine\ exhaust)} - h_{\ (condensate)})$$

$$Q_{\ rejected\ =} U \text{ x } A \text{ x } LMTD$$

Heat Transfer Area – This is calculated knowing the total heat transfer area of ACC tubes. For an SRC, the ratio of the air-side surface area and the total "face" area is approximately 120.

Outlet Air Temperature – The outlet air temperature is obviously less than the steam temperature and can be calculated from the following equation:

$$Q_{required} = m\grave{} \text{ x } C_{p\ air} \text{ x } (T_{air,\ out} - T_{air,\ in})$$

Face Velocity of the Air – The face velocity of the air, while not typically provided by the supplier, can be calculated from the mass of airflow rate, the air density, and the total face area of the ACC. Typical values will run from about 3 ft/sec (~1 m/s) to as much as 8–10 ft/sec (~3 m/s), with the average being about midway between those limits. Engineers who have performed velocity measurements at the ACC exit plane know that, while the average velocity may be in those limits, variations of a factor of five can occur at the outlet.

Fan Static Pressure – Fan static pressures will vary depending on whether the fan is a low noise or more standard design. Fan static pressure, which in essence is the force required to overcome the system resistance (with the required design airflow rate), will typically run from 0.3–0.5 inches of water (~100 Pa +/- 20%) for a standard fan and system design.

Fan Shaft Power or Brake Horsepower – Depending on the fan static efficiency, it is possible to calculate whether the fan system will deliver the appropriate amount of air.

Power Requirements – Total fan power can be calculated using the aforementioned information and assuming nominal gearbox efficiencies of approximately 97% and motor efficiencies of approximately 92–94%.

An example of this process is provided in Appendix B.

5.2 Pricing

This section is provided to indicate the relative costs of components associated with the SRC design. While providing general insight into the estimated cost and pricing of ACCs at the time of this writing, newer design approaches, manufacturing sources, and methods are constantly under development.

Table 5-1
Typical ACC Component Cost Breakdown

Component	% Cost	Est. $
Heat Exchanger Bundles	32.0%	$ 192,000
Structural Steel	16.0%	$ 96,000
Casing	0.5%	$ 3,000
Fan Inlet Bell	0.9%	$ 5,400
Ducting	6.0%	$ 36,000
Expansion Joints/Bellows	1.3%	$ 7,800
Piping	1.5%	$ 9,000
Mechanical Equipment	5.4%	$ 32,400
Air Removal Pumps	1.4%	$ 8,400
Valves and Instrumentation	0.5%	$ 3,000
Drain Pumps & Rupture Disc	0.1%	$ 600
Condensate Tank / “Decorator Dome”	0.2%	$ 1,440
Shipping (U.S. Destination)	11.0%	$ 66,000
Engineering/Project Mgmt.	5.0%	$ 30,000
Subtotal	81.8%	$ 491,040
Overhead, Contingency, Profit	18.2%	$ 108,960
Total	100.0%	$ 600,000

Information in Table 5-1 might be used as the basis to evaluate “adds” and “deducts” in the bid process. These are especially helpful when multiple suppliers of ACC components are being considered, such as may be the case if a vendor or contractor other than the ACC designer/supplier provides steam piping, condensate system components, etc. Table 5-1 data may also be used as a guideline for spare-parts inventory development and budgeting. However, it should not be used as a backdrop to second guess or negotiate with a prospective ACC supplier relative to the overall pricing of a proposed ACC.

Beyond the component cost breakdown, it is worth noting in this Table 5-1 example that the "Overhead, Contingency, Profit" line item (also referred to in many companies as "Gross Margin") of 18.2% reflects the competitive marketplace for capital equipment in the U.S. power industry today. Assuming that a supplier's overhead (or Sales, General, and Administrative Expense line item) runs on the order 12–15% of revenues, there is clearly little remaining for performance and execution contingencies, research and development, and profit. This perspective should be borne in mind when negotiating final contract terms and conditions. Again, as indicated in Section 1.0 of this specification, EPRI hopes that one byproduct of this document is the forging of a more open and cooperative relationship between the purchaser and the supplier.

6
PERFORMANCE AND ACCEPTANCE TESTING

6.1 Introduction

The material and methodology cited in this chapter was developed solely by Power Generation Technology, a division of Environmental Systems Corporation (ESC), through EPRI subcontracted efforts that began in late 2002. Karl R. Wilber, principal investigator, provided minor edits. ESC's principal scientist was David E. Wheeler, who has significant power plant testing and analysis experience, with emphasis on condensers and condenser cooling systems.

6.1.1 Scope

This section details the measured test parameters, instrumentation, test measurements, and data reduction procedure required for determination of the thermal capability of a dry ACC. While the procedure focuses on contractual acceptance testing of a new unit, the same procedure may be used for performance testing of an existing unit.

6.1.2 Basis

As of this writing, there is no current U.S. test code or procedure for performance and acceptance testing of ACCs. Both the Cooling Technology Institute (CTI) and ASME are currently working on performance test codes for this major plant component. In the absence of a controlling test code, several resources have been used in the preparation of this guideline. These include:

VGB – *Acceptance Test Measurements and Operation Monitoring of Air-Cooled Condensers under Vacuum* (1997) [a]

ASME PTC 12.2 – *1998 Steam Surface Condensers* [b]

CTI ATC-105 *Acceptance Test Code* (2000) [c]

ASME PTC 23 – *2003 Atmospheric Water Cooling Equipment* [d]

6.1.3 Test Plan

A test plan is a convenient vehicle for specification of responsible test participants, required preparations, measurement locations, test instrumentation, acceptable test conditions, anticipated deviations to the governing test code, adjustments to plant operations, calculation procedures, and expected test uncertainty. As an example, the measurement of steam flow and the estimation of steam quality will require the use of plant instruments, particularly flow elements. It is vital that such instruments be identified prior to the test so that any necessary calibrations can be performed. In addition, measurement of condensing pressure requires the installation of basket tips that may be different in number and location than those used by the plant for monitoring purposes. The preparation of a test plan, approved by manufacturer and the ACC purchaser prior to the test, is highly recommended.

6.1.4 Definitions and Nomenclature

Definitions:

Capability – A measure of ACC thermal capacity expressed as a ratio between the design steam flow and the predicted steam flow at the test conditions

Condenser Pressure – The condensing steam pressure at the ACC boundary

Predicted Steam Flow – The steam flow rate predicted by the ACC manufacturer for a given set of test conditions

Nomenclature

A	=	area, m^2 (ft^2)
C	=	condenser capability (%)
F_c	=	correction factor from test to guarantee conditions
NTU	=	number of heat transfer units
P	=	pressure, Pa (in. HgA, psia)
Q	=	heat transfer rate, Watts (Btu/hr)
T	=	temperature, °C (°F)
U	=	overall heat transfer coefficient, $W/m^2/°C$ ($Btu/hr/ft^2/°F$)
$\dot{V}$	=	volumetric flow rate m^3/s (ft^3/min)
W	=	power, W
X	=	steam quality
c_p	=	heat capacity at constant pressure, kJ/kg/°C (Btu/lbm/°F)
h	=	specific enthalpy, kJ/kg (Btu/lbm)
m_k	=	exponent for the correction of test fan motor power to guarantee conditions
s	=	specific entropy, kJ/kg/°C (Btu/lbm °F)
ΔT_{lm}	=	log mean temperature difference, °C (°F)
Φ	=	condenser effectiveness
Γ	=	condenser characteristic parameter
α	=	film heat transfer coefficient, $W/m^2/°C$ ($Btu/hr/ft^2/°F$)
ρ	=	density, kg/m^3 (lbm/ft^3)

Subscripts

a	=	air or atmospheric
c	=	condensate
e	=	exhaust or exit
l	=	liquid
G	=	guarantee
P	=	predicted
T	=	test
i	=	inlet
o	=	outlet
s	=	condensing steam
v	=	vapor

6.2 Conditions of Test

6.2.1 Test Witnesses

For acceptance testing, representatives of the owner and ACC manufacturer shall be given adequate notice prior to the test. The manufacturer shall be given permission, opportunity, and adequate notice to inspect the ACC and prepare the ACC for the test. In no case shall any directly involved party be barred from the test site.

6.2.2 Condition of the Equipment

At the time of the test, the ACC shall be in good operating condition.

Steam duct and condensate piping systems shall be essentially clear and free of foreign materials that may impede the normal flow of steam and condensate.

Mechanical equipment, including fans, gears, motors, pumps, air ejectors, etc., shall be clean and in good working order. Fans shall be rotating in the correct direction, with proper orientation of the leading and trailing edges. Fan blade pitch shall be set to a uniform angle that will yield within ±10% of the specified fan driver input power load, as measured at the motor switchgear.

Air in-leakage must be such that the vacuum equipment has 50% excess holding capacity during the test.

ACC air inlet perimeter area and discharge area shall be essentially clear and free from temporary obstructions that may impede normal airflow.

The air-side of the ACC fin tube bundles shall be essentially free of foreign material, such as pollen, dust, oil, scale, paper, animal droppings, etc.

Water level in the condensate hot well tank shall be at the normal operating level.

Representatives of the ACC purchaser and manufacturer shall agree prior to commencement of testing that the cleanliness and condition of the equipment is within the tolerances specified by the manufacturer. Prior establishment of cleanliness and condition criteria is recommended.

All emergency drain lines that have the potential for delivering superheated steam to the condenser shall be isolated. A closed valve shall be considered adequate isolation.

6.2.3 Operating Conditions

The test shall be conducted while operating as close to the operation/guarantee point(s) as possible. In any event, the test shall be conducted within the following limitations:

The test dry-bulb temperature shall be the inlet value.

The wind velocity shall be measured and shall not exceed the following:

Average wind velocity shall be less than or equal to 5 m/s (11 mph).

One-minute duration velocity shall be less than 7 m/s (15.6 mph).

The following variations from design conditions shall not be exceeded:

Dry-bulb temperature – ±10°C from design (18°F) but greater than 5°C (41°F)

Condensate Mass Flow – ±10% of the design value

Fan Motor Input Power – ±10% of the design value after air density correction (equation 6-8)

Steam turbine exhaust steam shall be distributed to all modules as recommended by the manufacturer. For the purposes of this procedure, a "module" is defined as the smallest ACC subdivision, bounded externally by fin tube bundles and internally by partition walls, which can function as an independent unit. Each module generally has a single fan.

There shall be no rain during the test period or in the one-hour period preceding the test period.

Steady-state operation of the ACC shall be achieved at least one hour before and maintained during the test. All fans should be at full speed.

6.2.4 Constancy of Test Conditions

For a valid test, variations in test conditions shall be within the following limits:

The variation in test parameter shall be computed as the slope of a least squares fit of the time plot of parameter readings. Condensate mass flow shall not vary by more than 2% during the tests.

The inlet dry-bulb temperature shall not vary by more than 3°C (6°F).

6.3 Duration of the Test

After reaching steady-state conditions, the requirements for the test duration shall be at least one hour. Longer test intervals are acceptable provided the constancy of test conditions is observed.

6.4 Frequency of Readings

Readings shall be taken at regular intervals and recorded in the units and to the number of significant digits shown in Table 6-1.

Table 6-1
Measurement Frequency

Parameter Measured	Minimum Readings Per Hour Per Station	Unit	Recorded to Nearest
ACC Condensate Mass Flow Rate	60	kg/h (lb/h)	0.1%
Condensate Hot Well Tank Level	60	m (ft)	0.01 (0.03)
Exhaust Steam Pressure	60	kPa (in. HgA)	0.005 (0.01)
Exhaust Steam Temperature (for comparison)	60	°C (°F)	0.05 (0.1)
Inlet Air Dry-Bulb Temperature	60	°C (°F)	0.01 (0.01)
Atmospheric Pressure	1	kPa (in. Hg)	0.2 (0.05)
Ambient Wind Velocity	60	m/s (mph)	0.1 (0.2)
Fan Power at Switchgear	1	kW (HP)	0.5%

Even when tested under the guidelines specified, the apparent performance of ACCs may vary with the following environmental conditions:

Wind speed

Wind direction

Atmospheric stability

To decrease the possibility of an anomalous test result, at least six tests shall be performed over a two-day period. The condenser capability shall be the average of the tests conducted where test conditions were within the limits specified in this guideline.

6.5 Test Measurements

The objective of the parameter measurements is to accurately and reproducibly measure ACC thermal performance for comparison against the manufacturer guarantee. The primary parameters to be measured or calculated are:

Condenser pressure

Steam quality (content)

Condensate flow rate

Condensate tank water level

Inlet air dry-bulb temperature

Barometric pressure

Fan motor input power

Wind speed

It is recommended that the following parameters be acquired for reference purposes:

Exhaust steam temperature

Condensate temperature

Air removal rate

Wind direction

6.5.1 Condenser Pressure

Condenser pressure shall be measured at the boundary of supply of the condenser manufacturer unless parties to the test agree upon another location. Four measurement points per inlet are required, unless the flow in a given inlet is less than 5% of the total steam flow. For inlets with flows that are less than 5% of the total steam flow, one pressure measurement is required. Steam inlets with less than 1% of the total steam flow need not be instrumented.

For ACCs with multiple steam inlets, the mass weighted average absolute pressure of the instrumented inlets shall be used as the condenser pressure on lookup charts, tables, and performance curves.

The pressure measurement points shall be located at positions 90° apart around the steam inlet. Pressure ports shall be bored holes in the wall of the steam inlet connected to basket tips or baffle plates, as illustrated in Figures 6-1 and 6-2. Separate pressure sensors shall be connected to each port. With this approach, a bad transmitter should not compromise the test. It is preferable to have at least two instruments. Scanning valves, which allow a single pressure device to make measurements on each port sequentially, are also acceptable. Provisions should be made for purging all pressure connections to keep them free of condensate.

Pressure sensors shall have a calibrated accuracy of 35 Pa (0.005 psia) or less. Steam temperature measurements should be taken in the thermal wells in the vicinity of the pressure measurements.

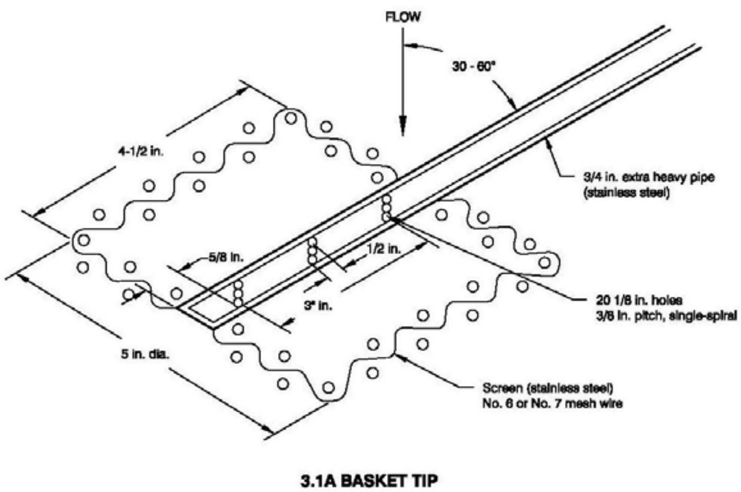

Figure 6-1
Schematic of Basket Tip

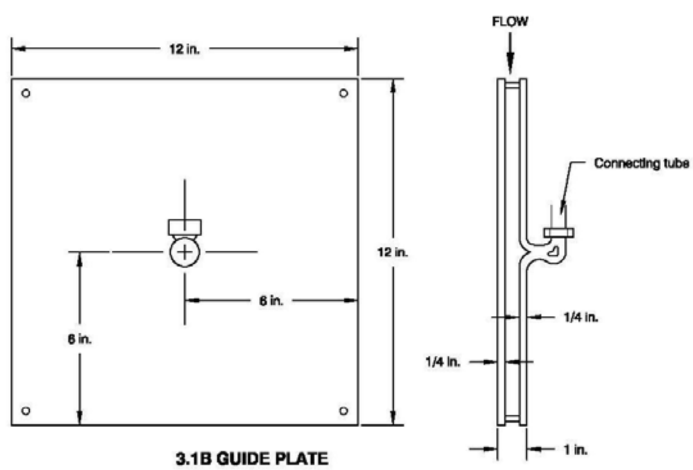

Figure 6-2
Schematic of Guide (Baffle) Plate

6.5.2 Steam Quality (Steam Content)

Since the steam at the condenser inlet will be in the wet steam range, measurement of the temperature and pressure is insufficient to determine its enthalpy. At present, there is no acceptable method for measuring the quality directly, thus the parties to the test must agree on a method for calculating this value. Some suggested methods include the following:

Energy Balance – If the temperature, pressure, and flow rate of all steam flows into the steam turbine are measured, and the power output is also measured, it is possible to determine the enthalpy of the exhaust steam by an energy balance. This method requires many measurements that are unlikely to be available unless a concurrent steam turbine test is being performed. If this method is used, the values of the steam flows should be verified by performing a mass balance around the condenser.

Expansion Line – The expansion for the low-pressure turbine is calculated from the unit heat balances or data from a previous steam turbine test. The quality of the turbine exhaust (condenser inlet) steam is calculated based on the calculated expansion line and measurement of the inlet temperature and pressure to the low-pressure turbine. A detailed procedure for the calculation is included in Section 6.11.

Cycle Model – With sufficient information from the steam turbine manufacturer, it is possible to construct a cycle model for the steam turbine. A cycle model can then be used to construct correction curves for steam quality based on measured cycle variables. Such a cycle model should be verified using the steam turbine manufacturer's thermal kit.

6.5.3 Condensate Flow

The condensate flow shall be measured downstream of the condensate pumps. The recommended devices for measuring the condensate flow are differential pressure producers (orifice plates, flow nozzles, venturis). The calibration records and construction details of the flow element shall be made available to all parties to the test. The pressure transmitter reading the differential pressure shall be calibrated prior to the test to an accuracy of not more than 0.25% of the expected differential at the design flow. The installation of the flow element shall conform to the specifications of ASME PTC 19.5 – *2004 Flow Measurement* [e].

A time-of-flight ultrasonic flow meter may be used by agreement between parties to the test. If used, the ultrasonic flow meter shall be calibrated in a pipe corresponding to the diameter and wall thickness of the pipe on which it will be installed. The calibration range shall cover the Reynolds number expected for the pipe at the design flow. The installation location shall have at least 16 pipe diameters of undisturbed length upstream of the meter and 4 pipe diameters of undisturbed length downstream of the meter. Readings shall be taken at six positions, 30º apart, around the circumference of the pipe. These readings shall be averaged to obtain the condensate flow. If the high and low readings differ by more than 2%, the cause shall be investigated. Use of an ultrasonic flow meter will result in a higher uncertainty for the condensate flow measurement than if an in-line flow element is used.

Steam flow to the condenser shall be calculated by a mass balance around the condenser, with consideration of any liquid flow streams upstream of the measurement point and the change in level of the tank during test. Design values may be used for flow streams representing less than 3% of the design steam flow.

6.5.4 Inlet Air Temperature

This guideline recommends ACC performance characterization based on the inlet air temperature as opposed to the ambient air temperature. Following are key considerations:

The results tend to be much more reproducible when the inlet air temperature is measured. ACCs are subject to recirculation (the re-entrainment of the exhaust air into the air inlet) and interference from other heat sources in the area. Slight changes in the wind speed or direction can greatly affect the amount of recirculation and interference, which in turn greatly increases the scatter in the test results if based on ambient temperature.

The test result tends to be a fairer representation of ACC performance. While the amount of recirculation is influenced by the condenser design chosen by the manufacturer, it is also governed by the condenser siting. Other structures or uneven topography in the vicinity can influence the amount of recirculation.

Ambient temperature in a power plant can be difficult to measure. It is difficult to find a location that is not influenced by other heat sources. Inlet temperatures to other equipment such as turbines or boilers are greatly influenced by heat sources in their surroundings. Temperature measurements at these locations alone should not be used to characterize ACC performance.

The air inlet temperature will be measured at the discharge location for each fan. At this location, the sensor is protected from solar radiation and exposed to a velocity of approximately 1000 ft/min. It is, therefore, not necessary to place the sensor in a psychrometer. The air temperature will be measured with a four-wire resistance temperature detector or thermistor with a calibrated accuracy of 0.05ºC (0.1ºF). At least one temperature sensor will be used for each cell, with a minimum of 12 sensors for the entire condenser.

6.5.5 Barometric Pressure

Barometric pressure will be measured at least once during each test period with a calibrated accuracy of 200 Pa (0.03 psia).

6.5.6 Fan Motor Input Power

The fan motor input power shall be determined by direct measurement of kilowatt input or by measurement of voltage, current, and power factor as described in American Society of Mechanical Engineers *PTC 6 REPORT - 1985 Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines* [f]. Measurement of the total input to all fan motors is acceptable if a measurement location isolated from other equipment can be established. The fan power measurement device shall have a maximum uncertainty of ±2% of reading.

6.5.7 Wind Speed

Wind speed shall be measured with a calibrated anemometer in an unobstructed location at a height relative to grade corresponding to the smaller of:

33 ft (10 m)

Half the air inlet height

If the wind speed is measured at a height lower than half the air inlet height, the measured wind speed shall be corrected to the midpoint of the air inlet height using the equation:

$$u^c = u(\frac{z_t}{z_m})^{0.2} \quad \textbf{Equation 6-1}$$

where

u^c	=	wind speed corrected to the midpoint of the air inlet
u	=	measured wind speed
z_m	=	vertical height of the wind speed station
z_t	=	vertical height of the midpoint of the air inlet

6.6 Evaluation of Test Data

6.6.1 Purpose

Section 6 develops a method for evaluation of the performance of an ACC from test data based on performance curves provided by the manufacturer.

6.6.2 Manufacturer's Data

The manufacturer shall submit a family of performance curves, consisting of a minimum of five curves, representing condenser pressure as function of dry-bulb temperature for steam flow rates of 80, 90, 100, 110, and 120% of design steam flow rate. One curve shall be provided for each design steam flow rate. Each curve shall be presented with dry-bulb temperature as the abscissa versus condensing steam pressure as the ordinate. Graphical scaling for pressure shall be incremented with a minimum resolution of 300 Pa (0.1 in. HgA) and a maximum resolution of 3000 Pa/2.5 cm (1.0 in. HgA). Dry-bulb temperature should be in the range of 0.2°C (0.5°F) and 0.1°C/mm (5°F/in.). **The curves shall be based on constant fan pitch.** An example performance curve is presented in Figure 6-3.

A table of values defining the curves shall also be provided. The table of values shall be sufficient to allow for the development of interpolation and curve fit equations that can be used in place of reading values off the curves during the performance test. Use of either the curve or the table of values should provide the same result, and either form of manufacturer-provided data is acceptable as the basis for capability calculations.

The design conditions – including steam mass flow rate, steam pressure, steam quality, fan motor input power, barometric pressure, and inlet dry-bulb temperature – shall be printed on the curves.

The effective area of the condenser and the volumetric airflow at design conditions shall also be included.

