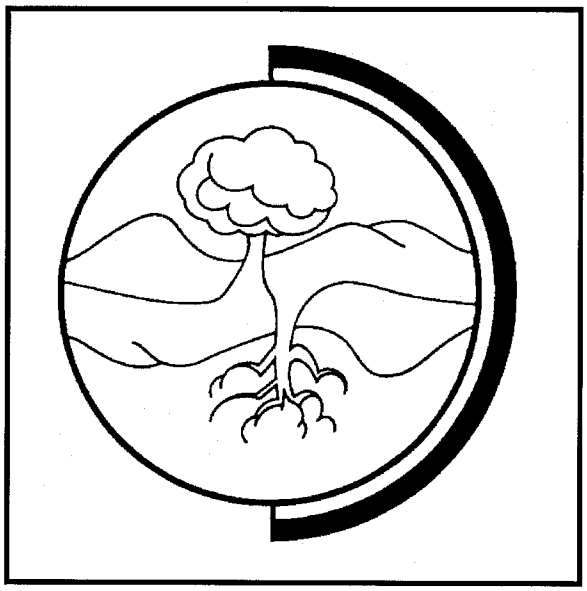
PNNL-10966
UC-202

A Comparative Assessment of Alternative Combustion Turbine Inlet Air Cooling Systems



February 1996

Prepared for the U.S. Department of Energy
under Contract DE-AC06-76RLO 1830

Pacific Northwest National Laboratory
Operated for the U.S. Department of Energy
by Battelle

Battelle

PNNL-10966

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency thereof, nor Battelle Memorial Institute, nor any of their employees, makes **any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights.** Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof, or Battelle Memorial Institute. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

PACIFIC NORTHWEST NATIONAL LABORATORY
operated by
BATTELLE MEMORIAL INSTITUTE
for the
UNITED STATES DEPARTMENT OF ENERGY
under Contract DE-AC06-76RLO 1830

Printed in the United States of America

Available to DOE and DOE contractors from the Office of Scientific and Technical Information, P.O. Box 62, Oak Ridge, TN 37831; prices available from (615) 576-8401.

Available to the public from the National Technical Information Service, U.S. Department of Commerce, 5285 Port Royal Rd., Springfield, VA 22161

The contents of this report are printed on recycled paper.

A Comparative Assessment of Alternative Combustion Turbine Cooling Systems

D.R. Brown
S. Katipamula
J.H. Koynenbelt

February 1996

Prepared for
The U.S. Department of Energy
Under Contract DE-AC06-76RLO 1830

Pacific Northwest National Laboratory
Richland, Washington 99352

Contents

Summary iii

1.0 Introduction 1.1

2.0 Concept Overview 2.1

3.0 Cooling System Selection 3.1

3.1 Chiller 3.2

3.2 Air Cooling Heat Exchanger 3.3

3.3 Heat Rejection 3.3

3.4 Storage 3.4

4.0 Application Condition Selection 4.1

4.1 Plant Type and Schedules 4.1

4.2 Inlet Air Temperature 4.2

4.3 Meteorology 4.6

5.0 Analytical Approach 5.1

6.0 Performance Analysis 6.1

6.1 Power Plants 6.1

6.2 Chillers 6.5

6.3 Storage 6.9

6.4 Fluid Circulation 6.9

6.5 Evaporative Cooling 6.10

6.6 Auxiliary System Cooling 6.10

6.7 Turbine Inlet and Exit Excess Pressure Drop 6.10

6.8 Combined-Cycle Steam Extraction Power Output Reduction 6.11

7.0 Cost Analysis 7.1

7.1 Chiller Capital Costs 7.1

7.2 Storage Capital Costs 7.2

7.3 Inlet Air Coil Capital Costs 7.2

7.4 Fluid Circulation Capital Costs 7.3

7.5 Miscellaneous Equipment Capital Costs 7.3

7.6 Annual Maintenance Costs 7.4

8.0 Economic Analysis 8.1

8.1 Incremental Power and Energy Calculation 8.1

8.2 Incremental Power and Energy Value 8.2

9.0 Results 9.1

9.1 Nomenclature 9.1

9.2 Industrial Turbine 9.1

9.3 Aeroderivative Turbine 9.4

9.4 Combined-Cycle 9.4

9.4.1 Intermediate Duty Plant; Cooling Hours Less Than Plant Hours 9.6

9.4.2 Intermediate Duty Plant; Cooling Hours Equal to Plant Hours 9.7

9.4.3 Baseload Duty Plant; Cooling Hours Less Than Plant Hours 9.7

9.4.4 Baseload Duty Plant; Cooling Hours Equal to Plant Hours 9.10

9.5 Climate Impacts 9.10

9.6 Net Present Value Components 9.12

10.0 Conclusions and Recommendations 10.1

11.0 References 11.1

12.0 Bibliography 12.1

Appendix A - Inlet Air Cooling Evaluation Model A.1

Appendix B - Component Sizing B.1

Appendix C - Complete Results C.1

Figures

2.1 Effect of Inlet Air Temperature on Combustion Turbine Performance 2.2

2.2 Generic Inlet Air Cooling System..... 2.3

Tables

3.1 Cooling System Technology Options..... 3.1

4.1 Power Plant and Cooling System Schedules 4.2

8.1 Power Plant Cost and Performance Assumptions..... 8.2

8.2 Power Plant Economic Assumptions..... 8.3

8.3 Incremental Power and Energy Values..... 8.3

9.1 Industrial Turbine Results-Nashville..... 9.2

9.2 Aeroderivative Turbine Results-Nashville 9.5

9.3 Combined-Cycle Results I-Nashville 9.6

9.4 Combined-Cycle Results II-Nashville..... 9.8

9.5 Combined-Cycle Results III-Nashville..... 9.9

9.6 Combined-Cycle Results IV-Nashville..... 9.10

9.7 Design Temperature Conditions..... 9.11

9.8 Climate Impacts on Economics..... 9.11

9.9 Net Present Value Components..... 9.13

1.0 Introduction

Interest in combustion turbine inlet air cooling (CTAC) has increased during the last few years as electric utilities continue to seek attractive sources of peaking power. Inlet air cooling increases the power production capacity and decreases the heat rate of a combustion turbine during hot weather when the demand for electricity is generally the greatest. Off-peak ice generation and storage using electrically-driven chillers has proven to be cost-effective under peak shaving conditions with a large difference between the value of peak and off-peak power (e.g., see Ebeling et al. 1992). The general applicability of the concept and the preference of specific concepts is still being debated in the literature, however.

For the past few years Rocky Research of Boulder City, Nevada has been funded by the U.S. Department of Energy to conduct research on complex compound (ammoniated salt) chill storage systems for low-temperature refrigeration applications. While the focus of this work has been on food processing applications, complex compound refrigeration could also be applied to CTAC. Advanced technologies such as this could improve CTAC economics and/or extend the range of cost-effective applications.

The Pacific Northwest National Laboratory[(a)] conducted this assessment of CTAC options for the U.S. Department of Energy's Office of Energy Management as part of the Thermal Energy Storage Program. The principal objectives of this study were as follows:

1. identify the preferred CTAC technology as a function of application conditions from currently available cooling technologies,

2. identify application conditions where currently available cooling technologies are not cost-effective, i.e., construction of additional power plant capacity without CTAC would be preferred,

3. determine the potential attractiveness of the ammoniated salt solid/vapor absorption cooling system proposed by Rocky Research for CTAC applications, and

4. determine the potential attractiveness of cooling combustion turbine inlet air to 0°F (in addition to cooling inlet air to a temperature of around 40 to 45°F, which is considered standard practice for CTAC).

Meeting the first two objectives identifies the fraction of combustion turbine applications that could be economically served by CTAC with currently available cooling technologies. It also establishes a baseline against which future development of CTAC technologies can be compared. Meeting the last two objectives directly measures the value of developing and demonstrating complex compound technology and/or cooling inlet air substantially below the freezing point.

To address the objectives established above, a screening study was designed to evaluate the principal cooling system technology options over a broad range of application conditions. Application variables

---

(a) Operated for the U.S. Department of Energy by Battelle under Contract DE-AC06-76RLO 1830.

included power plant type, power plant operating schedule, cooling system operating schedule, design inlet air temperature, and climate. Cooling system technology options included evaporative or refrigerative cooling, refrigerative chiller type, the inclusion of cool storage or not, and the type of storage (storage media type, daily or weekly cycle, and load-leveling or load-shifting design). A more explicit description of the application conditions and cooling system technologies evaluated is presented in the Sections that follow.

Altogether, approximately 3500 different cases were evaluated. Still, it's impossible to conduct a truly comprehensive analysis, with essentially endless variations of technology options and application conditions. Therefore, the results of this study are useful for establishing general rules-of-thumb regarding CTAC technology effectiveness, but better cooling system designs could always be proposed and alternative application conditions may yield different results.

The balance of the report documents the cooling systems and application conditions evaluated, key assumptions, and analysis leading up to the results, conclusions, and recommendations. An overview of the CTAC concept is presented in Section 2.0.

Note that English rather than metric units are used in this report. Although metric units are the standard for reports prepared for the U.S. Department of Energy, English units are used in this report to facilitate communication with cooling system vendors and users in the United States.

2.0 Concept Overview

Combustion turbines are constant volume machines, i.e., air intake is limited to a fixed volume of air regardless of ambient air conditions. As air temperature rises, its density falls. Thus, although the volumetric flow rate remains constant, the mass flow rate is reduced as air temperature rises. Power output is also reduced as air temperature rises because power output is proportional to mass flow rate. The conversion efficiency of the gas turbine also falls as air temperature rises because more power is required to compress the warmer air.

The impact of compressor inlet air temperature on mass flow rate, power output, and conversion efficiency is shown in Figure 2.1 for a typical "industrial" type turbine. Per custom, conversion efficiency is reported as a heat rate, which is the amount of fuel energy consumed per kWh of electricity produced. Thus, a rise in the heat rate is consistent with a drop in conversion efficiency. In general, the relationships are linear with temperature, or nearly so.

The performance curves show the dramatic effect of temperature on turbine performance, and the opportunity to improve performance via inlet air cooling. Cooling the inlet air improves both the power output and the heat rate, but the impact on power output is greater. Furthermore, the positive impacts from inlet air cooling increase with higher ambient temperatures, when utilities typically experience the highest demand for power. Therefore, incremental power production is greatest at the time it is most valued. *Usually, the primary objective of gas turbine inlet air cooling is to increase peak power output. The heat rate improvement is a significant, but secondary benefit.*

The relative importance of the two impacts depends mostly on the value of incremental kW and kWh production, the number of operating hours per year, and ambient air temperatures during the operating period. A greater number of annual operating hours will increase the importance of the heat rate impact, while fewer hours will emphasize the increase in power output. Applications with a higher ratio of average to peak ambient operating temperature will also increase the importance of the heat rate impact, all else equal.

In theory, power output could be further increased by cooling below the temperature range indicated in Figure 2.1. In practice, all turbines are designed around a maximum thrust level that sets a useful limit on the inlet air temperature. In particular, the maximum power output of "aeroderivative" type turbines often occurs around the freezing point and may actually decline below this temperature. Cooling near or below the freezing point also presents concerns associated with icing, which will be discussed in Section 4.0.

A block diagram of a generic combustion turbine inlet air cooling system, with and without storage, is shown in Figure 2.2. The basic building blocks are the chiller, its cooling tower, the air coil, and interconnecting piping. Cold fluid from the chiller is pumped through the air coil, where the coolant is heated and returned to the chiller, while the inlet air is cooled prior to entering the compressor. The cooling tower provides cooling water to the chiller condenser. Alternatively, an evaporative condenser might be used with some types of chillers. Including storage and its associated piping loop increases the number of system components, but allows the chiller and cooling tower components to be downsized, assuming that

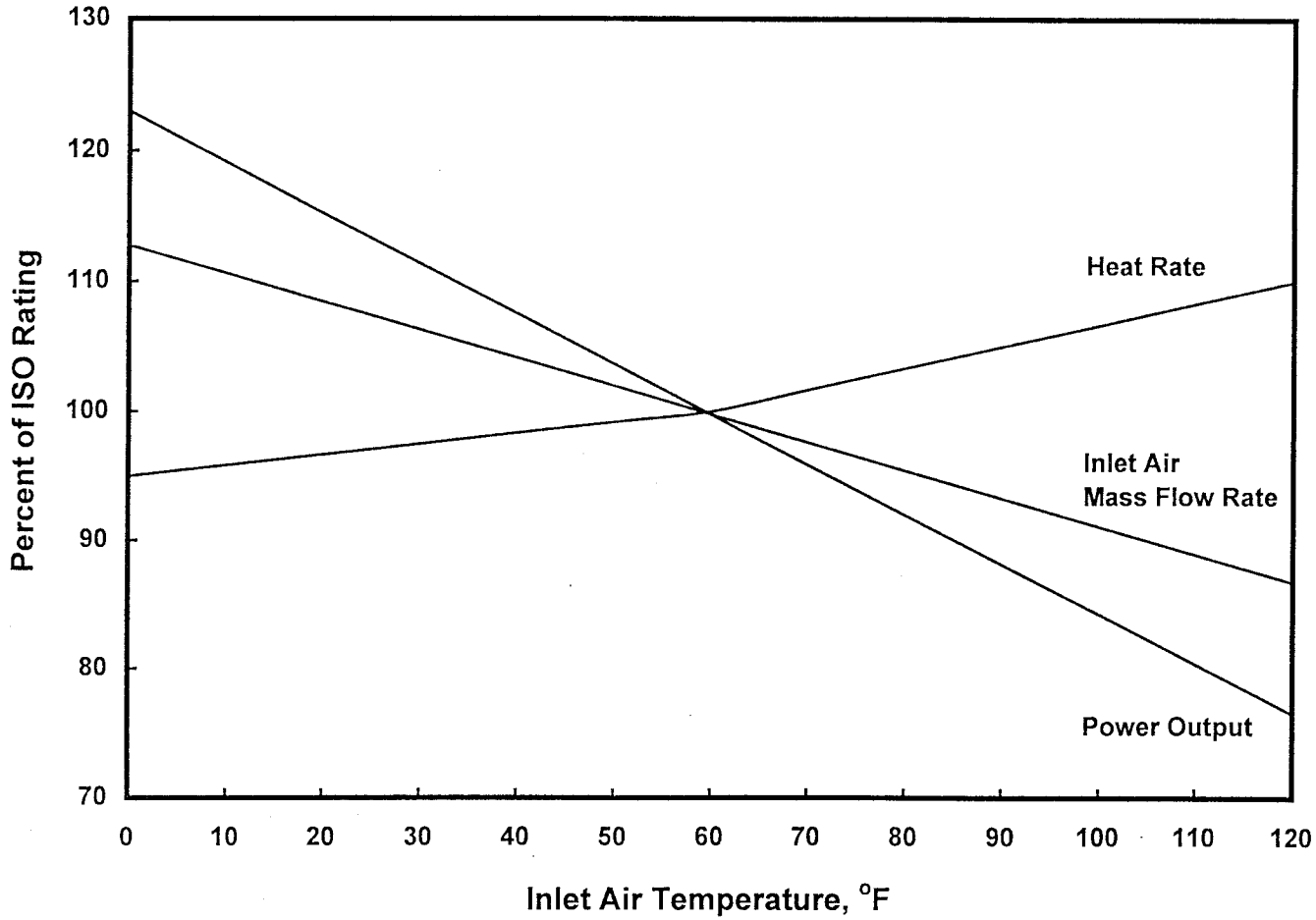


Figure 2.1. Effect of Inlet Air Temperature on Combustion Turbine Performance

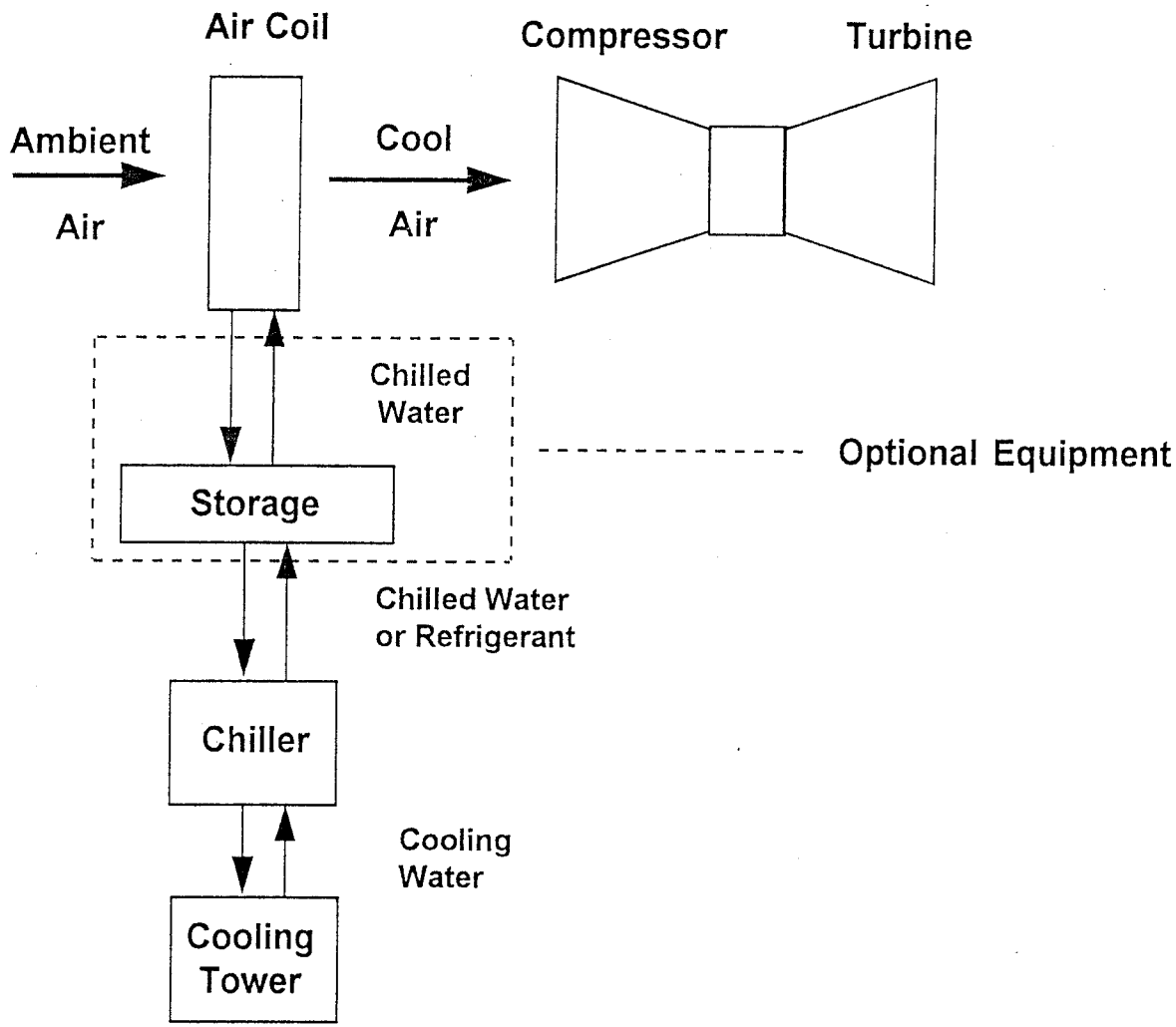


Figure 2.2. Generic Inlet Air Cooling System

cooling is not conducted 24 hours per day. Storage also significantly reduces on-peak parasitic power consumption for electrically-driven chiller systems.

The fundamental benefits of inlet air cooling were described above. The most obvious cost is the up-front investment and periodic maintenance of the inlet air cooling system hardware. The energy used to drive the chiller may also result in a significant expense, although the cost will vary significantly depending on whether the chiller is thermally or electrically driven and the source of thermal or electric energy. Inclusion of an air cooling coil within the inlet duct to the compressor will incur an additional pressure loss, with negative consequences to power output and heat rate, but the impacts will generally be less than 0.5%. These and other potential effects of inlet air cooling are discussed in more detail in the sections that follow.

The previous two paragraphs described the use of refrigerative systems for cooling turbine inlet air. A commonly used alternative, besides not cooling the inlet air at all, is to use evaporative cooling. Direct contact evaporative cooling, accomplished by passing the inlet air through a wetted media, can be particularly effective in drier climates. Any consideration of refrigerative inlet air cooling should also consider evaporative cooling as an option. Note that evaporative and refrigerative approaches should be considered independently. Direct evaporative cooling followed by refrigerative cooling would not reduce the refrigerative cooling load. It would only substitute latent load for sensible load.

3.0 Cooling System Selection

For a concept that may seem relatively straight-forward, a plethora of options complicates the inlet air cooling system component selection and system design process. A literature review was conducted to identify prospective cooling system technologies to be included in the study. A bibliography of CTAC-related literature is included at the end of this report. The initial list of alternatives developed from this process is presented in Table 3.1. Options that were quantitatively evaluated in this study are listed in bold print. The rationale for making these selections is discussed below.

Table 3.1. Cooling System Technology Options

Chiller
mechanical
electric, gas-fired, or steam turbine drivers
ammonia, CFC, or air working fluid
screw, reciprocating, or centrifugal compressor
absorption
lithium bromide, single or double stage
aqua-ammonia
complex compounds, single or double stage
steam jet
Thermal Source for Heat-Activated Chillers
add heat recovery steam generator to combustion turbines extract steam from combined-cycle or cogeneration plants
Air Cooling Heat Exchanger
evaporative cooling
direct/wetted media
indirect
evaporative/desiccant
refrigerative cooling
finned metal tubes
water or refrigerant in tubes
single or multiple evaporation temperatures
direct water spray
Heat Rejection
evaporative condenser
condenser and mechanical draft evaporative cooling tower
closed loop (finned tube/fan) cooler
Storage
storage type
ice/water
ice harvester or ice on tube
water
eutectic salts
brines
ice ponds
slush ponds
aquifer thermal energy storage (ATES)
storage cycle
daily or weekly
load-leveling or load shifting

3.1 Chiller

The chiller is the central piece of hardware in an inlet air cooling system. Its selection will determine or limit other equipment and design options. The two basic options are thermally-driven absorption chillers or mechanically-driven vapor compression chillers. Steam jet and mechanical air compression devices have also been suggested for use, but are significantly more expensive to own and/or operate.

Lithium bromide absorption chillers are commonly used to provide chilled water where steam or hot water is available. In these systems, lithium bromide is the absorbent and water is the refrigerant. A lithium bromide chiller operates at subatmospheric pressure in a hermetic vessel. Therefore, its evaporator cannot be directly placed in the path of the turbine inlet air. Thus, chilled water (limited to a minimum temperature of about 43°F) is produced and would be piped through a coil to cool the inlet air. Single-stage and double-stage units are available, which offers a trade-off between initial capital cost and improved coefficient of performance (COP).

Aqua-ammonia is another absorption chilling option. In these systems, water is the absorbent and ammonia is the refrigerant. The COP of an aqua-ammonia system is no better than a single-stage lithium bromide system, while its initial cost is several times higher. Aqua-ammonia systems do have the capability, however, of serving refrigeration applications at temperatures well below 32°F, where they would compete with complex compound or vapor compression chillers.

Complex compound technology is currently being developed by Rocky Research of Boulder City, Nevada for various refrigeration applications (Rockenfeller, Sarkisian, and Kirol 1992). The Rocky Research concept is a variation of absorption chillers that uses ammonia as the refrigerant and various proprietary solid salts as the "absorbent." The ammonia forms a complex compound with the salt, so this is not actually an absorption phenomena, but the complex compound cycle is similar to an absorption cycle. Like the aqua-ammonia systems, complex compounds can provide refrigeration at temperatures well below 32°F, but the systems being developed by Rocky Research are expected to cost much less.

The standard cooling technology for industrial refrigeration applications is mechanical vapor compression with ammonia refrigerant and a screw compressor. The preference for ammonia stems from its superior heat transfer properties and resulting high COP. Centrifugal compressors using CFC-22 have been widely used as water chillers for space cooling applications. Because of ozone depletion concerns, HCFC-134a may become more common for this application in the future. Safety concerns have precluded the use of ammonia for space cooling applications, however. The superior performance of ammonia makes it the preferred refrigerant for combustion turbine inlet air cooling (Kohlenberger 1994). Centrifugal compressors using CFC or HCFC refrigerants may offer lower first cost, but at lower efficiency and generally with less durability than the screw compressors designed for industrial refrigeration applications.

The mechanical vapor compression chiller can be powered by either electric motors, steam turbines, or gas-fired engines. Electric motors are the most common choice, with their advantages being lower first cost and simpler installation. For inlet air cooling, electric motors offer the flexibility of using off-peak electricity to run the chiller and charge a storage system, even when the combustion turbine is not running. On the other hand, the availability of low-cost or essentially free steam while the combustion turbine is running may make the steam turbine option attractive. Gas-fired engines would also offer the system charging flexibility

that electric motors provide.

3.2 Air Cooling Heat Exchanger

Refrigerative cooling systems must be compared to evaporative cooling, which is commonly applied to combustion turbines. The most common method of evaporative cooling uses a direct contact, wetted media design, which is the basis for evaluating evaporative inlet air cooling in this study. Indirect evaporative cooling, when coupled with direct evaporative cooling, can achieve lower air temperatures than direct evaporative cooling alone. However, the method consumes more water and causes a higher pressure drop, which makes it less desirable (Ondryas et al. 1991). Desiccant systems have also been proposed in conjunction with evaporative cooling, but the pressure drop for these systems is too great for combustion turbine applications.

In refrigerative cooling systems, the inlet air is generally cooled by passing it across a set of finned metal tubes containing either refrigerant or a secondary coolant such as water. The type of inlet air cooling coil used will depend on the type of chiller selected and whether or not storage is used. As described above, lithium bromide absorption chillers will always be coupled with an air/water coil. Vapor compression, aqua-ammonia, and complex compound chillers offer the option of evaporating the refrigerant directly in the coils, which reduces the air temperature for a given air coil approach temperature and also reduces equipment costs and pumping power compared to chilled water circulation systems.

If storage is included, the storage media is cooled and/or frozen by evaporating refrigerant in the storage vessel. In this case, a secondary coolant would be pumped from storage to the air cooling heat exchanger. If the secondary coolant is water it could be directly applied to the inlet air stream using evaporative cooling hardware. Routing the coolant through coils is generally preferred, however, because finned tubes suffer less pressure drop and water maintenance problems are minimized (Kohlenberger 1993).

Multiple coils, operating at sequentially declining evaporating temperatures can be used to lower the average "lift" and improve performance, but with increased capital costs. This design approach was used in this study for 0°F inlet air applications, but not for above-freezing applications.

3.3 Heat Rejection

As noted in Table 3.1, three different heat rejection approaches have been suggested in the literature. Either evaporative approach offers superior performance at a modest increase in cost compared to "dry cooling" with a closed-loop cooler. Industrial vapor compression refrigeration systems commonly use evaporative condensers while absorption systems are usually designed with standard condensers coupled with an evaporative cooling tower and connecting piping. The latter was applied to all cooling systems in this study to standardize the analysis. The impact on vapor compression system cost-effectiveness was assumed to be negligible.

3.4 Storage

Incorporating storage into the inlet air cooling system may be desirable to downsize the chiller and heat rejection components and significantly reduce on-peak parasitic power consumption for electrically-driven

systems. The chiller is usually the most expensive system component, so reducing its size and cost at the expense of adding storage and related piping can be cost-effective. Reducing on-peak parasitic power consumption is also important because increasing peak power output is usually the primary objective of combustion turbine inlet air cooling.

Chilled water or ice (really a mixture of water and ice) are the most obvious and effective storage media for inlet air cooling systems. Both are applicable to diurnal storage, and ice storage is applicable to weekly storage cycles as well. Seasonal storage of ice via engineered ice or snow ponds or chilled water in naturally occurring aquifers would also be possible, but these concepts suffer from site-specific limitations and have had only limited successful applications to date. Eutectic salts are another possibility, but the salts are more expensive than water, suffer availability losses on charge and discharge, and also suffer from limited application experience.

Steel or concrete cylindrical tanks can be used for water or ice storage. External insulation should be adequately thick to avoid condensation. Chilled water is added and removed from the bottom of water storage tanks, while warm water is added or removed from the top. This procedure forms a thermally stratified tank with a thermocline that rises on charge and falls on discharge. The preferred ice-making method uses a harvesting approach that periodically passes hot refrigerant from the compressor through the evaporator to release ice from the evaporator surface. The ice falls from the evaporator and makes a pile within the tank. Several evaporators are used to aid in distributing the ice. The alternative approach is to build up logs of ice around evaporator coils that run back and forth throughout the tank. Although the defrost cycle increases the effective cooling load by about 15%, the ice harvester is less costly to build because it requires much less evaporator surface and refrigerant inventory (Ebeling et al. 1992).

Selection of the storage media depends partly on the chiller type. Lithium bromide absorption chillers can only use water storage. Either water or ice storage is possible for vapor compression chillers. The principal advantage of ice is its greater chill storage density. The principal advantage of water is the mechanical simplicity of its storage system. Ice storage will generally allow the inlet air to be cooled to a lower temperature than with water storage, but ice generation requires a lower chiller evaporator temperature, which results in poorer chiller efficiency and higher chiller cost.

4.0 Application Condition Selection

The cost-effectiveness of CTAC is affected by the application conditions as much as by the choice of cooling system technology. Alternative conditions evaluated in this study include power plant type, power plant operating schedule, power plant cooling schedule, inlet air temperature, and climate.

4.1 Plant Type and Schedules

Combustion turbines are used in simple-cycle and combined-cycle electric power and cogeneration plants. Key differences in these applications for inlet air cooling systems are the presumed value of low-temperature thermal energy and the number of operating hours per day. For simple-cycle electric power plants, the energy in the exhaust gases is free (except for the cost of collection and assuming other uses of this energy are not perceived), while steam extracted from a combined-cycle or cogeneration power plant results in foregone electricity or steam sales, which has value. Simple-cycle electric plants tend to operate only a few hours per day, while combined-cycle and cogeneration plants tend to operate at least 12 hours per day. The inlet air cooling system may operate for fewer hours than the plant operates, of course, which adds to the number of system possibilities.

The availability of free thermal energy in the form of gas turbine exhaust suggests that a thermally-driven chiller may be preferred over an electrically-driven chiller in that situation. The use of storage, however, can significantly reduce the size and cost of the electrically-driven chiller and allow it to be driven by low-valued off-peak electricity. While storage could be used to reduce the size and cost of the thermally-driven chiller, there would be no free energy to drive the chiller during off-peak hours. For combined-cycle and cogeneration plants operating 24 hours per day, the availability of low-valued thermal energy during off-peak hours makes storage a more reasonable option for thermally-driven systems. Assignment of value to the thermal energy may also justify the expense of a more efficient thermally-driven chiller, whereas efficiency is of little concern when the energy is free.

The power plant types, plant operating schedules, and cooling system operating schedules evaluated in this study are summarized in Table 4.1. Simple-cycle applications were presumed to operate on schedules representative of peaking to intermediate duty while combined-cycle applications operate on intermediate to baseload schedules. Cooling system operation was presumed to always be coincident with power plant operation for simple-cycle applications, whereas the cooling period was allowed to be less than the plant operating period for combined-cycle applications. Thus, CTAC can be applied to a combined-cycle plant to produce incremental power on a peaking, intermediate, or baseload schedule, depending on the need. Both aeroderivative and industrial type turbines were evaluated for simple-cycle applications because both are commonly considered for this application and the two turbine types have significantly different cost and performance characteristics. Combined-cycle applications were based solely on an industrial type turbine. All applications produce electric power only, except for steam that is produced for powering thermally-activated chillers. CTAC is applicable to simple-cycle

Table 4.1. Power Plant and Cooling System Schedules

Plant Type	Plant Operating Schedule			
	Hours/Day	Time of Day	Days/Week	Months/Year
Simple Cycle[(a)]	4	2 to 6 PM	Mon. through Fri.	Apr. through Sept.
Simple Cycle[(a)]	8	Noon to 8 PM	Mon. through Fri.	Apr. through Sept.
Combined Cycle[(b)]	12	8 AM to 8 PM	Mon. through Fri.	Jan. through Dec.
Combined Cycle[(b)]	24	all day	Sun. through Sat.	Jan. through Dec.

(a) Cooling system and power plant schedules are the same.
(b) Planned shutdowns occur the first two weeks of April and October. Cooling schedules evaluated include the two simple-cycle schedules plus the first combined-cycle schedule for the 12 hr/day combined-cycle plant and both combined-cycle schedules for the 24 hr/day combined-cycle plant.

or combined-cycle cogeneration plants too, but the operating hours and value of thermal energy are similar to the combined-cycle electric power plant, so cogeneration applications were not explicitly evaluated.

4.2 Inlet Air Temperature

The inlet air cooling system must be designed to avoid icing at the compressor inlet or anywhere in the air intake structure. Ice fragments sucked into the compressor can cause serious structural damage. Icing is a potential problem anytime the ambient air temperature drops near the freezing mark. Compressor bleed air can be used to internally heat compressor inlet surfaces or directly injected into the inlet air stream (Dickson 1975). Exhaust from the turbine has also been routed back through the inlet air structure within closed heat exchanger surfaces, and electric heat tracing has been used as well (Wagar 1980).

The potential icing problem is exacerbated for inlet air cooling systems because warm ambient air will almost always be saturated after passing through the inlet air cooling coils. When the air is drawn into the mouth of the compressor, its velocity increases and its temperature drops further as air enthalpy is transformed into kinetic energy in an adiabatic process. Condensate icing can occur if the temperature drops below freezing. Equation (1) describes the relationship between air velocity and temperature. A temperature drop of about 10°F is common. Therefore, the design inlet air temperature should be at least 42°F to avoid potential icing problems.

$$V^2 / 2g = C_p * dT \tag{1}$$

where V = velocity
g = gravitational constant
C_p = heat capacity
dT = temperature drop.

The design inlet temperature may also be affected by the capabilities of the chiller. The minimum chilled water temperature available from lithium bromide absorption chillers is about 43°F. Therefore, practical approach temperatures at the air coil will limit the design inlet air temperature to about 50°F for this type of chiller. In this study, the design inlet air temperature was set at 52°F for lithium bromide absorption chillers and 42°F for other chiller types for above freezing applications.

The focus of inlet air cooling applications to date has been on above-freezing design conditions, and for good reason. As noted above, cooling the inlet air below freezing could lead to ice-caused damage of the compressor. At a minimum, periodic defrosting of the inlet air cooling coils would be required. In addition, while the performance benefit is linearly related to further reductions of the inlet air temperature (limited by the turbine's design maximum thrust limit), refrigeration costs per unit of cooling increase as the refrigeration temperature declines. Still, the potential benefit of cooling the inlet air below freezing is significant, and although the incremental cost per kW may be greater than for cooling to more moderate temperatures, it may still be less than installing a new power plant.

In addition to the above-freezing design conditions noted above, this study evaluated systems that cooled the inlet air to 0°F. Systems cooling to 0°F were split into two sections. The first section cools the inlet air to 42°F with a refrigerant evaporating at 32°F. The second section further cools the inlet air to 0°F with a refrigerant evaporating at -20°F. The design conditions for the first section avoid frost build-up while condensing a substantial portion of the moisture in the air before it can contribute to the icing problem. The approach temperature in the second section was doubled to allow for poorer heat transfer caused by icing of cooling coils.

Cooling the inlet air to 0°F will require periodic defrosting of the subfreezing cooling coils. In general, three half-size coil banks operating in parallel are proposed. Each half-size coil would be isolated from the others with metal ducting and louvered openings to allow defrosting to occur in one of the three half-size coils, while the inlet air would be cooled by the other two coils. Defrosting would occur on a rotating basis. The following design approaches were considered.

1. Hot Refrigerant Bypass. This approach is used for defrosting evaporative coils in most refrigeration applications. Hot refrigerant from the chiller is routed directly through the cooling coils, bypassing the condenser, to melt the ice from the coils. The same technique is also used in ice-making equipment. Hot refrigerant bypass has the advantage of a) being an established practice, b) allowing ice removal without having to melt all of the ice (like an ice-maker), and c) reduces condenser load. Its disadvantages include a) only being available when the chiller is running, which makes it impossible to incorporate with load-shifting storage, and b) requiring additional piping.

2. Turbine Exhaust. Part of the turbine exhaust could be routed past the frosted coils to melt away the ice. The principal advantage of this concept is the use of "free" waste heat. Disadvantages include a) requiring additional ducting, b) reducing turbine performance by increasing the exhaust pressure drop, and c) not being available in combined-cycle plants.

3. Electric Heat Tracing. Electric heat tracing might be externally applied to the cooling coils. The advantages of this concept are that it would a) always be available and b) allow ice removal without having to melt all of the ice. Its disadvantages include a) additional parasitic energy consumption, b) additional pressure drop across the coil, and c) questionable survival of the heat tracing in the operating

environment.

4. Extraction Steam. Extraction steam might be used to defrost the coils, if a suitable means of application could be developed. Low-grade thermal energy in the form of steam could be extracted from combined-cycle plants, but its disadvantages include a) not being available in simple-cycle plants, b) reduced power output from the steam turbine, c) piping costs, and d) the difficulty of applying the steam to the frost.

5. Ambient Air. Ambient air could be used to defrost the coils using an approach similar to that envisioned for using turbine exhaust. Compared to turbine exhaust, ambient air has the advantage of a) requiring minimal ducting and b) minimal incremental parasitic power consumption. Its principal disadvantage stems from the fact that it won't be nearly as warm as turbine exhaust, and won't necessarily be above freezing. On the other hand, if ice build-up is not great, say over a 4-hr cooling cycle, it might be possible to let the ice melt naturally during the rest of the day.

6. Turbine Operation. The coils could be defrosted by simply running the turbine without circulating coolant through the air coils until the coils are defrosted. This approach would not require a spare half-sized coil and therefore would have the advantages of incurring a) no incremental capital cost and b) no incremental operating costs to accomplish defrosting. This approach would have the disadvantages of a) reduced power output during the defrosting period and b) potential problems associated with cycling the inlet air temperature so frequently.

Hot refrigerant bypass (HRB) was chosen as the basis for defrosting the coil for systems cooling the inlet air to 0°F. This option is relatively simple to apply, is standard industry practice, and incurs minimal parasitic power consumption. Simply using ambient air may be practical for cooling applied in the summer only, but needs to be proven. Using turbine exhaust or electric heat tracing would be possible, but both would incur significant capital and operating costs. Note that HRB is only available while the chiller is running, so a load-leveling approach must be used if storage is incorporated.

As described above, after the inlet air has been cooled to 0°F by the cooling coils, it will accelerate and further cool according to the relationship described by Equation (1). Therefore, condensate icing will almost surely occur if no thermally protective measures are taken. The magnitude of the problem will have been substantially reduced, however, by the moisture removed in the cooling coils. Saturated air at 42°F contains 0.0056 lb of water per lb of dry air while this figure is reduced to 0.0008 lb at 0°F. At this latter condition, condensate icing may not be a problem, especially for applications that operate for only a portion of the day during the summer season.

In general, there are two basic options for minimizing or eliminating condensate icing: 1) reheating the air or 2) heating the surfaces of the intake structure. The following options were considered.

1. Compressor Bleed Air. Air from one of the early compressor stages is "bled" off of the main stream and routed past upstream inlet air structure surfaces to keep them relatively warm. Note that the surfaces don't have to be kept above freezing, just above the dew point, which would be 0°F after the cooling coils.

2. Compressor Cooling Air. Some air bled from the early compressor stages is used to cool hotter portions of the compressor. This air could be used as described above to keep upstream surfaces from icing.

3. Compressor Bleed Air Injection. Compressor bleed/cooling air could be injected immediately downstream of the cooling coils to reheat the inlet air.

4. Turbine Exhaust. Turbine exhaust could be routed back through the inlet air structure immediately after the cooling coils in a closed heat exchanger to reheat the inlet air.

5. Extraction Steam. Extraction steam could be routed back through the inlet air structure immediately after the cooling coils in a closed heat exchanger (or perhaps directly injected) to reheat the inlet air.

6. Electric Resistance Heating. Electric resistance heating could be used to warm the inlet air structure or to reheat the inlet air stream.

Reheating the air would avoid the formation of condensate icing, but would run counter to the purpose of cooling the inlet air. From a thermodynamic perspective, internally heating inlet air structure surfaces would be preferred because the air stream would not be reheated nearly as much. In addition, this approach is already used with compressor bleed air, although not necessarily at continuous icing conditions (Dickson 1975). In this study, if condensate icing is still a problem after moisture removal in the two cooling coils, then compressor cooling air is presumed to internally heat inlet air structure to further minimize ice build-up. If adequate compressor cooling air is available and inlet air structure heating channels are already incorporated into the design, then the marginal cost of avoiding condensate icing should be negligible. These assumptions have not been proven, however, and additional analysis and field testing will need to be conducted to confirm safe operating practices for subfreezing operating conditions.

4.3 Meteorology

The cost-effectiveness of CTAC will obviously depend on the reference operating conditions for an uncooled power plant. Warmer climates will show greater incremental power production, but will incur greater cooling loads and cooling system costs to meet those loads. Drier climates result in lower refrigerative cooling loads, but evaporative cooling also looks better in that situation.

Five climates, representing the range of climates found in the continental U.S., were evaluated. Cold, hot/cold, hot/humid, hot/arid, and temperate climate regions were defined in the *Small Office Building Handbook* (Holton et al. 1985). The cities selected to represent these five regions were Minneapolis, Nashville, Houston, Phoenix, and San Francisco, respectively. Minneapolis, Nashville, and Houston have similar peak design conditions, but their annual average conditions vary considerably. Phoenix is, of course, much warmer and drier, while San Francisco's weather is more moderate.