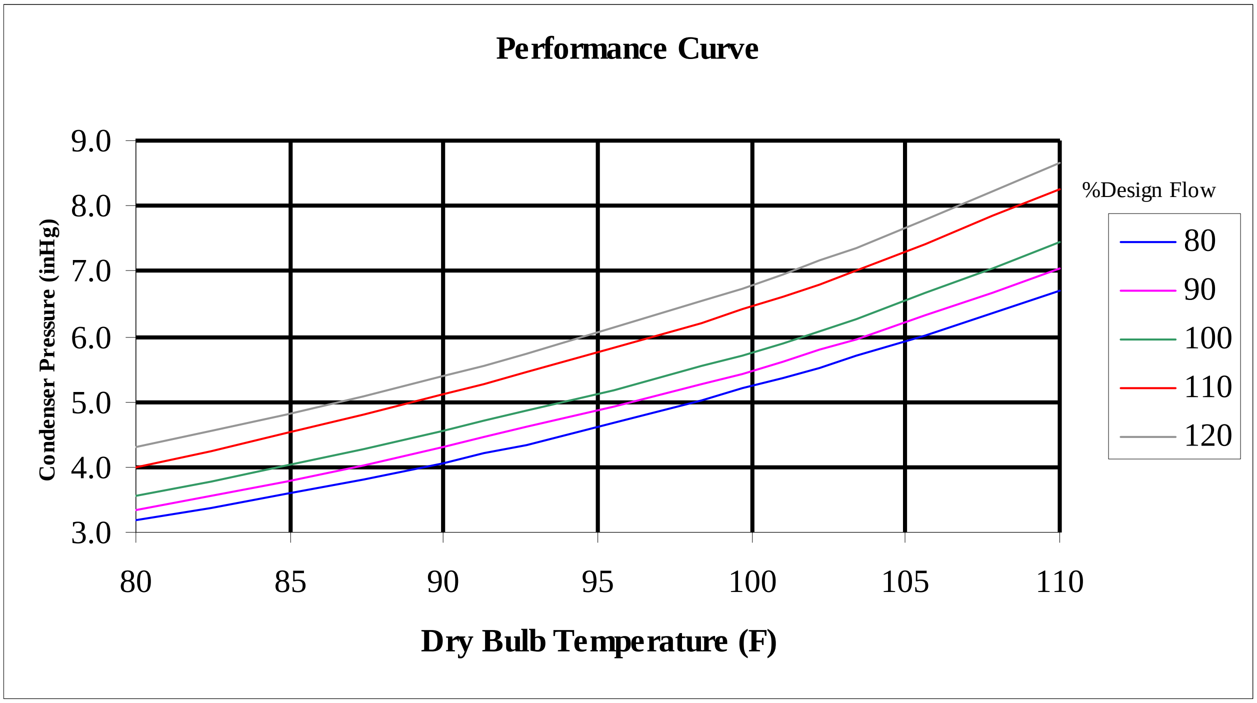

Figure 6-3
Example Performance Curve

6.6.3 Calculation of Condenser Capability

The condenser capability will be calculated by:

$$C = \frac{\dot{m}^{c}_{s,T}}{\dot{m}_{s,P}} \quad x \quad 100$$ **Equation 6-2**

where

C	=	condenser capability, percent
$\dot{m}^{c}{}_{s,T}$	=	corrected test mass flow of steam, kg/s (lbm/hr)
$\dot{m}_{s,P}$	=	predicted mass flow rate of steam at test conditions, kg/s (lbm/hr)

6.6.4 Predicted Steam Mass Flow Rate

The predicted condensing steam pressure at the measured inlet dry-bulb temperature shall be read from each of the performance curves. The resulting values shall be used to generate a plot of condensing steam pressure versus steam mass flow rate. The curve so generated is used to read the steam flow at the actual condensing steam pressure. This is the predicted steam mass flow rate, $\dot{m}_{s,pred}$, illustrated in Figure 6-4.

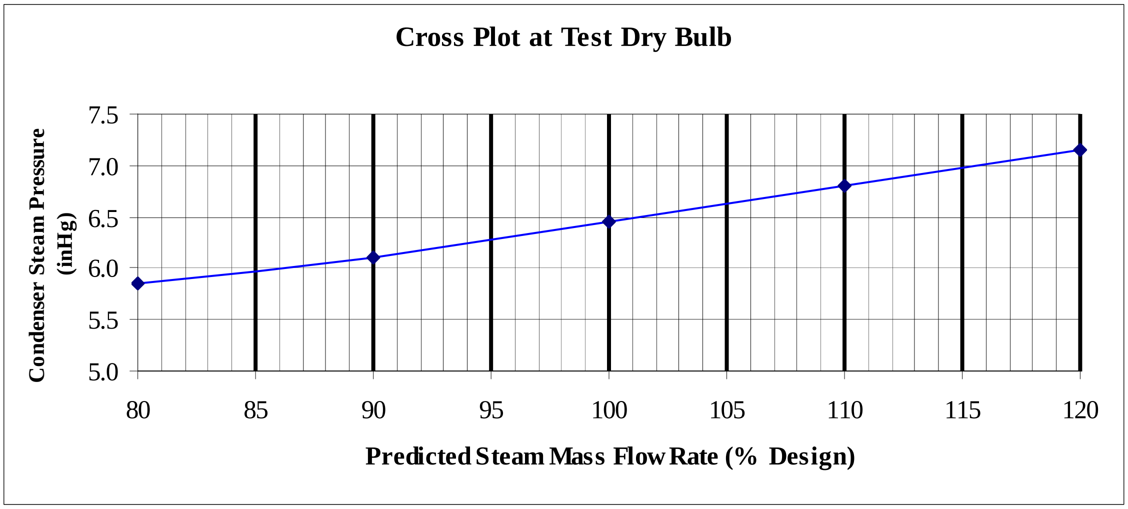

Figure 6-4
Cross Plot of Test Data

6.6.5 Corrected Test Steam Mass Flow Rate

The corrected test steam mass flow rate is calculated by

$$\dot{m}^{c}_{s,T} = \dot{m}_{s,T} F_c \qquad \textbf{Equation 6-3}$$

where

$\dot{m}_{s,T}$ = measured steam mass flow rate at test, kg/s (lbm/hr)

F_c = correction factor computed by

$$F_c = f_x f_p f_{fp} \qquad \textbf{Equation 6-4}$$

where

f_x = correction factor steam quality, dimensionless

f_p = correction factor for barometric pressure, dimensionless

f_{fp} = correction for fan power, dimensionless

The correction factor for steam quality is calculated by:

$$f_x = \frac{X_T}{X_G} \qquad \textbf{Equation 6-5}$$

where

X_T = steam quality at test conditions, kg/kg (lbm/lbm)

X_G = steam quality at guarantee conditions, kg/kg (lbm/lbm)

The correction factor for barometric pressure, f_p, shall be calculated by:

$$f_p = \left\{ \frac{P_T}{P_G}(1-\Gamma) + \Gamma(\frac{P_T}{P_G})^{m_k} \right\}^{-1}$$ **Equation 6-6**

where

P_T = test barometric pressure, kPa (psia)

P_G = design barometric pressure, kPa (psia)

Γ = constant factor based on design information; this factor can be calculated based on design information specified in Section 6.9

m_k = 0.45, unless otherwise specified by the manufacturer

The correction factor for fan power, f_{fp}, can be calculated by:

$$f_{fp} = \left(\frac{W_T^c}{W_G}\right)^{\frac{-1}{3}} \left\{ (1-\Gamma) + \Gamma\left(\frac{W_T^c}{W_G}\right)^{\frac{m_k - 1}{3}} \right\}^{-1}$$ **Equation 6-7**

W_T^c = test fan motor input power corrected for inlet air conditions, kW

W_G = guarantee fan motor input power, kW

The corrected fan motor power can be calculated by

$$W_T^c = \left\{ \frac{\rho_G}{\rho_T} \right\}$$ **Equation 6-8**

where

ρ_T = density of inlet air at test conditions, kg/m^3(lbm/ft^3)

ρ_G = density of inlet air at guarantee conditions, kg/m^3 (lbm/ft^3)

The average fan motor input power shall be corrected for any line losses between the measurement point and the boundary of supply for the condenser manufacturer.

The line loss can be calculated to one fan motor and applied to the other fan motors. This is because the line loss will be proportional to the length of the wire between the MCC and the fan motor, when the same size wire is used between the MCC and the boundary of supply of the ACC manufacturer.

6.7 Test Uncertainty

The purpose of the pretest uncertainty is to predict the uncertainty of the test results and to aid in the specification of test instrumentation that will achieve the test objective. The pretest uncertainty analysis should be documented in the test plan. The purpose of a posttest uncertainty analysis is to determine the accuracy or validity of the test result.

The following major uncertainty components are addressed in the ASME PTC 19.1 – *1998 Test Uncertainty* [g] test code:

Systematic uncertainty

Random uncertainty

Spatial uncertainty

Sensitivity coefficients

An overview of the uncertainty components is provided below.

Systematic Uncertainty – Systematic uncertainties are approximations of the fixed errors inherent in a measurement. These errors are also called bias errors. Systematic errors are typically the largest source of error in a condenser performance test. These uncertainties are primarily a result of the intrinsic accuracy of the instruments and the calibration procedures employed. Systematic uncertainties are estimated from review and analysis of the instrument manufacturer's specifications, independent parameter measurement by additional means, and examination of typical calibration data.

Spatial Systematic Uncertainty – Spatial systematic uncertainty errors occur during the measurement of a spatially diverse sample. Spatial error is defined as the difference between the true average value of a parameter and the average produced by an array of instruments used to measure the parameter. Spatial errors for a condenser performance test occur during the measurement of the inlet dry-bulb temperature. Spatial errors also occur during the measurement of condensate flow if an ultrasonic flow meter is used to measure the condensate flow. Spatial uncertainties are calculated from the average of local measurements in space. They are treated as constants for a given test period but may vary from one test period to another. For example, the spatial variation of the dry-bulb temperature may change from test to test due to changes in wind speed and direction, which in turn causes changes in recirculation and interference.

Random Uncertainty – Random uncertainty is also referred to as precision uncertainty. Random errors are evident by the scatter of data that results from repeated measurements of transient data (e.g., the variability in a dry-bulb temperature reading at a specific location). Precision errors can be reduced by increasing the number of measurement repetitions or by selecting data intervals with greater stability. Although it is possible to evaluate random uncertainty for a given test interval for each measured parameter, a more meaningful result is obtained by basing the random uncertainty on the variation of the condenser capability for the test periods.

Sensitivity Factors – Sensitivity factors relate a change in an independent measured parameter to the resulting change in the test result. These sensitivities may be calculated as the partial derivative of the test result with respect to the parameter of interest. However, it is usually more convenient to calculate the sensitivity factor numerically as the ratio of the change in the test result to the change in the test parameter. Sensitivity factors are used to combine the uncertainties for each test parameter into the uncertainty in the overall test result.

6.8 Basic Equations (A)

$$Q = UA\Delta T_{1_m} \qquad \text{A-1}$$

where

Q	=	heat duty
U	=	overall heat transfer coefficient
A	=	air-side heat transfer area for the condenser
ΔT_{lm}	=	log mean temperature difference

The log mean temperature difference, ΔT_{lm}, is defined as:

$$\Delta T_{lm} = \frac{(T_s - T_{a,i}) - (T_s - T_{a,o})}{\ln\left(\dfrac{T_s - T_{a,i}}{T_s - T_{a,o}}\right)} = \frac{T_{a,o} - T_{a,i}}{\ln\left(\dfrac{T_s - T_{a,i}}{T_s - T_{a,o}}\right)} \qquad \text{A-2}$$

where

T_s	=	condensing steam temperature
$T_{a,i}$	=	inlet dry-bulb temperature
$T_{a,o}$	=	outlet dry-bulb temperature

Note: Equation A-2 will generally give you a very different answer than Equation A-1 because of the difficulty in determining the exact condensing steam temperature. The true condensing temperature is not the same as the saturation temperature corresponding to the turbine backpressure and it can be as much as a few degrees higher than the actual condensing temperature in the ACC because of the pressure drop in the steam ducting.

The heat duty for the condenser is:

$$Q = \dot{m}_s (h_{s,i} - h_{c,o}) = \dot{m}_a c_{p,a} (T_{a,o} - T_{a,o}) \qquad \text{A-3}$$

where

$\dot{m}_s$	=	mass flow rate of steam
$h_{s,i}$	=	enthalpy of inlet steam
$h_{c,o}$	=	enthalpy of liquid condensate

$\dot{m}_a$	=	mass flow rate of air
$c_{p,a}$	=	heat capacity of air

The enthalpy of the inlet steam can be calculated by:

$$h_{s,i} = X_{s,i}h_s + (1 - X_{s,i})h_l \qquad \text{A-4}$$

where

$X_{s,i}$	=	quality of inlet steam
h_s	=	enthalpy of saturated steam at the condenser inlet pressure
h_l	=	enthalpy of condensate at the condenser inlet pressure

The mass flow rate of the inlet air can be calculated by:

$$\dot{m}_a = \dot{V}_{a,i}\rho_{a,i} \qquad \text{A-5}$$

where

$\dot{V}_{a,i}$	=	volumetric airflow
$\rho_{a,i}$	=	air density at inlet conditions

For constant pitch performance curves, the volumetric airflow rate is independent of air temperature and pressure. The outlet air temperature is calculated by:

$$T_{a,o} = T_{a,i} + \frac{Q}{\dot{m}_a c_{p,a}} \qquad \text{A-6}$$

From equation A-1

$$U = \frac{Q}{A\Delta T_{lm}} \qquad \text{A-7}$$

and from equations A-1, A-2, and A-7:

$$Q = \dot{m}_a c_{p,a}(T_s - T_{a,i}) \frac{e(\frac{UA}{\dot{m}_a c_{p,a}}) - 1}{e(\frac{UA}{\dot{m}_a c_{p,a}})} \qquad \text{A-8}$$

The number of heat transfer units is defined as:

$$NTU = \frac{UA}{\dot{m}_a c_{p,a}} \qquad \text{A-9}$$

and is equivalent to:

$$NTU = \frac{T_{a,o} - T_{a,i}}{\Delta T_{lm}} \qquad \text{A-10}$$

Therefore

$$Q = \dot{m}_a c_{p,a} (T_s - T_{a,i}) \frac{e^{NTU} - 1}{e^{NTU}} \qquad \text{A-11}$$

The effectiveness of the condenser is defined as:

$$\Phi = \frac{T_{a,o} - T_{a,i}}{T_s - T_{a,i}} \qquad \text{A-12}$$

and is equivalent to

$$\Phi = NTU \frac{\Delta T_{lm}}{T_s - T_{a,i}} \qquad \text{A-13}$$

For the case of isothermal condensation:

$$\Phi = 1 - e^{-NTU} \qquad \text{A-14}$$

The ratio of the test to guarantee effectiveness is:

$$\frac{\Phi_T}{\Phi_G} = \left(\frac{1 - e^{-NTU_T}}{1 - e^{-NTU_G}} \right) = 1 + \frac{1 - e^{-(NTU_T - NTU_G)}}{e^{NTU_G} - 1} \qquad \text{A-15}$$

For small values of δ(-0.15<δ<0.15)

$$e^{-\delta} \approx 1 - \delta \qquad \text{A-16}$$

Therefore,

$$\frac{\Phi_T}{\Phi_G} = 1 + \frac{NTU_T - NTU_G}{e^{NTU_G} - 1} = 1 + \frac{NTU_G}{e^{NTU_G} - 1} \frac{NTU_T}{NTU_G} - \frac{NTU_G}{e^{NTU_G} - 1} \qquad \text{A-17}$$

Defining

$$\Gamma = \frac{NTU_G}{e^{NTU_G} - 1} \qquad \text{A-18}$$

$$\frac{\Phi_T}{\Phi_G} = 1 + \Gamma \frac{NTU_T}{NTU_G} - \Gamma \qquad \text{A-19}$$

From equations A-4 and A-13,

$$\frac{Q_T}{Q_G} = \frac{\Phi_T}{\Phi_G} \frac{\dot{m}_{a,T}}{\dot{m}_{a,G}} \qquad \text{A-20}$$

From equation A-10

$$\frac{NTU_T}{NTU_G} = \frac{U_T}{U_G} \frac{\dot{m}_{a,G}}{\dot{m}_{a,T}} \qquad \text{A-21}$$

Since

$$\dot{m}_a = \rho_a \dot{V}_a \qquad \text{A-22}$$

where

$\dot{V}_a$ = volumetric flow rate of air

ρ_a = density of air

$$\frac{NTU_T}{NTU_G} = \frac{U_T}{U_G} \frac{\rho_{a,i,G} \dot{V}_{a,i,G}}{\rho_{a,i,T} \dot{V}_{a,i,T}} \qquad \text{A-23}$$

The air-side heat transfer coefficient, α_a, is a function of Reynolds number

$$\alpha_a \propto \mathrm{Re}^{m_k} \propto \left(\rho_q \dot{V}_a\right)^{m^k} \qquad \text{A-24}$$

Since the overall heat transfer resistance is dominated by the air-side resistance

$$\frac{U_T}{U_G} \approx \frac{\alpha_T}{\alpha_G} \approx \left(\frac{\rho_{a,i,T}}{\rho_{a,i,G}}\right)^{m_k} \left(\frac{\dot{V}_{a,i,T}}{\dot{V}_{a,i,G}}\right)^{m_k} \qquad \text{A-25}$$

Therefore,

$$\frac{NTU_T}{NTU_G} = \left(\frac{\rho_{a,i,G}}{\rho_{a,i,T}}\right)^{1-m_k} \left(\frac{\dot{V}_{a,i,G}}{V_{a,i,T}}\right)^{1-m_k} \qquad \text{A-26}$$

From equations A-19, A-20, and A-26:

$$\frac{Q_T}{Q_G} = \frac{\rho_{a,i,T}}{\rho_{a,i,G}} \frac{\dot{V}_{a,i,T}}{\dot{V}_{a,i,G}} \left[1 - \Gamma + \Gamma \left(\frac{\rho_{a,i,G}}{\rho_{a,i,T}} \right)^{1-m_k} \left(\frac{\dot{V}_{a,i,G}}{\dot{V}_{a,i,T}} \right)^{1-m^k} \right] \tag{A-27}$$

From fan affinity laws:

$$\frac{\dot{V}_T}{\dot{V}_G} = \left(\frac{W_T^c}{W_G} \right)^{\frac{1}{3}} \tag{A-28}$$

where

W_G = fan power at guarantee conditions

W_T^c = test fan power corrected to design temperature and pressure

Substituting in equation A-27 yields:

$$\frac{Q_T}{Q_G} = \frac{\rho_{a,i,T}}{\rho_{a,i,G}} \left(\frac{W_T^c}{W_G} \right)^{\frac{1}{3}} \left[1 - \Gamma + \Gamma \left(\frac{\rho_{a,i,G}}{\rho_{a,i,T}} \right)^{1-m^k} \left(\frac{W_T^c}{W_G} \right)^{\frac{m^k - 1}{3}} \right] \tag{A-29}$$

The correction factor for fan power is:

$$f_{fp} = \frac{\dot{m}_{s,G}}{\dot{m}_{s,T}} = \frac{Q_G}{Q_T} = \left(\frac{W_T^c}{W_G} \right)^{\frac{-1}{3}} \left[1 - \Gamma + \Gamma \left(\frac{W_T^c}{W_G} \right)^{\frac{m_k - 1}{3}} \right]^{-1} \tag{A-30}$$

From the ideal gas law:

$$\rho = \frac{PM}{RT_A} \tag{A-31}$$

where

P = absolute pressure, Pa (psia)

M = molecular weight of gas, 28.945 kg/kg-mole (lbm/lbm-mole)

R = universal gas constant, $8.3143\text{x}10^3 \dfrac{Pa\ m^3}{kg - mole\ K}$ $\left(10.73\ \dfrac{psi\ \ ft^3}{lbm - mole\ {}^\circ R}\right)$

T_A = absolute temperature, K (°R)

The change in volumetric flow with inlet temperature is included in the performance curves. Therefore, the correction factor for barometric pressure is

$$f_p = \frac{\dot{m}_{s,G}}{\dot{m}_{s,T}} = \frac{Q_G}{Q_T} = \frac{P_{a,G}}{P_{a,T}}\left[1 - \Gamma + \Gamma\left(\frac{P_{a,G}}{P_{a,T}}\right)^{1-m_k}\right]^{-1} \qquad \text{A-32}$$

where

$P_{a,G}$	=	atmospheric pressure at guarantee conditions
$P_{a,T}$	=	atmospheric pressure at test conditions

6.9 Calculation of Condenser Characteristics (B)

Calculate guarantee heat transfer rate using equation A-4

$$Q = \dot{m}_s (h_{s,i} - h_{c,o}) \qquad \text{B-1}$$

Calculate the inlet air density at guarantee conditions using equation A-30

$$\rho = \frac{PM}{RT_A} \qquad \text{B-2}$$

Calculate the air mass flow rate using equation A-5

$$\dot{m}_a = \dot{V}_{a,i}\rho_{a,i} \qquad \text{B-3}$$

Calculate outlet air temperature using equation A-7

$$T_{a,o} = T_{a,i} + \frac{Q}{\dot{m}_a c_{p,a}} \qquad \text{B-4}$$

Calculate log mean temperature difference using equation A-2

$$\Delta T_{lm} = \frac{T_{a,o} - T_{a,i}}{\ln\left(\frac{T_s - T_{a,i}}{T_s - T_{a,o}}\right)} \qquad \text{B-5}$$

Calculate NTU using equation A-11

$$NTU = \frac{T_{a,o} - T_{a,i}}{\Delta T_{lm}} \qquad \text{B-6}$$

Calculate the heat transfer characteristic using equation A-18

$$\Gamma = \frac{NTU_G}{e^{NTU_G} - 1} \qquad \text{B-7}$$

6.10 Example Air-Cooled Condenser Capability Calculations

6.10.1 Design and Test Data

ACC design and test conditions are presented in Table 6-2.

Table 6-2
Air-Cooled Condenser Test Data

Parameter	Units	Design	Test
Steam flow	lbm/hr	1,250,000	1,250,834
Steam quality	%	94	94.6
Condenser pressure	in. HgA	3.00	3.86
Inlet air temperature	ºF	65.0	75.49
Atmospheric pressure	in. HgA	28.85	28.56
Fan power	hp	6115	5634
Wind speed	mph	10.0	8.6
Condensate outlet temperature	ºF	113.0	121.2
Volumetric airflow	acfm	3.775×10^{7}	----
Calculated Values			
Condensing steam temperature	ºF	115.0	124.1
Enthalpy of vapor at condenser pressure	Btu/lbm	1111.1	1114.9
Enthalpy of liquid at condenser pressure	Btu/lbm	83.0	92.1
Enthalpy of condensing steam	Btu/lbm	1049.8	1060.0
Enthalpy of condensate at outlet temperature	Btu/lbm	81.0	89.2

Condensing steam temperature and specific enthalpy values for steam and liquid water were calculated by using steam properties software.

6.10.2 Performance Curves

The performance curves supplied by the manufacturer are presented in Figure 6-5 and Table 6-3.

Table 6-3
Performance Curves

		Condensing Pressure (in. HgA)	

Air Temp	120% Flow	110% Flow	100% Flow	90% Flow	80% Flow
85.0	6.39	5.68	5.05	4.48	3.96
80.0	5.65	5.02	4.44	3.92	3.48
75.0	4.98	4.42	3.90	3.44	3.03
70.0	4.39	3.89	3.42	3.02	2.62
65.0	3.87	3.41	3.00	2.63	2.30
60.0	3.40	2.98	2.63	2.30	2.01
55.0	2.99	2.62	2.30	2.01	1.74
50.0	2.63	2.31	2.02	1.75	1.51
45.0	2.31	2.02	1.77	1.53	1.32
75.5	5.05	4.48	3.95	3.49	3.07

Design data for condenser pressure as function of dry-bulb temperature for steam flow rates between 80% and 120% of the design flow rate were provided by the manufacturer. The predicted condenser pressure at the test inlet air temperature of 75.5ºF was calculated by nonlinear interpolation for each flow rate.