5.0 Analytical Approach

The evaluation compared the benefit of incremental power and energy production with the cost of owning and operating the inlet air cooling system. Costs and benefits were denominated in dollars and the resulting net present value (NPV) was calculated for each combination of cooling system technology and application condition evaluated. In addition, the capital cost per incremental power output was calculated for each combination evaluated. Refrigerative and evaporative cooling system economics were calculated based on reference plants that were uncooled. Refrigerative cooling system economics were also calculated based on reference plants that were evaporatively cooled.

The evaluation of each case proceeded as follows. The starting point is a set of data characterizing a specific combination of cooling system technology and application conditions being evaluated. From this initial input, power plant power and energy production, and cooling system loads are calculated. Next, the cooling system components are sized. This allows the calculation of component costs (initial capital and annual operating and maintenance costs) and parasitic power consumption. Finally, the cooling system investment per kW of incremental power production and its NPV are calculated. All calculations are conducted within a computer model that allows the cooling loads and component performance to be simulated on an hourly basis.

Detailed assumptions regarding the cost, performance and economic analyses are presented in Sections 6.0, 7.0, and 8.0. Computer modeling and component sizing details are presented in Appendices A and B.

6.0 Performance Analysis

The performance analysis included calculation of the power output and fuel consumption of the combustion turbine, with and without inlet air cooling, plus the parasitic power consumption of inlet air cooling system components. The latter includes compressor power for vapor compression chillers, steam and pump power for absorption chillers, pump power for the various fluid circulation systems, and fan power for the cooling tower.

Annual performance data were calculated by a computer model (see Appendix A) with the ability to aggregate performance calculated for each operating hour of the year, based on hourly weather data. Performance assumptions were developed based on a combination of information collected from equipment vendors and published literature. Specific assumptions are presented below for the power plants and cooling system components.

6.1 Power Plants

Three power plant types were evaluated in this study: 1) simple-cycle industrial turbine, 2) simple-cycle aeroderivative turbine, and 3) combined-cycle (based on an industrial-type turbine). Analysis of inlet air cooling systems requires knowledge of the power plant's (combustion turbine *and* steam turbine, for combined-cycle plants) performance (power output, heat rate, and mass flow rate) as a function of various operating conditions (altitude, dry-bulb temperature, humidity, turbine inlet pressure drop, and turbine exit pressure drop). The impact of operating conditions on power output and heat rate directly affect inlet air cooling economics, of course, while understanding the impact on mass flow rate is crucial to sizing cooling system components.

Power output, heat rate, and inlet air mass flow rate at ISO conditions (59°F dry-bulb temperature, 60% relative humidity, and sea level elevation), and the correction factors for calculating these performance characteristics at non-ISO conditions are presented below for each power plant type. Up to five adjustment factors (F1 for altitude, F2 for dry-bulb temperature, F3 for humidity, F4 for excess inlet pressure drop, and F5 for excess exhaust pressure drop) are multiplied by the ISO performance to calculate performance at non-ISO conditions.

Simple-Cycle Industrial Frame Combustion Turbine

ISO power output = 100 MW
ISO heat rate = 11,700 Btu/kWh based on HHV
ISO inlet air mass flow rate = 2,342,346 lb/hr.

Power Adjustment Factors:

F1 = 1.0 - 3.1875 E-5 * h; h = altitude, feet
F2 = 1.23 - 3.875 E-3 * T; T = inlet air dry-bulb temperature, °F
F3 = 1.0

F4 = 1.0 - 4.333 E-3 * P_i; P_i = excess inlet air pressure loss, inches H_2O
F5 = 1.0 - 1.634 E-3 * P_e; P_e = excess exhaust air pressure loss, inches H_2O.

Heat Rate Adjustment Factors:

F1 = 1.0
F2 = 0.95 + 8.333 E-4 * T for T <= 60°F; F2 = 1.0 + 1.666 E-3 * (T - 60) for T > 60°F
F3 = 1.0
F4 = 1.0 + 1.666 E-3 * P_i
F5 = 1.0 + 1.666 E-3 * P_e.

Inlet Air Mass Flow Rate Adjustment Factors:

F1 = 1.0 - 3.1875 E-5 * h
F2 = 1.128 - 2.17 E-3 * T
F3 = 1.0
F4 = 1.0 - 2.333 E-3 * P_i
F5 = 1.0.

Simple-Cycle Aeroderivative Combustion Turbine

ISO power output = 45 MW
ISO heat rate = 10,000 Btu/kWh based on HHV
ISO inlet air mass flow rate = 1,093,482 lb/hr.

Power Adjustment Factors:

F1 = 1.0 - 3.3125 E-5 * h
F2 = 1.125 for T <= 32°F; F2 = 1.125 - 5.06 E-3 * (T - 32) for T > 32°F
F3 = 1.0
F4 = 1.0 - 3.6 E-3 * P_i
F5 = 1.0 - 1.4 E-3 * P_e.

Heat Rate Adjustment Factors:

F1 = 1.0
F2 = 0.94 + 1.00 E-3 * T for T <= 60°F; F2 = 1.0 + 1.666 E-3 * (T - 60) for T > 60°F
F3 = 1.0
F4 = 1.0 + 1.2 E-3 * P_i
F5 = 1.0 + 1.1875 E-3 * P_e.

Inlet Air Mass Flow Rate Adjustment Factors:

F1 = 1.0 - 3.3125 E-5 * h
F2 = 1.15 - 1.875 E-3 * T for T <= 32°F; F2 = 1.09 - 3.21 E-3 * (T - 32) for 32°F < T < 60°F;
F2 = 1.0 - 4.67 E-3 * (T - 60) for T >= 60°F
F3 = 1.0
F4 = 1.0 - 2.55 E-3 * P_i
F5 = 1.0 - 3.125 E-3 * P_e.

Combined-Cycle

ISO power output = 225 MW
Combustion turbine portion = 155 MW
Steam turbine portion = 70 MW
ISO heat rate = 7300 Btu/kWh based on HHV
ISO inlet air mass flow rate = 3,365,475 lb/hr.

Power Adjustment Factors for Combustion Turbine Portion:

F1 = 1.0 - 3.33 E-5 * h
F2 = 1.22 - 3.7 E-3 * T
F3 = 1.0
F4 = 1.0 - 3.2 E-3 * P_i
F5 = 1.0 - 1.2 E-3 * P_e.

Power Adjustment Factors for Steam Turbine Portion:

The power adjustment factors for the steam turbine require a more complicated format. In general, steam turbine power is presumed to be proportional to steam production from the heat recovery steam generator (HRSG), with an adjustment for the impact of ambient temperature on condenser temperature. The general form for steam turbine power adjustment is as follows:

P/P_{ISO} = exhaust flow factor * exhaust temperature factor * condenser temperature factor.

Each of these three factors is defined below.

Exhaust Flow Factor = F1*F2*F3*F4*F5, where

F1 = 1.0 - 3.33 E-5 * h
F2 = 1.12 - 2.15 E-3 * T
F3 = 1.0
F4 = 1.0 - 2.5 E-3 * P_i
F5 = 1.0.

Exhaust Temperature Factor = ($T_{exhaust}$ - 550)/542, where

$T_{exhaust}$ = 1054 + 0.517 * T + P_i + 0.8 * P_e for T <= 60°F or
$T_{exhaust}$ = 1085 + 0.850 * (T - 60) + P_i + 0.8 * P_e for T > 60°F.

Condenser Temperature Factor =

1 - 1.9 E-4 * (T_{wb} - 54) for T_{wb} <= 54°F or
1 - 9.0 E-4 * (T_{wb} - 54) for T_{wb} > 54°F, and

T_{wb} = ambient wet-bulb temperature, °F.

Heat Rate (HR) Adjustment Factors for Combustion Turbine (CT) Portion:

F1 = 1.0
F2 = 0.94 + 1.00 E-3 * T for T <= 60°F; F2 = 1.0 + 1.5 E-3 * (T - 60) for T > 60°F
F3 = 1.0
F4 = 1.0 + 1.7 E-3 * P_i
F5 = 1.0 + 1.45 E-3 * P_e.

Heat Rate Adjustment Factors for Combined-Cycle (CC):

Overall heat rate = CT heat rate * CT power / CC power
7300 = CT heat rate * 155/225; Reference CT heat rate = 10,597 Btu/kWh
Adjusted CT heat rate = 10,597 * F1 * F2 * F3 * F4 * F5
Adjusted CC power = Adj. CT power + Adj. ST power
Adjusted CC heat rate = Adj. CT heat rate * Adj. CT power / Adj. CC power.

Inlet Air Mass Flow Rate Adjustment Factors:

F1 = 1.0 - 3.33 E-5 * h
F2 = 1.12 - 2.15 E-3 * T
F3 = 1.0
F4 = 1.0 - 2.5 E-3 * P_i
F5 = 1.0.

6.2 Chillers

The cost and performance of the chiller are key to the cost-effectiveness of inlet air cooling systems. Chiller performance is characterized by its coefficient of performance (COP), "relative capacity," heat rejection rate, and, for absorption-type chillers, the steam conditions and additional electric power consumption, if any. Performance assumptions for complex compound, vapor compression, aqua-ammonia, and lithium bromide chillers are presented below.

The COP, of course, is the amount of cooling obtained per unit of input energy, with the input and output measured in the same units. The COP varies considerably between chiller types and for the same chiller type, depending on the evaporating and condensing temperature, with the difference between these two temperatures defined as the "lift." In general, COP increases as the lift is reduced. Absorption-type chillers also come in single- and double-stage versions. Double-stage chillers use waste heat from the higher temperature stage to drive the lower temperature stage. A higher COP results, but the double-stage unit is more expensive. COP equations are presented here as a function of wet-bulb temperature for explicit chiller types, number of stages, and evaporator temperatures. The COP equations assume the condenser temperature is 15°F higher than the wet-bulb temperature. In general, the wet-bulb temperature is limited to a minimum of 25°F for COP modeling purposes, with more restrictive assumptions listed below for lithium bromide chillers.

Like the COP, the cooling capacity of a chiller is a function of the condenser temperature, and is expressed here as a function of the wet-bulb temperature. The chiller cost equations presented in Section 8 are based on chiller capacity at "rated" temperature conditions. Capacity is typically less at temperature conditions corresponding to the peak cooling load. Thus, the "relative" capacity equation must be applied to adjust the peak chiller capacity to the rated chiller capacity.

All energy absorbed by a chiller at the evaporator and input to the chiller in the form of steam or electricity must ultimately be rejected at a cooling tower. Thus, heat rejection requirements are directly related to chiller COP and vary considerably, especially when absorption systems are compared to vapor compression systems.

Complex Compound (CC) Chillers:

CC chiller 1 (CC-1); single-stage, 20°F evaporator:

COP = 0.496 - 7.0 E-4 * Twb
Relative capacity = 1.652 - 0.0079 * Twb
Steam temperature = 228°F; pressure = 19 psia; h_{fg} = 960 Btu/lb
Heat rejection rate = 39,335 Btu/hr per ton
Parasitic power = 0

CC chiller 2 (CC-2); single-stage, 32°F evaporator:

COP = 0.496 - 7.0 E-4 * Twb
Relative capacity = 1.415 - 0.0051 * Twb
Steam temperature = 228°F; pressure = 19 psia; h_{fg} = 960 Btu/lb
Heat rejection rate = 39,211 Btu/hr per ton
Parasitic power = 0

CC chiller 3 (CC-3); double-stage, 20°F evaporator:

COP = 0.867 - 2.5 E-3 * Twb
Relative capacity = 1.566 - 0.0069 * Twb
Steam temperature = 351°F; pressure = 137 psia; h_{fg} = 870 Btu/lb
Heat rejection rate = 29,991 Btu/hr per ton
Parasitic power = 0

CC chiller 4 (CC-4); double-stage, 32°F evaporator:

COP = 0.850 - 1.85 E-3 * Twb
Relative capacity = 1.334 - 0.0041 * Twb
Steam temperature = 351°F; pressure = 137 psia; h_{fg} = 870 Btu/lb
Heat rejection rate = 29,094 Btu/hr per ton
Parasitic power = 0

CC chiller 5 (CC-5); single-stage, -20°F evaporator:

COP = 0.42 - 6.75 E-4 * Twb
Relative capacity = 1.231 - 0.003 * Twb
Steam temperature = 293°F; pressure = 60 psia; h_{fg} = 915 Btu/lb
Heat rejection rate = 44,787 Btu/hr per ton
Parasitic power = 0

CC chiller 6 (CC-6); single-stage, storage type, -20°F evaporator:

COP = 0.52
Relative capacity = 1
Steam temperature = 213°F; pressure = 15 psia; h_{fg} = 969 Btu/lb
Heat rejection rate = 21,624 Btu/hr per ton for sizing heat rejection equipment
Heat rejection rate = 35,076 Btu/hr per ton for calculating annual heat rejection load
Parasitic power = 0

The storage version of the complex compound chiller is actually an integrated chiller/storage device that is sized based on ton-hours rather than tons.

Ammonia Vapor Compression (VC) Chillers:

VC chiller 1 (VC-1); 20°F evaporator:

COP = 14.1277 exp(-0.0144 * Twb)
Relative capacity = 1.08 - 0.001 * Twb
Heat rejection rate = 14,734 Btu/hr per ton
Parasitic power = 0

VC chiller 2 (VC-2); 32°F evaporator:

COP = 20.1417 exp(-0.0171 * Twb)
Relative capacity = 1.265 - 0.0033 * Twb
Heat rejection rate = 14,334 Btu/hr per ton
Parasitic power = 0

VC chiller 3 (VC-3); -20°F evaporator:

COP = 5.863 exp(-0.0123 * Twb)
Relative capacity = 1.008 - 0.001 * Twb
Heat rejection rate = 17,444 Btu/hr per ton
Parasitic power = 0

Aqua-Ammonia (AA) Chillers

AA chiller 1 (AA-1); 20°F evaporator:

COP = 0.820 - 0.00374 * Twb
Relative capacity = 1.5625 - 0.0075 * Twb
Steam temperature = 252°F; pressure = 31 psia; h_{fg} = 944 Btu/lb
Heat rejection rate = 34,263 Btu/hr per ton
Parasitic power = 0

AA chiller 2 (AA-2); 32°F evaporator:

COP = 0.857 - 0.00372 * Twb
Relative capacity = 1.5625 - 0.0075 * Twb
Steam temperature = 240°F; pressure = 25 psia; h_{fg} = 952 Btu/lb
Heat rejection rate = 32,754 Btu/hr per ton
Parasitic power = 0

AA chiller 3 (AA-3); -20°F evaporator:

COP = 0.8672 - 0.00635 * Twb
Relative capacity = 1.5625 - 0.0075 * Twb
Steam temperature = 315°F; pressure = 83 psia; h_{fg} = 899 Btu/lb
Heat rejection rate = 42,691 Btu/hr per ton
Parasitic power = 0

Lithium Bromide (LiBr) Chillers

LiBr chiller 1 (LiBr-1); single-stage:

COP = 0.671 - 9.7 E-4 * $(Twb - 65)^{1.5}$
Relative capacity = 1.22 - 0.024 * (Twb - 65)
Steam temperature = 244°F; pressure = 27 psia; h_{fg} = 950 Btu/lb
Heat rejection rate = 29,200 Btu/hr per ton
Parasitic power = 0.0136 kW/ton

The minimum cooling water temperature allowable for single-stage LiBr chillers is 75°F. Therefore, Twb is limited to a minimum of 65°F.

LiBr chiller 2 (LiBr-2); double-stage, 32°F evaporator:

COP = 1.237 - 4.258 E-3 * $(Twb - 68)^{1.2}$
Relative capacity = 1.074 -0.0255 * (Twb - 68)
Steam temperature = 347°F; pressure = 130 psia; h_{fg} = 873 Btu/lb
Heat rejection rate = 21,800 Btu/hr per ton
Parasitic power = 0.028 kW/ton

The minimum cooling water temperature allowable for double-stage LiBr chillers is 78°F. Therefore, Twb is limited to a minimum of 68°F.

6.3 Storage

Storage is a significant component of a CTAC system, but can be effectively modeled with a few simple inputs. Thermal losses through the walls of the storage tank are typically no more than 1%, but other loss mechanisms have a much greater impact. Water storage relies on the creation of a thermocline to separate the hot and cold portions of the water. The difference between hot and cold temperatures, hence hot and cold buoyancies, is not much in this application of water/thermal storage. Therefore, losses across the thermocline are relatively high. The combined water storage efficiency was assumed to be 85% (ASHRAE 1987; EPRI 1988; Zwillenberg et al. 1991). Ice storage is really a mixture of water and ice. During discharge, warm return water is sprayed on top of the ice to facilitate melting. Here, mixing is not only beneficial, but required. During the charging cycle, however, periodic defrosting of the ice generator is required to release the ice from the evaporator surfaces. The defrost energy increases the refrigeration load and chiller capacity, but, unlike the water storage losses or the 1% loss through the tank walls, it does not affect the capacity of the storage tank. The defrost efficiency factor was presumed to be 85% (i.e., the chiller load and refrigeration capacity are increased by a factor of 1/0.85 to account for the defrost energy that must be removed), which is separate from the ice storage efficiency, which was presumed to be 99% (Babcock and Wilcox 1994).[(a)]

6.4 Fluid Circulation

CTAC systems include several fluid circulation loops, with the specific number and type depending on the type of chiller and storage. Included here are the ammonia loop connecting the chiller evaporator and other chiller components (except for LiBr chillers), the chilled water loop connecting LiBr chillers and storage with the inlet air cooling coils, the cooling water loop connecting the chiller with the cooling tower, the steam loop connecting the steam source with absorption-type chillers, and storage water loops connecting relatively warm storage water with evaporators in the ice generator or water chiller. The following pumping power rules-of-thumb were established from a collection of published data describing CTAC systems.

(a) Personal communication with representatives of Lincoln Electric System and Henry Vogt Company.

Ammonia loop power: 0.005 kW/ton or 2.0 E-5 kW/lb per hr ammonia

Chilled water loop power: 0.03 kW/gpm water

Cooling water loop power: 0.02 kW/gpm water

Steam loop power: 0.03 kW/gpm or 6.0 E-5 kW/lb per hr steam

Storage charging water loop power

Ice generation: 0.03 kW/ton
Chilled water generation: 0.02 kW/ton.

6.5 Evaporative Cooling

Evaporative cooling towers are used by every chiller to reject heat. In addition, evaporative inlet air cooling is an alternative to refrigerative inlet air cooling. Water consumption for both types of evaporators is calculated based on the thermodynamic requirement plus a 10% allowance for blowdown and carryover. The inlet air evaporator consumes 0.01 kW/gpm of water circulating through the cooler or 0.3 kW/gpm evaporating. The cooling tower fan consumes 1.25 kW/MMBtu per hour of heat rejected.

6.6 Auxiliary System Cooling

Cooling the inlet air of the turbine will increase its shaft power output, but care must be taken not to exceed the operating capabilities of other power plant components, specifically the generator, transformer, and lube oil system. Depending on the capabilities of these components, supplemental cooling may be required. This study assumed that supplemental cooling equal to 5% of the inlet air cooling load would be required (Babcock and Wilcox 1994).[(a)] Chillers and storage components were oversized by this factor and a separate cooling loop and heat exchanger was provided to cool the auxiliary equipment.

6.7 Turbine Inlet and Exit Excess Pressure Drop

Adding an inlet air cooling device increases the compressor pressure drop, resulting in decreased power output and increased heat rate. Although these losses are relatively small compared to the improvement in these same factors achieved from cooling the air, they are significant enough to warrant inclusion in the analysis. A similar effect occurs if a heat recovery steam generator is added to a simple-cycle turbine to provide steam for an absorption-type chiller. The increase in exhaust pressure

drop also reduces power output and increases heat rate, but by a relatively small amount. The specific assumptions used for excess inlet and exhaust pressure drop are shown below.

Inlet Air Cooling Coil Air-Side Pressure Drop:

Evaporative cooling: 1.0 in. of water
Refrigerative cooling to 42°F: 0.5 in. of water
Refrigerative cooling to 0°F: 1.3 in. of water

HRSG Air-Side Pressure Drop:

Maximum pressure drop for full heat recovery = 10 in. of water
Actual pressure drop = 10 in. * $(1000 - T_r)/(1000-300)$
T_r = calculated stack reject temperature, °F.

6.8 Combined-Cycle Steam Extraction Power Output Reduction

Absorption-type chillers require a source of low pressure steam. For simple-cycle power plants, an HRSG and related steam/condensate piping are added to provide the steam. In combined-cycle plants, steam is presumed to be extracted from the lower pressure stages of the steam turbine. While no additional fuel is required in either case (except for the effect of increasing the exhaust pressure when adding an HRSG), extracting steam from a combined-cycle plant's steam turbine does incur a loss of power production. The loss increases as the extraction pressure increases and is described by the following relationship:

kW/lb per hr steam = $-0.0079 + 0.0172 \ln (P)$.

where P = steam pressure, psia.

(a) Personal communication with representatives of Lincoln Electric System.

7.0 Cost Analysis

Initial capital cost and annual operating and maintenance cost estimating equations were developed for each cooling system component. Cost data acquired from equipment vendors, published literature describing inlet air cooling systems, and cost estimating reference books were the basis for developing the equations. The equations were applied to equipment sizing results to produce system cost estimates peculiar to each case evaluated.