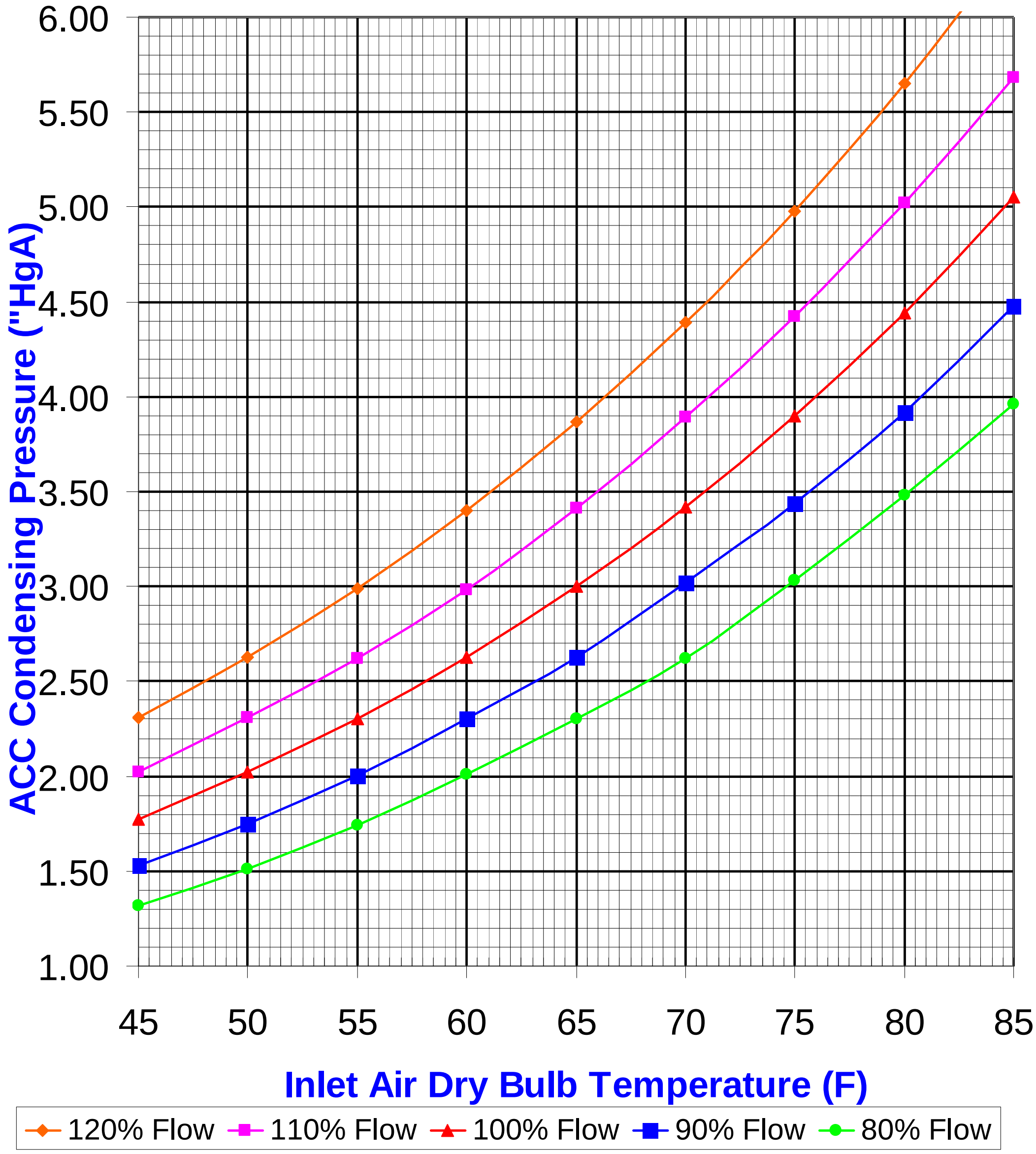

Figure 6-5
Air-Cooled Condenser Performance Curves

6.10.3 Predicted Steam Flow at Test Conditions

The data required for calculation of the predicted flow at test conditions are summarized in Table 6-4.

Table 6-4
Condenser Pressure at Test Conditions

Steam Flow	Condenser Pressure
% of Guarantee	in. HgA
80%	3.07
90%	3.49
100%	3.95
110%	4.48
120%	5.05
98.01%	3.86

The predicted percentage steam flow rate at test pressure of 3.86 in. HgA was calculated by nonlinear interpolation from the data in Table 6-4. The predicted steam flow rate at test conditions is

$$m_{s,P} = (0.9801)x(1{,}250{,}000) = 1{,}225{,}069 lbm/hr$$

6.10.4 Calculation of Condenser Characteristics

1. Calculate guarantee heat transfer rate using equation B-1

$$Q = \dot{m}_s(h_{s,i} - h_{c,o})$$

$$Q = 1.25x10^6(1049.8 - 83.0) = 1.208x10^9 Btu/hr$$

2. Calculate the inlet air density at guarantee conditions using equation B-2

$$\rho = \frac{PM}{RT_A}$$

$$\rho_G = \rho_{a,i} = \frac{(28.85)(0.491)(28.945)}{(10.73)(65.0 + 459.7)} = 0.0729 lbm/ft^3$$

3. Calculate the air mass flow rate using equation B-3

$$\dot{m}_a = \dot{V}_{a,i}\rho_{a,i}$$

$$\dot{m}_a = (3.775x10^7)(0.0729)(60) = 1.6508x10^8 \, lbm/hr$$

4. Calculate outlet air temperature using equation B-4

$$T_{a,o} = T_{a,i} + \frac{Q}{\dot{m}_a c_{p,a}}$$

$$T_{a,o} = 65.0 + \frac{1.208x10^9}{1.65x10^8 \, (0.24)} = 95.5$$

5. Calculate log mean temperature difference using equation B-5

$$\Delta T_{lm} = \frac{T_{a,o} - T_{a,i}}{\ln\left(\frac{T_s - T_{a,i}}{T_s - T_{a,o}}\right)}$$

$$\Delta T_{lm} = \frac{95.5 - 65.0}{\ln\left(\frac{115.0 - 65.0}{115.0 - 95.5}\right)} = 32.4$$

6. Calculate NTU using equation B-6

$$NTU = \frac{T_{a,o} - T_{a,i}}{\Delta T_{lm}}$$

$$NTU = \frac{95.5 - 65.5}{32.4} = 0.942$$

7. Calculate the heat transfer characteristic using equation B-7

$$\Gamma = \frac{NTU_G}{e^{NTU_G} - 1} \qquad \Gamma = \frac{0.942}{e^{0.942} - 1} = 0.602$$

6.10.5 Correction to Guarantee Conditions

1. Calculate correction for steam quality using equation 6-5

$$f_x = \frac{X_T}{X_G}$$

$$f_x = \frac{94.6}{94.0} = 1.007$$

2. Calculate correction for barometric pressure using equation 6-6

$$f_p = \left\{ \frac{P_T}{P_G}(1-\Gamma) + \Gamma(\frac{P_T}{P_G})^{m_k} \right\}^{-1}$$

$$f_p = \left\{ \frac{28.56}{28.85}(1-0.602) + 0.602(\frac{28.56}{28.85})^{0.45} \right\}^{-1} = 1.007$$

3. Calculate air inlet density at test conditions using $\rho = \frac{PM}{RT_A}$

$$\rho_T = \frac{(28.56)(0.491)(28.945)}{(10.73)(75.5+459.7)} = 0.0707 \text{lbm}/\text{ft}^3$$

4. Calculate corrected fan power using equation 6-8

$$W_T^c = \frac{\rho_T}{\rho_G} W_T$$

$$W_T^c = \frac{0.0707}{0.0729} 5634 = 5464 \text{hp}$$

5. Calculate fan power correction using equation 6-7

$$f_{fp} = \left(\frac{W_T^c}{W_G} \right)^{\frac{-1}{3}} \left\{ (1-\Gamma) + \Gamma \left(\frac{W_T^c}{W_G} \right)^{\frac{m_k - 1}{3}} \right\}^{-1}$$

$$f_{fp} = \left(\frac{5464}{6115} \right)^{\frac{-1}{3}} \left\{ (1-0.602) + 0.602 \left(\frac{5464}{6115} \right)^{\frac{0.45-1}{3}} \right\}^{-1} = 1.012$$

6. Calculate test steam flow correction factor using equation 6-4

$$F_c = f_x f_p f_{fp}$$

$$F_c = (1.007)(1.007)(1.012) = 1.025$$

7. Calculate corrected condenser steam flow using equation 6-3

$$\dot{m}_{s,T}^{c} = \dot{m}_{s,T} F_c$$

$$\dot{m}_{s,T}^{c} = (1{,}250{,}834)(1.025) = 1{,}263{,}931\, lbm/hr$$

6.10.6 Calculation of Condenser Capability

Calculate condenser capability using equation 6-2

$$C = \frac{\dot{m}_{s,T}^{c}}{\dot{m}_{s,P}}(100)$$

$$C = \frac{1{,}263{,}931}{1{,}225{,}069}(100) = 103.2\%$$

6.11 Calculation of Steam Quality

Procedure for Calculation of Steam Quality at Turbine Exhaust

The procedure that follows assumes that the slope of the enthalpy versus entropy line for the low-pressure steam turbine is independent of the exhaust pressure, inlet temperature, pressure, and flow. This is equivalent to assuming a constant isentropic efficiency for the low-pressure turbine. Studies using cycle models have indicated that the error involved with calculating the steam quality based on this assumption is less than 1%.

1. From the turbine heat balance diagram corresponding to the ACC design conditions, obtain the inlet temperature and pressure for the low-pressure turbine as well as the turbine exhaust enthalpy and pressure.

2. Using steam tables or equivalent software look up (or calculate) the specific enthalpy and specific entropy of the low-pressure turbine inlet steam.

3. Calculate the quality of the turbine exhaust steam by:

$$X_d = \frac{h_{e,d} - h_{l,d}}{h_{v,d} - h_{l,d}}$$

where

X_d	=	moisture fraction of the turbine exhaust at the heat balance conditions
$h_{v,d}$	=	specific enthalpy of saturated vapor at the exhaust pressure
$h_{e,d}$	=	specific enthalpy of the exhaust steam
$h_{l,d}$	=	specific enthalpy of saturated liquid at the exhaust pressure

This value should correspond to the guarantee condition for the condenser.

4. Calculate the entropy of the turbine exhaust steam by:

$$s_{e,d} = (1 - X_d)s_{v,d} + X_d s_{l,d}$$

where

s_e	=	specific entropy of turbine exhaust steam
$s_{v,d}$	=	specific entropy of saturated vapor at the turbine exhaust pressure
$s_{l,d}$	=	specific entropy of saturated liquid at the turbine exhaust pressure

5. Calculate the slope of the expansion line by:

$$m_e = \frac{h_{i,d} - h_{e,d}}{s_{i,d} - s_{e,d}}$$

where

m_e	=	slope of the expansion line
$h_{i,d}$	=	enthalpy of the low-pressure turbine inlet steam
$s_{i,d}$	=	entropy of the low-pressure inlet steam

Note 1: The termination point of this expansion line is the used energy end point (UEEP), rather than the expansion line end point (ELEP). The UEEP represents the actual enthalpy of the exhaust steam. The ELEP is a constructed quantity to allow for calculation of the enthalpy value for the extraction steam to the low-pressure condensate heaters (if any), which may be saturated.

Note 2: If a turbine test on the unit has been performed, the slope of the expansion line may be calculated by substituting actual values from the turbine test for the design values in steps 1 through 5.

6. From the temperature and pressure of the turbine inlet steam at test conditions, determine the enthalpy, h_i and entropy, s_i, of the exhaust steam at test conditions.

7. Calculate the quality of the steam at the test condition by:

$$X_T = \frac{(h_i - h_l) + m_e(s_i - s_l)}{(h_v - h_l) + m_e(s_v - s_l)}$$

where

X_T	=	steam quality at the turbine exhaust at test conditions
$h_{t,i}$	=	specific enthalpy of the inlet steam for the low-pressure turbine
s_i	=	specific entropy of the inlet steam for the low-pressure turbine
h_l	=	specific enthalpy of liquid water at the turbine exhaust pressure

h_v = specific enthalpy of vapor at the turbine exhaust pressure
Se = specific entropy of liquid water at the turbine exhaust pressure

6.12 References

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c) Cooling Technology Institute. *Acceptance Test Code*. ATC-105 (00). Houston, Texas: CTI, Revised February 2000.

d) American Society of Mechanical Engineers. *PTC 23 – 2003 Atmospheric Water Cooling Equipment*. New York, New York: ASME, 2003.

e) American Society of Mechanical Engineers. *PTC 19.5 – 2004 Flow Measurement.* New York, New York: ASME, 2004.

f) American Society of Mechanical Engineers. *PTC 6 REPORT - 1985 Guidance for Evaluation of Measurement Uncertainty in Performance Tests of Steam Turbines.* New York, New York: ASME, 1985, reaffirmed 1997.

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7
AIR-COOLED CONDENSER INSTALLATION AND COMMISSIONING ISSUES

7.1 Overview

This chapter is intended to summarize major ACC installation and commissioning issues. It is based on the assumption that, in most cases, the ACC supplier is *not* responsible for unloading of ACC components, temporary storage, protection of components, and the erection and initial check out of the ACC. It assumes that a free on board (FOB) job site protocol characterizes the contractual relationship between the purchaser and the supplier. However, it is common that the ACC supplier would provide guidance, both written and via on-site technical and commissioning assistance, to promote proper offloading, storage, construction, and commissioning.

7.2 Unloading and Storage

7.2.1 Shipping and Receiving

ACC components are typically manufactured and preassembled in sizes and weights to facilitate shipment via truck from point of manufacture to the job site. It can be the case that rail, or in rare cases, cargo ships and trucking are used in combination. In any event, the responsible party or parties (which may be dictated by ownership of record and attendant insurance coverage) should receive the shipments. At that time, they should note unusual transfer of weights, damaged packaging, etc., documenting any aberrations via photographs and written correspondence.

Perhaps the most vulnerable ACC components are the finned-tube sections. Particular scrutiny of these sections should be made to assess whether damage to them has occurred prior to unloading. Signs of damage to protected or galvanized surfaces should be noted. In addition, surfaces should be inspected for evidence of grease, dirt, or other contaminants. Finally, any evidence of separation of finned tubes from collection headers should be noted and reported to the shipper and supplier.

7.2.2 Unloading of Air-Cooled Condenser Components

The purchaser should solicit specific unloading procedures from the ACC supplier. This would include such areas as:

Tie down and bolting arrangements and removal of the same

Component lists, including descriptions, weights, etc.

Location of lifting lugs

Recommendations for support or spreader beams, jigs, straps, etc. for lifting

Precautions on lifting procedures

Notations of unique supporting requirements (e.g., spreader bars) for specific components, etc.

7.2.3 Storage of Air-Cooled Condenser Components

Depending on plant construction and shipping sequencing schedules, ACC components may be placed in lay down and/or storage areas for an extended period. If this is anticipated, specific instructions from the supplier for storage and protection of ACC components should be solicited.

7.3 Erection of the Air-Cooled Condenser

ACC erection should be performed in strict accordance with instructions and procedures provided by the supplier. Therefore, the ACC specification should solicit the following:

A list of all components to be installed (e.g., structural steel, A-frames, finned-tubes, steam and condensate headers, wind walls and division walls, steam ducting, ejector skids, condensate tanking, interconnecting piping, gearboxes, motors, fans, fan rings, walkways, ladders, cleaning systems, instrumentation and control sensors and systems, etc.

A reiteration of foundation requirements, connections, etc.

Documentation on the sequence of erection, noting individual and cumulative tolerances

Special tools, jigs, spreader bars, etc. that might be required or would prove valuable in construction

Welding and fitting procedures, including torque settings, etc.

7.4 Startup and Commissioning Tests

7.4.1 Pressure Testing

Pressure testing of the assembled ACC system(s), is an essential part of commissioning and startup . This testing verifies the integrity of the total ACC system, including finned-tube bundles, steam headers, dephlegmator sections, and connecting piping. Pressure and integrity testing should also extend to the condensate system, including piping, drains, valving, etc.

The ACC should be equipped with a blanking plate spool piece that can be used to isolate the system for pressure testing. To conduct pressure testing, a compressor capable of pressurizing the total ACC system to 15 psig should be provided along with pressure relief valving. Once the system is fully pressurized, leak-test fluid can be administered to weld joints, fittings, and other connections that may be suspect, based on supplier insights from previous installations.

7.4.2 In-leakage Testing

Air in-leakage testing can also be performed, obviously after the ACC is in service and sufficient vacuum exists on the ACC system. In performing such tests, a tracer gas is typically administered around welded connections, valving, fittings, etc. Monitoring for that same gas at the exhaust of the air ejector system reveals the presence of leaks and the need for remedial action. Helium injection, a procedure used in the 1970s and 1980s has been replaced with the more common and sensitive sulfur hexafluoride (SF_6), which requires lower concentration limits for detection. Testing contractors specializing in such services can typically offer insights and experience for efficient administration of leak-test gases. Areas where leaks typically occur – such as around weld joints, valves, and test ports – should be well known to testing services suppliers.

7.4.3 Internal Cleaning and System Inspections

Experience indicates that the ACC can be a collection or disposal point for debris, including carryover of trace metals and contaminants from the steam system. For this reason, it is recommended that inspections and internal purging of the ACC be performed during outages and prior to performance and acceptance testing of the equipment. The ACC manufacturer should spearhead those efforts as part of the commissioning, startup, and acceptance testing.

7.4.4 Rotating Equipment and Vibration Assessments

Before the fans, motors, and gearboxes are operated, the gearboxes should be filled with the proper type and level of lubricant. In addition, all bolts and connections should be checked for proper torque.

Fans and fan shafts can be rotated by hand to ensure proper clearance and to confirm that there are no obstructions in the path of the fan blades. During initial startup of the rotating equipment, vibration switches can be set to their minimum sensitivity to assess the vibration behavior of the rotating equipment. Fan tip clearance, tracking, and blade pitch settings should be verified for each fan assembly and blade. This information should be recorded to ensure that these assessments were made and to reference during future outages and verifications.

Once the fans, motors and gearboxes are energized, the following areas should be assessed:

Motor voltage and current along with ambient air temperatures

Gearbox lube oil temperatures, which should remain at ~190–210ºF (~90–100ºC)

Motor bearing temperatures to determine if they remain within specified limits

With the fans still in operation, the vibration switch sensitivities can be increased to a trip point and then backed off slightly.

7.4.5 Walk-Through Inspection

With the ACC in its initial operation, a walk-through inspection should be performed with the following areas in mind:

Is each bay free of debris, grease, or other contaminants?

Are the bays and doorways properly installed and providing cell-to-cell isolation?

Are there any signs of finned-tube bundle corrosion in this early post-construction stage?

Are the walkways, fan bridge, ladders, inlet screens and supports, and access platforms properly installed and plumb?

Are there any obvious signs of excessive or differential vibrations from cell to cell?

Is all hardware, valving, and instrumentation installed per the specification and supplier's proposal? Is the equipment operating properly and supplying reasonable outputs and functionality?

Is the condensate tank system installed properly, and are the condensate pumps operating properly?

Has the rupture disc been installed properly and the blind plate been removed following pressure testing?

Is the air ejector system operating properly?

A
AIR-COOLED CONDENSER INSTALLATIONS

Appendix A.1: GEA Power Cooling, Inc., Direct Air-Cooled Condenser Installations

STATION OWNER (A/E)	SIZE MWe (1)	STEAM FLOW [Lb/Hr]	TURBINE Back Pressure [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 Generating Station **Braintree Electric Light Dept.** Braintree, MA (R.W. Beck)	20	190,000	3.5	50	1975	Combined Cycle
Benicia Refinery **Exxon Company, U.S.A.** Benicia, 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 (Pennsylvania Engineering)	7.5	50,340	4.0	79	1985	WTE

STATION OWNER (A/E)	SIZE MWe (1)	STEAM FLOW [Lb/Hr]	TURBINE Back Pressure [In. HgA]	DESIGN TEMP. [Deg. F]	YEAR	REMARKS
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)	4	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®, 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®, 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)

STATION OWNER (A/E)	SIZE MWe (1)	STEAM FLOW [Lb/Hr]	TURBINE Back Pressure [In. HgA]	DESIGN TEMP. [Deg. F]	YEAR	REMARKS
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 Ltd.** 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 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

STATION OWNER (A/E)	SIZE MWe (1)	STEAM FLOW [Lb/Hr]	TURBINE Back Pressure [In. HgA]	DESIGN TEMP. [Deg. F]	YEAR	REMARKS
Samalayuca II Power Station **Comisión 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 Recovery 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 Ltd.** North Bay, Ontario, Canada	30	245,000	2.0	53.6	1994	Combined Cycle
Kapuskasing Plant **TransCanada Pipelines Ltd.** 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 (Supplied & Erected)
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

STATION OWNER (A/E)	SIZE MWe (1)	STEAM FLOW [Lb/Hr]	TURBINE Back Pressure [In. HgA]	DESIGN TEMP. [Deg. F]	YEAR	REMARKS
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 (Supplied & Erected)
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 (Supplied & Erected)
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 (Supplied & Erected)
Rumford Power Project ***Rumford Power Associates, Ltd.*** Rumford, ME (Stone & Webster Engineering Corp)	80	545,800	5.0	90	1998	Combined Cycle

STATION OWNER (A/E)	SIZE MWe (1)	STEAM FLOW [Lb/Hr]	TURBINE Back Pressure [In. HgA]	DESIGN TEMP. [Deg. F]	YEAR	REMARKS
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 Plant
Bajio Power Project **InterGen** *Querétaro, Guanajuato, Mexico* (Bechtel International)	150	1,307,000	3.54	71.4	1999	Combined Cycle
University of Alberta *Edmonton, Alberta, Canada* *(Sandwell)*	25	277,780	9.15	59	1999	Gas-Fired Plant Cogeneration
Monterrey Cogeneration Project **Enron Energía Industrial de México** *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	Combined Cycle Cogeneration
Front Range Power Project **Front Range Power Company** *Fountain, Colorado* (TIC/UE Front Range JV)	150	1,266,477	3.57	80	2000	Combined Cycle
Goldendale Energy Project **Goldendale Energy Inc.** *Goldendale, Washington* (NEPCO)	110	678,000	4.5	90	2000	Combined 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 (Supplied & Erected)
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) (Supplied & Erected)
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)

STATION OWNER (A/E)	SIZE MWe (1)	STEAM FLOW [Lb/Hr]	TURBINE Back Pressure [In. HgA]	DESIGN TEMP. [Deg. F]	YEAR	REMARKS
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 (Utility Engineering)	277	1,501,332	3.47	74	2001	Combined Cycle
Spalding Energy Center **InterGen** *Spalding, United Kingdom* (Bechtel Power Corporation)	358	1,998,093	3.12	49	2002	Combined Cycle
Jordan Rehab Power Station **CEGCO** *Amman, Jordan* (Doosan Heavy Industries)	100	833,422	8.94	96.8	2003	Combined Cycle
Currant Creek Project **PacifiCorp** Mona, Utah (Shaw Stone & Webster)	200	1,552,100	6.52	87	2004	Combined Cycle
Snowflake **Pinnacle West** Snowflake, Arizona (REM Engineering)	3	36,000	4.0	95	2004	WTE (Wood-Fired Plant)
32nd Street Naval Station **NORESCO, LLC.** San Diego, California (University Mechanical)	5	84,000	3.0	80	2004	Cogeneration
El Encino **CFE** Chihuahua, Mexico (Dragados)	72	437,165	4.43	98.6	2004	Combined Cycle
Ft Wainwright CHP Plant **U.S. Army** Ft Wainwright, Alaska (Haskell)	3 x 5	3 x 68,500	5.0	82.0	2004	Combined Cycle Cogeneration
Fibrominn Biomass Power Plant **Fibrowatt LLC** Benson, Minnesota (SNC-Lavalin)	55	350,650	3.0	72	2004	WTE

NOTES: (1) Steam side of cycle only

Appendix A.2: SPX Cooling Technologies Reference List

BALCKE HAMON DRY COOLING Marley

POWER PLANTS

AIR COOLED CONDENSERS

INDIRECT COOLING SYSTEMS

INDIRECT COOLING SYSTEMS

Client	Location	Thermal Load MW	Year Install.	Type
ESKOM / Grootvlei 6	South Africa	335	1978	MR, A
ESKOM / Kendal	South Africa	6 x 895	1986/1992	MR, G
VEW / Schmehausen	Germany	438	1977	MR, G
ENEL / Trino Vercellese	Italy	2 x 266.3	1995/1996	MR, A

AIR COOLED CONDENSERS

Client	Location	Steam Load t/h	Year Install.	Type
BBC Mannheim / TOUSS Power Station	Iran	4 x 360	1984	MR, G
ISCOR Vanderbijlpark	South Africa	150	1985	MR, G
ABB Baden Switzerland	Afghanistan	101.5	1987	MR, G
Indeck Energy, Silversprings	USA	55	1990	MR, G
CRS Sirrine, Lowell	USA	73	1991	MR, G
ABB / PPC / Chania	Greece	160	1993	SR, A
Siemens KWU, Offenbach / Rye House Power Station	Great Britain	852	1993	MR, G
ABB Stal / Gas Edon Erica	Netherlands	85	1995	SR, A

Code : SR, A = Single row condenser with aluminium fins
MR, G = Multiple row condenser with galvanised steel fins
MR, A = Multiple row condenser with aluminium fins

See also our references lists for: Waste to Energy, Industry.

SPX Cooling Technologies
REFERENCE LIST

POWER PLANTS

AIR COOLED CONDENSERS
INDIRECT COOLING SYSTEMS

Client	Location	Steam Load t/h	Year Install.	Type
ABB Stal / Gas Edon Klazienaveen	Netherlands	85	1995	SR, A
ABB Stal / Pgem / Borculo	Netherlands	38	1995	SR, A
Billings Generation / Montana	USA	210	1995	MR, G
Edison / San Quirico	Italy	180	1995	SR, A
Mitsubishi / Jandar	Syria	2 x 432	1995	SR, A
Bechtel, Crocket	USA	275	1996	MR, G
Centro Energia / FWI / Comunanza	Italy	195	1996	SR, A
Siemens / East. Elect. / King's Lynn	United Kingdom	330	1996	SR, A
Centro Energia / Fwi / Teverola	Italy	203	1997	SR, A
Electrabel-Spe / TBL / Brugge	Belgium	572	1997	SR, A
Electrabel-Spe / TBL / Gent	Belgium	351	1997	SR, A
Fiat Avio / Coastal Habibullah / Quetta	Pakistan	145	1997	SR, A
Kepco / Halim	Korea	132	1997	SR, A
Mitsubishi Takasago / MHI	Japan	325	1997	SR, A
Anaconda / ABB Power / Murrin Murrin	Australia	98.5	1998	SR, A
Electrabel / Gec Alsthom / Baudour	Belgium	343	1998	SR, A
Sondel / Celano	Italy	185	1998	SR, A

Code : SR, A = Single row condenser with aluminium fins
MR, G = Multiple row condenser with galvanised steel fins
MR, A = Multiple row condenser with aluminium fins

See also our reference list :Waste to Energy, Industry.

POWER PLANTS

AIR COOLED CONDENSERS
INDIRECT COOLING SYSTEMS

Client	Location	Steam Load t/h	Year Install.	Type
Thomassen Power Systems NL / Esenyurt	**Turkey**	**177**	**1998**	**MR, G**
ABB Baden / Enfield	**Great Britain**	**346**	**1999**	**MR, G**
ABB Baden / Monterrey	**Mexico**	**2 x 244,4**	**1999**	**MR, G**
Enron / Stone&Webster / Sutton Bridge	**Great Britain**	**2 x 350**	**1999**	**SR, A**
Kanagawa / Fujisawa No.2 Pst	**Japan**	**75,1**	**1999**	**SR,A**
Kanagawa / Samukawa Pst	**Japan**	**75,6**	**1999**	**SR,A**
Mitsubishi, Japan / Chihuahua	**Mexico**	**450**	**1999**	**MR, G**
ABB Baden, Schweiz / Blackstone, Mass.	**USA**	**2 x 256**	**2000**	**MR, G**
ABB Baden, Schweiz / Hays, TX	**USA**	**2 x 256**	**2000**	**MR, G**
ABB Baden, Schweiz / Lake Road, CT	**USA**	**3 x 256**	**2000**	**MR, G**
ABB Baden, Schweiz / Midlothian, Texas	**USA**	**4 x 255**	**2000**	**MR, G**
Babcock Borsig Power, Oberhausen / Debrecen	**Hungary**	**127**	**2000**	**MR, G**
EDF / Rio Bravo	**Mexico**	**516**	**2000**	**MR, G**
Entergy / Mitsubishi / Damhead Creek	**Great Britain**	**2 x 370**	**2000**	**SR, A**

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SPX Cooling Technologies
REFERENCE LIST

POWER PLANTS

AIR COOLED CONDENSERS
INDIRECT COOLING SYSTEMS

Client	Location	Steam Load t/h	Year Install.	Type
FLS, Denmark / Elean	**Great Britain**	**100**	**2000**	**MR, G**
Zorlu Enerji / Bursa II	**Turkey**	**60**	**2000**	**SR, A**
ABB Alstom, Switzerland / Bellingham, Mass.	**USA**	**2 x 256**	**2001**	**MR, G**
ABB Alstom, UK / Shotton	**Great Britain**	**250**	**2001**	**MR, G**
ABB Baden, Schweiz / Midlothian, Texas	**USA**	**2 x 255**	**2001**	**MR, G**
Babcock Borsig / Al Taweelah	**Abu Dhabi**	**1,000.8**	**2001**	**MR, G**
Bechtel / Hsin Tao	**Taiwan**	**635**	**2001**	**SR, A**
Calpine / Bechtel / Sutter	**USA**	**573**	**2001**	**SR, A**
EDF / Saltillo	**Mexico**	**230**	**2001**	**SR, A**
Edison / Jesi	**Italy**	**200**	**2001**	**SR, A**
Electrabel / Alstom / Esch-s-Alzette	**Luxembourg**	**360**	**2001**	**SR, A**
Hyundai E.C. / Baria	**Vietnam**	**220**	**2001**	**SR, A**
Nevada Power Services, APEX, Nevada	**USA**	**657**	**2001**	**MR, G**
Parsons Energy, Houston / Chehalis, WA	**USA**	**490**	**2001**	**MR, G**

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MR, G = Multiple row condenser with galvanised steel fins
MR, A = Multiple row condenser with aluminium fins

See also our reference list :Waste to Energy, Industry.

POWER PLANTS

AIR COOLED CONDENSERS
INDIRECT COOLING SYSTEMS

Client	Location	Steam Load t/h	Year Install.	Type
Babcock Borsig Power / Tarragona	Spain	247,2	2002	MR, G
KeySpan / Ravenswood, NY	USA	278	2002	MR, G
Sithe / Raytheon / Fore River	USA	1 x 658	2002	SR, A
Sithe / Raytheon / Mystic	USA	2 x 658	2002	SR, A
Toshiba / Sanix	Japan	298,3	2002	SR, A
Utashinai / Hokkaido	Japan	34,8	2002	SR, A
Abener / El Sauz	Mexico	390	2003	MR, G
CWEME / Zhangshan Unit 1 + 2	China	2 x 669,3	2003	MR, G
EDF / Rio Bravo III	Mexico	515	2003	MR, G
Hydro / Sluiskil	Netherlands	117	2003	SR, A
Reliant / Sargent & Lundy / Big Horn	USA	464	2003	SR, A
Siemens Offenbach / Hagit	Israel	350	2003	MR, G
Zorlu Enerji / Bursa III	Turkey	63	2003	SR, A
Abener Energia S.A. / Hermosillo	Mexico	251,3	2004	MR, G

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MR, G = Multiple row condenser with galvanised steel fins
MR, A = Multiple row condenser with aluminium fins

See also our reference list :Waste to Energy, Industry.

SPX Cooling Technologies

REFERENCE LIST

POWER PLANTS

AIR COOLED CONDENSERS
INDIRECT COOLING SYSTEMS

Client	Location	Steam Load t/h	Year Install.	Type
Altek / Kirklarreli	**Turkey**	**97**	**2004**	**SR, A**
Aluminium Bahrain / Alstom / Alba	**Bahrain**	**2 x 525**	**2004**	**SR, A**
EDF / Rio Bravo IV	**Mexico**	**515**	**2004**	**MR, G**
Endesa / Duro Felguera / Son Reus	**Spain**	**262**	**2004**	**SR, A**
Enipower / Snamprogetti / Ferrera	**Italy**	**2 x 360** **1 x 264**	**2004**	**SR, A**
Enipower / Snamprogetti / Mantova	**Italy**	**2 x 360**	**2004**	**SR, A**
Fisia Italimpianti / Accerra	**Italy**	**325**	**2004**	**MR, G**
Genwest LLC / Silverhawk	**USA**	**742**	**2004**	**SR, A**
ICA / Fluor Daniel / La Laguna 2	**Mexico**	**470**	**2004**	**SR, A**
Nypa / GE / Sargent & Lundy / Poletti	**USA**	**490**	**2004**	**SR, A**
Shanxi / Yushe Unit 1 + 2	**China**	**2 x 669,5**	**2004**	**MR, G**
Shanxi Huaze Aluminum / Hejin	**China**	**2 x 671**	**2004**	**SR, A**
Shanxi Pingshuo Meiganshi Power Generation / Shou Zhou	**China**	**2 x 157,5**	**2004**	**MR, G**
Siemens, NL / Mymensingh	**Bangladesh**	**297**	**2004**	**MR, G**

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MR, G = Multiple row condenser with galvanised steel fins
MR, A = Multiple row condenser with aluminium fins

See also our reference list :Waste to Energy, Industry.

SPX Cooling Technologies
REFERENCE LIST

POWER PLANTS

AIR COOLED CONDENSERS
INDIRECT COOLING SYSTEMS

Client	Location	Steam Load t/h	Year Install.	Type
Star Energy / Toshiba / TCIC / Fong Der	**Taiwan**	**2 x 502**	**2004**	**SR, A**
Sun Ba / Toshiba / TCIC / Chang Bin	**Taiwan**	**502**	**2004**	**SR, A**
Transalta / Delta Hudson / Chihuaha III	**Mexico**	**326**	**2004**	**SR, A**
Astoria Energy / Shaw Group / Astoria	**USA**	**460**	**2005**	**SR, A**
CWEME / Guijao Unit 1 + 2	**China**	**2 x 678**	**2005**	**MR, G**
Foster Wheeler / Teverola	**Italy**	**363**	**2005**	**SR, A**
M Project	**Japan**	**107,9**	**2005**	**SR, A**
Mitsubishi / Castelnou	**Spain**	**720**	**2005**	**SR, A**
Shanxi Zhaoguang Electric Power / Huozhou 2	**China**	**2 x 678**	**2005**	**SR, A**
Inner Mongolia / Fengzhen	**China**	**2 x 1317**	**2006**	**SR, A**
Inner Mongolia Shangdu Power Co., Ltd. / Zhenglan	**China**	**2 x 1321**	**2006**	**SR, A**
Shanxi / Datong	**China**	**4 x 164**	**2006**	**MR, G**
Shanxi / Wuxiang	**China**	**2 x 1332**	**2006**	**MR, G**
Shanxi / Xishan	**China**	**3 x 167**	**2006**	**MR, G**

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MR, G = Multiple row condenser with galvanised steel fins
MR, A = Multiple row condenser with aluminium fins

See also our reference list :Waste to Energy, Industry.

SPX Cooling Technologies
REFERENCE LIST

POWER PLANTS

AIR COOLED CONDENSERS
INDIRECT COOLING SYSTEMS

Client	Location	Steam Load t/h	Year Install.	Type
Shanxi / Yanggao	**China**	**2 x 147**	**2006**	**MR, G**
Shanxi / Baode	**China**	**2 x 330**	**2006**	**MR, G**
Hebei Guodian Longshan Power Plant / Longshan	**China**	**2 x 1325**	**2006**	**SR, A**
Inner Mongolia Sangdu Power Co / Zhenglan 2	**China**	**2 x 1321**	**2006**	**SR, A**
CWEME / Dalate	**China**	**2 x 1308**	**2006**	**SR, A**

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MR, G = Multiple row condenser with galvanised steel fins
MR, A = Multiple row condenser with aluminium fins

See also our reference list :Waste to Energy, Industry.

Appendix A.3: Combined ACC Installations as of September 2003

Combined ACC Installations as of September 2003

Vendor	Customer	Project/Site	Country	Steam Load	Steam Load	Condenser Load	Turbine Capacity	perating Pressur	Operating Pressure	Condensing Temp.	Service
				t/h	lb/hr	MWth	MWe	barA	in. Hga	F	Date
Marley/BDT	Babcock Borsig / Al Taweelah, Abu Dhabi		Abu Dhabi	1,000.80	2,201,760	648		2.8	82.67		2001
Marley/BDT	ABB Baden, Schweiz		Afghanistan	101.5	223,300	66		0.12	3.54	121.37	1987
GEA	Blohm & Voss	Buenos Aires	Argentina	2.5	5,500	2	0.5	0.14	4.13	126.84	1963
GEA	KKK	Rosario	Argentina	13	28,600	8		1	29.53		1980
GEA	Pluspetrol Energy S.A.	Tucuman	Argentina	521.6	1,147,520	338	150	0.169	4.99	133.54	1997
Marley/BDT	Oschatz		Argentina	8.5	18,700	6		6	177.16		1971
Marley/BDT	Oschatz, Essen		Argentina	8.4	18,480	5		1.5	44.29		1976
Marley/BDT	Siemens		Argentina	21	46,200	14		5.5	162.39		1980
Marley/BDT	Siemens Österreich		Arnoldstein	27	59,400	17		0.1	2.95	114.83	2003
Marley/BDT	Bechtel, Canada /Kwinana		Australia	36	79,200	23		0.2	5.91	139.85	1969
GEA	Western Mining Corp.		Australia	50	110,000	32		0.3	8.86	156.45	1978
GEA	Elliott	Shell Geelong	Australia	86.2	189,640	56	25.6	0.2	5.91	139.85	1990
GEA	Energy Development Ltd.	Pine Creek	Australia	42.2	92,840	27	10	0.123	3.63	122.25	1994
GEA	InterGen / Shell Coal	Millmerran	Australia	2 x 930	2 x 2,046,000	1,204	2x420	0.184	5.43	136.66	2000
Marley/BDT	Chemserv, Austria / AGRO, Linz		Austria	21.5	47,300	14		0.28	8.27	153.93	1992
Marley/BDT	Chemserv, Austria / Chemie Linz		Austria	15.2	33,440	10		0.115	3.40	119.85	1992
Hamon	Anaconda/ABB Power/Murrin Murrin		Austria	2 x 91.2	2 x 200,600	118	75				1998
GEA	Albatross Refinery	Antwerpen	Belgium	34.5	75,900	22		0.29	8.56	155.28	1967
Marley/BDT	Bayer / Shell		Belgium	2.2	4,840	1		1.25	36.91		1970
GEA	Serete	Progil Antwerpen	Belgium	14.1	31,020	9		5	147.63		1972
Marley/BDT	S.E.I.B., Brüssel		Belgium	90	198,000	58		7	206.68		1976
Marley/BDT	BASF, Antwerpen		Belgium	8.7	19,140	6		0.9	26.57		1976
Marley/BDT	BASF Antwerpen		Belgium	8.8	19,360	6		1.3	38.38		1978
Marley/BDT	BASF, Antwerpen		Belgium	3.7	8,140	2		4	118.10		1979
Marley/BDT	BASF, Antwerpen		Belgium	1.9	4,180	1		1.1	32.48		1980
Marley/BDT	SERT / MVA Harelbeke		Belgium	20	44,000	13		0.05	1.48	90.68	1985
Marley/BDT	FABRICOM / MVA Pont de Loup		Belgium	18.8	41,360	12		0.45	13.29	112.07	1985
GEA	Uhde	BASF Antwerpen	Belgium	45	99,000	29		0.2	5.91	139.85	1989
Marley/BDT	BASF, Antwerpen		Belgium	13.5	29,700	9		1	29.53		1989
Hamon	RMZ/Houthalen		Belgium	24.7	54,400	16	8				1996
Hamon	Electrabel-SPE/TBL/Brugge		Belgium	537.8	1,183,200	348	460				1997
Hamon	Electrabel-SPE/TBL/Gent		Belgium				350				1997
Hamon	Seghers/Indaver		Belgium	86.5	190,400	56	25				1997
Hamon	Electabel/GEC Alsthom/Baudour		Belgium	330.7	727,600	214	350				1998
Hamon	Thumaide/CNIM Ipalle		Belgium	64.9	142,800	42	15				2001
GEA	Mc Lellan	Belco	Bermuda	31	68,200	20		0.2	5.91	139.85	1985
GEA	Mc Lellan	Belco	Bermuda	31	68,200	20		0.2	5.91	139.85	1985
GEA	Brefcon London	Cochabamba	Bolivia	28.5	62,700	18		0.411	12.14	141.44	1976
GEA	Brefcon London	Cochabamba	Bolivia	64	140,800	41		0.221	6.53	143.86	1976
GEA	Brefcon London	Santa Cruz	Bolivia	17.5	38,500	11		0.411	12.14	141.44	1976
GEA	Bophuthatswana Power Corp.	Mmammatsuve	Bophuthatswana	195.5	430,100	127	62	0.25	7.38	149.12	1986
GEA	Botswana Power Corp.	Morupule 1-4	Botswana	4x100.6	4 x 221,320	260	4x33	0.15	4.43	129.28	1986
GEA	AKZ		Brazil	10	22,000	6		0.3	8.86	156.45	1979
GEA	CEEE. Porto Allegre	Candiota	Brazil	764	1,680,800	494	2x160	0.2	5.91	139.85	1982
GEA	Polimex-Cekop Klöckner INA	Socamé	Cameroon	11	24,200	7		1.2	35.43		1974
GEA	Procofrance	Sonara	Cameroon	6.8	14,960	4		0.17	5.02	133.75	1979
GEA	Procofrance	Victoria	Cameroon	6.8	14,960	4		0.165	4.87	132.68	1980
GEA	TransCanada Pipeline Ltd.	Nipigon	Canada	77	169,400	50	15	0.1	2.95	114.83	1990
GEA	TransCanada Pipeline Ltd.	Kapuskasing	Canada	111.1	244,420	72	30	0.067	1.98	100.48	1994
GEA	TransCanada Pipeline Ltd.	North Bay	Canada	111.1	244,420	72	30	0.067	1.98	100.48	1994
GEA	Potter Station Power Partnership	Potter	Canada	82.5	181,500	53	20	0.129	3.81	123.94	1994
GEA	University of Alberta	Edmonton	Canada	126	277,200	82	25	0.31	9.15	157.41	2000
GEA	Montenay	Burnaby	Canada	54	118,800	35	10	0.07	2.07	102.01	2003
Hamon	ABB Kesselanlagen/Kezo/Hinwil		CH	30.9	68,000	20	10				1995
GEA	Las Ventanas		Chile	26	57,200	17	5.4	0.07	2.07	102.01	1963
GEA	Turbinenfabrik J.Nadrowski	Valparaiso	Chile	1.1	2,420	1		0.5	14.76		1975

GEA	Methanex Chile Ltd.	Cabo Negro	Chile	33.8	74,360	22	6	0.135	3.99	125.55	1995
GEA	AEG	Shanghai	China	10.7	23,540	7		0.2	5.91	139.85	1980
GEA	AEG	Shanghai	China	7	15,400	5		0.2	5.91	139.85	1980
GEA	AEG	Shanghai	China	10.3	22,660	7		0.2	5.91	139.85	1980
GEA	AEG	Shanghai	China	7.4	16,280	5		0.2	5.91	139.85	1980
GEA	Borsig	Nanking	China	33	72,600	21		0.15	4.43	129.28	1981
GEA	AEG	Nanking	China	12.3	27,060	8		0.2	5.91	139.85	1981
GEA	AEG	Nanking	China	10.5	23,100	7		0.2	5.91	139.85	1981
GEA	AEG	Nanking	China	28	61,600	18		0.2	5.91	139.85	1981
GEA	AEG	Nanking	China	16.3	35,860	11		0.2	5.91	139.85	1981
Marley/BDT	Zhenhai, China		China	42	92,400	27		0.25	7.38	149.12	2002
GEA	State Power	Datong No. 1	China	465	1,023,000	301	160	0.16	4.72	131.57	2003
Marley/BDT	CWEME / Zhangshan Unit 1 + 2		China	2 x 669.3	2 x 1,472,460	866		0.34	10.04	158.44	2003
Marley/BDT	Alstom, Nürnberg, Delitzsch		Delitzsch	62	136,400	40		0.1	2.95	114.83	2003
GEA	Shell Refinery	Frederica	Denmark	8	17,600	5	1.8	0.2	5.91	139.85	1965
GEA	Shell Refinery	Frederica	Denmark	14.6	32,120	9	2.5	0.115	3.40	119.85	1965
Marley/BDT	Burmeister & Wain		Denmark	14	30,800	9		1.6	47.24		1969
Marley/BDT	Burmeister & Wain		Denmark	14	30,800	9		1.6	47.24		1969
GEA	von Roll	Nyborg	Denmark	22.6	49,720	15		11.5	339.55		1974
GEA	von Roll	Nyborg	Denmark	3.6	7,920	2		11.5	339.55		1974
GEA	AEG	Novopan	Denmark	20	44,000	13		1.1	32.48		1980
GEA	Widmer u. Ernst	Nyborg	Denmark	26.6	58,520	17		3.5	103.34		1981
GEA	Kommunekemi	Nyborg	Denmark	23	50,600	15		0.54	15.94		1986
Hamon	Blohm & Voss/Schwerin		Denmark	72.6	159,800	47	20				1994
Hamon	ABB Turbine/Fichtner/Gera-Nord		Denmark	46.4	102,000	30	75				1995
Hamon	Schworerhaus/Hohenstein		Denmark	12.4	27,200	8	3				1996
Hamon	ABB Turbinen/Frankfurt-Oder		Denmark	46.4	102,000	30	45				1997
Hamon	Linde/BASF Ludwigshafen		Denmark	17.0	37,400	11	8				1997
Hamon	Est-Geko/HKW Meuselwitz		Denmark	17.0	37,400	11	1				1997
Hamon	Lentjes Energietechnik/Wurzburg		Denmark				14				1998
GEA	Petro Industrial	Esmeraldas	Ecuador	56	123,200	36	15	0.152	4.49	129.75	1996
GEA	SOLLAC	Ebange	France	40	88,000	26	10	4.5	132.87		1958
GEA	SOLLAC	Ebange	France	53.5	117,700	35	10	2	59.05		1963
GEA	Esso Refinery	Port Jerome	France	73	160,600	47		0.153	4.52	129.98	1966
GEA	Esso Refinery	Port Jerome	France	50.4	110,880	33		0.153	4.52	129.98	1966
GEA	Esso Refinery	Port Jerome	France	14.9	32,780	10		0.153	4.52	129.98	1966
GEA	Comp. Francaise de Raffinage		France	30.6	67,320	20		0.138	4.07	126.33	1967
GEA	Esso Refinery	Port Jerome	France	11.4	25,080	7		0.204	6.02	140.62	1969
GEA	Tunzini	Toulouse	France	20.5	45,100	13		3.1	91.53		1969
GEA	St. Gobain-Pechiney		France	16	35,200	10		0.18	5.31	135.85	1969
GEA	CFR Haucancourt		France	21.5	47,300	14		0.156	4.61	130.67	1969
GEA	Stone & Webster	Total Chimie	France	150	330,000	97	22.5/18.4	0.204	6.02	140.62	1970
GEA	CRF La Mede		France	31.3	68,860	20		0.17	5.02	133.75	1970
GEA	Linde	Naphta Chemie	France	2x46.4	2 x 102,080	2 x 46.4	2x9.5	0.2	5.91	139.85	1971
GEA	Shell	Petit Couronne	France	31.9	70,180	21	15	0.2	5.91	139.85	1971
GEA	Foster Wheeler	SNPA Gonfreville	France	16.5	36,300	11	1.22	27	797.20		1971
GEA	Technip	Ugine Kuhlmann	France	15	33,000	10		1	29.53		1972
GEA	Tunzini	Toulouse	France	20.5	45,100	13		2	59.05		1973
GEA	Technip Raffinerie des Flandres		France	23.3	51,260	15		0.117	3.45	120.47	1973
GEA	Rhone Progil	Zuid Chemie	France	12	26,400	8		1	29.53		1974
GEA	Stone & Webster	ATO Chimie	France	175	385,000	113		0.2	5.91	139.85	1975
Marley/BDT	Oschatz, Essen		France	76	167,200	49		30	885.78		1977
GEA	Technip Raffinerie des Flandres		France	15.4	33,880	10		0.135	3.99	125.55	1981
GEA	Technip Raffinerie des Flandres		France	35	77,000	23		0.2	5.91	139.85	1982
GEA	Armand Interch. SA	Cabot	France	75.7	166,540	49	20	0.16	4.72	131.57	1987
Marley/BDT	GEC Alstom / DINAN		France	30.6	67,320	20		0.15	4.43	129.28	1997
Marley/BDT	CNIM / Monthyon		France	50	110,000	32		0.145	4.28	128.08	1998
Marley/BDT	ABB Alstom, France / Blois		France	27.7	60,940	18		0.16	4.72	131.57	1999
Marley/BDT	ABB Alstom, France / Maubeuge		France	28	61,600	18		0.13	3.84	124.22	2000
Marley/BDT	Alstom Paris, France / Villers St. Paul		France	51.2	112,640	33		0.1	2.95	114.83	2001
Marley/BDT	Alstom Paris, France / Douchy		France	27.5	60,500	18		0.13	3.84	124.22	2002