All costs were estimated in mid-1994 dollars and are intended to include all costs associated with equipment and material purchase and installation. All components except for the complex compound chiller are mature, commercially available products. Complex compound chiller costs were estimated based on a mature production scenario, i.e., the first unit would probably be more expensive.

Note that no equations for estimating annual operating labor costs are presented. No incremental operating labor is presumed to be required based on advice received from personnel at Lincoln Electric System, which has operated an inlet air cooling system on one of their turbines since 1992.

7.1 Chiller Capital Costs

The most striking observation for chillers is the extremely high cost of the aqua-ammonia systems compared to the other options. Otherwise, all chillers increase in cost as the evaporator temperature is lowered, and double-stage units cost more than single-stage units. Absorption chiller costs are less sensitive to evaporator temperature than vapor compression chillers, resulting in a significant cost advantage for complex compound chillers at the lower evaporating temperature required for cooling inlet air to 0°F.

LiBr Absorption Chillers:

44°F water, single stage = $480/ton
44°F water, double stage = $650/ton

Complex Compound Absorption Chillers:

-20°F evaporator temperature, single stage = $490/ton
20°F evaporator temperature, single stage = $390/ton
20°F evaporator temperature, double stage = $550/ton
32°F evaporator temperature, single stage = $380/ton
32°F evaporator temperature, double stage = $530/ton
-20°F evaporator temperature, single stage storage = $310/ton-hr

Note: The storage version of the complex compound absorption chiller integrates the chiller and storage into a single component. The cost for both is based on the required storage size.

Aqua-Ammonia Absorption Chillers:

-20°F evaporator temperature = $3000/ton
20°F evaporator temperature = $2800/ton
32°F evaporator temperature = $2700/ton

Ammonia Vapor Compression Chillers:

-20°F evaporator temperature = $780/ton
20°F evaporator temperature = $520/ton
32°F evaporator temperature = $480/ton.

7.2 Storage Capital Costs

Except for the LiBr units, the chiller costs presented above do not include the cost of the evaporator. Without storage, refrigerant (ammonia) evaporation for the other units occurs directly in the inlet air cooling coils. Otherwise, refrigerant is evaporated in an ice or chilled water generator, depending on the storage type selected. Note that chilled water can be generated in a relatively simple and inexpensive, shell-and-tube type heat exchanger, while ice generation requires more complex and costly ice harvesting equipment. Water and ice storage tanks are essentially the same, but ice storage (actually an ice/water mixture) is less costly per ton-hr because of its higher chill energy density.

Ice Generator = $590/ton

Chilled Water Generator = $50/ton

Ice Storage = $163.6 * (ton-hrs)$^{0.75}$

Water Storage = $572.4 * (ton-hrs)$^{0.75}$.

7.3 Inlet Air Coil Capital Costs

The cooling coils include the coil, its enclosure, and foundation. For refrigerative cooling, the same cost is assumed for either water or ammonia fluid circulating through the coils. The higher cost for the 0°F air cooling coils is based on having three 50% capacity banks (with two cooling and one defrosting at any point in time) and the extra cost of dampers required to isolate each of the banks for defrosting.

Evaporative Cooling Coils = $3/kW of ISO turbine capacity

Refrigerative Cooling Coils:

For 42 or 52°F air = $0.20/UA
For 0°F air = $0.33/UA

where UA = heat transfer surface conductance, Btu/hr - °F.

7.4 Fluid Circulation Capital Costs

Individual estimates for pumps and piping were aggregated to develop the water circulation cost

equations shown below. Unit cost differences are directly related to the length of the piping run and pumping head. An insulation premium was also added for the steam/condensate pipe. Ammonia circulation costs are much lower than water circulation because ammonia passes through a phase change, so its mass flow rate is much lower for the same thermal duty. Note that ammonia circulation costs are doubled for 0°F inlet air temperature applications to allow for hot refrigerant bypass for defrosting.

Water Circulation:

Steam/condensate = $\$320,000 * (gpm/10,000)^{0.6}$
Chilled water = $\$290,000 * (gpm/10,000)^{0.6}$
Cooling water = $\$260,000 * (gpm/10,000)^{0.6}$
Storage charging water = $\$100,000 * (gpm/10,000)^{0.6}$.

Ammonia Circulation:

For 42 or 52°F air = $\$54,000 * (lb/hr/15,000)^{0.6}$
For 0°F air = $\$108,000 * (lb/hr/15,000)^{0.6}$.

7.5 Miscellaneous Equipment Capital Costs

Capital cost estimating equations for other cooling system components are listed below. Heat recovery steam generators are only used for simple-cycle power plants with absorption-type chillers.

Cooling Tower = $\$662,000 * (MMBtu/hr/192)^{0.876}$

System Controls = $250,000 (fixed for all systems)

Plant Electrical = $100,000 (fixed for all systems)

Transformer = $\$5,553 * (peak\ auxiliary\ kW)^{0.4359}$.

Heat Recovery Steam Generator:

$\$10,980 * (UA)^{0.8}$ +
$\$9,658 * (SF)$ +
$\$918 * (EF)^{1.2} * (1000 - TR)/300$

where UA = heat transfer surface conductance, Btu/s - °F
SF = steam flow, lb/s
EF = turbine exhaust flow, lb/s
TR = exhaust gas reject temperature after HRSG, °F.

Generator Cooling Loop Heat Exchanger = $\$865 * (HTA)^{0.511}$

where HTA = heat exchanger heat transfer area, square feet.

7.6 Annual Maintenance Costs

Annual maintenance costs were estimated as a percentage of initial capital cost for each inlet air cooling component. In addition, make-up water, for water consumed at either an evaporative inlet air cooler or a cooling tower, was estimated to cost $1/1000 gallons. Maintenance cost estimating percentages are listed below.

Component	Percentage
LiBr chiller	5.0
NH_3/H_2O chiller	4.0
Vapor compression chiller	4.0
Complex compound chiller	3.0
Ice generator	2.0
Chilled water generator	1.0
Ice storage	1.5
Water storage	1.5
Direct contact evaporative air cooler	2.0
Inlet air coil	
42°F and 52°F air	2.0
0°F air	3.0
Circulation (pumps and pipes)	3.0
Cooling tower	2.0
Controls	6.0
Plant electrical	2.0
Transformer	2.0
HRSG	3.0
Generator cooling heat exchanger	1.0

8.0 Economic Analysis

The economic analysis compared the benefit of incremental power and energy production with the cost of owning and operating the inlet air cooling system. With costs and benefits denominated in dollars, economic feasibility was determined by calculating the net present value (NPV) of the investment. The preferred cooling systems were, of course, those with the highest NPVs for a given set of application conditions. Refrigerative and evaporative cooling systems were compared against reference plants that were uncooled, and refrigerative cooling systems were also compared against reference plants that were already evaporatively cooled.

The value of incremental power and energy production varies depending on ownership perspective. From a utility perspective, value is defined by the alternative to inlet air cooling for providing additional capacity. The most likely alternative would be to build a new turbine without refrigerative inlet air cooling. The value of incremental power would be equal to the expected fixed costs (capital and fixed operation and maintenance [O&M]), while the value of incremental energy would be equal to the expected variable costs (fuel and variable O&M). From an industrial perspective, the value of incremental power and energy is set more specifically by the prevailing contract with the utility for power purchases or sales. This contract will also probably contain requirements that will dictate the period the inlet air cooling system will need to operate to achieve the maximum credit for increased generating capacity. This study takes the utility perspective.

Obviously, the value of incremental power and energy not only varies with the general differences between a utility and an industrial perspective, but also because of all the differences that exist between individual utilities and industrial companies. While this will have a significant effect on the magnitudes of the NPVs calculated for each case, it shouldn't significantly affect the rankings of the NPVs for each case. Given the uncertainty in the values for incremental power and energy, the capital cost per incremental power output has also been calculated for each case. While this is an inferior investment decision criteria, it is independent of any assumed values for incremental power and energy and can be quickly compared to the cost of adding capacity via another means.

8.1 Incremental Power and Energy Calculation

Incremental power and energy production is simply the difference between the cooled and uncooled output of the power plant. The difference increases as the ambient temperature climbs. For incremental energy, hourly differences are aggregated over all cooling hours to calculate the annual incremental energy production. For incremental power, the figure of interest to utilities is the increase in capacity (or incremental power) occurring when the utility experiences its peak power demand. For most utilities, peak demand occurs during the summer, and is driven by air conditioning loads. While peak incremental power production will always occur at the peak dry-bulb temperature, peak air conditioning loads and peak power demand correspond more closely to peak air enthalpy conditions. Thus, the peak hourly air enthalpy was selected as the design point for calculating incremental power. Peak enthalpy tends to occur at higher wet-bulb, but lower dry-bulb temperatures. As a result, the incremental power calculated is less than if the basis were the peak dry-bulb temperature. A utility-specific analysis would allow the timing of peak demand to

be more accurately defined, but was beyond the scope of this study.

8.2 Incremental Power and Energy Value

Taking the utility perspective described above, the value of incremental power and energy depends mostly on the cost and performance characteristics of the power plant being displaced by inlet air cooling and the peak power demand temperature and humidity conditions. Power plant cost and performance characteristics and utility economic assumptions presented in Tables 8.1 and 8.2 were coupled with the levelized cost methodology described in Brown et al. (1987) to calculate the incremental power and energy figures shown in Table 8.3. The same economic assumptions and levelized cost methodology were also used to calculate the NPV for each case once the incremental costs and revenues (from incremental power and energy production) were calculated. The key inputs affecting the differences in the value of power and energy are the plant capital cost and heat rate. Climate also has an interesting impact. The severe peak demand conditions in Phoenix reduce power plant output, causing the cost per kW to rise. The opposite is found for a mild climate like that found in San Francisco.

In general, the power and energy values shown in Table 8.3 were applied to the incremental power and energy generated by each plant type while the cooling system was operating. However, when the combined-cycle plant was cooled on a simple-cycle schedule, its incremental power and energy production were valued at the industrial turbine rates. Off-peak energy to drive vapor compression chillers for power plants running less than 24 hours/day was presumed to come from a coal-fired power plant. The value of this energy was set at $0.0195/kWh based on a heat rate of 10,500 Btu/kWh, variable O&M equal to $0.004/kWh, and coal at $1.36/MMBtu in 1994 dollars, escalating at 0.5% per year in excess of general inflation.

Table 8.1. Power Plant Cost and Performance Assumptions

	Industrial Simple-Cycle	Aeroderivative Simple-Cycle	Industrial Combined-Cycle
Capital cost, $/kW	449	749	602
Fixed annual O&M, $/kW/yr	10.9	15.6	27.6
Variable O&M, $/kWh	0.0002	0.0002	0.0008
Heat rate, Btu/kWh	11,700	10,000	7,300
Source: Technical Assessment Guide (EPRI 1993)			

Table 8.2. Power Plant Economic Assumptions

Economic life	30 years
Depreciable life	20 years
Income tax rate	38%
Nominal discount rate	9.25%
General inflation rate	4.1%
Property tax and insurance	2%
Price year	1994
1st operating year	1997
Construction period	1-2 years
Natural gas price	$2.25/MMBtu
Natural gas inflation rate	6.4%

Sources: Technical Assessment Guide (EPRI 1993)
Annual Energy Outlook 1995 (EIA 1995a)
Monthly Energy Review (for March 1995) (EIA 1995b)

Table 8.3. Incremental Power and Energy Values

City	Indust. Turbine		Aero. Turbine		Combined-Cycle	
	$/kW	$/kWh	$/kW	$/kWh	$/kW	$/kWh
Houston	63	0.038	110	0.032	98	0.024
Minneapolis	64	0.037	109	0.032	98	0.024
Nashville	64	0.038	110	0.032	98	0.024
Phoenix	69	0.039	123	0.033	105	0.024
San Francisco	62	0.037	106	0.031	95	0.024

9.0 Results

The net present value (NPV) and capital cost per incremental kW ($/kW) were calculated for about 3500 cases. This includes 44 industrial turbine simple-cycle cases, 24 aeroderivative turbine simple-cycle cases and 288 combined-cycle cases, with each evaluated for five different cities and with the reference case being either an uncooled or evaporatively cooled plant. A complete listing of these results is presented in Appendix C. This section presents and discusses selected results.

9.1 Nomenclature

Each case is distinguished by its case number, chiller type, plant operating hours per day, cooling hours per day, and storage type. The case number uniquely defines one of 356 cases. Industrial turbine cases begin with an "i," aeroderivative turbine cases with an "a," and combined-cycle cases with a "c,", "d," "e," or "f." The four combined-cycle groups represent different combinations of plant operating hours and cooling system operating hours. The second letter of the case number, an "e" or an "r," designates a case as being evaporatively or refrigeratively cooled. The chiller type can be either aqua-ammonia (AA), complex compound (CC), lithium bromide (LiBr), or vapor compression (VC). The chiller numbers refer to the specific versions of each chiller type defined in Section 6.0. Plant operating hours and cooling system operating hours are designated by two numbers separated by a backslash. For example, "12/4" would mean the plant runs 12 hours per day while the cooling system runs 4 hours per day. The specific power plant and cooling system schedule assumptions were defined in Section 4.0. The storage type is defined by a four letter code. The first letter refers to ice (I) or water (W) storage, the second letter to a daily (D) or weekly (W) cycle, and the final two letters to a load-shifting (LS) or load-leveling (LL) design basis.

9.2 Industrial Turbine

NPV and $/kW results for cooling an industrial turbine in a simple-cycle application are presented in Table 9.1 for the Nashville climate. The magnitude of the NPV and $/kW figures are affected by the climate, but technology rankings generally are not. A comparison of results across climates is presented later.

A quick review of the figures presented in Table 9.1 identifies the aqua-ammonia systems as the only ones with negative NPVs for the uncooled base and installed costs that are far too high for this technology to be cost-effective. The poor results for the aqua-ammonia system stem directly from the high capital cost for its chiller. Evaporative cooling yields the lowest $/kW for the uncooled base by more than a factor of two, but its NPV is relatively low. Thus, some form of refrigerative cooling is preferred.

Table 9.1. Industrial Turbine Results-Nashville

Case	Chiller	OpHrs	Storage	Uncooled Base		Evap. Cooled Base	
				$/kW	NPV, $M	$/kW	NPV, $M
ie1	None	4/4	None	68	2.86	--	--
ie2	None	8/8	None	68	3.10	--	--
ir1	AA-2	4/4	None	1075	-14.00	1400	-17.00
ir2	AA-2	8/8	None	1075	-13.40	1400	-16.60
ir3	CC-2	4/4	None	353	3.75	462	0.76
ir4	CC-2	8/8	None	353	4.36	462	1.13
ir5	LiBr-1	4/4	None	397	2.05	564	-0.95
ir6	LiBr-1	8/8	None	397	2.53	564	-0.71
ir7	VC-2	4/4	None	307	4.07	424	1.08
ir8	VC-2	8/8	None	307	4.51	424	1.27
ir9	VC-1	4/4	IWLS	158	9.09	203	6.10
ir10	VC-1	4/4	IWLL	155	8.92	200	5.92
ir11	VC-1	4/4	IDLS	182	8.48	234	5.48
ir12	VC-1	4/4	IDLL	175	8.28	227	5.29
ir13	VC-1	8/8	IWLS	249	7.49	319	4.25
ir14	VC-1	8/8	IWLL	225	7.59	293	4.35
ir15	VC-1	8/8	IDLS	323	5.64	413	2.40
ir16	VC-1	8/8	IDLL	271	6.26	357	3.02
ir17	VC-2	4/4	WWLS	221	6.25	305	3.26
ir18	VC-2	4/4	WWLL	214	6.24	298	3.24
ir19	VC-2	4/4	WDLS	200	6.56	276	3.56
ir20	VC-2	4/4	WDLL	192	6.46	269	3.47
ir21	VC-2	8/8	WWLS	339	4.60	468	1.36
ir22	VC-2	8/8	WWLL	305	4.99	428	1.75
ir23	VC-2	8/8	WDLS	311	4.91	430	1.68
ir24	VC-2	8/8	WDLL	268	5.32	382	2.09
ir25	CC-2/CC-5	4/4	None	413	4.48	474	1.48
ir26	CC-2/CC-5	8/8	None	413	5.78	474	2.54
ir27	AA-2/AA-3	4/4	None	1126	-28.00	1289	-30.70
ir28	AA-2/AA-3	8/8	None	1126	-26.00	1289	-29.59
ir29	VC-2/VC-3	4/4	None	367	4.20	441	1.20
ir30	VC-2/VC-3	8/8	None	367	4.50	441	1.26
ir31	VC-1/VC-3	4/4	IWLS	254	9.60	297	6.60
ir32	VC-1/VC-3	8/8	IWLS	316	7.76	369	4.52
ir33	VC-1/VC-3	4/4	IWLL	253	9.91	297	6.92
ir34	VC-1/VC-3	8/8	IWLL	303	8.76	357	5.52
ir35	VC-1/VC-3	4/4	IDLS	270	8.95	316	5.96

Case	Chiller	OpHrs	Storage	Uncooled Base		Evap. Cooled Base	
				$/kW	NPV, $M	$/kW	NPV, $M
ir36	VC-1/VC-3	8/8	IDLS	368	5.79	430	2.55
ir37	VC-1/VC-3	4/4	IDLL	268	9.22	315	6.23
ir38	VC-1/VC-3	8/8	IDLL	337	7.29	399	4.05
ir39	VC-2/CC-5	4/4	None	410	4.41	478	1.08
ir40	VC-2/CC-5	8/8	None	410	5.10	478	1.87
ir41	VC-1/CC-5	4/4	IWLL	309	9.27	353	6.27
ir42	VC-1/CC-5	8/8	IWLL	352	8.42	404	5.18

For the single chiller cases without storage, ammonia vapor compression cooling produces the lowest $/kWs and highest NPVs, although the complex compound systems are competitive on an NPV basis. Note that the $/kW for a given chiller type is the same for either 4 or 8 cooling hours per day. This result occurs because the peak design condition is the same for the two periods. The NPV increases for the 8-hour cases because the marginal cost of running the system is less than the marginal benefit of improving the heat rate over an extended operating period.

The importance of storage for reducing $/kW and increasing NPV are clearly shown by comparing the results of cases ir9 through ir24 with ir1 through ir8. Comparing the best cases with and without storage shows that storage can reduce the $/kW and increase the NPV by a factor of two or more. Storage reduces the size and cost of the most expensive cooling system component (the chiller), while also reducing the consumption of on-peak parasitic power for vapor compression chillers. Note that storage was only considered for vapor compression systems. With cooling required for the duration of the plant operating period, there is no "off-peak" waste heat source available to drive absorption-type chillers. Natural gas firing was presumed to be too expensive, but this assumption should probably be verified by additional analysis.

Closer examination of cases ir9 through ir24 shows that ice storage is preferred to water storage and that weekly storage is preferred for ice while daily storage is preferred for water. Weekly storage allows the greatest reduction in chiller size and cost at the expense of increased storage size and cost. The high cost of the ice generator compared to the chilled water generator drives this difference in cycle preference. A load-leveling storage design results in the lowest $/kW because chiller size is reduced the most, but load-shifting reduces on-peak parasitic power consumption the most, and sometimes has the higher NPV. Storage is most cost-effective for plants being cooled only a few hours a day, where the benefits of downsizing the chiller are maximized.

Cases ir25 through ir42 examine the benefits of cooling the inlet air to 0°F. Various ice storage configurations are incorporated on the first cooling stage for some cases. Cooling to 0°F increases the $/kW, but also increases the NPV. The increased $/kW can be attributed to a linear increase in the generating capacity and heat rate benefits, but increasing unit costs and decreasing COPs at lower evaporator temperatures for chillers. Increasing chiller unit costs are exacerbated by the lack of storage for the subfreezing second stage. Various brines or other media could potentially be used for storage at this lower temperature, but were presumed to be too costly. Additional analysis of prospective low-temperature

storage media may be warranted to test this assumption.

Refrigerative cooling is less attractive, but still cost-effective if an evaporative cooling system is already installed. In this case, the incremental power and energy production is reduced, while the refrigerative cooling system cost remains the same. Note that evaporative cooling followed by refrigerative cooling would not reduce the refrigerative cooling load, hence the size and cost of the refrigerative cooling system. Evaporative cooling just exchanges sensible cooling load for latent cooling load. Refrigerative cooling followed by evaporative cooling makes no sense either, because the air is almost always saturated after refrigerative cooling.[(a)]

9.3 Aeroderivative Turbine

Inlet air cooling results for the aeroderivative turbine are presented in Table 9.2, again for the Nashville climate. In general, the results are similar to that for the industrial turbine. For example, the preferred cooling system uses ammonia vapor compression chillers and weekly ice storage. There are several interesting differences, however. Note that 0°F inlet air (two-chiller) cases were not evaluated because power output is constant for below freezing temperatures for this type of turbine. The $/kW figures have dropped by about 10-20% for the non-storage cases, which can be attributed directly to the greater sensitivity of aeroderivative turbine power output to inlet air temperature. This effect is masked for the storage cases for two reasons: 1) the benefits of downsizing the chiller while adding storage are larger for the industrial turbine because it was assumed to be a larger machine, and 2) adding storage increases peak power output per ISO kW output the same for both turbine types, which benefits the industrial turbine the most on a percentage basis. Finally, the increase in NPVs for the aeroderivative cases is driven by its higher value per incremental kW.