Marley/BDT	SMITOM, France / Haguenau		France	19	41,800	12		0.15	4.43	129.28	2002
Marley/BDT	CNIM / France Lasse		France	36.1	79,420	23		0.114	3.37	119.54	2003
Marley/BDT	CT Environment, France / Dunkerque		France	25.1	55,220	16		0.087	2.57	109.79	2003
GEA	Wirus Werke	Gütersloh	Germany	5.5	12,100	4	1	0.075	2.21	104.46	1939
GEA	Dynamit Nobel AG	Friedland	Germany	each 7.7/10	16,940/22,000	5/6.5	3 x 1.5	0.065	1.92	99.42	1940
GEA	Kohlenbergwerke	Marienstein	Germany	5.6	12,320	4	1.5	0.075	2.21	104.46	1950
GEA	Arenberg-Bergbau	Bottrop	Germany	40	88,000	26	6.5	0.075	2.21	104.46	1953
GEA	Geha-Möbelwerke	Hövelhof	Germany	3.5	7,700	2	0.6	0.2	5.91	139.85	1954
GEA	Vereinigte Papierwerke	Heroldsberg	Germany	4	8,800	3	2.5	0.075	2.21	104.46	1955
GEA	Berlin	Gütersloh	Germany	3	6,600	2	0.9	0.09	2.66	111.02	1956
GEA	Hochhaus Le Corbusier	Berlin	Germany	1.8	3,960	1	0.5	0.7	20.67		1957
GEA	Bochumer Verein	Bochum	Germany	3.5	7,700	2		6	177.16		1957
GEA	Pfaff-Werke	Kaiserslautern	Germany	12	26,400	8	3	1.2	35.43		1958
GEA	Technische Hochschule	Karlsruhe	Germany	4.5	9,900	3	1	0.075	2.21	104.46	1958
GEA	Horremer Brikettfabrik	Köln	Germany	20	44,000	13	28	5	147.63		1958
GEA	Vereinigte Glanzstoffwerke	Kelsterbach	Germany	8	17,600	5	5.62	0.07	2.07	102.01	1959
GEA	Daimler Benz AG	Sindelfingen I	Germany	13.5	29,700	9	9.6	0.08	2.36	106.77	1959
GEA	Daimler Benz AG	Sindelfingen I	Germany	18	39,600	12	11	0.08	2.36	106.77	1959
GEA	Piasten Schokoladenfabrik	Forchheim	Germany	6	13,200	4	2.5	0.05	1.48	90.68	1960
GEA	Portlandzementwerke	Gütersloh	Germany	22	48,400	14	5.6	0.08	2.36	106.77	1960
GEA	Kraftwerk Hausham	Hausham	Germany	79.5	174,900	51	40	0.058	1.71	95.53	1960
GEA	Roser-Feuerbach	Stuttgart	Germany	6	13,200	4	3.1	0.09	2.66	111.02	1960
GEA	Volkswagenwerk AG	Wolfsburg	Germany	2x110	2 x 242,000	2 x 110	2x40	0.09	2.66	111.02	1960
GEA	Volkswagenwerk AG	Wolfsburg	Germany	130	286,000	84	48	0.09	2.66	111.02	1960
Marley/BDT	Ingenieurschule Duisburg		Germany	2.8	6,160	2		2.2	64.96		1960
GEA	Ingenieurschule	Darmstadt	Germany	0.35	770	0		0.5	14.76		1961
GEA	Dürrwerke	Ratingen	Germany	0.19	418	0		11	324.79		1961
GEA	Daimler Benz AG	Sindelfingen	Germany	18.29	40,238	12	15	0.08	2.36	106.77	1961
Marley/BDT	DEW / Werhohl		Germany	10	22,000	6		1.1	32.48		1961
GEA	NEAG	Celle	Germany	13	28,600	8	2.8	0.07	2.07	102.01	1962
GEA	Röhm & Haas	Worms	Germany	12	26,400	8	5	0.08	2.36	106.77	1962
Marley/BDT	Kübel, Worms		Germany	8.5	18,700	6		1.5	44.29		1962
GEA	CONDEA	Brunsbüttelkoog	Germany	3.43	7,546	2	0.6	0.08	2.36	106.77	1963
GEA	Ytong Grube Messel	Darmstadt	Germany	10	22,000	6	3.6	0.07	2.07	102.01	1963
GEA	Shell Raffinerie	Ingolstadt	Germany	11.5	25,300	7	2.4	0.2	5.91	139.85	1963
GEA	Preussag AG	KW Ibbenbüren	Germany	304	668,800	197	150	0.042	1.24	85.36	1964
GEA	Union Rheinische Kraftstoff AG	Wesseling	Germany	4.8	10,560	3		0.1	2.95	114.83	1964
GEA	Daimler Benz AG	Bruchsal	Germany	4	8,800	3		1.5	44.29		1965
GEA	Erdölchemie	Dormagen	Germany	75	165,000	49	10.3	0.117	3.45	120.47	1965
GEA	Degussa Chemie	Kalscheuren	Germany	20	44,000	13	7.6	0.08	2.36	106.77	1965
GEA	Volkswagenwerk AG	Kassel	Germany	31.63	69,586	20	3.3	0.105	3.10	116.59	1965
GEA	Volkswagenwerk AG	Kassel	Germany	42.19	92,818	27	3.6	0.088	2.60	110.21	1965
GEA	Badische Anilin- u. Sodafabrik	Ludwigshafen	Germany	40	88,000	26	38	0.055	1.62	93.77	1965
GEA	Union Rheinische Kraftstoff AG	Wesseling	Germany	2x4.8	2 x 10,560	2 x 4.8	2x1.25	0.1	2.95	114.83	1965
GEA	Rheinische Olefin-Werke AG	Wesseling	Germany	44	96,800	28	9	0.065	1.92	99.42	1965
GEA	Volkswagenwerk AG	Wolfsburg	Germany	110	242,000	71	40	0.09	2.66	111.02	1965
GEA	Volkswagenwerk AG	Wolfsburg	Germany	130	286,000	84	48	0.076	2.24	104.93	1965
Marley/BDT	Kübel, Worms		Germany	10.5	23,100	7		1.5	44.29		1965
Marley/BDT	Aicher, Rosenheim		Germany	5	11,000	3		2.5	73.82		1965
GEA	Koppers-Wistra	Bad Godesberg	Germany	10	22,000	6		11	324.79		1966
GEA	Berliner Stadtreinigung	Berlin	Germany	20	44,000	13		2	59.05		1966
GEA	Shell Raffinerie	Godorf	Germany	15.7	34,540	10		0.387	11.43		1966
GEA	Vereinigte Kesselwerke	MVA Hagen	Germany	2x16	2 x 35,200	2 x 16		15	442.89		1966
GEA	C. Freudenberg	Weinheim	Germany	10	22,000	6		1.03	30.41		1966
Marley/BDT	AEG-Kanis / Cabot		Germany	25	55,000	16		0.76	22.44		1966
Marley/BDT	Holzwerke Bähre		Germany	10	22,000	6		1.3	38.38		1966
Marley/BDT	Metallwerke Bähre		Germany	7	15,400	5		1.7	50.19		1966
Marley/BDT	Zeche Friedrich Heinrich		Germany	6.3	13,860	4		16	472.42		1966
GEA	Schulte	Düsseldorf	Germany	7.2	15,840	5		3.5	103.34		1967
GEA	Frisia Raffinerie	Emden	Germany	12	26,400	8		0.5	14.76		1967
GEA	Rheinische Olefin-Werke AG	Godorf	Germany	11.5	25,300	7		0.1	2.95		1967

GEA	Stadtwerke Darmstadt		Germany	30.5	67,100	20		43	1269.62		1967
Marley/BDT	Stadt Iserlohn		Germany	20	44,000	13		2.4	70.86		1967
Marley/BDT	Dillinger Hütte		Germany	3.4	7,480	2		1.5	44.29		1967
Marley/BDT	Saline Ludwigshafen		Germany	0.3	660	0		1	29.53		1967
GEA	Papiermühle	Inden	Germany	6	13,200	4		3	88.58		1968
GEA	Stadtwerke Bremen		Germany	100	220,000	65		21	620.05		1968
GEA	Stadtwerke Bremen		Germany	30	66,000	19		2.3	67.91		1968
Marley/BDT	Holtkamp		Germany	3	6,600	2		1.5	44.29		1968
Marley/BDT	Thyssen AG		Germany	31.4	69,080	20		10	295.26		1968
Marley/BDT	Glanzstoff AG, Köln		Germany	13	28,600	8		1	29.53		1968
Marley/BDT	Holzwerke Osterwald		Germany	5	11,000	3		1.5	44.29		1968
Marley/BDT	Wirus-Werke		Germany	2.1	4,620	1		1	29.53		1968
GEA	Caliqua	Darmstadt	Germany	10	22,000	6		1.2	35.43		1969
GEA	Grüner Bräu	Fürth	Germany	2	4,400	1		0.6	17.72		1969
GEA	Esso	Ingolstadt	Germany	75	165,000	49		0.2	5.91	139.85	1969
GEA	Gesellschaft f. Kernforschung	Karlsruhe	Germany	4	8,800	3		1.1	32.48		1969
GEA	BASF Ludwigshafen	Ludwigshafen	Germany	40.8	89,760	26		0.126	3.72	123.11	1969
GEA	Erdölchemie	Worringen	Germany	2x66.8	2 x 146,960	2 x 66.8		0.1	2.95	114.83	1969
GEA	BASF Ludwigshafen		Germany	35.3	77,660	23		0.126	3.72	123.11	1969
GEA	BASF Ludwigshafen		Germany	34.2	75,240	22		0.126	3.72	123.11	1969
GEA	BASF Ludwigshafen		Germany	10.1	22,220	7		0.126	3.72	123.11	1969
Marley/BDT	Rheinstahl Hattingen		Germany	55.6	122,320	36		15	442.89		1969
Marley/BDT	Thyssen AG, Beeckerswerth		Germany	31.5	69,300	20		10	295.26		1969
Marley/BDT	Thyssen AG		Germany	31.3	68,860	20		20	590.52		1969
Marley/BDT	Stadtwerke Solingen		Germany	17.5	38,500	11		0.15	4.43	129.28	1969
GEA	Marathon	Burghausen	Germany	35	77,000	23		0.34	10.04	158.44	1970
GEA	RMV	Duisburg	Germany	5	11,000	3		11	324.79		1970
GEA	Benze	Einbeckhausen	Germany	4	8,800	3		1.5	44.29		1970
GEA	Piasten Schokoladenfabrik	Forchheim	Germany	16.3	35,860	11		0.15	4.43	129.28	1970
GEA	WMF	Geislingen	Germany	30	66,000	19		3.5	103.34		1970
GEA	Milchzentrale	Karlsruhe	Germany	7	15,400	5		5	147.63		1970
GEA	Daimler Benz AG	Sindelfingen	Germany	34	74,800	22		0.078	2.30	105.86	1970
Marley/BDT	Stadtwerke Kassel		Germany	25.3	55,660	16		3.5	103.34		1970
GEA	CONDEA	Brunsbüttelkoog	Germany	1.95	4,290	1		0.08	2.36	106.77	1971
GEA	BBC	Daimler Benz AG	Germany	30.5	67,100	20	17	0.1	2.95	114.83	1971
GEA	GHH	Rheinstahl Henrichshütte	Germany	19.85	43,670	13	6	0.17	5.02	133.75	1971
GEA	von Roll	Stadtwerke Landshut	Germany	5.1	11,220	3		0.1	2.95	114.83	1971
GEA	Toyo Engineering. Japan		Germany	2x47.43	2 x 104,346	2 x 47.4	2x9.2	0.32	9.45	158.09	1971
GEA	Toyo Engineering. Japan		Germany	2x24	2 x 52,800	2 x 24	2x4.7	0.32	9.45	158.09	1971
GEA	Toyo Engineering. Japan		Germany	2x18.7	2 x 41,140	2 x 18.7	2x3.6	0.32	9.45	158.09	1971
GEA	Toyo Engineering. Japan		Germany	2x13	2 x 28,600	2 x 13	2x2.1	0.32	9.45	158.09	1971
GEA	Toyo Engineering. Japan		Germany	2x77.5	2 x 170,500	2 x 77.5	2x15.5	0.36	10.63	157.03	1971
GEA	Toyo Engineering. Japan		Germany	2x57	2 x 125,400	2 x 57	2x11	0.32	9.45	158.09	1971
GEA	Stadtwerke Frankfurt		Germany	12	26,400	8		0.5	14.76		1971
Marley/BDT	Stadtwerke Oberhausen		Germany	137	301,400	89		15.5	457.65		1971
Marley/BDT	Lurgi / BEB		Germany	20.4	44,880	13		1.5	44.29		1971
Marley/BDT	GHH / Rottka		Germany	20	44,000	13		0.1	2.95	114.83	1971
Marley/BDT	Stadt Hagen		Germany	16	35,200	10		15	442.89		1971
Marley/BDT	KHD, Köln		Germany	3.2	7,040	2		0.4	11.81		1971
GEA	Siemens Werke AG	Brigitta Elverath	Germany	58.9	129,580	38	12	0.31	9.15	157.41	1972
GEA	Siemens Werke AG	Brigitta Elverath	Germany	36.5	80,300	24	8	0.324	9.57	158.28	1972
GEA	Hoechst	Münchsmünster	Germany	20	44,000	13		6	177.16		1972
GEA	Volkswagenwerk AG	Wolfsburg	Germany	110	242,000	71	40	0.09	2.66	111.02	1972
GEA	Volkswagenwerk AG	Wolfsburg	Germany	130	286,000	84	48	0.076	2.24	104.93	1972
GEA	Deutsche Marathon Petr. GmbH		Germany	11.55	25,410	7		0.2	5.91	139.85	1972
Marley/BDT	AEG Kanis / Hamburg		Germany	50	110,000	32		0.14	4.13	126.84	1972
Marley/BDT	VKW / MVA Iserlohn		Germany	29	63,800	19		2.7	79.72		1972
Marley/BDT	Kübel, Worms		Germany	12	26,400	8		1.5	44.29		1972
Marley/BDT	Du Pont, Uentrop		Germany	2.9	6,380	2		1	29.53		1972
GEA	Caliqua	Industriewerke Hamberger	Germany	5.28	11,616	3		2.5	73.82		1973
GEA	Terry GmbH	Oberhausen	Germany	2.6	5,720	2		1	29.53		1973
GEA	Gebr. Aicher. Holzindustrie	Rosenheim	Germany	6.5	14,300	4		2	59.05		1973

GEA	Stadtwerke Landshut		Germany	2.7	5,940	2		0.1	2.95	114.83	1973
Marley/BDT	Lurgi / BEB		Germany	20.8	45,760	13		1.5	44.29		1973
Marley/BDT	Grefrath Velour		Germany	5.5	12,100	4		5	147.63		1973
GEA	HKG Hamm Uentrop	KKW Schmehausen	Germany	710	1,562,000	459	330	0.08	2.36	106.77	1974
GEA	Degussa Chemie	Werk Kalscheuren	Germany	20	44,000	13		0.08	2.36	106.77	1974
Marley/BDT	Babcock / Krupp		Germany	98	215,600	63		18.5	546.23		1974
Marley/BDT	VKW / MVA Göppingen		Germany	42	92,400	27		0.15	4.43	129.28	1974
Marley/BDT	VKW / MVA Kiel		Germany	13.9	30,580	9		15	442.89		1974
Marley/BDT	Du Pont, Uentrop		Germany	3.3	7,260	2		1	29.53		1974
Marley/BDT	Degussa		Germany	2.8	6,160	2		2.5	73.82		1974
GEA	Krantz Wärmetechnik	Fürstenfeldbruck	Germany	20	44,000	13		8	236.21		1975
GEA	Matth. Hohner AG	Trossingen	Germany	5	11,000	3		1.3	38.38		1975
GEA	Vereinigte Kesselwerke	Wuppertal	Germany	96.1	211,420	62		0.12	3.54	121.37	1975
GEA	Refuse incinerationsanlage	Wuppertal	Germany	96.1	211,420	62		0.12	3.54	121.37	1975
GEA	AEG Kanis		Germany	6	13,200	4		0.3	8.86	156.45	1975
GEA	Glasfabrik Heye		Germany	2.55	5,610	2		0.07	2.07	102.01	1975
Marley/BDT	Stadt Bremerhaven		Germany	80	176,000	52		0.47	13.88		1975
Marley/BDT	Borsig / Ruhrgas		Germany	54	118,800	35		0.22	6.50	143.67	1975
Marley/BDT	PWA Stockstadt		Germany	50	110,000	32		3.5	103.34		1975
Marley/BDT	Preussag		Germany	10.8	23,760	7		7	206.68		1975
GEA	Siemens Werke AG	Brigitta Elverath	Germany	58.9	129,580	38	12	0.31	9.15	157.41	1976
GEA	Thyssen Rheinstahl Technik	Wupperverband	Germany	9.9	21,780	6		1.5	44.29		1976
Marley/BDT	Widmer und Ernst / MVA Hamburg		Germany	81	178,200	52		0.12	3.54	121.37	1976
Marley/BDT	Babcock /Mannesmann		Germany	60	132,000	39		46	1358.20		1976
Marley/BDT	Stadtwerke Frankfurt		Germany	24	52,800	16		0.5	14.76		1976
Marley/BDT	Lurgi / BEB		Germany	20.5	45,100	13		5	147.63		1976
Marley/BDT	Widmer und Ernst / MVA Fürth		Germany	16.5	36,300	11		5	147.63		1976
Marley/BDT	Borsig / Ruhrgas		Germany	15.2	33,440	10		3	88.58		1976
Marley/BDT	Kübel, Worms		Germany	15	33,000	10		1.5	44.29		1976
Marley/BDT	SSK von Schaewen		Germany	8.1	17,820	5		1	29.53		1976
Marley/BDT	VKW / Behring Marburg		Germany	3.6	7,920	2		11	324.79		1976
Marley/BDT	Du Pont, Uentrop		Germany	2	4,400	1		1	29.53		1976
GEA	Linde AG	EC-Worringen	Germany	2x74.5	2 X 163,900	96		0.1	2.95	114.83	1977
GEA	Wester GmbH	MVA Bamberg	Germany	32	70,400	21		7	206.68		1977
GEA	Südhessische Gas-u.Wasser AG	MVA Darmstadt	Germany	35	77,000	23		7	206.68		1977
GEA	Saarberg Fernwärme GmbH	MVA Neunkirchen	Germany	30	66,000	19		0.5	14.76		1977
GEA	Mobil Oil	Neag	Germany	1	2,200	1		1.05	31.00		1977
Marley/BDT	VKW / MVA Krefeld		Germany	54	118,800	35		3.7	109.25		1977
Marley/BDT	AKZO, Köln		Germany	8.1	17,820	5		1.7	50.19		1977
GEA	Omnical	Gießen	Germany	5	11,000	3		0.25	7.38	149.12	1978
GEA	Thyssen	Hattingen	Germany	5.5	12,100	4		0.17	5.02	133.75	1978
Marley/BDT	Cabot, Hanau		Germany	13.6	29,920	9		0.2	5.91	139.85	1979
Marley/BDT	Siemens / Globus		Germany	11.1	24,420	7		3.9	115.15		1979
Marley/BDT	Dr. Pauli		Germany	6	13,200	4		1.2	35.43		1979
Marley/BDT	Schmidt'sche Heißdampf, Kassel		Germany	6	13,200	4		1.8	53.15		1979
GEA	HAVG	Biebesheim	Germany	40.6	89,320	26		2.6	76.77		1980
GEA	MVA Bielefeld	Bielefeld	Germany	135	297,000	87		0.2	5.91	139.85	1980
GEA	Schlotterer	Bodelshausen	Germany	10	22,000	6		6.5	191.92		1980
GEA	Widmer u. Ernst	Herten	Germany	70	154,000	45		0.2	5.91	139.85	1980
GEA	Stadtwerke Landshut	Landshut	Germany	12.7	27,940	8		0.29	8.56	155.28	1980
GEA	Stadtwerke Pforzheim	Pforzheim	Germany	60	132,000	39		0.7	20.67		1980
GEA	AEG	Seitz Filter	Germany	6	13,200	4		6.5	191.92		1980
Marley/BDT	GHH / Henrichshütte, Hattingen		Germany	32.5	71,500	21		0.18	5.31	135.85	1980
Marley/BDT	Klingele		Germany	24.4	53,680	16		3	88.58		1980
Marley/BDT	Goepfert & Reimer / v		Germany	5	11,000	3		0.5	14.76		1980
GEA	Stadtwerke Bremen	Bremen	Germany	50	110,000	32		22	649.57		1981
GEA	Raschka für Kläranlage	Karlsruhe	Germany	3.7	8,140	2		23	679.10		1981
Marley/BDT	BBC Berlin / MVA Krefeld		Germany	59.3	130,460	38		0.18	5.31	135.85	1981
Marley/BDT	Siemens		Germany	20	44,000	13		3.5	103.34		1981
Marley/BDT	VEBA-Oel		Germany	0.63	1,386	0		1.5	44.29		1981
GEA	MVA Bonn	Bad Godesberg	Germany	12	26,400	8		1.7	50.19		1982

Marley/BDT	Widmer & Ernst / MVA Ingolstadt		Germany	26.3	57,860	17		0.12	3.54	121.37	1982
Marley/BDT	Technische Werke Ludwigshafen		Germany	18	39,600	12		0.1	2.95	114.83	1982
Marley/BDT	Babcock Krauss Maffei Imperial / MPA Burgau		Germany	12	26,400	8		0.2	5.91	139.85	1982
Marley/BDT	Standard Messo / MVA Stapelfeld		Germany	8	17,600	5		0.09	2.66	111.02	1982
GEA	Omnical	Ewersbach	Germany	4	8,800	3		1.5	44.29		1983
GEA	Volz Holzwerke	Friedenweiler	Germany	3	6,600	2		1.5	44.29		1983
GEA	von Roll	Kempten	Germany	54	118,800	35		2	59.05		1983
GEA	SW Würzburg		Germany	51	112,200	33		0.12	3.54	121.37	1983
GEA	Megal	Waidhaus	Germany	50.5	111,100	33		0.075	2.21	104.46	1984
Marley/BDT	BBC Mannheim / MVA Geiselbullach		Germany	33	72,600	21		0.13	3.84	124.22	1984
Marley/BDT	BBC Mannheim / MVA Neustadt		Germany	26	57,200	17		0.12	3.54	121.37	1984
GEA	Zweckverband Hamm		Germany	72.4	159,280	47	25	0.18	5.31	135.85	1985
Marley/BDT	Deutsche Babcock Anlagen AG / MVA Leverkusen		Germany	70	154,000	45		2	59.05		1985
Marley/BDT	Stadtw. Frankfurt / MVA Frankfurt		Germany	25	55,000	16		0.5	14.76	40.74	1985
GEA	SW Marktoberdorf		Germany	12	26,400	8		17	501.94		1986
GEA	SVA Schwabach		Germany	21.4	47,080	14	5	1.5	44.29		1987
GEA	SVA Schöneiche		Germany	15.8	34,760	10	3.8	1.8	53.15		1987
Marley/BDT	Blohm u. Voss / MVA Pinneberg		Germany	31	68,200	20		0.2	5.91	139.85	1987
Marley/BDT	BASF, Münster		Germany	12.6	27,720	8		11	324.79		1987
GEA	von Roll	MVA Darmstadt	Germany	46	101,200	30		0.3	8.86	156.45	1988
GEA	Steinmüller	RZR Herten IM-II	Germany	23.5	51,700	15		0.19	5.61	137.87	1988
GEA	Bremer Wollkämmerei		Germany	30	66,000	19	18.3	0.21	6.20	141.78	1988
Marley/BDT	Märkischer Kreis / AMK Iserlohn		Germany	53	116,600	34		2.7	79.72		1988
Marley/BDT	Blohm und Voss / MVA Beselich		Germany	6.2	13,640	4		0.16	4.72	131.57	1988
Marley/BDT	Bayer, Uerdingen		Germany	4.5	9,900	3		1.05	31.00		1988
GEA	Heye Glas	Germersheim	Germany	7.4	16,280	5		0.1	2.95	114.83	1989
GEA	Stadtwerke Landshut	MVA Landshut	Germany	13	28,600	8		0.12	3.54	121.37	1989
GEA	MAN GHH	Rostock	Germany	2x22	2 X 48,400	28		0.137	4.05	126.07	1989
GEA	MAN GHH	Rostock	Germany	2x16	2 X 35,200	21		3.5	103.34		1989
GEA	MAB Lentjes	RZR Herten SM II	Germany	47	103,400	30		0.19	5.61	137.87	1989
Marley/BDT	MBA Bremerhaven		Germany	42	92,400	27		0.47	13.88	88.81	1989
Marley/BDT	Schlotterer		Germany	12	26,400	8		2.3	67.91		1989
Marley/BDT	Rütgerswerke		Germany	4	8,800	3		0.17	5.02	133.75	1989
Marley/BDT	Chemische Fabrik Budenheim		Germany	2.7	5,940	2		0.06	1.77	96.68	1989
Marley/BDT	Oberrheinische Mineralölwerke		Germany	0.42	924	0		2	59.05		1989
GEA	ABB	Brilon	Germany	15.1	33,220	10		0.2	5.91	139.85	1990
GEA	Esso AG	Ingolstadt	Germany	75	165,000	49		0.2	5.91	139.85	1990
GEA	Reining Heisskühlung Hoesch		Germany	2.1	4,620	1	1	1.1	32.48		1990
GEA	Rohrbach Zementwerk		Germany	28	61,600	18		0.123	3.63	122.25	1990
Marley/BDT	ABB / BASF		Germany	52.2	114,840	34		0.2	5.91	139.85	1990
Marley/BDT	Siemens / MHKW Weissenhorn		Germany	38	83,600	25		0.15	4.43	129.28	1990
Marley/BDT	Pfleiderer		Germany	20	44,000	13		6	177.16		1990
Marley/BDT	Gnettner		Germany	6	13,200	4		1.8	53.15		1990
Marley/BDT	Spillingwerk		Germany	3	6,600	2		1.4	41.34		1990
GEA	BASF	BASF KW Nord	Germany	152.5	335,500	99	50	0.185	5.46	136.87	1991
GEA	Heye Glas	Germersheim	Germany	17	37,400	11	11	0.1	2.95	114.83	1991
Marley/BDT	Siemens / MVA Schwandorf		Germany	120	264,000	78		0.12	3.54	121.37	1991
Marley/BDT	Turbon Tunzini / Sophia Jacoba		Germany	13.2	29,040	9		3.5	103.34		1991
Marley/BDT	Blohm und Voss / Hornschuh		Germany	8	17,600	5		5	147.63		1991
Marley/BDT	Rauschert		Germany	0.56	1,232	0		0.3	8.86	156.45	1991
GEA	STEAG AG	Kaiserstuhl	Germany	76.7	168,740	50	47	0.15	4.43	129.28	1992
Marley/BDT	ABB Nürnberg / AVA Augsburg		Germany	100	220,000	65		8.2	242.11		1992
Marley/BDT	Zell- u. Papierfabrik Rosenthal		Germany	27.3	60,060	18		3.5	103.34		1992
Marley/BDT	B S R, Berlin		Germany	21	46,200	14		29	856.25		1992
Marley/BDT	ABB Nürnberg / AVA Augsburg		Germany	9	19,800	6		2.7	79.72		1992
GEA	ABB/Egger GmbH	Brilon II	Germany	39	85,800	25	12.5	0.25	7.38	149.12	1993
GEA	Siemens. Erlangen	Burgkirchen	Germany	27	59,400	17	12.7	0.117	3.45	120.47	1993
GEA	Siemens	Kempten	Germany	26	57,200	17		0.15	4.43	129.28	1993
Marley/BDT	ABB Nürnberg / AVA Augsburg		Germany	56	123,200	36		0.12	3.54	121.37	1993

Marley/BDT	MAN GHH / GSB Ebenhausen		Germany	32	70,400	21		0.21	6.20	141.78	1993
Marley/BDT	Krupp Stahl, Bochum		Germany	16.5	36,300	11		1.3	38.38		1993
Marley/BDT	Rauma, Düsseldorf / BASF Ludwigs.		Germany	2.92	6,424	2		0.19	5.61	137.87	1993
Hamon	ABB/PPC/Chania		Germany	154.5	340,000	100	130				1993
GEA	ABB	Ingolstadt	Germany	72	158,400	47		0.18	5.31	135.85	1994
GEA	AMK	Iserlohn	Germany	51	112,200	33		0.11	3.25	118.26	1994
GEA	ABB	Köln	Germany	161	354,200	104	56	0.116	3.43	120.16	1994
GEA	ABB	Velsen	Germany	71	156,200	46		0.143	4.22	127.59	1994
Marley/BDT	ESP Heizwerke GmbH/ Sulzbach-Rosenberg		Germany	18.8	41,360	12		0.2	5.91	139.85	1994
GEA	Hornitex	Beeskow	Germany	25	55,000	16		0.1	2.95	114.83	1995
GEA	ESP GEKO GmbH	Hagenow	Germany	21	46,200	14		0.12	3.54	121.37	1995
GEA	ZSVM	Schwabach	Germany	3	6,600	2		0.18	5.31	135.85	1995
GEA	ABB	Ulm	Germany	41	90,200	27	15	0.143	4.22	127.59	1995
Marley/BDT	ML Ratingen / MVA Offenbach *		Germany	34	74,800	22		0.12	3.54	121.37	1995
Marley/BDT	Blohm + Voss / SAVA Brunsbüttel *		Germany	14	30,800	9		0.12	3.54	121.37	1995
GEA	Rettenmeier	Wilburgstetten	Germany	24	52,800	16		0.2	5.91	139.85	1996
Marley/BDT	Siemens KWU / AEZ Kreis Wesel		Germany	75	165,000	49		0.1	2.95	114.83	1996
Marley/BDT	EAG Krefeld		Germany	75	165,000	49		0.18	5.31	135.85	1996
Marley/BDT	Siemens KWU / SBA Fürth *		Germany	47.2	103,840	31		0.138	4.07	126.33	1996
Marley/BDT	Stadtwerke Kiel / MVA Kiel		Germany	20.5	45,100	13		3.5	103.34		1996
Marley/BDT	Deutsche Babcock Anlagen, Oberhausen / VERA, Hamburg *		Germany	15	33,000	10		0.2	5.91	139.85	1996
GEA	ABB	Böblingen	Germany	43	94,600	28	12	0.175	5.17	134.81	1997
Marley/BDT	ML Ratingen / MHKW Pirmasens *		Germany	65.1	143,220	42		0.11	3.25	118.26	1997
Marley/BDT	ESP GEKO / HKW Dresden		Germany	29	63,800	19		1.2	35.43		1997
Marley/BDT	ESP GEKO / HKW Feldberg		Germany	20	44,000	13		0.2	5.91	139.85	1997
Marley/BDT	Addinol, Osterrode / KRUMPA		Germany	9.6	21,120	6		0.1	2.95	114.83	1998
GEA	Rettenmeier	Ullersreuth/Gaildorf	Germany	2 x 24	2 X 52,800	31	2 x 10	0.2	5.91	139.85	1999
Marley/BDT	Babcock Kraftwerks-technik, Berlin / Brand-Erbisdorf		Germany	33	72,600	21		0.19	5.61	137.87	1999
Marley/BDT	LEG Lurgi Entsorgung / Neunkircher		Germany	19.6	43,120	13		0.096	2.83	113.35	1999
Marley/BDT	ETT Bünde / Günzburg		Germany	2.2	4,840	1		1.1	32.48		1999
GEA	Horn	Hornitex	Germany	50	110,000	32	24.4	0.1	2.95	114.83	2000
Marley/BDT	ANO Bremen		Germany	40	88,000	26		2.3	67.91		2000
Marley/BDT	HKW Glückstadt		Germany	40	88,000	26		5	147.63		2000
Marley/BDT	MVA Stapelfeld		Germany	20	44,000	13		0.3	8.86	156.45	2000
Marley/BDT	EST-EABG / Steinbach		Germany	6.5	14,300	4		1.5	44.29		2000
Marley/BDT	Babcock SIK / Eisenberg		Germany	24	52,800	16		0.25	7.38	149.12	2001
Marley/BDT	AE Energietechnik, Wien, Austria / T A. Lauta		Germany	68.73	151,206	44		100	2952.60		2003
Marley/BDT	Alstom, Nürnberg / Zolling		Germany	61.5	135,300	40		0.1	2.95	114.83	2003
Marley/BDT	Alstom, Nürnberg / Landesbergen		Germany	61.5	135,300	40		0.1	2.95	114.83	2003
Marley/BDT	Kraftanlagen München / MHKW Mainz		Germany	17	37,400	11		0.15	4.43	129.28	2003
GEA	Rhone Poulenc	Petrochemie f. SICNG	Greece	16	35,200	10		0.2	5.91	139.85	1976
Marley/BDT	Babcock Borsig Power, Oberhausen / Debrecen		Hungary	127	279,400	82		0.25	7.38	149.12	2000
GEA	MSEB/Siemens	Uran	India	2x508	2 X 1,117,600	657	2x120	0.28	8.27	153.93	1992
GEA	Hitech Carbon	Hitech Carbon	India	30	66,000	19	6	0.226	6.67	144.79	1997
Marley/BDT	EID Parry, India / Chennai		India	16	35,200	10		2.8	82.67		2001
GEA	Lurgi for Pertamina		Indonesia	28.3	62,260	18		0.2	5.91	139.85	1983
Marley/BDT	Blohm+Voss, Hamburg, Batam, Indonesien		Indonesia	26.1	57,420	17		0.45	13.29	112.07	1993
GEA	Stork Boilers	Sugar Mill	Iran	60	132,000	39		1	29.53		1976
Marley/BDT	BBC Mannheim / TOUSS Power Station		Iran	4 x 360	4 x 792,000	932		0.27	7.97	152.43	1983
GEA	AEG Kanis	NPC Shiraz	Iran	9.1	20,020	6	3	9.2	271.64		1987

GEA	Tavanir/Siemens	Gilan	Iran	3x550	3x1,210,000	1,068	1170	0.2	5.91	139.85	1991
GEA	Skodaexport/1.Brünner Maschine	Iranshahr	Iran	4x180	4 x 396,000	466	4x64	0.237	7.00	146.81	1993
GEA	ABB	Ghom	Iran	404	888,800	261	2x100	0.245	7.23	148.24	1994
GEA	Siemens	Huntstown	Ireland	330	726,000	214	120	0.09	2.66	111.02	2001
GEA	Trepel		Israel	18.6	40,920	12		2.52	74.41		1969
GEA	ABB	Ramat Hovav	Israel	382	840,400	247	110	0.18	5.31	135.85	1996
GEA	Alstom	Hagit	Israel	2 x 382	2 x 840,400	494	2 x 110	0.18	5.31	135.85	2001
Marley/BDT	Siemens, Offenbach, Hagit		Israel	350	770,000	226		0.165	4.87	132.68	2003
GEA	S.M.T.	Trasimeno	Italy	2x68	2 x 149,600	2 x 44	2x29	0.06	1.77	96.68	1956
GEA	S.M.T.	Trasimeno	Italy	86	189,200	56	36	0.06	1.77	96.68	1956
GEA	Prada		Italy	8.5	18,700	6	1.4	0.1	2.95	114.83	1968
GEA	Philips Carbon Black		Italy	17.2	37,840	11		0.35	10.33	157.99	1969
GEA	Philips Carbon Black		Italy	7.9	17,380	5		0.35	10.33	157.99	1969
GEA	Cementerie Calabro	Lucane	Italy	90	198,000	58	30	0.08	2.36	106.77	1971
GEA	Pirelli		Italy	2.5	5,500	2	699	0.069	2.04	101.51	1971
GEA	De Nora		Italy	18.2	40,040	12		0.16	4.72	131.57	1971
GEA	von Roll	Bologna	Italy	2x21	2 x 46,200	27		19	560.99		1972
GEA	von Roll	Livorno	Italy	2x11.4	2 x 25,080	15		17	501.94		1972
GEA	von Roll	Bologna	Italy	25	55,000	16		19	560.99		1973
GEA	Cementerie Calabro	Lucane	Italy	90	198,000	58	30	0.08	2.36	106.77	1975
GEA	Columbian Carbon		Italy	8.3	18,260	5		1.04	30.71		1976
GEA	AMOCO	Cremona	Italy	18	39,600	12		0.2	5.91	139.85	1981
GEA	Industria Petroli Chemi	Rho	Italy	17.1	37,620	11		0.13	3.84	124.22	1982
GEA	Foster Wheeler	Porcari	Italy	90	198,000	58	30	0.18	5.31	135.85	1994
Hamon	Edison/San Quirico		Italy	173.1	380,800	112	130				1995
Hamon	Centro Energia/FWI/Comunanza		Italy	188.5	414,800	122	150				1996
Hamon	Centro Energia/FWI/Teverola		Italy	188.5	414,800	122	150				1997
Hamon	Sondel/Celano		Italy	177.7	391,000	115	132				1998
Hamon	Edison/Jesi		Italy	177.7	391,000	115	130				2001
Marley/BDT	Fisia Italimpianti / Italy for Accerra		Italy	325	715,000	210		0.12	3.54	121.37	2003
Hamon	Enipower/Snamprogetti/Ferrera		Italy				1010				2004
Hamon	NSC/Iisuka City		Japan	9.3	20,400	6	1				1996
Hamon	Mitsubishi/Takasago		Japan	324.5	714,000	210	350				1997
GEA	Jordan Electric Authority	Breda/Mailand	Jordan	2x109	2 x 239,800	141	2x33	0.27	7.97	152.43	1976
GEA	Jordan Electric Authority	Breda/Mailand	Jordan	113.6	249,920	74	33	0.276	8.15	153.34	1977
GEA	AEG Kanis	Zarqa Refinery	Jordan	68.18	149,996	44	18	0.218	6.44	143.30	1978
GEA	Fuji Electric	Zarqa	Jordan	2x214	2 x 470,800	277	2x66	0.276	8.15	153.34	1979
GEA	Jordan Electric Authority	Hussein 7	Jordan	211.5	465,300	137	56	0.28	8.27	153.93	1984
GEA	Kellog London		Lebanon	19.5	42,900	13		0.207	6.11	141.20	1970
GEA	Ewbank and Partners		Lebanon	39	85,800	25	8	0.242	7.15	147.71	1971
Marley/BDT	Hildebrand		Liberia	12	26,400	8		2.2	64.96		1975
Marley/BDT	UHDE, Dortmund		Libya	21.9	48,180	14		3.5	103.34		1975
Marley/BDT	UHDE, Dortmund		Libya	21.9	48,180	14		3.5	103.34		1975
GEA	ARBED	Dudelange	Luxembourg	50	110,000	32	13	0.075	2.21	104.46	1955
Hamon	Electrabel/Alstom/Esch-S-Alzette		Luxembourg	340.0	748,000	220	350				2001
GEA	Tamatave Refinery	Tamatave	Madagascar	6.3	13,860	4		0.85	25.10		1965
GEA	JGC Corp./Shell	Bintulu	Malaysia	309.7	681,340	200	100	0.4	11.81	146.63	1990
GEA	JGC Corp./Shell	Bintulu	Malaysia	134.9	296,780	87	60	0.4	11.81	146.63	1990
GEA	JGC Corp./Shell	Bintulu	Malaysia	119.3	262,460	77	40	0.4	11.81	146.63	1990
Marley/BDT	Petronas, Malaysia / Kuantan		Malaysia	54	118,800	35		0.256	7.56	150.15	1999
Marley/BDT	Petronas, Malaysia / Kuantan		Malaysia	49	107,800	32		0.256	7.56	150.15	1999
GEA	Tenaga Nasional Berhad	Gelugor	Malaysia	429.4	944,680	278	120	0.23	6.79	145.53	2002
GEA	Comisión Federal de Electricidad	Samalayuca II	Mexico	588.3	1,294,260	381	210	0.237	7.00	146.81	1995
Marley/BDT	ABB Baden, Schweiz / Monterrey		Mexico	2 x 244.4	2 x 537,680	316		0.16	4.72	131.57	1999
Marley/BDT	Mitsubishi, Japan / Chihuahua		Mexico	450	990,000	291		0.093	2.75	112.20	1999
GEA	InterGen	Bajio	Mexico	593	1,304,600	384	150	0.12	3.54	121.37	2000
Marley/BDT	EdF, France / Rio Bravo		Mexico	516	1,135,200	334		0.12	3.54	121.37	2000
Hamon	EDF/Saltillo		Mexico	227.2	499,800	147	220				2001
GEA	Enron Energia / Tractebel	Monterrey	Mexico	304.8	670,560	197	80	0.2	5.91	139.85	2002
Marley/BDT	EdF, France / Rio Bravo III		Mexico	515	1,133,000	333		0.126	3.72	123.11	2003
Marley/BDT	EdF, France / Rio Bravo IV		Mexico	515	1,133,000	333		0.126	3.72	123.11	2003
Marley/BDT	Abener, Spain for El Sauz		Mexico	390	858,000	252		0.08	2.36	106.77	2003

GEA	SWAWEK	Windhoek	Namibia	2x94.8	2 x 208,560	123	2x30	0.138	4.07	126.33	1971
GEA	SWAWEK	Windhoek	Namibia	94.8	208,560	61	30	0.138	4.07	126.33	1972
GEA	SWAWEK	Windhoek	Namibia	95	209,000	61	30	0.2	5.91	139.85	1978
GEA	Esso Rotterdam		Netherlands	20	44,000	13		0.17	5.02	133.75	1968
GEA	Neratoom		Netherlands	118.2	260,040	76		40	1181.04		1969
GEA	Neratoom		Netherlands	44.4	97,680	29		15	442.89		1969
Marley/BDT	Didier		Netherlands	2.1	4,620	1		0.35	10.33	157.99	1977
GEA	AEG Kanis	Nijmwegen	Netherlands	30	66,000	19		3.5	103.34		1978
Marley/BDT	Stork Boilers		Netherlands	41	90,200	27		0.1	2.95	114.83	1981
GEA	Ijssel Centrale	Enschede	Netherlands	72	158,400	47	18	0.065	1.92	99.42	1985
GEA	DSM Geleen		Netherlands	2x67.6	2 x 148,720	87		0.1	2.95	114.83	1985
GEA	DSM Geleen		Netherlands	17.9	39,380	12		0.1	2.95	114.83	1986
Marley/BDT	KEMA / Engergiebedrijf Leiden		Netherlands	31.8	69,960	21		1.1	32.48		1986
GEA	DSM Geleen		Netherlands	19.9	43,780	13		0.1	2.95	114.83	1987
GEA	Royal Schelde	Alkmaar	Netherlands	142	312,400	92		0.08	2.36	106.77	1993
GEA	Stork	Eerbeek	Netherlands	40	88,000	26	18	0.1	2.95	114.83	1993
GEA	ABB Stal	Helmond	Netherlands	45	99,000	29	28	0.1	2.95	114.83	1993
GEA	Esso. Rotterdam	Hydrocracker	Netherlands	12	26,400	8		0.17	5.02	133.75	1993
GEA	ABB Stal	s'Hertogenbosch	Netherlands	45	99,000	29	28	0.1	2.95	114.83	1993
GEA	ABB Stal	Bergen op Zoom	Netherlands	42	92,400	27		0.13	3.84	124.22	1994
GEA	Royal Schelde	Eindhoven	Netherlands	58	127,600	38		0.15	4.43	129.28	1994
Marley/BDT	Stork Ketels / Wapenveld		Netherlands	46.8	102,960	30		0.1	2.95	114.83	1995
Hamon	ABB Stal/Gas Edon Erica		Netherlands	80.4	176,800	52	70				1995
Hamon	ABB Stal/Gas Edon Klazienaveen		Netherlands	80.4	176,800	52	70				1995
Hamon	ABB Stal/PGEM/Borculo		Netherlands	37.1	81,600	24	30				1995
GEA	Royal Schelde	AVIRA/Arnheim	Netherlands	24	52,800	16		0.08	2.36	106.77	1996
Marley/BDT	AVI Twente, Hengelo / Twente		Netherlands	88.2	194,040	57		0.085	2.51	108.95	1996
Hamon	ML-Gavi Wijster		Netherlands	154.5	340,000	100	54				1996
Marley/BDT	Siemens KWU / Cuijk *		Netherlands	77.4	170,280	50		0.1	2.95	114.83	1999
GEA	DSM	NAK 4	Netherlands	2 x 76	2 x 167,200	98	2 x 25	0.133	3.93	125.03	2000
Marley/BDT	Siemens Österreich		Niklasdorf	42	92,400	27		0.5	14.76	40.74	2003
GEA	Maschinenfabrik Eßlingen	Oslo	Norway	2x15	2 x 33,000	2 x 15		6	177.16		1966
GEA	Siemens	NRL Pakistan	Pakistan	23	50,600	15		0.2	5.91	139.85	1994
Hamon	Fiat Avio/Coastal Habibulla/Quetta		Pakistan	136.0	299,200	88	130				1997
Marley/BDT	Krupp, Essen		Poland	20	44,000	13		0.8	23.62		1974
Marley/BDT	Zimmer, Frankfurt		Poland	12	26,400	8		1.2	35.43		1981
GEA	Petrochimica Refinery		Portugal	26.5	58,300	17		0.17	5.02	133.75	1967
GEA	Petrochimica Refinery		Portugal	26.5	58,300	17		0.17	5.02	133.75	1967
GEA	Petrochimica Refinery		Portugal	7.5	16,500	5		0.17	5.02	133.75	1967
GEA	Linde AG	Sines	Portugal	140	308,000	91		0.15	4.43	129.28	1978
GEA	Uhde	Tosi	Portugal	12.1	26,620	8		0.12	3.54	121.37	1980
GEA	Lurgi / Quimigal	Lavrados	Portugal	21.9	48,180	14		0.123	3.63	122.25	1981
GEA	Lurgi / Quimigal	Lavrados	Portugal	38.7	85,140	25		0.123	3.63	122.25	1981
GEA	SCECO	Riyadh #9	Saudi Arabia	4x438.5	4 x 964,700	1,135	4x107	0.56	16.53		1996
GEA	Heilborn GmbH	Sierra Leone	Sierra Leone	6.4	14,080	4		0.2	5.91	139.85	1983
GEA	BBC	Ula Pandan	Singapore	100	220,000	65		1.25	36.91		1978
GEA	Deutsche Babcock	Ula Pandan	Singapore	33	72,600	21		1.25	36.91		1982
GEA	GHH	Iscor	South Africa	118	259,600	76		0.21	6.20	141.78	1975
Marley/BDT	DB Thermal, Johannesburg		South Africa	80	176,000	52		5	147.63		1976
GEA	ESCOM	Matimba	South Africa	6x1588	6 x 3,493,600	6,165	6x665	0.22	6.50	143.67	1985
GEA	AECI Midland		South Africa	70	154,000	45	29	0.175	5.17	134.81	1985
GEA	ESCOM	Majuba	South Africa	3x1525	3 x 3,355,000	2,960	3x665	0.156	4.61	130.67	1990
Hamon	KEPCO/Halim		South Korea	126.7	278,800	82	105				1997
GEA	Uhde	Encaso I	Spain	23.5	51,700	15		0.2	5.91	139.85	1968
GEA	Uhde	Encaso II	Spain	23.5	51,700	15		0.2	5.91	139.85	1968
GEA	Uhde	Encaso III	Spain	18.2	40,040	12		0.164	4.84	132.46	1968
GEA	Siemens / Union Termica S.A.	Utrillas	Spain	314.1	691,020	203	160	0.1	2.95	114.83	1968
GEA	Repesa		Spain	17.5	38,500	11		0.225	6.64	144.61	1968
GEA	C.E.P.S.A.	Cadiz	Spain	56	123,200	36	7.6	0.25	7.38	149.12	1971
GEA	Robur	Mallorca	Spain	14	30,800	9		18	531.47		1978
GEA	Siemens	Almeria	Spain	5	11,000	3		0.157	4.64	130.90	1980
GEA	Siemens Spain	Solar Almeria	Spain	5.1	11,220	3		0.16	4.72	131.57	1980