9.4 Combined-Cycle

The combined-cycle analysis was more complicated than the industrial or aeroderivative simple-cycle analyses because two plant operating schedules were evaluated and the cooling system operating cycle was allowed to be different than the plant operating cycle. When the cooling period is shorter than the plant operating period, storage is possible for the heat-activated absorption-type chillers. Thus, the number of cases evaluated increased significantly. Not all cases evaluated are presented here to focus on the more relevant results. In particular, the aqua-ammonia results have been excluded because of their extremely poor economics. A complete list of the results for all cases is presented in Appendix C.

(a) These comments apply to direct evaporative cooling and not to indirect methods of evaporative cooling.

Table 9.2. Aeroderivative Turbine Results-Nashville

Case	Chiller	OpHrs	Storage	Uncooled Base		Evap. Cooled Base	
				$/kW	NPV, $M	$/kW	NPV, $M
ae1	None	4/4	None	52	2.91	--	--
ae2	None	8/8	None	52	3.00	--	--
ar1	AA-2	4/4	None	881	-6.92	1153	-3.63
ar2	AA-2	8/8	None	881	-4.31	1153	-3.46
ar3	CC-2	4/4	None	313	7.66	411	4.73
ar4	CC-2	8/8	None	313	7.91	411	4.88
ar5	LiBr-1	4/4	None	350	5.60	500	2.67
ar6	LiBr-1	8/8	None	350	5.79	500	2.76
ar7	VC-2	4/4	None	273	7.19	374	4.26
ar8	VC-2	8/8	None	273	7.36	374	4.35
ar9	VC-1	4/4	IWLS	159	10.41	206	7.47
ar10	VC-1	4/4	IWLL	156	10.21	203	7.27
ar11	VC-1	4/4	IDLS	179	10.12	231	7.18
ar12	VC-1	4/4	IDLL	172	9.84	224	6.90
ar13	VC-1	8/8	IWLS	238	9.49	308	6.46
ar14	VC-1	8/8	IWLL	217	9.34	284	6.31
ar15	VC-1	8/8	IDLS	298	8.61	385	5.58
ar16	VC-1	8/8	IDLL	251	8.57	333	5.54
ar17	VC-2	4/4	WWLS	225	7.66	314	4.72
ar18	VC-2	4/4	WWLL	219	7.59	307	4.65
ar19	VC-2	4/4	WDLS	203	7.85	284	4.91
ar20	VC-2	4/4	WDLL	196	7.70	277	4.76
ar21	VC-2	8/8	WWLS	335	6.65	468	3.62
ar22	VC-2	8/8	WWLL	302	6.75	427	3.72
ar23	VC-2	8/8	WDLS	304	6.89	424	3.86
ar24	VC-2	8/8	WDLL	263	6.91	376	3.88

9.4.1 Intermediate Duty Plant; Cooling Hours Less Than Plant Hours

The intermediate duty combined-cycle plant operates 12 hours per day, 5 days per week. The results for the best cases (lowest $/kW and/or highest NPV) for each combination of chiller type and daily cooling period are presented in Table 9.3 for a combined-cycle plant operating 12 hours per day while the cooling system operates only 4 or 8 hours per day. Note that 4 and 8 hour cooling follows the same schedule as the simple-cycle plants and the incremental power and energy is valued the same as the simple-cycle industrial turbine application.

Inlet air cooling affects the performance of the steam turbine as well as the combustion turbine in a combined-cycle plant. Inlet air cooling reduces the exhaust temperature, but increases the exhaust flow rate, with the net result being an increase in steam turbine power output in addition to the increase in combustion turbine power output. As a result, $/kW results are generally lower than for corresponding industrial turbine simple-cycle cases. NPV results are generally higher for the combined-cycle cases, but this is primarily because of the larger size of the combined-cycle plant.

Table 9.3. Combined-Cycle Results I-Nashville

Case	Chiller	OpHrs	Storage	Uncooled Base		Evap. Cooled Base	
				$/kW	NPV, $M	$/kW	NPV, $M
ce1	None	12/4	None	65	4.40	--	--
ce2	None	12/8	None	65	4.90	--	--
cr14	LiBr-2	12/4	WDLL	215	8.87	311	4.19
cr16	LiBr-2	12/8	WDLL	358	4.50	555	-0.19
cr19	VC-1	12/4	IWLS	127	16.61	163	11.42
cr20	VC-1	12/4	IWLL	123	16.36	159	11.17
cr24	VC-1	12/8	IWLL	180	15.28	235	10.09
cr70	CC-2	12/4	WDLL	195	9.72	281	5.04
cr74	CC-1	12/4	IDLL	214	11.48	284	6.80
cr68	CC-2	12/8	None	266	8.74	374	4.06
cr57	VC-1/VC-3	12/4	IWLL	201	21.41	234	16.20
cr58	VC-1/VC-3	12/8	IWLL	240	21.72	281	16.50
cr81	CC-1/CC-6	12/4	IDLL	227	21.04	261	16.33
cr78	CC-2/CC-5	12/8	None	314	9.80	385	5.10
cr82	CC-1/CC-6	12/8	IDLL	339	14.80	397	10.09
cr85	VC-1/CC-6	12/4	IWLL	213	23.09	244	18.39
cr84	VC-2/CC-5	12/8	None	299	11.95	363	7.24
cr86	VC-2/CC-6	12/8	IWLL	380	13.37	438	8.66

The preferred cooling system is still an ammonia vapor compression cooling system with weekly ice storage, which has the lowest $/kW and highest NPV for the above-freezing (single chiller) applications (cr20 and cr24). For 0°F inlet air applications, ammonia vapor compression also has the lowest $/kW, but a combination vapor compression and complex compound system (cr85) has the highest NPV when cooling for only 4 hours per day.

Storage improves the economics of the heat-activated chillers, but not as much as it improves the economics of the vapor compression chiller because heat is still only available for 12 hours per day. Thus, a load-leveling storage design is generally preferred to a load-shifting design for heat-activated chillers for this application. In addition, storage looks much more attractive when cooling only 4 hours per day, as would be expected.

Single-stage (CC-1 and CC-2) and double-stage (CC-3 and CC-4) complex compound chillers were investigated. Double-stage chillers are more efficient, but cost more to build and consume higher pressure steam, which increases the extraction loss effect in the steam turbine. The results of this analysis show the single-stage complex compound chillers to be preferred, which suggests that single-stage lithium bromide chillers may also be better than the double-stage lithium bromide chillers (cr11 through cr16) that were evaluated.

9.4.2 Intermediate Duty Plant; Cooling Hours Equal to Plant Hours

Table 9.4 presents the results for cooling and operating a combined-cycle power plant 12 hours per day. Again, the ammonia vapor compression system with weekly ice storage is the preferred system for both above-freezing and 0°F inlet air temperature applications. Storage results in only a moderate advantage for the vapor compression system because the chiller size can at most be cut in half. Storage is not possible for the heat-activated systems.

Note that the $/kW for the evaporative cooler has increased from $68 for cooling 4 or 8 hours per day to $77 for 12 hours per day. This occurs because the design condition (maximum ambient air enthalpy) for the longer cooling cycle is slightly more humid.

9.4.3 Baseload Duty Plant; Cooling Hours Less Than Plant Hours

The baseload duty combined-cycle plant operates 24 hours per day, 7 days per week. The results for the best cases (lowest $/kW and/or highest NPV) for each combination of chiller type and daily cooling period are presented in Table 9.5. Again, note that 4- and 8-hour cooling follows the same schedule as the simple-cycle plants, and the incremental power and energy is valued the same as the simple-cycle industrial turbine application.

Table 9.4. Combined-Cycle Results II-Nashville

Case	Chiller	OpHrs	Storage	Uncooled Base		Evap. Cooled Base	
				$/kW	NPV, $M	$/kW	NPV, $M
ce3	None	12/12	None	77	5.95	--	--
dr1	CC-4	12/12	None	435	4.45	656	-1.66
dr2	LiBr-2	12/12	None	520	3.62	810	-2.49
dr4	VC-2	12/12	None	265	15.81	345	9.70
dr5	VC-1	12/12	IWLS	309	18.31	381	12.20
dr6	VC-1	12/12	IWLL	246	18.98	310	12.88
dr7	VC-1	12/12	IDLS	465	12.10	573	5.99
dr8	VC-1	12/12	IDLL	310	15.69	395	9.59
dr9	VC-2	12/12	WWLS	396	12.44	521	6.33
dr10	VC-2	12/12	WWLL	320	13.68	430	7.58
dr11	VC-2	12/12	WDLS	386	12.23	507	6.13
dr12	VC-2	12/12	WDLL	274	13.95	374	7.85
dr13	CC-4/CC-5	12/12	None	430	4.79	533	-1.66
dr15	VC-2/VC-3	12/12	None	301	23.73	349	17.59
dr16	VC-1/VC-3	12/12	IWLL	285	27.89	327	21.74
dr17	CC-2	12/12	None	287	12.90	384	6.79
dr18	CC-2/CC-5	12/12	None	330	13.66	392	7.52
dr19	VC-2/CC-5	12/12	None	314	16.71	370	10.57
dr20	VC-2/CC-5	12/12	IWLL	297	19.83	344	13.69

Operation of the plant 24 hours per day significantly improves the economics of heat-activated chiller systems because storage systems can be charged around the clock. Thus, the $/kW is lower and the NPV is higher for most heat-activated chiller systems when the results for the best cases from the 24 hour per day combined-cycle plant are compared to the best cases from the 12 hour per day combined-cycle plant. On the other hand, the results for the vapor compression systems are about the same. Still, the ammonia vapor compression systems still produce the lowest $/kWs and highest NPVs for above-freezing and 0°F temperatures at all daily cooling hours. The complex compound systems produce similar, but never better results. The lithium bromide systems are less attractive, but the single-stage lithium bromide chiller should also be run for this application to see if it provides better results.

Table 9.5. Combined-Cycle Results III-Nashville

Case	Chiller	OpHrs	Storage	Uncooled Base		Evap. Cooled Base	
				$/kW	NPV, $M	$/kW	NPV, $M
ce4	None	24/4	None	65	3.71	--	--
ce5	None	24/8	None	65	4.21	--	--
ce6	None	24/12	None	77	5.89	--	--
er19	LiBr-2	24/4	WDLS	168	11.71	233	7.71
er20	LiBr-2	24/4	WDLL	162	11.39	229	7.39
er22	LiBr-2	24/8	WDLL	240	9.80	349	5.30
er24	LiBr-2	24/12	WDLL	324	10.25	453	4.20
er28	VC-1	24/4	IWLS	128	17.14	165	13.14
er29	VC-1	24/4	IWLL	123	16.95	159	12.95
er33	VC-1	24/8	IWLL	180	16.68	235	12.18
er27	VC-2	24/12	None	265	15.78	345	9.72
er142	CC-1	24/4	IWLS	129	16.83	166	12.83
er143	CC-1	24/4	IWLL	125	16.51	162	12.51
er145	CC-1	24/8	IWLL	183	15.86	239	11.86
er126	CC-2	24/12	None	287	12.52	384	6.46
er138	CC-1	24/12	IDLL	315	13.08	406	7.02
er115	VC-1/VC-3	24/4	IWLL	201	23.26	234	19.24
er116	VC-1/VC-3	24/8	IWLL	240	26.15	281	21.62
er75	VC-2/VC-3	24/12	None	301	23.10	349	17.01
er154	CC-1/CC-6	24/4	IDLS	215	25.63	244	21.60
er140	CC-2/CC-5	24/8	None	314	14.88	385	10.85
er155	CC-1/CC-6	24/8	IDLS	396	17.53	451	13.51
er141	CC-2/CC-5	24/12	None	330	11.05	392	4.96
er163	VC-1/CC-6	24/4	IWLL	207	26.00	236	21.98
er161	VC-2/CC-5	24/8	None	299	17.02	363	13.00
er162	VC-2/CC-5	24/12	None	314	14.48	370	8.38

9.4.4 Baseload Duty Plant; Cooling Hours Equal to Plant Hours

Storage was not considered when both the plant and cooling system operate 24 hours per day, so the number of cases considered for this application was limited. It would be possible to incorporate load-leveling storage, but the results in Table 9.5 show that storage is not usually desirable even when the inlet air is being cooled only 12 hours per day.

The results for cooling 24 hours per day are shown in Table 9.6. While inlet air cooling is cost-effective, the results are not as attractive as when storage can be effectively incorporated. Otherwise, the technology rankings are the same as seen for the other combined-cycle applications.

9.5 Climate Impacts

The peak hourly design conditions have a significant impact on the economics of inlet air cooling. Design conditions are affected by the climate and the cooling period selected, as shown in Table 9.7, which presents the design dry-bulb and wet-bulb temperatures for each of the combinations of plant operating hours, cooling operating hours, and city evaluated. As described in Section 8.0, the peak hourly enthalpy was selected as the design condition, which is essentially the same as the hour with the peak wet-bulb temperature.

The impact of climate and the design conditions on $/kW and NPV is illustrated in Table 9.8 for selected cases. In general, climate affects the magnitude of the $/kW and NPV results, but does not affect the ranking of the alternative cooling systems. Both evaporative and refrigerative cooling are most attractive in warm, dry climates. Note that turbine generating capacity and heat rate are driven by the dry-bulb temperature, but the cooling load is driven by the wet-bulb temperature. $/kW results track directly with the design conditions, while NPV results are also affected by average operating conditions. Thus, Houston looks the poorest, while Phoenix or San Francisco look the best in terms of $/kW. On the other hand, San Francisco isn't nearly as attractive compared to Phoenix on a NPV basis. Yet,

Table 9.6. Combined-Cycle Results IV-Nashville

Case	Chiller	OpHrs	Storage	Uncooled Base		Evap. Cooled Base	
				$/kW	NPV, $M	$/kW	NPV, $M
ce7	None	24/24	None	77	6.17	--	--
fr2	LiBr-2	24/24	None	520	2.71	857	-6.09
fr8	CC-2	24/24	None	287	11.49	389	3.61
fr4	VC-2	24/24	None	265	15.78	352	8.99
fr7	VC-2/VC-3	24/24	None	301	22.00	354	13.43
fr9	CC-2/CC-5	24/24	None	330	3.29	397	-6.55
fr10	VC-2/CC-5	24/24	None	314	7.83	375	-0.81

Table 9.7. Design Temperature Conditions

Plant Hour/Cooling Hour Pairs	Dry-Bulb/Wet-Bulb Temperature Pairs (F)				
	Nashville	Houston	Minneapolis	Phoenix	San Francisco
4/4	92/79	91/80	92/78	100/75	71/63
8/8	92/79	87/80	90/79	92/75	84/65
12/4	92/79	91/80	92/78	100/75	71/63
12/8	92/79	87/80	90/79	92/75	84/65
12/12	91/80	85/80	90/79	92/75	80/66
24/4	92/79	91/80	92/78	100/75	71/63
24/8	92/79	87/80	90/79	92/75	84/65
24/12	91/80	82/81	90/79	92/75	85/67
24/24	91/80	82/81	90/79	92/75	85/67

Table 9.8. Climate Impacts on Economics

Case	Chiller	OpHrs	Storage	Nashville	Houston	Minneapolis	Phoenix	San Francisco
				$/kW				
ce1	None	12/4	None	65	82	61	34	119
ce2	None	12/8	None	65	145	78	54	44
ce3	None	12/12	None	77	236	78	54	60
ce4	None	24/4	None	65	82	61	34	119
ce5	None	24/8	None	65	145	78	54	44
ce6	None	24/12	None	77	(a)	78	54	46
ce7	None	24/24	None	77	(a)	78	54	46
cr17	VC-2	12/4	None	245	264	234	169	204
cr18	VC-2	12/8	None	245	296	259	203	148
cr20	VC-1	12/4	IWLL	123	136	114	96	127
cr24	VC-1	12/8	IWLL	180	223	172	167	115
				NPV ($M)				
ce1	None	12/4	None	4.40	3.36	4.71	10.28	1.98
ce2	None	12/8	None	4.90	2.21	4.12	8.21	6.62
ce3	None	12/12	None	5.95	1.89	5.77	10.09	7.38
ce4	None	24/4	None	3.71	2.69	4.03	9.61	1.31
ce5	None	24/8	None	4.21	1.54	3.44	7.54	5.94
ce6	None	24/12	None	5.89	-0.53	5.62	10.39	9.47
ce7	None	24/24	None	6.17	-0.24	5.84	11.55	9.69
cr17	VC-2	12/4	None	9.46	8.45	9.57	17.07	6.60
cr18	VC-2	12/8	None	10.82	8.17	9.34	15.88	12.14
cr20	VC-1	12/4	IWLL	16.36	15.53	16.30	22.54	9.30
cr24	VC-1	12/8	IWLL	15.28	12.48	14.41	19.10	14.30

(a) At the design conditions for this case (82°F, dry-bulb and 81°F, wet-bulb) evaporative cooling provides essentially no benefit, so its $/kW approaches infinity.

its important to note that the results show inlet air cooling to be economically attractive for all of the climates investigated, which represent the majority of the climate range found in the U.S. outside of Alaska.

9.6 Net Present Value Components

The NPVs for selected combined-cycle cases are broken into revenue (positive cash flows) and cost (negative cash flows) components in Table 9.9. As expected, incremental power output is always the principal benefit, but incremental energy output is also quite important. Incremental energy output increases in importance as the number of cooling hours per day increases, although this effect is partly masked by a change in the value of incremental power and energy when comparing 4 or 8 hour per day cases to the 12 hour per day cases. Incremental energy output is also generally more important for cases that cool the inlet air to 0°F (those with two chillers). Costs are dominated by fuel and capital. Even though the heat rate decreases with inlet air cooling, the power output increases more, so fuel consumption is greater. As expected, fuel consumption costs are greatest for longer daily cooling hours.

Table 9.9. Net Present Value Components

Case	Chiller	OpHrs	Storage	Revenue Fractions		Cost Fractions			
				On-peak kW	On-peak kWh	Off-Peak kwH	O&M	Fuel	Capital
ce1	None	12/4	None	0.88	0.12	0.00	0.10	0.43	0.47
ce2	None	12/8	None	0.74	0.26	0.00	0.07	0.63	0.30
er20	LiBr-2	24/4	WDLL	0.81	0.19	0.06	0.16	0.27	0.50
er22	LiBr-2	24/8	WDLL	0.66	0.34	0.07	0.15	0.34	0.43
er24	LiBr-2	24/12	WDLL	0.71	0.29	0.10	0.13	0.41	0.36
er29	VC-1	24/4	IWLL	0.77	0.23	0.06	0.12	0.38	0.43
er33	VC-1	24/8	IWLL	0.62	0.38	0.06	0.11	0.46	0.37
er27	VC-2	24/12	None	0.67	0.33	0.00	0.12	0.57	0.31
er143	CC-1	24/4	IWLL	0.77	0.23	0.08	0.13	0.37	0.42
er145	CC-1	24/8	IWLL	0.62	0.38	0.08	0.12	0.44	0.36
er138	CC-1	24/12	IDLL	0.66	0.34	0.10	0.12	0.46	0.31
er115	VC-1/VC-3	24/4	IWLL	0.70	0.30	0.07	0.15	0.34	0.45
er116	VC-1/VC-3	24/8	IWLL	0.53	0.47	0.07	0.12	0.46	0.35
er75	VC-2/VC-3	24/12	None	0.58	0.42	0.00	0.11	0.64	0.25
er154	CC-1/CC-6	24/4	IDLS	0.74	0.26	0.04	0.18	0.29	0.49
er155	CC-1/CC-6	24/8	IDLS	0.57	0.43	0.04	0.18	0.31	0.47
er141	CC-2/CC-5	24/12	None	0.59	0.41	0.00	0.13	0.64	0.24
er163	VC-1/CC-5	24/4	IWLL	0.73	0.27	0.04	0.17	0.31	0.48
er161	VC-2/CC-5	24/8	None	0.53	0.47	0.00	0.15	0.48	0.37
er162	VC-2/CC-5	24/12	None	0.59	0.41	0.00	0.12	0.64	0.24

10.0 Conclusions and Recommendations

The principal objectives of this study were presented in the Introduction. These are repeated below along with the corresponding conclusions.

1. Identify the preferred CTAC technology as a function of application conditions from currently available cooling technologies.

   Ammonia vapor compression chillers with weekly ice storage were generally found to be preferred for most application conditions.

2. Identify application conditions where currently available cooling technology is not cost-effective.

   CTAC was found to be more cost-effective than simply building additional uncooled power plant capacity for all application conditions investigated. This included relatively moderate climates such as San Francisco and applications where storage was limited (e.g, cooling 24 hours per day).

3. Determine the potential attractiveness of the complex compound (ammoniated salt solid/vapor "absorption") cooling system.