GEA	Montcada		Spain	14.39	31,658	9		1.2	35.43		1984
GEA	Saica	Zaragoza	Spain	43.2	95,040	28		5	147.63		1987
GEA	RESA	MVA Tarragona	Spain	40	88,000	26		0.25	7.38	149.12	1988
GEA	Alstom	Son Reus	Spain	279	613,800	181	75	0.111	3.28	118.59	2001
Marley/BDT	Ghesa ESP / La Loma		Spain	56.63	124,586	37		0.108	3.19	117.60	2001
Marley/BDT	Ghesa ESP / Enemansa		Spain	56.63	124,586	37		0.108	3.19	117.60	2001
Marley/BDT	Babcock Borsig Power, Oberhausen / Tarragona		Spain	247.2	543,840	160		0.06	1.77	96.68	2003
Marley/BDT	ISCOR, Vanderbijlpark		Südafrika	150	330,000	97		0.25	7.38	149.12	1985
GEA	von Roll	Göteborg	Sweden	3x44	3 x 96,800	85		19	560.99		1971
Marley/BDT	Oschatz		Sweden	29	63,800	19		7.31	215.84		1972
Marley/BDT	Oschatz		Sweden	29	63,800	19		7	206.68		1975
GEA	von Roll	Norrtorp	Sweden	20	44,000	13		1.6	47.24		1982
GEA	Stadtwerke Biehl		Switzerland	8.33	18,326	5		21	620.05		1967
GEA	Stadtwerke Neuenburg		Switzerland	17.9	39,380	12	4.3	0.1	2.95	114.83	1969
Marley/BDT	von Roll, Zürich		Switzerland	62.4	137,280	40		6	177.16		1969
GEA	KVA Winterthur	Winterthur	Switzerland	15	33,000	10		1.2	35.43		1970
GEA	von Roll	Schaffhausen	Switzerland	2x10.5	2 x 23,100	14		21	620.05		1972
GEA	von Roll	Stadtwerke Zürich	Switzerland	2x19	2 x 41,800	14		0.09	2.66	111.02	1973
GEA	Widmer u. Ernst	Werdenberg Lichtenstein	Switzerland	16.9	37,180	11		2.7	79.72		1973
Marley/BDT	Sulzer AG		Switzerland	25.3	55,660	16		6	177.16		1974
Marley/BDT	Ciba-Geigy		Switzerland	11.3	24,860	7		5.2	153.54		1974
GEA	Müra	Biel	Switzerland	10.25	22,550	7		26	767.68		1975
Marley/BDT	Martin /KVA Bazenheid		Switzerland	29.4	64,680	19		21	620.05		1975
GEA	BBC. Baden	KVA Winterthur	Switzerland	37.2	81,840	24		0.143	4.22	127.59	1977
Marley/BDT	Müra, Biel		Switzerland	11	24,200	7		0.65	19.19		1978
GEA	BBC	Stadtwerke Zürich	Switzerland	35	77,000	23		0.13	3.84	124.22	1979
GEA	Widmer u. Ernst	Werdenberg	Switzerland	20.4	44,880	13		0.1	2.95	114.83	1982
Marley/BDT	Kringelen / KVA Linthgebiet		Switzerland	26.7	58,740	17		0.13	3.84	124.22	1983
Marley/BDT	Wehrle Werke / MVA Buchs		Switzerland	20	44,000	13		4.7	138.77		1983
Marley/BDT	Kühnle, Kopp & Kausch / MVA Bazenheid		Switzerland	30	66,000	19		1.15	33.95		1984
Marley/BDT	Wehrle / KVA Buchs		Switzerland	21	46,200	14		4.7	138.77		1985
GEA	SW St. Gallen		Switzerland	24.7	54,340	16		0.16	4.72	131.57	1986
Marley/BDT	Lonza, Basel		Switzerland	4	8,800	3		7	206.68		1987
GEA	KVA Werdenberg		Switzerland	12	26,400	8		0.1	2.95	114.83	1989
Marley/BDT	Lurgi / KVA Bazenheid		Switzerland	1.14	2,508	1		0.12	3.54	121.37	1989
GEA	SAIOD	Cottendart	Switzerland	18.4	40,480	12	6	0.1	2.95	114.83	1990
Marley/BDT	Caliqua		Switzerland	16	35,200	10		0.8	23.62		1990
Marley/BDT	ABB / KVA Oftringen		Switzerland	27.7	60,940	18		0.1	2.95	114.83	1991
Marley/BDT	Siemens, Erlangen / KVA Müra Biel		Switzerland	16	35,200	10		7	206.68		1991
GEA	von Roll	Buchs	Switzerland	38	83,600	25		0.1	2.95	114.83	1993
GEA	Blohm & Voss	Winterthur	Switzerland	28.8	63,360	19	17	0.14	4.13	126.84	1993
Marley/BDT	GEKAL / KVA Buchs		Switzerland	39	85,800	25		0.2	5.91	139.85	1993
GEA	AWZ Zürich	KVA Josefstraße II	Switzerland	37.5	82,500	24	10	0.13	3.84	124.22	1994
Marley/BDT	Caliqua Basel / KVA Thurgau		Switzerland	72	158,400	47		1.7	50.19		1994
Marley/BDT	Caliqua Basel / KVA Thurgau		Switzerland	23	50,600	15		0.15	4.43	129.28	1994
Marley/BDT	ABB Enertech AG / KVA Niederurnen		Switzerland	26.5	58,300	17		0.1	2.95	114.83	1996
Marley/BDT	Caliqua, Basel / KVA Gamsen		Switzerland	17.6	38,720	11		0.1	2.95	114.83	1996
Marley/BDT	Caliqua, Basel / KVA Basel III		Switzerland	46	101,200	30		5	147.63		1999
Marley/BDT	Caliqua, Schweiz / KVA Thun		Switzerland	47	103,400	30		0.12	3.54	121.37	2002
GEA	Kraftwerk Aleppo	Aleppo	Syria	50.5	111,100	33	13.8	0.095	2.80	112.97	1960
Hamon	Mitsubishi/Jandar		Syria	772.7	1,700,000	500	670				1995
GEA	Siemens Werke AG		Taiwan	8	17,600	5		0.3	8.86	156.45	1978
GEA	Lurgi		Taiwan	8	17,600	5		0.3	8.86	156.45	1981
GEA	MHI	Hsintien	Taiwan	60	132,000	39	20	0.18	5.31	135.85	1991
GEA	MHI	Shulin	Taiwan	87	191,400	56	30	0.2	5.91	139.85	1991
Marley/BDT	EPA Taiwan / Chung-Hsin Electric &		Taiwan	140	308,000	91		0.15	4.43	129.28	1997

Marley/BDT	EPA Taiwan / Chung-Hsin Electric &		Taiwan	94	206,800	61		0.15	4.43	129.28	1997
Hamon	Tuntex		Taiwan	85.0	187,000	55	20				1997
GEA	Lurgi AG	Taoyuan	Taiwan	141.4	311,080	91	35	0.15	4.43	129.28	2000
Hamon	Bechtel/Hsin Tao		Taiwan	608.9	1,339,600	394	600				2001
GEA	Siemens	Kuo Kuang	Taiwan	469	1,031,800	303	160	0.2	5.91	139.85	2002
Marley/BDT	Babcock Borsig Power, Gummersbach / Yungkang		Taiwan	98	215,600	63		0.177	5.23	135.23	2002
Hamon	Star Energy/Toshiba/TCIC/Fong Der		Taiwan	989.1	2,176,000	640	980				2004
Hamon	Sun Ba/Toshiba/TCIC/Chang Bin		Taiwan	494.5	1,088,000	320	490				2004
GEA	Stone & Webster	Aliaga	Turkey	15	33,000	10		0.191	5.64	138.07	1979
Marley/BDT	NEMA, Netzschkau / Izmit *		Turkey	19.5	42,900	13		0.078	2.30	105.86	1995
GEA	Siemens	Sise Cam	Turkey	34	74,800	22	10	0.1	2.95	114.83	1996
GEA	Zorlu Enerji	Bursa	Turkey	38	83,600	25	10	0.119	3.51	121.08	1997
Marley/BDT	Thomassen Power Systems NL / Esenyurt		Turkey	177	389,400	115		0.253	7.47	149.63	1998
Hamon	Zorlu Enerji/Bursa		Turkey	54.1	119,000	35	55				2000
GEA	Edison Mission Energy	Esenyurt	Turkey	78	171,600	50	75	0.25	7.38	149.12	2001
GEA	Shell Refinery	Haven	UK	19.5	42,900	13		0.138	4.07	126.33	1965
GEA	Foster Wheeler	Killingholme	UK	21	46,200	14		0.126	3.72	123.11	1968
GEA	Mobil Oil		UK	18.1	39,820	12		0.224	6.61	144.42	1968
GEA	Foster Wheeler	Shell Stanlow	UK	35	77,000	23	8	0.15	4.43	129.28	1971
GEA	Foster Wheeler	Shell Stanlow	UK	5.65	12,430	4	1	0.15	4.43	129.28	1971
GEA	W.H. Allen Sons. British Steel		UK	23.6	51,920	15		0.112	3.31	118.91	1971
GEA	W.H. Allen Sons. British Steel		UK	5.49	12,078	4		0.112	3.31	118.91	1971
GEA	Kellog		UK	60.14	132,308	39		0.2	5.91	139.85	1979
GEA	Kellog		UK	60.14	132,308	39		0.081	2.39	107.22	1979
GEA	Kellog		UK	49.7	109,340	32		0.2	5.91	139.85	1979
GEA	Kellog		UK	49.7	109,340	32		0.081	2.39	107.22	1979
GEA	Caloric		UK	2.9	6,380	2		10	295.26		1981
GEA	Aalborg Ciserv	Eye Power	UK	13	28,600	8	14	0.09	2.66	111.02	1991
GEA	Hawker Siddeley Pow. Eng.	Corby	UK	410	902,000	265	120	0.08	2.36	106.77	1992
GEA	Aalborg Ciserv	Eye Power	UK	47.9	105,380	31	28	0.09	2.66	111.02	1992
GEA	Hawker Siddeley Pow. Eng.	Peterborough	UK	410	902,000	265	120	0.08	2.36	106.77	1992
Marley/BDT	Siemens KWU, Offenbach / Rye House Power Station		UK	852	1,874,400	551		0.092	2.72	111.81	1992
GEA	Aalborg Ciserv	Glanford	UK	48.4	106,480	31	28	0.09	2.66	111.02	1993
GEA	NNC	Sheffield	UK	50	110,000	32	6.8	0.2	5.91	139.85	1996
Hamon	Siemens/East. Elect./King's Lynn		UK	315.3	693,600	204	360				1996
GEA	AES Electric Ltd.	Barry CHP	UK	270.7	595,540	175	100	0.1	2.95	114.83	1997
Marley/BDT	CNIM / Stoke-on-Trent Municipal Waste Plant		UK	61.6	135,520	40		0.08	2.36	106.77	1997
Marley/BDT	CNIM / Wolverhampton Municipal Waste Plant		UK	35	77,000	23		0.083	2.45	108.09	1997
Marley/BDT	CNIM / Dudley Municipal Waste Plant		UK	30	66,000	19		0.082	2.42	107.66	1997
Marley/BDT	Taymel / Thetford Biomass Plant		UK	137	301,400	89		0.08	2.36	106.77	1998
Hamon	Enron/Stone&Webster/Sutton Bridge		UK	684.6	1,506,200	443	780				1999
Marley/BDT	ABB Baden, Schweiz / Enfield		UK	346	761,200	224		0.085	2.51	108.95	1999
Hamon	Entergy/Mitsubishi/Damhead Creek		UK	700.1	1,540,200	453	780				2000
Marley/BDT	FLS, Denmark / Elean, UK		UK	100	220,000	65		0.065	1.92	99.42	2000
Hamon	EPR Scotland/Abengoa/Westfield		UK	38.6	85,000	25	12				2000
GEA	InterGen	Coryton Energy	UK	743	1,634,600	481	250	0.085	2.51	108.95	2001
Marley/BDT	ABB Alstom, UK / Shotton		UK	250	550,000	162		0.06	1.77	96.68	2001
Marley/BDT	CEL Intern. UK / Coventry Waste		UK	57	125,400	37		0.15	4.43	129.28	2001
GEA	InterGen	Spalding Energy	UK	906.3	1,993,860	586	358	0.105	3.10	116.59	2003
GEA	Black Hills Power & Light Co.	Neil Simpson I	USA	76	167,200	49	20	0.152	4.49	129.75	1968
GEA	Exxon	Benicia	USA	22.2	48,840	14		0.322	9.51	158.19	1975
GEA	Braintree Electric Light Dept.	Norton P. Potter	USA	86	189,200	56	20	0.118	3.48	120.78	1975
GEA	Black Hills Power/Pacific Power	Wyodak	USA	855	1,881,000	553	330	0.203	5.99	140.43	1977
GEA	Chugach Electric	Beluga	USA	217	477,400	140	65	0.19	5.61	137.87	1979

GEA	BBC	Beluga	USA	0.22	484	0		1	29.53		1979
GEA	Pacific Gas & Electric	Gerber	USA	23.6	51,920	15	3.7	0.07	2.07	102.01	1981
GEA	Miami Intl. Airport	Miami	USA	6.8	14,960	4		10.5	310.02		1982
GEA	Pacific Ultrapower	Chinese Station	USA	82.5	181,500	53	22.4	0.203	5.99	140.43	1984
GEA	Energy Factors / Sithe Energies	North Island NAS	USA	29.5	64,900	19	4	0.17	5.02	133.75	1984
GEA	Energy Factors / Sithe Energies	NTC	USA	18.1	39,820	12	2.6	0.17	5.02	133.75	1984
GEA	Dutchess County	Dutchess County RRF	USA	22.8	50,160	15	7.5	0.135	3.99	125.55	1985
GEA	Olmsted County	Olmsted County RRF	USA	18.14	39,908	12	4	0.186	5.49	137.07	1985
GEA	Wheelabrator Sherman Energy	Sherman	USA	56.9	125,180	37	20	0.067	1.98	100.48	1985
GEA	Chicago Northwest	Chicago	USA	19	41,800	12	1	1.05	31.00		1986
GEA	American Ref-Fuel	SEMASS	USA	184.8	406,560	120	54	0.12	3.54	121.37	1986
GEA	Ogden Martin Systems	Haverhill	USA	159.6	351,120	103	47	0.17	5.02	133.75	1987
GEA	ABB	Hazleton	USA	190.5	419,100	123	67.5	0.125	3.69	122.83	1987
GEA	TBG Cogen	Grumman	USA	48	105,600	31	13	0.183	5.40	136.46	1988
GEA	RAM Enterprises	National City	USA	4.1	9,020	3		0.7	20.67		1988
GEA	Oxford Energy	Exeter	USA	90	198,000	58	30	0.1	2.95	114.83	1989
GEA	Wheelabrator Environ. Sys.	Spokane	USA	70	154,000	45	26	0.067	1.98	100.48	1989
GEA	Intercontinental Energy	Bellingham	USA	324	712,800	210	100	0.1	2.95	114.83	1990
GEA	Cogen Technologies, Inc.	Linden	USA	867	1,907,400	561	285	0.083	2.45	108.09	1990
GEA	Falcon Seaboard	Norcon-Welsh	USA	68	149,600	44	20	0.085	2.51	108.95	1990
GEA	Energy America Southeast	North Branch	USA	282	620,400	182	80	0.237	7.00	146.81	1990
GEA	Intercontinental Energy	Sayreville	USA	324	712,800	210	100	0.1	2.95	114.83	1990
Marley/BDT	Indeck Energy		USA	55	121,000	36		0.085	2.51	108.95	1990
Hamon	Ogden/Martin/Huntington		USA	98.9	217,600	64	35				1991
GEA	University of Alaska	Fairbanks	USA	21	46,200	14	10	0.2	5.91	139.85	1991
GEA	Odgen Martin Systems	Union County	USA	161.9	356,180	105	50	0.27	7.97	152.43	1991
Marley/BDT	CRS Sirrine, Lowell		USA	73	160,600	47		0.11	3.25	118.26	1991
Marley/BDT	CNF Constructors		USA	58	127,600	38		0.12	3.54	121.37	1991
GEA	Black Hills Power & Light	Neil Simpson II	USA	248.7	547,140	161	80	0.2	5.91	139.85	1992
GEA	Odgen Martin Systems	Onondaga County	USA	117	257,400	76	50	0.1	2.95	114.83	1992
GEA	Falcon Seaboard	Saranac	USA	334.2	735,240	216	80	0.17	5.02	133.75	1992
GEA	Dutchess County	Dutchess Co. Extension	USA	22.5	49,500	15	15	0.17	5.02	133.75	1993
GEA	Mission Energy	Gordonsville	USA	2x158.4	2 x 348,480	205	2x50	0.2	5.91	139.85	1993
Marley/BDT	Bechtel, Rochester		USA	100	220,000	65		0.12	3.54	121.37	1993
GEA	Browning Ferris Gas Serv. Inc.	Arbor Hill	USA	39.6	87,120	26	9	0.1	2.95	114.83	1994
GEA	Municipal Electric Utility	Cedar Falls	USA	111.6	245,520	72	40	0.12	3.54	121.37	1994
GEA	Ogden Martin Systems of Haverhill	Haverhill Extension	USA	20.2	44,440	13	46.9	0.17	5.02	133.75	1994
GEA	MacArthur Res. Recovery Agency	Islip	USA	18.1	39,820	12	11	0.16	4.72	131.57	1994
GEA	Browning Ferris Gas Serv. Inc.	Pine Bend Landfill	USA	26.4	58,080	17	6	0.1	2.95	114.83	1994
Marley/BDT	Billings Generation / Billings, MT		USA	210	462,000	136		0.253	7.47	149.63	1995
GEA	Browning Ferris Gas Serv. Inc.	Mallard Lake Landfill	USA	46	101,200	30	9	0.1	2.95	114.83	1996
Marley/BDT	Bechtel, Crockett		USA	275	605,000	178		0.067	1.98	100.48	1996
GEA	Energy Management Inc.	Dighton	USA	200	440,000	129	60	0.186	5.49	137.07	1998
GEA	Sempra Energy / Reliant Energy	El Dorado	USA	483.3	1,063,260	313	150	0.085	2.51	108.95	1999
GEA	Rumford Power Associates	Rumford	USA	247.5	544,500	160	80	0.169	4.99	133.54	1999
GEA	Tiverton Power Associates	Tiverton	USA	249.5	548,900	161	80	0.169	4.99	133.54	1999
Marley/BDT	ABB Baden, Schweiz / Midlothian, TX		USA	4 x 255	4 x 561,000	660		0.12	3.54	121.37	2000
Marley/BDT	ABB Baden, Schweiz / Lake Road, CT		USA	3 x 256	3 x 563,200	497		0.12	3.54	121.37	2000
Marley/BDT	ABB Baden, Schweiz / Blackstone, MA		USA	2 x 256	2 x 563,200	331		0.12	3.54	121.37	2000
Marley/BDT	ABB Baden, Schweiz / Hays, TX		USA	2 x 256	2 x 563,200	331		0.12	3.54	121.37	2000
Hamon	Calpine/Bechtel/Sutter		USA	511.5	1,125,400	331	500				2001
GEA	Front Range Power	Front Range	USA	574.5	1,263,900	372	150	0.12	3.54	121.37	2001
Marley/BDT	ABB Alstom, Schweiz / Bellingham, MA		USA	2 x 256	2 x 563,200	331		0.12	3.54	121.37	2001
Marley/BDT	ABB Alstom, Schweiz / Midlothian, TX		USA	2 x 255	2 x 561,000	330		0.12	3.54	121.37	2001
Hamon	Sithe/Raytheon/Mystic		USA	1282.7	2,822,000	830	1600				2002
Hamon	Sithe/Raytheon/Fore River		USA	641.4	1,411,000	415	800				2002

GEA	PG & E Generating	Athens	USA	3x340	3 x 748,000	660	2x120	0.169	4.99	133.54	2002
GEA	Reliant Energy	Choctaw County	USA	766.6	1,686,520	496	350	0.156	4.61	130.67	2002
GEA	Calpine	Goldendale Energy	USA	307.5	676,500	199	110	0.15	4.43	129.28	2002
GEA	Reliant Energy	Hunterstown	USA	766.6	1,686,520	496	350	0.156	4.61	130.67	2002
GEA	Duke Energy Moapa LLC	Moapa	USA	2x780	2 x 1,716,000	1,009	2x200	0.21	6.20	141.78	2002
GEA	Black Hills Generation	Wygen 1	USA	248.7	547,140	161	80	0.2	5.91	139.85	2002
Marley/BDT	Mirant - Nevada Power Services, USA / APEX, Nevada		USA	657	1,445,400	425		0.338	9.98	158.48	2002
Hamon	Reliant/Sargent & Lundy/Big Horn		USA	463.6	1,020,000	300	650				2003
GEA	Calpine	Otay Mesa	USA	680.9	1,497,980	441	277	0.117	3.45	120.47	2003
Marley/BDT	Parsons Energy, Houston / Chehalis WA		USA	490	1,078,000	317		0.067	1.98	100.48	2003
Marley/BDT	KeySpan NY, USA / Ravenswood, NY		USA	278	611,600	180		0.183	5.40	136.46	2003
Hamon	Genwest LLC/Silverhawk		USA	712.5	1,567,400	461	500				2004
Hamon	NYPA/GE/Sargent & Lundy/Poletti		USA	488.4	1,074,400	316	450				2004
Hamon	Transalta/Delta Hudson/Chihuaha III		USA	316.8	697,000	205	350				2004
GEA	Maui Electric	Maalaea. Unit 15	USAS	72	158,400	47	30	0.2	5.91	139.85	1990
GEA	Toyo Engineering. Japan		USSR	72.6	159,720	47	14.5	0.32	9.45	158.09	1970
GEA	Toyo Engineering. Japan		USSR	54.6	120,120	35	10.6	0.32	9.45	158.09	1970
GEA	Toyo Engineering. Japan		USSR	52.7	115,940	34	10.2	0.32	9.45	158.09	1970
GEA	Toyo Engineering. Japan		USSR	18	39,600	12	3.5	0.32	9.45	158.09	1970
GEA	Toyo Engineering. Japan		USSR	13	28,600	8	2.1	0.32	9.45	158.09	1970
GEA	Toyo Engineering. Japan		USSR	3x72.6	3 x 159,720	141	14.5	0.32	9.45	158.09	1972
GEA	Toyo Engineering. Japan		USSR	54.6	120,120	35	10.6	0.32	9.45	158.09	1972
GEA	Toyo Engineering. Japan		USSR	3x54.6	3 x 120,120	106	10.6	0.32	9.45	158.09	1972
GEA	Toyo Engineering. Japan		USSR	3x52.7	3 x 115,940	102	10.2	0.32	9.45	158.09	1972
GEA	Toyo Engineering. Japan		USSR	18.7	41,140	12	3.6	0.32	9.45	158.09	1972
GEA	Toyo Engineering. Japan		USSR	3x18.4	3 x 40,480	36	3.6	0.32	9.45	158.09	1972
GEA	Toyo Engineering. Japan		USSR	80	176,000	52		0.32	9.45	158.09	1972
GEA	Toyo Engineering. Japan		USSR	54.6	120,120	35	10.6	0.32	9.45	158.09	1973
GEA	Toyo Engineering. Japan		USSR	54.6	120,120	35	10.6	0.32	9.45	158.09	1973
GEA	Toyo Engineering. Japan		USSR	18.7	41,140	12	3.6	0.32	9.45	158.09	1973
GEA	Toyo Engineering. Japan		USSR	18.7	41,140	12	3.6	0.32	9.45	158.09	1973
GEA	Toyo Engineering. Japan		USSR	80	176,000	52		0.32	9.45	158.09	1973
GEA	Toyo Engineering. Japan		USSR	80	176,000	52		0.32	9.45	158.09	1973
GEA	Technimont		USSR	30.3	66,660	20	8.5	0.2	5.91	139.85	1976
GEA	Technimont		USSR	30.3	66,660	20		0.2	5.91	139.85	1978
GEA	Technimont		USSR	30.3	66,660	20		0.2	5.91	139.85	1978
GEA	Davy Powergas	Gubaha	USSR	57.5	126,500	37		0.28	8.27	153.93	1979
GEA	Davy Powergas	Gubaha	USSR	54.4	119,680	35		0.28	8.27	153.93	1979
GEA	Davy Powergas	Gubaha	USSR	53	116,600	34		0.28	8.27	153.93	1979
GEA	Davy Powergas	Tomsk	USSR	57.5	126,500	37		0.28	8.27	153.93	1979
GEA	Davy Powergas	Tomsk	USSR	54.4	119,680	35		0.28	8.27	153.93	1979
GEA	Davy Powergas	Tomsk	USSR	53	116,600	34		0.28	8.27	153.93	1979
Marley/BDT	Salzgitter		USSR	21.8	47,960	14		1.35	39.86		1979
GEA	KSB		USSR	55.5	122,100	36		0.32	9.45	158.09	1981
GEA	Machinoimport	Moskau	USSR	5x77.8	5 x 171,160	252		0.32	9.45	158.09	1982
Marley/BDT	Oxidor		Venezuela	4.5	9,900	3		7	206.68		1974
GEA	Linde AG	El Tablazo	Venezuela	120	264,000	78		0.21	6.20	141.78	1990
Hamon	Hyundai E.C./Baria		Venezuela	217.9	479,400	141	160				2001
GEA	TDK / Mitsui	Ba Ria	Vietnam	217	477,400	140	60	0.178	5.26	135.43	1997
GEA	Lurgi Paris	Lendava	Yugoslavia	19.5	42,900	13		0.306	9.03	157.05	1977
GEA	Siemens Werke AG	Banja Luka	Yugoslavia	10	22,000	6		4.9	144.68		1978
GEA	Pancero Refinery		Yugoslavia	26.4	58,080	17		0.211	6.23	141.97	1981
Marley/BDT	Shanxi, Yushe		Yushe	2 x 669.5	2 x 1,472,900	866		0.34	10.04	158.44	2004
GEA	Foster Wheeler	Conoco		6.42	14,124	4		0.345	10.19	158.27	1976

B

EXAMPLE OF AIR-COOLED CONDENSER DESIGN CHECK

Customer	: Howden Cooling Fans
Project	: Project
Project ref.	: 12345
Howden ref.	: 01
File	: C:\CF-P20\CFP20600\SI-Def.HCF
Page	: 1
Date	: 14 Feb 2005
Time	: 12:55
Name	: CF-Sales

Howden

Howden Cooling Fans

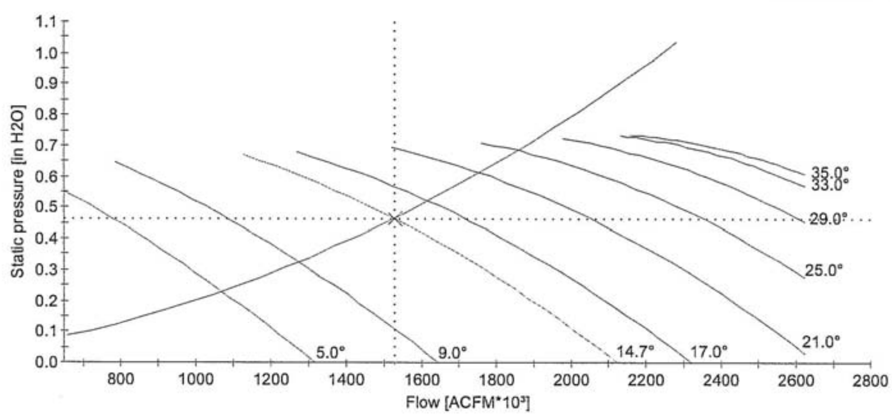
Lansinkesweg 4, 7553 AE Hengelo
P.O.Box 975, 7550 AZ Hengelo
The Netherlands

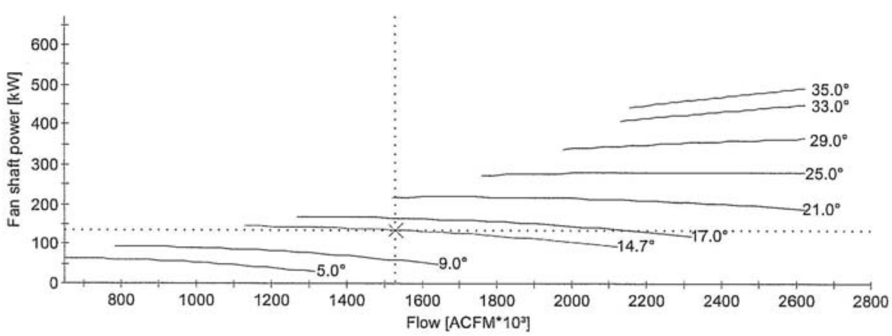
Tel : +31 (0)74 255 60 00
Fax : +31 (0)74 255 60 60
E-Mail : Cooling.Fans@Howden.nl

Selection input data

Air flow	: 1528.9	[ACFM*10³]	Application	: Air cooled condensor	
Static pressure	: 0.420	[in H2O]	Installation type	: Forced Draught	
Inlet temperature	: 98.6	[°F]	Mounting orientation	: vertical shaft; hub at inlet	
Air humidity	: 60	[%]	Fan inlet shape	: Bell, L/D = 0.15	
Altitude correction	: 2952.0	[ft]	Forced draught installation; no diffuser present		
Fan diameter	: 34 [ft]		Obstacle position	: at fan inlet:	at fan outlet:
Fan blade type	: ELF		Obst. area/housing area	: --	--
Blade number	: 7		Obst. frontal area	: --	7.22 [m²]
Fan speed	: 82	[R/min]	Ratio dist.X/housing dia	: --	0.057
			Fan clearance	: 0.01	(2S/FanDia)
			Crosswind	: 0.0	[m/s]

Selection graph





CF-P20 V6.00 Date: 02 Feb 2004 | **Serial no: HCF-600inet** | CFP20600.DLL V6.00 Date: 02 Feb 2004

Customer : Howden Cooling Fans
Project : Project
Project ref. : 12345
Howden ref.: 01
File : C:\CF-P20\CFP20600\SI-Def.HCF

Page : 1
Date : 14 Feb 2005
Time : 12:53
Name : CF-Sales

Howden Cooling Fans

Lansinkesweg 4, 7553 AE Hengelo
P.O.Box 975, 7550 AZ Hengelo
The Netherlands

Tel : +31 (0)74 255 60 00
Fax : +31 (0)74 255 60 60
E-Mail : Cooling.Fans@Howden.nl

Selection input data

Air flow	: 1528.9	[ACFM*10³]	Application	: Air cooled condensor
Static pressure	: 0.420	[in H2O]	Installation type	: Forced Draught
Inlet temperature	: 98.6	[°F]	Mounting orientation	: vertical shaft; hub at inlet
Air humidity	: 60	[%]	Fan inlet shape	: Bell, L/D = 0.15
Altitude correction	: 2952.0	[ft]	Forced draught installation; no diffuser present	

Fan diameter	: 34 [ft]	Obstacle position	: at fan inlet:	at fan outlet:
Fan blade type	: ELF	Obst. area/housing area	: --	--
Blade number	: 7	Obst. frontal area	: --	7.22 [m²]
Fan speed	: 82 [R/min]	Ratio dist.X/housing dia	: --	0.057
		Fan clearance	: 0.01	(2S/FanDia)
		Crosswind	: 0.0	[m/s]

Selection detailed results for 34 ELF 7

Calculated air density	: 1.016	[kg/m³]
Blade tip speed	: 44.5	[m/s]
Fan speed	: 82.0	[R/min]
Blade tip angle	: 14.7	[°]
Static efficiency	: 62.1	[%]
Total efficiency	: 81.3	[%]
Fan shaft power	: 134.3	[kW] 180HP
Pressure margin [%]	: 54.6¹ / 43.9²	
Air flow margin [%]	: 24.3¹ / -26.0²	

¹ according to API
² at selected blade angle

Sound power spectrum

Octave [Hz]	PWL [dB]	PWL(A) [dB(A)]	Tolerance [dB]
63	104.8	78.6	5
125	104.8	88.7	3
250	100.8	92.2	2
500	97.8	94.6	2
1000	94.8	94.8	2
2000	86.8	88.0	2
4000	82.8	83.8	2
8000	78.8	77.7	2

Tolerance on sound level values ± 2 [dB(A)]

Impeller Sound Power Level	: 98.5	[dB(A)]
SPL 1m beside fan (A)	: 68.9	[dB(A)]
SPL at 45° 1m below fan (B)	: 71.3	[dB(A)]
SPL 1m below fan (C)	: 76.9	[dB(A)]

Reflections not considered!

! SPL levels for guidance only !
See Howden Cooling Fans Manual for explanation of SPL calculations; norm 03-07.334.

Dynamical pressure	: 35.7	[Pa]
Corrected Fan Static Pressure	: 115.5	[Pa]
Axial thrust	: 12665	[N]
Sound due to blade angle	: 0.3	[dB(A)]
Sound due to inlet shape	: 0.0	[dB(A)]
Sound due to inlet obstacles	: 0.0	[dB(A)]
Sound due to outlet obstacles	: 1.2	[dB(A)]
Total Sound Power Level	: 99.7	[dB(A)]

Max. allowed tip speed	: 60.0	[m/s]
Max. allowed blade angle	: 35.0	[°]
Max. allowed blade temp.	: 149.0	[°F]
Impeller mass	: 1540	[kg]
Impeller moment of inertia	: 8330	[kg.m²]
Force due to loss of 1 blade(s)	: 24733	[N]
Residual imbalance force (G6.3)	: 83	[N]
Blade natural frequency	: 3.9	[Hz]
Blade operating natural frequency	: 4.3	[Hz]
Operating frequency	: 1.4	[Hz]
Blade passing frequency	: 9.6	[Hz]
Dimensionless flow Cf	: 0.192	
Dimensionless pressure Cp	: 0.115	

CF-P20 V6.00 Date: 02 Feb 2004   Serial no: HC[illegible]   [illegible] V6.00 Date: 02 Feb 2004

EXAMPLE OF A DESIGN CHECK

Data given by a vendor

1. **Geometric Data**

Total No. Modules	20	Total Bare Surface Area	504,182 ft^2
No.Streets x Mod./Str.	4 x 5	Total Finned Area	7,900,525 ft^2
Width x Length	166 x 227 ft.	Fan Deck Height	64 ft.
Total Height	100 ft.		

2. **Performance Data**

Heat Duty	691.3 MMBTU/h	Steam Flow	719,060 lbs/h
U Bare	85.2 BTU/hft^2°F	Turb.B.P.	4.87 in.HgA
U Finned	5.44 BTU/hft^2°F	Pr.at Inlet	4.713 in.HgA
MTD	16.1 °F	Temp.In/Out	132.7/ 130.5 °F
		Enthalpy	1061.9 BTU/lb
Air Flow	114,025,725 lbs/h	Temp. In.	98.6 °F
ACFM/Fan	1,528,870	Temp. Out	123.9°F
DP Static	0.42 in.WG	Elevation	2,952 ft.
		Min. Amb.	15 °F

3. **Construction Data**

No.primary Mod.	16	Tube	8 5/8x3/4 in.
No. Sec. Mod.	4	Thk.	0.059 in.
Module WXL	41.5x45.4 ft.	Pitch	2 17/64 in.
No.Tube Rows	1	Length Pri.	36 ft.
No. Bun./Mod.	12	Length sec.	32.8 ft.
No.Primary Bun.	192	No.Tub/Bun	40
No.Sec. Bun.	48	Fin	7 7/8x ¾ in.
Duct Dia	177 in.	Fin Thk.	0.012 in.
Manifold	4 x 91 in.	Fins/ Inch	11
Cond.header	24 in.		

4. **Mechanical Equipment**

Fan	ELF	Motor	Elec.	Gearbox	Helical
Diameter	34 ft.	RPM	1800/900	Reduction	21.3
No.Blades	7	Rating	200 HP	Serv.Factor	2.0
RPM	82	V/ F/ Ph	480/60/3		
Fan Shaft HP	171				

EXAMPLE OF A DESIGN CHECK

Steam Duct

Bottom Exhaust + 65 ft. Straight length + 1x Tee Junction + 2x 83 ft. Distribution Lines + 90 ft. Riser + 227 ft. Manifold.

Velociy Head		**ρ* V²/2gċ**	
Total VH (Btm. Exh. + 3x90° Bends + 1x Tee)		**3.4**	
Steam Velocity		**164.8 ft/sec**	
(719060*0.944/3600*147.2/(π/4*(175.25/12)²			**sp.vol=147.2**
Duct friction loss	**f*L/D(V²/2gċ/sp.vol.)**		**f = 0.007**
Total Duct Loss		**0.137 + 0.01 = 0.147 in.Hg**	
Duct Loss given			**0.157 OK**

Modules	**20**			
Primary	**16**	**Secondary**	**4**	
Bun x Tub	**192 x 40**		**48 x 40**	
Bare surface	**18.1/12*35.48*40*192**		**18.1/12*32.2*40*48**	
Area ft²	**411,000**		**93,251**	
Total	**504,251**	**Given**	**504,182**	**OK**
Face Area ft²	**40*2.2656/12*35.48*192**		**40*2.2656/12*32.2*48**	
	51,445		**11,672**	
Total Face Area	**63,117**			
Air Flow lbs/h	**114,025,725 (given)**			
Mass Flow Rate	**114025725/3600/63117 = 0.502 lbs/ft²s**			
Temp.Rise Air °F	**691.3E6/114025725/0.24 = 25.3 °F**		**sp.heat air = 0.24 BTU/lb**	
Air Temp. In		**98.6**		
Air Temp. Out		**123.9 °F**		**OK**

Pressure Drop Steam Side (Primary):

Pr. At Tube Inlet	**4.713 in.HgA**	**sp.vol.= 151.8**		
DP Tube entrance	**0.015**			
DP Tube	**0.071**			
DP Outlet	**0.001**			
	....			
Total	**0.087 in.HgA**			
Pr. At Outlet	**4.626 in.HgA**	**Temp.= 130.8 °F**	**Given = 130.5**	**OK**
MTD	**16.06 °F (based on 130.5 °F) Checks w/given value**			**OK**
Assume U given is correct				
Q Primary	**563.663 MMBTU/h**			
Q to Secondary	**127.640 MMBTU/h**	**Dry Steam**	**125,331 lbs/h**	
Cond.Header	**24 in. (given) Thk = 0.375 in.**		**ID = 23.25 in.**	
No. of Lines	**4 Roofs x 2 lines per roof x 2 (steam from both sides of Sec.Module)**			
Steam Velocity	**114 ft/s**			
Pr. Loss	**0.015 + 0.029 = 0.044 in.HgA**			
Pr. at Sec.Inlet	**4.582 in.HgA**		**sp.vol = 155.9**	

Pr. Loss sec.	**0.031 in.Hg**				
Pr. Out of Sec.	**4.551 in.HgA**	**Temp.= 130.2 °F**			
MTD	**15.75 °F**				
Heat Load Sec.	**125.133 MMBTU/h**				

Total Heat Load **688.8 MMBTU/h** **vs** **691.3 MMBTU/h** **-0.4%** **OK**

FAN

ACFM/Fan **Air Flow/60/No.Fans/ρair** **ρair at 98.6 °F&2952 ft.**
= 0.06339 lbs/ft³
1,527,722 **given 1,528,870** **OK**

Static pr.loss **0.42 in.WG** **given**

Fan checked by Fan Mfr. Program (attached)

Fan Shaft HP		**180**		
Gearbox Efficiency	**0.97**			
Motor efficiency	**0.94**			
Power at Motor Terminal		**197.4**		
Motor Rating		**200 HP**		**OK**
Gearbox Ratio		**1750/82**	**21.3**	**OK**

C
TERMINOLOGY

Glossary of Air-Cooled Condenser Components

C.1 Steam Side

Dogbone Expansion Joint – This rectangular connection is welded to the turbine exhaust to minimize forces from steam duct to turbine.

Transition Piece - Located below the dogbone joint (which is rectangular), this piece transitions to a circular shape.

Main Duct – This circular piping transports steam from turbine exhaust to the ACC.

Steam Distribution Line – This piping is placed across the width of the ACC to distribute steam to the A-frame streets.

Risers – These individual vertical ducts transport steam from the steam distribution line to the top of the tube bundles.

Gimbal Expansion Bellows – This connective system is used to absorb the movement in the main duct, distribution line, and riser due to thermal expansion.

Steam Distribution Manifolds – These manifolds distribute steam to the primary bundles.

Hinged Expansion Bellows – These connective systems absorb the movement in the manifolds due to thermal expansion.

Drain Pot – This system collects water from the steam duct, with water from the drain pot pumped to the condensate receiver tank.

Balance Line – This connecting line between main steam duct and condensate receiver tank equalizes pressure during startup and provides steam for reheat and deaeration during normal operation.

Rupture Disc/s Assembly – This pressure relief system is mounted on the main steam duct to protect the unit from over pressure.

C.2 Tube Bundles

Primary Stage or "Kondensator" – This heat exchange tube bundle system first receives steam from the turbine exhaust. The tube bundle consists of finned tubes arranged in single or multiple rows. The tubes are welded to the tubesheet at the top and bottom.

Second Stage or "Dephlegmator" – Steam not condensed in the primary stage is condensed in the second-stage tube bundles. These bundles are typically the same configuration as primary stage bundles except the tube length is often shorter.

C.3 Steam/Condensate Carryover Lines

The carryover lines are located at the bottom of the tube bundles and transport steam from the primary bundles to the secondary bundles. They also collect the condensate from all tube bundles.

C.4 Condensate Side

Drain Line – The drain line drains the condensate from the carryover lines to the condensate receiver tank (CRT).

Condensate Receiver Tank – The CRT is a cylindrical vessel, normally located underneath or adjacent to the ACC. Condensate from the ACC, makeup water required for the plant operation, and other drains flow into the tank.

Deaerator – The deaerator is located on top of the CRT to reheat and release oxygen from the condensate and makeup water.

C.5 Air Take Off Side

Air Take Off Line – These connecting lines convey noncondensables (mainly air) leaking into the turbine exhaust steam from the top portion of the secondary bundles to the air removal equipment.

Air Removal Equipment – Steam jet air ejectors or liquid ring vacuum pumps are used to remove noncondensables from the ACC and release them to the atmosphere. In the case of steam jet ejectors, a hogging ejector is used for air evacuation during startup and a holding ejector is used for removal of air during normal operation.

C.6 Mechanical Equipment

Fans – Axial induced draft fans deliver cooling airflow.

Electric Motors – Fans are driven by electric motors, with the speed dependent on the ambient temperature range of operation and noise level criteria. Motors normally are built to the following specifications: totally enclosed fan-cooled (TEFC), NEMA4 enclosure for outdoor installation, Class B temperature rise, and Class F insulation.

Speed Reducers – Typically parallel shaft, spiral bevel type gearboxes are used as speed reducers.

Couplings – Flexible couplings, used to connect motors to gearboxes, are intentionally designed to be the first failure point in the event of a drive system failure.

Vibration Switches – Cut off switches are used to shut a motor off in case of excess vibration of mechanical equipment.

C.7 Structural Steel

Structural steel is used in columns, bracings, beams, fan decks, fan rings, fan bells, module partition plates, motor bridges, walkways and platforms, stairs, ladders, windwall bracings, and sheeting.

C.8 Drain Pot System

The drain pot pump circulates water from the steam duct to the CRT. Normally, 2 x 100% pumps are provided, with one in operation and one in standby mode.

The Electric Power Research Institute (EPRI)

The Electric Power Research Institute (EPRI), with major locations in Palo Alto, California, and Charlotte, North Carolina, was established in 1973 as an independent, nonprofit center for public interest energy and environmental research. EPRI brings together members, participants, the Institute's scientists and engineers, and other leading experts to work collaboratively on solutions to the challenges of electric power. These solutions span nearly every area of electricity generation, delivery, and use, including health, safety, and environment. EPRI's members represent over 90% of the electricity generated in the United States. International participation represents nearly 15% of EPRI's total research, development, and demonstration program.

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