   The complex compound systems were found to be competitive, but generally not quite as attractive as ammonia vapor compression systems. Complex compound systems were most competitive (within a few % of the vapor compression systems' $/kW and NPV figures) for applications with combined-cycle plants running 24 hours per day, which allowed storage for heat-activated chillers to be charged up to 24 hours per day.

4. Determine the potential attractiveness of cooling combustion turbine inlet air to 0°F.

   Cooling the inlet air to 0°F was found to be more cost-effective (higher NPV, although the $/kW was also higher) than limiting cooling to above-freezing temperatures.

Additional conclusions resulting from the study are as follows:

5. Lithium bromide absorption chillers were generally less attractive than either complex compound or ammonia vapor compression systems.

6. Ammonia vapor compression, complex compound, and lithium bromide cooling systems all yielded cost-effective (positive NPV) results.

7. Aqua-ammonia absorption chillers have significantly higher capital costs, were rarely cost-effective, and were never the preferred system.

8. Single-stage complex compound systems were more cost-effective than double-stage complex compound systems. The double-stage systems cost less and use less steam, but require higher pressure steam, which results in a greater loss of power when extracted from the steam turbine in the combined-cycle plant.

9. Storage significantly improves the cost-effectiveness of inlet air cooling, especially for applications requiring cooling for about one-third of the day or less.

10. Limitations on the application of storage significantly hamper the cost-effective application of CTAC. This included a) no storage for heat-activated chillers in simple-cycle applications (no waste heat source when the power plant isn't running), b) limited storage for heat-activated chillers running less than 24 hours per day, and c) no cost-effective storage media for below freezing applications.

11. Refrigerative cooling was found to be cost effective even if evaporative cooling was already in place.

12. Incremental power output is most important, but incremental energy output was found to contribute from about 20 to 45% of the incremental revenues resulting from CTAC, depending on the application conditions and the cooling system used.

13. Evaporative cooling results in a lower $/kW, but also a lower NPV than refrigerative cooling systems. The evaporative cooling capital cost per incremental peak power output was less than $100/kW for most application conditions. The best refrigerative cooling application yielded an incremental cost of about $100/kW for inlet air cooling provided 4 hours per day. The best application cost rose to about $200/kW for cooling provided 24 hours per day.

The results of the study provide the basis for making the following recommendations:

1. Inlet air cooling should be considered a standard practice to be incorporated with combustion turbine installation.

2. The potential cost-effectiveness of cooling the inlet air to near 0°F warrants further investigation (modeling and testing) of options for defrosting cooling coils and minimizing or eliminating condensate icing.

3. Off-peak operation with natural gas should be evaluated to determine the cost-effectiveness of storage with heat-activated chillers for simple-cycle applications and for combined-cycle plants operating less than 24 hours per day.

4. Single-stage lithium bromide chillers should be evaluated for combined-cycle plants in addition to the double-stage lithium bromide chillers that were evaluated.

5. Potential storage media for subfreezing applications should be identified and evaluated.

6. Better complex compound systems (lower cost and/or higher efficiencies) need to be identified before development and demonstration of this concept for inlet air cooling is warranted.

7. CTAC NPVs should be evaluated from an industrial perspective with a broader range of values for power, on-peak energy, and off-peak energy.

8. The computer model developed for this study should be upgraded to allow the investigation of a) variable inlet air temperatures, b) multiple evaporation temperatures above and below the freezing point, and c) variable fractions of the peak cooling load. These three capabilities would allow further investigation of optimum CTAC systems.

11.0 References

Aho, V. A., B. W. Kernigan, and P. J. Weinberger. 1988. *The AWK Programming Language*, Addison-Wesley Publishing Company, New York.

American Society of Heating, Refrigeration, and Air Conditioning (ASHRAE). 1987. *1987 ASHRAE Handbook, HVAC Systems and Applications*. Atlanta, Georgia.

Babcock and Wilcox. 1994. *Ice Thermal Energy Storage for Gas Turbine Inlet Air Cooling*. New York.

Brown, D. R., J. A. Dirks, M. K. Drost, G. E. Spanner, and T. A. Williams. 1987. *An Assessment Methodology for Thermal Energy Storage Evaluation*. PNL-6372. Pacific Northwest Laboratory. Richland, Washington.

Close D. B., A. D. Robbins, P. H. Rubin, and R. Stallman. 1993. *The GAWK Manual*, version 2.15, ed. 0.15, Free Software Foundation, Cambridge, Massachusetts.

Dickson, J. 1975. "Reducing the Hazards of Intake Icing." *Gas Turbine International*. Vol. 16, No. 4.

Ebeling, J. A., R. Halil, D. Bantam, B. Bakenhus, H. Schreiber, and R. Wendland. 1992. "Peaking Gas Turbine Capacity Enhancement Using Ice Storage for Compressor Inlet Air Cooling." *Proceedings, International Gas Turbine and Aeroengine Congress and Exposition, 1992*. The American Society of Mechanical Engineers. New York.

Electric Power Research Institute. 1988. *Commercial Cool Storage; Reduced Cooling Costs With Off-Peak Electricity*. Palo Alto, California.

Electric Power Research Institute. 1993. *Technical Assessment Guide, Electricity Supply-1993*. EPRI TR-102276-V1R7. Palo Alto, California.

Energy Information Administration. 1995a. *Annual Energy Outlook 1995, With Projections to 2010*. DOE/EIA-0383(95). Washington, D.C.

Energy Information Administration. 1995b. *Monthly Energy Review, March 1995*. DOE/EIA-0035(95/3). Washington D.C.

Hall I. J., R. R. Prarie, H. E. Anderson and E. C. Boes. 1981. "Generation of typical meteorological years for 26 SOLMET stations," Appendix to TMY User's Manual, National Climatic Center, Asheville, North Carolina.

Holton, J. K., T. D. Swanson, R. Gulati, and G. K. Hart. 1985. *Small Office Building Handbook*. Van Nostrand Reinhold. New York.

Kohlenberger, C. 1993. "A Review of Inlet Air Cooling Systems for Increasing Gas Turbine Performance." Published by the American Society of Mechanical Engineers, Los Angeles Section Power Division.

Kohlenberger, C. 1994. "A Review of Inlet Air Cooling Systems for Increasing Gas Turbine Performance." *Proceedings, 1994 Power-Gen Conference*. Pennwell Conferences and Exhibitions. Houston, Texas.

Ondryas, I. S., D. A. Wilson, M. Kawamoto, and G. L. Haub. 1991. "Options in Gas Turbine Power Augmentation Using Inlet Air Chilling." *Journal of Engineering for Gas Turbines and Power*. Vol. 113, pp. 203-211.

Rockenfeller, U., P. Sarkisian, and L. D. Kirol. 1992. "Coordinative Complex Compounds for Efficient Storage of Polar Refrigerants and Gases." *Proceedings, 1992 International Energy Conversion Engineering Conference*. SAE International. Warrendale, Pennsylvania.

Wagar, S. N. 1980. "Indirect Heating System for Turbine Anti-Icing." *Diesel & Gas Turbine Worldwide*. Vol. 12, No. 2.

Zwillenberg, M. L., D. Smith, A. Cohn, I. Oliker, and W. Major. 1991. "Assessment of Refrigeration-Type Cooling of Inlet Air for Essex Unit No. 9." *Proceedings, 1991 International Power Generation Conference*. The American Society of Mechanical Engineers. New York.

12.0 Bibliography

Andreport, J. S. 1994. "Performance and Economics of CT Inlet Air Cooling using Chilled Water Storage." *ASHRAE Transactions 1994*. Vol. 100, Pt. 1. American Society of Heating, Refrigeration, and Air-Conditioning. Atlanta, Georgia.

Babcock and Wilcox. 1994. *Ice Thermal Energy Storage for Gas Turbine Inlet Air Cooling*. New York.

Bacigalupo, E., L. Tasso, and R. G. Zinnari. 1993. "Power Augmentation Using An Inlet Air Chilling System in a Cogenerative Power Plant Equipped With a Heavy Duty Gas Turbine." Published in the proceedings of the *1993 ASME Cogen-Turbo Conference*. American Society of Mechanical Engineers. New York.

Cross, J. K., W. A. Beckman, J. W. Mitchell, D. T. Reindl, and D. E. Knebel. 1995. "Modeling of Hybrid Combustion Turbine Inlet Air Cooling Systems." *ASHRAE Transactions 1995*. Vol. 101, Pt. 2. American Society of Heating, Refrigeration, and Air-Conditioning. Atlanta, Georgia.

Davis, T. L., A. Hobbs, J. Shell, K. McAllister, L. Beaty, and R. Reid. 1995. *Combustion Turbine Capacity Augmentation Utilizing Inlet Air Direct Contact Cooling and Compression*. Carolina Power and Light. Raleigh, North Carolina.

DeLucia, M., R. Bronconi, and E. Carnevale. 1993. "Performance and Economic Enhancement of Cogeneration Gas Turbines Through Compressor Inlet Air Cooling." *Proceedings, International Gas Turbine and Aeroengine Congress and Exposition, 1993*. The American Society of Mechanical Engineers. New York.

Dickson, J. 1975. "Reducing the Hazards of Intake Icing." *Gas Turbine International*. Vol. 16, No. 4.

Ebeling, J. A. 1991. "Combustion Turbine Inlet Air Cooling With Thermal Energy Storage." Published in the proceedings of *POWER-GEN '91*. Pennwell Conferences and Exhibitions Company. Houston, Texas.

Ebeling, J. A., R. Halil, D. Bantam, B. Bakenhus, H. Schreiber, and R. Wendland. 1992. "Peaking Gas Turbine Capacity Enhancement Using Ice Storage for Compressor Inlet Air Cooling." *Proceedings, International Gas Turbine and Aeroengine Congress and Exposition, 1992*. The American Society of Mechanical Engineers. New York.

Ebeling, J. A., L. H. Beaty, and P. A. Garrett. 1994. "TIP^2 Capacity Performance Test Results for the Butler Warner Generation Plant." Published in the proceedings of *POWER-GEN '94*. Pennwell Conferences and Exhibitions Company. Houston, Texas.

Ebeling, J., R. Balsbaugh, S. Blanchard, and L. Beaty. 1994. "Thermal Energy Storage and Inlet Air

Cooling for Combined Cycle." Published in the proceedings of the *International Gas Turbine and Aeroengine Congress and Exposition*. ASME-94-GT-310. American Society of Mechanical Engineers. New York.

Electric Power Research Institute. 1988. *Commercial Cool Storage; Reduced Cooling Costs With Off-Peak Electricity*. Palo Alto, California.

Gillingham, G. R. 1976. *A New System for Preventing Icing of Gas Turbine Inlets*. ASME-76-GT-84. American Society of Mechanical Engineers. New York.

Giourof, A. 1995. "Gas-turbine inlet air cooling: You can almost pick your payback." *Power*. May, 1995. pp. 56-58. McGraw-Hill Publishing Company. New York.

Guinn, G. R. 1993. "Evaluation of Combustion Gas Turbine Inlet Air Precooling for Time Varying Annual Climatic Conditions." Published in the proceedings of the *1993 ASME Cogen-Turbo Conference*. American Society of Mechanical Engineers. New York.

Hall, A. D., J. C. Stover, and R. L. Breisch. 1994. "Gas Turbine Inlet-Air Chilling at a Cogeneration Facility." *ASHRAE Transactions 1994*. Vol. 100, Pt. 1. American Society of Heating, Refrigeration, and Air-Conditioning. Atlanta, Georgia.

Herold, K. E. 1995. "Design challenges in absorption chillers." *Mechanical Engineering*. October, 1995. pp. 80-83. American Society of Mechanical Engineers. New York.

Hufford, P. E. 1991. "Absorption Chillers Maximize Cogeneration Value." *ASHRAE Transactions: Symposia.* NY-91-2-3. American Society of Heating, Refrigeration, and Air-Conditioning. Atlanta, Georgia.

Kohlenberger, C. 1993. *A Review of Inlet Air Cooling Systems for Increasing Gas Turbine Performance*. Published by the American Society of Mechanical Engineers, Los Angeles Section Power Division, Los Angeles, California.

Kohlenberger, C. 1994. "A Review of Inlet Air Cooling Systems for Increasing Gas Turbine Performance." *Proceedings, 1994 Power-Gen Conference*. Pennwell Conferences and Exhibitions. Houston, Texas.

Kolp, D. A., and D.J. Moeller. 1989. "World's First STIG LM5000 Installed at Simpson Paper Company." *Journal of Engineering for Gas Turbines and Power*. April, 1989. Vol. 111. pp. 200-210.

Kolp, D. A., W. M. Flye, and H. A. Guidotti. 1995. "Advantages of Air Conditioning and Supercharging an LM6000 Gas Turbine Inlet." *Journal of Engineering for Gas Turbines and Power*. July, 1995. Vol.

117 pp. 513-527.

MacCracken, C. D. 1994. "An Overview of the Progress and the Potential of Thermal Storage in Off-Peak Turbine Inlet Cooling." *ASHRAE Transactions 1994*. Vol. 100, Pt. 1. American Society of Heating, Refrigeration, and Air-Conditioning. Atlanta, Georgia.

Mackie, E. I. 1994. "Inlet Air Cooling for a Combustion Turbine Using Thermal Storage." *ASHRAE Transactions 1994*. Vol. 100, Pt. 1. American Society of Heating, Refrigeration, and Air-Conditioning. Atlanta, Georgia.

Moftah, M. 1971. *Gas Turbine Performance Under Varying Ambient Temperature*. ASME 71-GT-57. American Society of Mechanical Engineers. New York.

Nasser, A.E.M., and M. A. El-Kalay. 1991. "A Heat-Recovery Cooling System to Conserve Energy in Gas-Turbine Power Stations in the Arabian Gulf." *Applied Energy*. Vol. 38. pp. 133-142.

Ondryas, I. S. 1991. "Go beyond evaporative coolers to stretch gas-turbine output." *Power*. July, 1991. pp. 27-29.

Ondryas, I. S., D. A. Wilson, M. Kawamoto, and G. L. Haub. 1991. "Options in Gas Turbine Power Augmentation Using Inlet Air Chilling." *Journal of Engineering for Gas Turbines and Power*. Vol. 113, pp. 203-211.

Rockenfeller, U., P. Sarkisian, and L. D. Kirol. 1992. "Coordinative Complex Compounds for Efficient Storage of Polar Refrigerants and Gases." *Proceedings, 1992 International Energy Conversion Engineering Conference*. SAE International. Warrendale, Pennsylvania.

Rose, D. T., W. A. Ryan, and H. Perez-Blanco. 1995. "Adopting steam-driven absorption cooling." *Mechanical Engineering*. October, 1995. pp. 76-78. American Society of Mechanical Engineers. New York.

Sengupta, U., and G. Soroka. 1989. "An Inlet Air Washer/Chiller System for Combined Cycle Plant Repowering." Published in the proceedings of the *American Power Conference*. Illinois Institute of Technology. Chicago, Illinois.

Sundbom, T. A., and J. A. Ebeling. 1994. "Increasing Capacity Sales with Inlet Air Cooling." In the proceedings of the *1994 ASME Cogen-Turbo Conference*. American Society of Mechanical Engineers. New York.

Sullivan, J., and J. A. Ebeling. 1994. "Selling Cold Air DSM Incentives Lead to Combustion Turbine Inlet Air Cooling Opportunities for ESCOs." In the proceedings of the *1994 ASME Cogen-Turbo Conference*. American Society of Mechanical Engineers. New York.

Tanabe, K., and R. Okugawa. 1976. *Evaluation of Improved Cycles for Marine Application of an Aircraft Derivative Gas Turbine*. ASME-76-GT-114. American Society of Mechanical Engineers. New York.

Wagar, S. N. 1980. "Indirect Heating System for Turbine Anti-Icing." *Diesel & Gas Turbine Worldwide*. Vol. 12, No. 2.

Zwillenberg, M. L., D. Smith, A. Cohn, I. Oliker, and W. Major. 1991. "Assessment of Refrigeration-Type Cooling of Inlet Air for Essex Unit No. 9." *Proceedings, 1991 International Power Generation Conference*. The American Society of Mechanical Engineers. New York.

Appendix A

Inlet Air Cooling Evaluation Model

Appendix A

Inlet Air Cooling Evaluation Model

The combustion turbine inlet air cooling performance evaluation model is a collection of *awk* and UNIX™ shell scripts. The *awk* is a powerful file processing programming language originally developed in 1977 for the UNIX™ operating system. The name *awk* comes from the initials of its designers: Alfred V. Aho, Peter J. Weinberger, and Brian W. Kernigan. In 1985 a new version made the programming language more powerful, introducing user-defined functions, multiple input streams, and computed regular expressions (Aho et al. 1988). The GNU[(a)] version of *awk* implementation, called *gawk*, was used for modeling the performance of inlet air cooling technologies (Close et al. 1993). The *gawk* is a free program and is available for UNIX, MS-DOS and MAC operating systems.

The *awk* language was chosen because of its simplicity and convenience of programming a wide variety of computing and data-manipulation tasks. Because *awk* is an interpreted language (like BASIC), the development time is minimal. To write a program to do the same task in a language such as C, Pascal, or FORTRAN would not only be time-consuming but would need many more lines of code. In spite of its simplicity, the structure of the *awk* language is similar to C; therefore, *awk* scripts can be converted to C with minimal effort.

A.1 Computer Requirements

The model was developed on a SUN SPARCStation™ 10 UNIX™ workstation running SunOS™ 4.2 operating system. Although *awk* scripts are portable across operating systems (UNIX™, MAC™, or MS-DOS™), the accompanying UNIX™ shell scripts are not. These scripts enable post-processing of the output for detailed analysis and run a series of simulations in an automated way. However, the model itself can be run on any platform. It takes less than 1 minute to run on the UNIX™ workstation, but it may take several minutes on a PC with an Intel 486™ CPU.

(a) GNU, which stands for Gnu's Not UNIX, is the name for the complete UNIX-compatible software system being developed by the Free Software Foundation (FSF). The FSF is a group started by Richard Stallman to embody his ideas of personal freedom and how software should be produced. The free software includes a UNIX shell, machine language, compilers, editors and many other things which have, in the past, been considered part of the operating system. These are very well known and respected programs and can be found at the “prep.ai.mit.edu” ftp site.

A.2 Model Structure

The model consists of seven major modules and several small functions. The first module initializes all the variables, assigns values to constants and reads the command line inputs (input file name, parameter file name, etc.). The second module reads parameter inputs from the parameter file and calculates the outdoor condition independent corrections factors for gas turbine power, heat rate, air mass flow rate, etc. The third module reads the hourly TMY[(a)] weather file (Hall et al. 1981) and performs the hourly simulation. The fourth module calculates minimum and maximum values of certain variables for the storage cases and also sizes the chiller and storage tanks. The fifth module calculates daily/weekly and annual chiller energy consumption. The sixth module sizes various components and also calculates the costs of those components. The seventh module calculates the incremental power generation cost and the net present value (NPV) of the investment. Figure A.1 shows schematically the combustion turbine inlet air cooling performance evaluation model structure.

A.3 Model Capabilities

In general, the model is capable of assessing the economic feasibility of cooling the inlet air to combustion turbines for site-specific conditions by simulating the performance of the gas turbine with inlet air cooling, sizing necessary components, calculating the cost of the components, and calculating the monetary value of the incremental power and the NPV of the investment. Sample output from the model is presented in Table A.1.

The model is capable of evaluating the performance of cooling the inlet air to turbines in simple-cycle (industrial frame and aeroderivative type) or combined-cycle plant configurations. The air to the turbine can either be cooled refrigeratively or evaporatively. Thermal energy storage is an option for refrigeratively cooled systems. The model allows either daily or weekly ice or water storage with load-shifting or load-leveling storage designs. At the present time, the model can only handle three different inlet air temperatures: 1) 52°F, 2) 42°F, and 3) 0°F. For 0°F cooling, the model cools the air in two stages with two different chillers: 1) from the ambient condition to 42°F, and 2) from 42°F to 0°F. The chillers can either be electrically-driven or thermally-activated; for thermally-activated chillers, the turbine exhaust gas (simple-cycle) or steam from the combined-cycle plant is used.

(a) The Typical Meteorological Year (TMY) is a "typical" year of weather prepared from a large data base of the SOLMET (SOLar and METeorological) data and is available for 239 stations in the U.S. The data include 1 complete year of hourly values (8760 values) for each of the following parameters: direct (beam) solar radiation, total horizontal solar radiation, dry-bulb (temperature), wet-bulb (temperature), dew-point (humidity), wind speed and cloud cover. These data are provided in a single file in ASCII format. The SOLMET hourly data base was first prepared for 26 cities that had measured hourly solar radiation data. Then it was extended to 222 additional locations by using models to generate synthetic solar radiation measures for these locations.

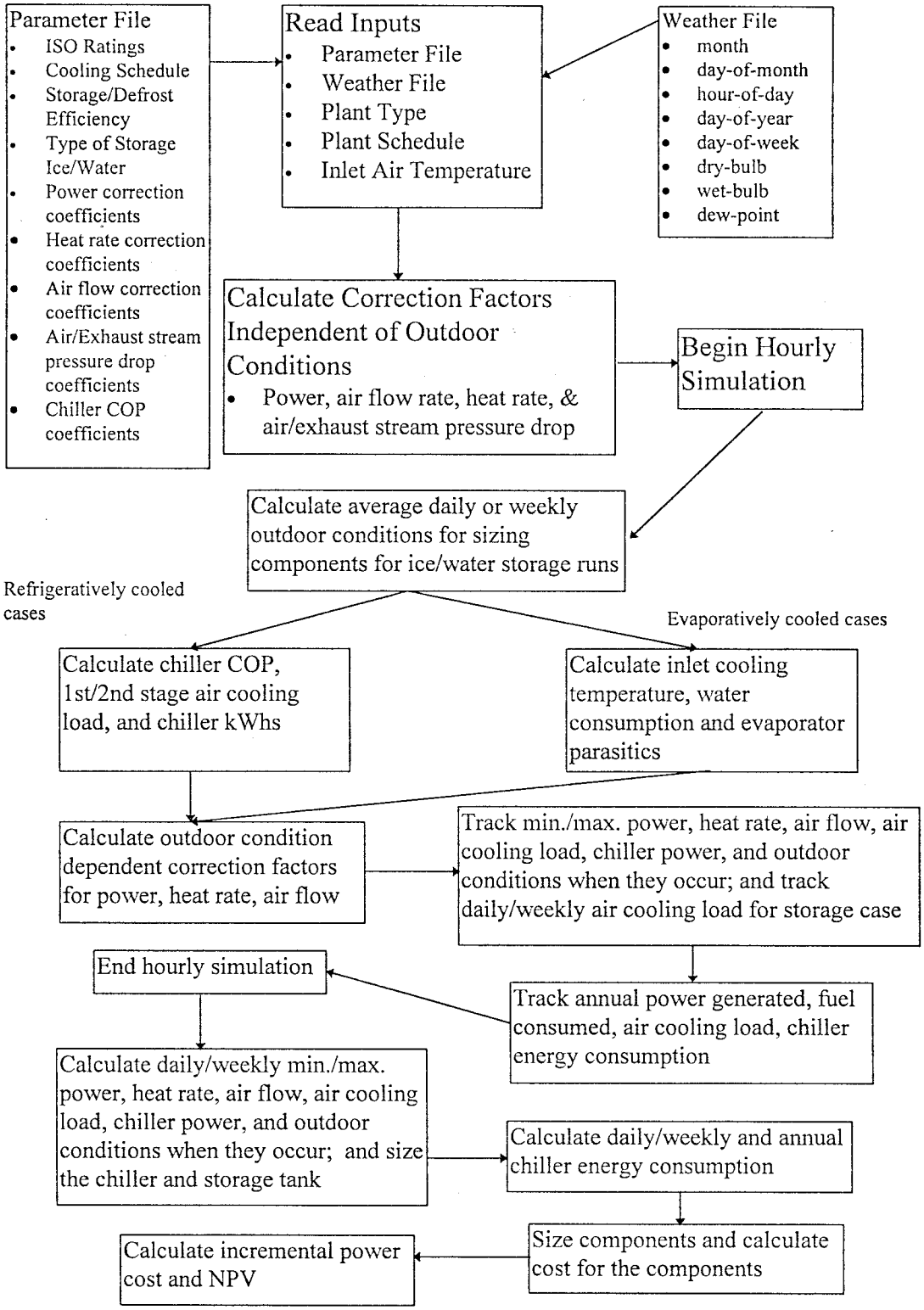


Figure A.1. Schematic of the Combustion Turbine Inlet Air Cooling Performance Evaluation Method

A.4 Model Limitations

Although the model is a powerful analysis tool, it has the following limitations: 1) the model does not allow for entering the input values interactively because the user interface is inadequate; in addition, the shell scripts that automate the parametric runs are not portable to MS-DOS or MAC operating systems, 2) the range checks are not performed on all input values, i.e., the inputs are not checked if they are within the acceptable range, 3) the inlet air temperature is fixed to three values (52°F, 42°F, and 0°F), 4) storage media is limited to ice and water, 5) cooling in multiple stages is not possible; currently, the model cools the air in a single stage when the air is cooled to 42°F or 52°F and in two stages when the air is cooled to 0°F, and 6) the component sizing basis is limited to meeting 100% of the peak design conditions.

Table A.1. Sample Model Output

Parameter File	cr51
Weather File	nashville.tmy
Plant Type	CC Plant
Schedule Information	
Weekends are not simulated	
Beginning for the operation period (inclusive)	9
End for the operation period (inclusive)	20
Beginning month (inclusive)	1
End month (inclusive)	12
Inlet air is cooled to (F)	0
Generator Cooling Factor	1.050
ISO Power (MW)	225
ISO GT Power (MW)	155
ISO ST Power (MW)	70
ISO Heat Rate (Btu/kWh)	7300
ISO Air Flow Rate(lbs/h)	3365475
Beginning of cooling period (inclusive)	15
End of cooling period (inclusive)	18
Beginning month for cooling (inclusive)	4
End month for cooling (inclusive)	9
Daily load shifting storage being used	
1st stage daily storage efficiency	0.990
1st stage ice generation (defrost) efficiency	0.850
2nd stage daily storage efficiency	1.000
2nd stage ice generation (defrost) efficiency	1.000
Ice is the storage media	
Power Coefficients Echo	
Intercept for Altitude	1.0000e+00
Intercept for Dry-Bulb Temperature/T_evap: First Range	1.2200e+00
Intercept for Dry-Bulb Temperature/T_evap: Second Range	1.0000e+00
Intercept for Absolute Humidity	1.0000e+00
Intercept for Incremental Inlet Air Pr. Drop in of water	1.0000e+00
Intercept for Incremental Inlet Exhaust Pr. Drop in of water	1.0000e+00
Slope for Altitude	-3.3300e-05
Slope for Dry-Bulb Temperature/T_evap: First Range	-3.7000e-03
Slope for Dry-Bulb Temperature/T_evap: Second Range	0.0000e+00
Slope for Absolute Humidity	0.0000e+00
Slope for Incremental Inlet Air Pr. Drop in of water	-3.2000e-03
Slope for Incremental Inlet Exhaust Pr. Drop in of water	-1.2000e-03

Heat Rate Coefficients Echo	
Intercept for Altitude	1.0000e+00
Intercept for Dry-Bulb Temperature/T_evap: First Range	9.4000e-01
Intercept for Dry-Bulb Temperature/T_evap: Second Range	1.0000e+00
Intercept for Absolute Humidity	1.0000e+00
Intercept for Incremental Inlet Air Pr. Drop in of water	1.0000e+00
Intercept for Incremental Inlet Exhaust Pr. Drop in of water	1.0000e+00
Slope for Altitude	0.0000e+00
Slope for Dry-Bulb Temperature/T_evap: First Range	1.0000e-03
Slope for Dry-Bulb Temperature/T_evap: Second Range	1.5000e-03
Slope for Absolute Humidity	0.0000e+00
Slope for Incremental Inlet Air Pr. Drop in of water	1.7000e-03
Slope for Incremental Inlet Exhaust Pr. Drop in of water	1.4500e-03
Air Flow Coefficients Echo	
Intercept for Altitude	1.0000e+00
Intercept for Dry-Bulb Temperature/T_evap: First Range	1.1200e+00
Intercept for Dry-Bulb Temperature/T_evap: Second Range	1.0000e+00
Intercept for Dry-Bulb Temperature/T_evap: Third Range	1.0000e+00
Intercept for Absolute Humidity	1.0000e+00
Intercept for Incremental Inlet Air Pr. Drop in of water	1.0000e+00
Intercept for Incremental Inlet Exhaust Pr. Drop in of water	1.0000e+00
Slope for Altitude	-3.3300e-05
Slope for Dry-Bulb Temperature/T_evap: First Range	-2.1500e-03
Slope for Dry-Bulb Temperature/T_evap: Second Range	0.0000e+00
Slope for Dry-Bulb Temperature/T_evap: Third Range	0.0000e+00
Slope for Absolute Humidity	0.0000e+00
Slope for Incremental Inlet Air Pr. Drop in of water	-2.5000e-03
Slope for Incremental Inlet Exhaust Pr. Drop in of water	0.0000e+00
Exhaust Flow Factor Echo	
Intercept for Altitude	1.0000e+00
Intercept for Dry-Bulb Temperature/T_evap: First Range	1.1200e+00
Intercept for Dry-Bulb Temperature/T_evap: Second Range	1.0000e+00
Intercept for Absolute Humidity	1.0000e+00
Intercept for Incremental Inlet Air Pr. Drop in of water	1.0000e+00
Slope for Altitude	-3.3300e-05
Slope for Dry-Bulb Temperature/T_evap: First Range	-2.1500e-03
Slope for Dry-Bulb Temperature/T_evap: Second Range	0.0000e+00
Slope for Absolute Humidity	-2.5000e-03
Slope for Incremental Inlet Air Pr. Drop in of water	0.0000e+00

Condenser Temperature Factor Echo	
Intercept for Twb < 54°F	1.0000e+00
Intercept for Twb > 54°F	1.0000e+00
Slope for Twb < 54°F	-1.9000e-04
Slope for Twb > 54°F	-9.0000e-04
Echo Adjustments	
Altitude	577
Air Stream Pressure Drop (in of H_20)	1.300
Exhaust Stream Pressure Drop (in of H_20)	0.000
Chiller Coefficients Echo	
Chiller Data	
Chiller Type is	Complex Compound Chiller
1st stage chiller intercept	8.6700e-01
1st stage chiller slope	-2.5000e-03
2nd stage chiller intercept	5.2000e-01
2nd stage chiller slope	-0.0000e+00
Echo Storage Script	
Parameter file	cr51
Input file	cr51.day
Plant Type	CC Plant
Inlet air is cooled to (F)	42
Chiller Coefficients Echo	
Chiller Type is	Complex Compound Chiller
1st stage chiller intercept	8.6700e-01
1st stage chiller slope	-2.5000e-03
2nd stage chiller intercept	5.2000e-01
2nd stage chiller slope	-0.0000e+00
Echo Constants Post-Processor ...	
Cooling Media NH3	
1st Stage	
Chilled water/NH3 supplied temperature (F)	34
Chilled water/NH3 return temperature (F)	46
1st Stage Generator	
Cooling hx lmtd (F)	10
U-Value of heat exchanger (Btu/h-F-ft2)	225
2nd Stage Generator	
Cooling hx lmtd (F)	60
U-Value of heat exchanger (Btu/h-F-ft2)	225
GT exhaust gas temperature (F)	1000
GT exhaust gas cp (Btu/lb-F)	0.253
Cooling water delta T range (F)	10

Chilled water circulation power (kW/gpm)	0.0300
NH3 circulation power (kW/lb/h)	2.00000e-05
Steam/Water piping power (kW/lb/h)	6.00000e-05
Cooling water piping power (kW/gpm)	0.0200
Cooling tower fan power (kW/mmBtu/h)	1.2500
Ice generator power (kW/ton)	0.0300
Ice generator loop sizing (gpm/ton)	3.0000
Water chiller generator power (kW/ton)	0.0200
Water chiller generator loop sizing (gpm/ton)	2.0000
First stage	
Feed water temperature to HRSG (F)	351
Steam pressure (psia)	137
Latent heat of steam at HRSG outlet condition (Btu/lb)	870
Chiller parasitic power (kW/ton)	0.0000
Chiller heat rejection rate (Btu/h/ton cooling)	29991
Second stage	
Feed water temperature to HRSG (F)	213
Steam pressure (psia)	15
Latent heat of steam at HRSG outlet condition (Btu/lb)	969
Chiller parasitic power (kW/ton)	0.0000
Chiller heat rejection rate (Btu/h/ton cooling)	35076
Chiller heat rejection rate: Sizing (Btu/h/ton cooling)	21624
1st Stage Relative cost equation	
Intercept	1.566
Slope	-6.90000e-03
2nd Stage Relative cost equation	
Intercept	1.000
Slope	-0.00000e+00
Output Results with cr51 Parameter File	
Number of plant operating hours	2868
Number of inlet cooling hours	480
A. GT Energy Output (MWh)	448135.525
Average power output (MW)	156.254
Net GT MW @max cooling load condition (uncooled)	133.718
Average GT heat rate (Btu/kWh)	10524.276
A. CC Energy Output (MWh)	642002.833
Average power output (MW)	223.850
Ref. power at max. enthalpy (uncooled) 8 20 15 232 5 92 79	197.853
Base power (cooled) 8 20 15 232 5 92 79	254.541
A. CC Fuel Input (mmBtu)	4716301.796
Average CC heat rate (Btu/kWh)	7346.232

P. Hourly GT Power (MW)	4 16 15 106 5 52 52	184.695
P. Hourly ST Power (MW)	5 1 15 121 6 60 45	71.576
P. Hourly CC Power (MW)	5 1 15 121 6 60 45	256.271
P. Hourly GT Heat Rate (Btu/kWh)	6 30 14 181 3 94 75	11161.823
P. Hourly CC Heat Rate (Btu/kWh)	7 24 12 205 6 91 80	7511.692
P. Hourly Flow Rate CC (lbs/h)	4 16 15 106 5 52 52	3684892.728
Min. Hourly GT Power (MW)	6 30 14 181 3 94 75	132.042
Min. Hourly ST Power (MW)	7 24 12 205 6 91 80	64.128
Min. Hourly CC Power (MW)	6 30 14 181 3 94 75	196.306
Min. Hourly GT Heat Rate (Btu/kWh)	4 16 15 106 5 52 52	9982.981
Min. Hourly CC Heat Rate (Btu/kWh)	5 1 15 121 6 60 45	7194.749
Min. Hourly CC Flow Rate (lbs/h)	6 30 14 181 3 94 75	3019966.996
Air Cooling Load Output		
P. Hourly 1st Stage CL (mmBtuh)	8 20 15 232 5 92 79	96.948
A. 1st Stage CL (mmBtu)		29954.180
Average hourly 1st Stage CL (Btu/h)		62.405
P. Hourly 2nd Stage CL (mmBtuh)	4 16 15 106 5 52 52	56.420
A. 2nd Stage CL (mmBtu)		26995.434
Average hourly 2nd Stage CL (Btu/h)		56.240
P. Daily 1st Stage CL (mmBtu)	202 85 77	358.095
P. Daily 2nd Stage CL (mmBtu)	106 50.5 50	225.681
Chiller Peak Energy Non-Storage Case (adjusted by gencoolfac)		
P. Hourly 1st Stage ChE (mmBtuh)	8 20 15 232 5 92 79	152.047
P. Hourly 2nd Stage ChE (mmBtuh)	4 16 15 106 5 52 52	113.925
P. Hourly 1st Stage ChE (kW)	8 20 15 232 5 92 79	44549.500
P. Hourly 2nd Stage ChE (kW)	4 16 15 106 5 52 52	33379.867
A. 1st Stage ChE (mmBtu		45517.187
A. 2nd Stage ChE (mmBtu)		54510.010
A. 1st Stage ChE (kWh)		13336415.726
A. 2nd Stage ChE (kWh)		15971289.321
Chiller Sizing Output (adjusted by gencoolfac)		
Non-Storage 1st Stage ChC (mmBtu)	8 20 15 232 5 92 79	101.796
Non-Storage 2nd Stage ChC (mmBtu)	4 16 15 106 5 52 52	59.241
Adjusted for generating cooling and storage losses ...		
D. LS 1st Stage ChC (mmBtu)	202 85 77	55.853
D. LS 2nd Stage ChC (mmBtu)	106 50.5 50	29.621

Storage Sizing Output Adjusted (gencoolfac and storage eff.)	
D. LS 1st Stage SC (mmBtu)	379.798
D. LS 2nd Stage SC (mmBtu)	236.965
Results: Post-Processing Storage Script	
1st Stage Peak Chiller Energy Consumption Storage Case (adjusted by gencoolfac and storage efficiency)	
Hourly 1st Stage ChE (mmBtu) 82.25 77.50	82.960
Hourly 1st Stage ChE (kWh)	24306.966
Annual 1st Stage ChE (mmBtu)	54009.325
Annual 1st Stage ChE (kWh)	15824589.937
2nd Stage Peak Chiller Energy Consumption (adjusted by gencoolfac)	
Hourly 2nd Stage ChE (mmBtu) 82.25 77.50	56.963
Hourly 2nd Stage ChE (kWh)	16689.934
Annual 2nd Stage ChE (mmBtu)	27255.005
Annual 2nd Stage ChE (kWh)	7985644.661
Maximum air side exhaust pressure drop in HRSG (in of H_20)	10
Results From Post-Processing....	
1st Stage Ice Storage	
Cooling coil UA [42] (Btu/F-h)	4462682.955
Generator cooling heat exchanger area (sf)	2154.407
Chilled water circulation loop size [42/52 F] (gpm)	16145.134
Chilled water generator loop size [42/52 F] (gpm)	807.257
Chilled water pump peak kW	484.354
Chilled water pump annual energy consumption (kWh)	149651.279
Generator cooling loop pump peak kW	24.218
Generator cooling loop annual pumping energy (kWh)	7482.564
Ammonia loop (chiller-to-storage)	
Circulation loop size (lb/h)	101735.155
Peak pumping power (kW)	2.035
Annual pumping energy (kWh)	1361.603
2nd Stage Ammonia	
Cooling coil UA [0 F] (Btu/F-h)	1519855.557
Generator cooling heat exchanger area (sf)	208.964
Ammonia circulation loop size [0 F] (lb/h)	96775.643
Generator circulation loop size [0 F] (lbh)	4838.782
Peak pumping power (kW)	1.936
Annual pumping energy (kWh)	926.086
Peak pumping power [generator] (kW)	0.097
Annual pumping energy [generator] (kWh)	46.304
1st Stage ice generator sizing	
Circulation loop size (gpm)	13963.150
Peak pumping power (kW)	139.631

Annual pumping energy (kWh)	93440.018
1st Stage Cooling Tower	
Peak heat rejection (Btu/h)	139589610.550
Water pipe sizing (gpm)	27895.606
Peak water pumping power (kW)	557.912
Annual water pumping energy (kWh)	373349.264
Chiller parasitic power (kW)	0.000
Annual chiller parasitic energy (kWh)	0.000
Peak fan power (kW)	174.487
Annual fan energy (kWh)	116764.982
Annual water consumption (gals)	11623137.460
2nd Stage Cooling Tower	
Peak heat rejection (Btu/h)	53376321.200
Water pipe sizing (gpm)	10666.731
Peak water pumping power (kW)	213.335
Annual water pumping energy (kWh)	331146.810
Chiller parasitic power (kW)	0.000
Annual chiller parasitic energy (kWh)	0.000
Peak fan power (kW)	66.720
Annual fan energy (kWh)	103566.165
Annual water consumption (gals)	6355571.691
1st Stage	
Steam extraction (lb/h)	95355.977
Power reduction (kW)	7316.061
2nd Stage	
Steam extraction (lb/h)	58785.036
Power reduction (kW)	2273.715
1st stage annual energy loss (kWh)	4762979.097
2nd stage annual energy loss (kWh)	1087906.627
484.354 1.93551 24.2177 0.0967756 213.335 66.7204 0	
Cost Information	
1st Stage Chiller	
Type:	Complex Compound Chiller
Cost ($/ton):	550.00
O&M cost (as % of initial cost):	5.00
Cost ($):	2482337.78
O&M Costs ($):	124116.89
2nd Stage Chiller	
Type:	Complex Compound Chiller
Cost ($/ton):	0.00
O&M cost (as % of initial cost):	5.00

Cost ($):	0.00
O&M Costs ($):	0.00
2nd Stage Chiller/Storage Cost	
Chiller/Storage cost ($/ton-hr):	310.00
Chiller/Storage O&M cost as (% of initial cost):	4.00
Chiller/Storage cost ($/ton-hr):	6121595.83
Chiller/storage O&M cost (as % of initial cost):	244863.83
1st Stage Ice Generator/Storage	
Generator cost ($/ton):	590.00
Generator O&M cost as (% of initial cost):	2.00
Ice storage cost ($/ton-hrs)^0.75:	163.60
Ice storage O&M cost (as % of initial cost):	1.50
Generator cost ($):	2746086.17
Generator O&M costs ($):	54921.72
Ice storage cost ($):	388205.67
Ice storage O&M costs ($):	5823.08
1st Stage Chilled Water Generator	
Generator cost ($/ton):	50.00
Generator O&M cost as (% of initial cost):	1.00
Generator cost ($):	0.00
Generator O&M costs ($):	0.00
Water storage cost ($/ton)^0.75:	572.40
Water storage O&M cost (as % of initial cost):	1.50
Water storage cost ($):	0.00
Water storage O&M costs ($):	0.00
1st Stage air cooling coil	
Cost ($/[Btu/h-f]):	0.20
O&M cost (as % of initial cost):	2.00
Cost ($):	892536.59
O&M costs ($):	17850.73
2nd Stage air cooling coil	
Cost ($/[Btu/h-f]):	0.33
O&M cost (as % of initial cost):	3.00
Cost ($):	501552.33
O&M costs ($):	15046.57
1st Stage generator cooling coil	
Cost ($/ft^2)^0.511:	865.00
O&M cost (as % of initial cost):	1.00
Cost ($):	43686.43
O&M costs ($):	436.86
2nd Stage generator cooling coil	

Cost ($):	13260.88
O&M costs ($):	132.61
1st Chilled water circulation cost	
Cost (gpm/10,000)^0.6:	290000.00
O&M cost (as % of initial cost):	3.00
Cost ($):	386565.41
O&M costs ($):	11596.96
Cost ($) [generator loop]:	64062.66
O&M costs ($) [generator loop]:	1921.88
1st Ammonia circulation cost	
Cost (lb/h/15,000)^0.6:	54000.00
O&M cost (as % of initial cost):	3.00
Cost ($):	0.00
O&M costs ($):	0.00
Cost ($) [generator loop]:	0.00
O&M costs ($) [generator loop]:	0.00
2nd Ammonia circulation cost	
Cost (lb/h/15,000)^0.6:	108000.00
Cost ($):	330543.51
O&M costs ($):	9916.31
Cost ($) [generator loop]:	27389.28
O&M costs ($) [generator loop]:	821.68
1st stage storage water charging cost (ice loop)	
Cost (gpm/10,000)^0.6:	100000.00
O&M cost (as % of initial cost):	3.00
Cost ($):	122177.16
O&M costs ($):	3665.31
1st stage storage water charging cost	
Cost (gpm/10,000)^0.6:	100000.00
O&M cost (as % of initial cost):	3.00
Cost ($):	0.00
O&M costs ($):	0.00
1st Ammonia circulation cost (chiller-to-storage)	
Cost (lb/h/15,000)^0.6:	54000.00
O&M cost (as % of initial cost):	3.00
Cost ($):	170302.74
O&M costs ($):	5109.08
Water circulation cost (chiller-to-storage)	
Cost (gpm/10,000)^0.6:	100000.00
O&M cost (as % of initial cost):	3.00
Cost ($):	0.00

O&M costs ($):	0.00
1st Steam/Condensate cost	
Cost (gpm/10,000)^0.6:	320000.00
O&M cost (as % of initial cost):	3.00
Cost ($):	29728.18
O&M costs ($):	891.85
1st stage HRSG cost	
Cost (UA)^0.8:	10980.00
Cost (SF):	9658.00
Cost (EF)^1.2:	918.00
O&M cost (as % of initial cost):	3.00
Cost ($):	0.00
O&M costs ($):	0.00
2nd Steam/Condensate cost	
Cost ($):	22239.23
O&M costs ($):	667.18
2nd stage HRSG cost	
Cost ($):	0.00
O&M costs ($):	0.00
1st Stage cooling tower circulation cost	
Cost (gpm/10,000)^0.6:	260000.00
O&M cost (as % of initial cost):	3.00
Cost ($):	481165.88
O&M costs ($):	14434.98
2nd Stage cooling tower circulation cost	
Cost ($):	270266.46
O&M costs ($):	8107.99
1st Stage cooling tower cost	
Cost (MMBtu/h-192)^0.876:	662000.00
O&M cost (as % of initial cost):	2.00
Cost ($):	500699.81
O&M costs ($):	10014.00
2nd Stage cooling tower cost	
Cost ($):	215696.71
O&M costs ($):	4313.93
Cost of makeup water is $1/1000 gals	
1st stage makeup water costs	11623.14
2nd stage makeup water costs	6355.572
System control cost	
Fixed Cost ($):	250000.00
O&M cost (as % of initial cost):	6.00

O&M costs ($):	15000.00
Plant electric cost	
Fixed Cost ($):	100000.00
O&M cost (as % of initial cost):	2.00
O&M costs ($):	2000.00
Peak transformer cost	
Cost ($ kW)^0.4359:	5553.00
O&M cost (as % of initial cost):	2.00
Cost ($):	106352.24
O&M costs ($):	2127.04
Peak auxiliary power consumption (kW)	790.66
Off-Peak auxiliary power consumption (kW)	874.07
Peak auxiliary annual energy consumption (kWh)	591893.12
Off-Peak auxiliary annual energy consumption (kWh)	584915.87
Total capital cost	16266450.96
Total O&M cost	571759.21
Echo Input NPV Routine	
Reading Parameters from file	npv.cr51
Net kWh reference uncooled plant	619825508
Net mmBtu reference uncooled plant	4573613.7
kW/kWh multiplier	10.391
O&M and off-peak kWh multiplier	10.39100
Fuel multiplier	14.32600
Capital cost multiplier	1.00970
Peak kW cost ($/kW)	64.00
Peak kWh cost ($/kWh)	0.03800
Off-Peak kWh cost ($/kWh)	0.02400
Fuel cost ($/mmBtu)	2.25
Reading Inputs from file	cr51.npv
Gross kW cooled plant	254541
Net kW @max cooling load condition	197853
Peak auxiliary kW	790.7
Peak auxiliary kW reduction due to steam extraction	0.0
Gross kWh cooled plant	642003000
Net kWh reference uncooled plant	619825508
Peak auxiliary kWh	591893
Peak auxiliary kWh lost due to steam extraction	0
Off-Peak auxiliary kWh	584916
Off-Peak auxiliary kWh lost due to steam extraction	5850890
Fuel input	4716300.000
Net mmBtu reference uncooled plant	4573613.731

Total capital costs	16266500.00
Total O&M costs	571759.00
Output from NPV routine	
Net on-peak kW	55897.341
Net on-peak kWh	21585599.000
Net off-peak kWh	6435806.000
Net fuel	142686.269
NPV	17126622.037
Incremental kW cost	291.007
On-Peak kW fraction of NPV	37173073.301
On-Peak kWh fraction of NPV	8523246.450
Off-Peak kWh fraction of NPV	-1604987.044
O&M fraction of NPV	-5941147.769
Fuel fraction of NPV	-4599277.852
Capital fraction of NPV	-16424285.050

Appendix B

Component Sizing

Appendix B

Component Sizing

The equations and assumptions used to size cooling system components are presented in this Appendix.

B.1 Chiller Sizing

For direct air cooling applications (without storage) the chiller is sized to meet the peak air cooling load (pacl) and the cooling load required to cool the generator:

$$1^{st} \text{ Stage Chiller Size (Btu/h)} = 1^{st} \text{ Stage pacl x gcf}$$

where "gcf," the generator cooling factor, is the amount of additional cooling needed for the generator (a value of 1.05 was for all cases). When the air is cooled to 0°F, two chillers are sized. The first chiller meets the peak cooling load required to cool air from the ambient condition to 42°F (1st stage). The second chiller meets the peak cooling load to cool the air from 42°F to 0°F (2nd stage).

$$2^{nd} \text{ Stage Chiller Size (Btu/h)} = 2^{nd} \text{ Stage pacl x gcf}$$

When daily storage is used, the total load the chiller is required to provide is the sum of the peak daily air cooling load, the daily generator cooling load, and the load required to offset the storage and defrost (ice storage only) losses. For load-shifting storage, the chiller is sized to generate the total load while operating only during the period when there is no inlet air cooling:

$$\text{Chiller Size, Daily Load-Shifting (Btu/h)} = (\text{pacl x gcf x seff x deff})/\text{nop}$$

where "seff" and "deff" are storage and defrost efficiencies, respectively, and "nop" is the number of hours the chiller can operate when inlet air is not being cooled. For example, if the inlet air is cooled 4 hours per day with an electrically-driven chiller, "nop" is 20 hours. For thermally-activated chillers "nop" is the difference between the number of plant operating hours and the number of inlet air cooling hours (because the thermal energy required to run the chiller is extracted from the power plant). For load-leveling storage the chiller is sized to provide the total daily load while operating 24 hours a day (for electrically-driven chillers) or while the plant is operating (for thermally-activated chillers).

Chiller Size, Daily Load-Leveling (Btu/h) = (pacl x gcf x seff x deff)/op

where "op" is either 24 or the number of plant operating hours.

When weekly storage is used, the total load the chiller is required to provide is the sum of the peak weekly air cooling load, the weekly generator cooling load and load required to offset the storage and defrost (ice storage only) losses. For load-shifting, the chiller is sized to generate the total load while operating only during the period when there is no inlet air cooling. The same equations used to size chillers for daily load-shifting and load-leveling are used of weekly load-shifting and load-leveling as well. The "nop" for weekly storage is either multiplied by 5 or 7, for thermally- activated chillers, depending on whether the plant operates on weekends or not. For electrically-driven chillers with the plant operating on weekends the daily "nop" is multiplied by 7; when the plant is not operating on weekends the weekly "nop" is:

weekly nop = (daily nop x 5) + 48

B.2 Storage Sizing

The storage size for daily load-shifting is:

Daily Storage Size, Load-Shifting (Btu) = (pdcl x gcf)/seff

where "pdcl" is the peak daily cooling load. The storage size for daily load-leveling is:

Daily Storage Size, Load-Leveling (Btu) = (pdcl x gcf)/seff -
(chiller size x noc x deff)

where "noc" is the number of inlet cooling hours. The storage size for weekly load-shifting is:

Weekly Storage Size, Load-Shifting (Btu) = (pwcl x gcf)/seff -
(chiller size x nop x 4 x deff)

where "pwcl" is the peak weekly cooling load. The storage size for weekly load-leveling is:

Weekly Storage Size, Load-Leveling (Btu) = (pwcl x gcf)/seff - (chiller size x
(noc + 24 x 4) x deff)

B.3 Inlet Air Cooling Coil Sizing

Both the 1st and 2nd stage inlet air cooling coil UAs are sized as follows:

Coil UA Size (Btu/h-°F) = pacl/?T_{lm}

where "?T_{lm}" is the log mean temperature difference. The heat exchangers required to cool the generator are sized as follows:

$$\text{Generator Cooling Coil Size (sf)} = \text{pacl} \times (\text{gcf} - 1) / (?T_{lm} \times U)$$

The values used for "?T_{lm}" and "UA" are shown in Table B.1.

B.4 Chilled Water Circulation System Sizing

The chilled water (CW) circulation loop is sized as follows:

$$\text{CW Circulation System Size (gpm)} = \text{pacl} / (500.4 \times [T_o - T_i])$$

For LiBr chillers or water storage cases T_o and T_i are 56°F and 44°F, respectively; and for ice storage T_o and T_i are 46°F and 34°F, respectively. The circulation system needed to cool the generator is calculated similarly:

$$\text{CW Circulation System Size Generator Loop (gpm)} = \text{pacl} \times (\text{gcf} - 1) / (500.4 \times [T_o - T_i])$$

The peak pumping pump required to pump water is calculated as follows:

$$\text{Peak Pumping Power (kW)} = \text{CW Circulation System Size} \times \text{kW_gpm}$$

where "kW_gpm" is the power (kW) required to pump 1 gallon of CW water per minute. The same equation is used for calculating the generator peak pumping power as well. Annual pumping energy for both the main CW loop and the generator cooling loop is calculated as follows:

$$\text{Annual CW Pumping Energy (kWh)} = \text{Peak Pumping Power} \times \text{acl/pacl}$$

where "acl" is the annual cooling load.

Table B.1. ?T_{lm} and U Values Used for Sizing Generator Cooling Coil

Mode	?T_{lm}	U
Inlet air cooling to 42°F or 52°F	35	225
Inlet air cooling to 0°F		
1st Stage	10	225
2nd Stage	60	225

B.5 Ammonia Circulation System Sizing

The ammonia circulation loop is sized as follows:

Ammonia Circulation System Sizing (lbs/h) = pacl / Ammonia hfg

where "Ammonia hfg" is the latent heat of vaporization for ammonia. For ammonia chillers operating at an evaporator temperature of 32°F, the hfg is 543 Btu/lb, and for chillers operating at 20°F and -20°F evaporator temperatures, hfg values are 549 Btu/lb and 583 Btu/lb, respectively. Typically, the first 1st stage chiller operates at 32°F or 20°F evaporator temperature, while the 2nd stage chiller operates at -20°F evaporator temperature. When ice or water storage is used, an additional ammonia loop between the chiller and storage tank is sized similarly:

Ammonia Chiller-to-Storage Sizing (lb/h) = Storage Size / Ammonia hfg

The circulation loop needed to cool the generator is calculated as follows:

Ammonia Circulation Size for Generator Loop (lb/h) = pacl x (gcf - 1) / Ammonia hfg

The peak pumping pump required to pump ammonia is calculated as follows:

Peak Pumping Power (kW) = Ammonia Circulation System Size x kW_lbh

where "kW_lbh" is the power (kW) required to pump 1 pound of ammonia per hour. The same equation is used for calculating the generator peak pumping power as well. Annual pumping energy for the main ammonia loop, chiller-to-storage loop, and the generator loop is calculated as follows:

Annual Ammonia Pumping Energy (kWh) = Peak Pumping Power x acl/pacl

B.6 Ice Generator Sizing

The size of the ice generator is same as the chiller size. The circulation loop for the ice generator is sized as follows:

Ice Generator Circulation Loop Sizing (gpm) = Chiller Size x ice_gpmpt / 12000

where "ice_gpmpt" is the circulation loop flow rate in gallons per minute per ton of chilling capacity. The peak pumping power is calculated as follows:

Peak Pumping Power for Ice Generator (kW) = Chiller Size x ikW_ton / 12000

where "ikW_ton" is the kW per ton of chilling capacity. Annual pumping power is calculated as follows:

Annual Pumping Energy (kWh) = Peak Pumping Power x acl x gcf / (seff x Chiller Size)

B.7 Chilled Water Generator Sizing

The size of the chilled water generator is same as the chiller size. The circulation loop for the chilled water generator is sized as follows:

CW Generator Circulation Loop Sizing (gpm) = Chiller Size / (12000 x h2o_gpmpt)

where "h2o_gpmpt" is the circulation loop flow rate in gallons per minute per ton of chilling capacity. The peak pumping power is calculated as follows:

Peak Pumping Power for CW Generator (kW) = Chiller Size / (12000 x wkW_ton)

where "wkW_ton" is the kW per ton of chiller capacity. Annual pumping power is calculated as follows:

Annual Pumping Power (kWh) = Peak Pumping Power x acl x gcf / (seff x Chiller Size)

B.8 Cooling Tower Sizing

The peak heat rejection rate at the cooling tower for both 1st and 2nd stage is calculated as follows:

Peak Heat Rejection Rate (Btu/h) = Chiller Size x Chiller Heat Rejection Rate / 12000

where the chiller heat rejection rate varies by chiller technology. The water pipe sizing for the cooling tower is:

Cooling Water Sizing (gpm) = Peak Heat Rejection Rate / (500.4 x $?T_e$)

where, $?T_e$, is assumed to be 10°F. The peak pumping power is calculated as follows:

Peak Pumping Power (kW) = Cooling Water Sizing x ckW_gpm

where "ckW_gpm" is 0.02. Annual pumping energy is calculated as follows:

Annual Pumping Energy (kWh) = Peak Pumping Power x acl x gcf / (seff x Chiller Size)

The peak fan power for the cooling tower is calculated as follows:

Peak Fan Power (kW) = Peak Heat Rejection Rate x kW_mmBtu/1E6

where "kW_mmBtu" is 1.25. The annual fan energy consumption is calculated as follows:

Annual Fan Energy (kWh) = (Peak Fan Power x acl x gcf)/(seff x Chiller Size)

B.9 Chiller Parasitic Energy

The LiBr chiller has additional parasitic consumption, which is calculated as follows:

Annual Parasitic (kWh) = (Chiller Size / 12000 x kW_ton) x acl x gcf / Chiller Size

where "kW_ton" is 0.0136 kW/ton for a single stage LiBr chiller and 0.028 kW/ton for a double stage LiBr chiller.

B.10 Heat Recovery Steam Generator (HRSG) Sizing

The heat recovery steam generator for thermally-activated chillers with a simple-cycle power plant is sized as follows:

HRSG UA (Btu/h-°F) = Peak Chiller Energy Consumption/?T_{lm}

where "?T_{lm}" is the log mean temperature difference:

?T_{lm} = ([gt_fex - feed_t] - [gt_ex - feed_t])/(log[{gt_fex - feed_t}/{gt_ex - feed_t}])

where "gt_ex" is the temperature of the exhaust gas leaving the gas turbine (1000°F) and "gt_fex" is the temperature of the exhaust gas leaving the HRSG. The exhaust gas leaving the HRSG is calculated as follows:

gt_fex (°F) = gt_ex - Peak Chiller Energy Consumption/(mass x C_p Gas)

where "mass" is the mass flow rate of the exhaust gas and a value of 0.25 Btu/°F-lb is assumed for the exhaust gas C_p. The required steam flow is calculated as follows:

Steam Flow (lb/h) = Peak Chiller Energy Consumption/(Steam hfg)

A maximum pressure drop equivalent to 10 inches of water is assumed to occur in the HRSG, if all the energy from the exhaust gas is extracted to generate steam (i.e., if gt_fex = 300°F). However, only a fraction of the exhaust flow energy is required in most cases to meet the peak energy requirements of the thermally-activated chiller:

Fraction of Energy Required = (gt_ex - gt_fex)/(gt_ex - 300)

The required exhaust air flow to meet the peak chiller energy consumption is calculated as follows:

Exhaust Flow Rate (lbs/h) = Chiller Energy Consumption/(Cp Gas x [gt_ex - gt_fex])

B.11 Power Reduction Due to Steam Extraction

Power is lost when a thermally-activated chiller is used with a combined-cycle power plant because the steam extracted from the combined-cycle to run the chiller would otherwise be used to generated electricity. The foregone electricity production is calculated as follows:

Power Reduction (kW) = (-0.0079 + 0.0172 x $\log_e$ [Steam Pressure, psia])

The amount of steam that has to be extracted is calculated as follows:

Steam Flow (lb/h) = Peak Chiller Energy Consumption/(Steam hfg)

Appendix C

Complete Results

Appendix C

Complete Results

NPV and $/kW results are presented for each case evaluated in the tables that follow. The nomenclature used to define each case is the same as that described in Section 9, but is repeated here.

Each case is distinguished by its case number, chiller type, plant operating hours per day, cooling hours per day, and storage type. The case number uniquely defines one of 356 cases. Industrial turbine cases begin with an "i," aeroderivative turbine cases with an "a," and combined-cycle cases with a "c,", "d," "e," or "f." The four combined-cycle groups represent different combinations of plant operating hours and cooling system operating hours. The second letter of the case number, an "e" or an "r," designates a case as being evaporatively or refrigeratively cooled. The chiller type can be either aqua-ammonia (AA), complex compound (CC), lithium bromide (LiBr), or vapor compression (VC). The chiller numbers refer to the specific versions of each chiller type defined in Section 6. Plant operating hours and cooling system operating hours are designated by two numbers separated by a backslash. For example, "12/4" would mean the plant runs 12 hours per day while the cooling system runs 4 hours per day. The specific power plant and cooling system schedule assumptions were defined in Section 4. The storage type is defined by a four letter code. The first letter refers to ice or water storage, the second letter to a daily or weekly cycle, and the final two letters to a load-shifting or load-leveling design basis.

Table C.1. Aeroderivative Turbine $/kW; Uncooled Base

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
ae1	NA	4/4	NA	52	65	49	27	95
ae2	NA	8/8	NA	52	115	62	43	35
ar1	AA-2	4/4	NA	881	941	845	628	708
ar2	AA-2	8/8	NA	881	1034	921	738	530
ar3	CC-2	4/4	NA	313	331	304	232	306
ar4	CC-2	8/8	NA	313	364	328	273	221
ar5	LiBr-1	4/4	NA	350	379	334	233	387
ar6	LiBr-1	8/8	NA	350	430	372	286	220
ar7	VC-2	4/4	NA	273	291	262	193	254
ar8	VC-2	8/8	NA	273	326	288	231	180
ar9	VC-1	4/4	IWLS	159	173	150	128	179
ar10	VC-1	4/4	IWLL	156	170	147	125	177
ar11	VC-1	4/4	IDLS	179	191	180	139	198
ar12	VC-1	4/4	IDLL	172	184	173	133	192
ar13	VC-1	8/8	IWLS	238	287	230	224	167
ar14	VC-1	8/8	IWLL	217	263	209	203	152
ar15	VC-1	8/8	IDLS	298	350	315	268	203
ar16	VC-1	8/8	IDLL	251	301	267	225	172
ar17	VC-2	4/4	WWLS	225	251	207	173	267
ar18	VC-2	4/4	WWLL	219	244	201	167	261
ar19	VC-2	4/4	WDLS	203	220	205	151	269
ar20	VC-2	4/4	WDLL	196	212	198	144	261
ar21	VC-2	8/8	WWLS	335	423	320	312	210
ar22	VC-2	8/8	WWLL	302	385	288	280	191
ar23	VC-2	8/8	WDLS	304	371	326	269	215
ar24	VC-2	8/8	WDLL	263	326	283	231	185

Table C.2. Combined-Cycle $/kW; Uncooled Base; 12 hr/day plant; 4 or 8 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
ce1	NA	12/4	NA	65	82	61	34	119
ce2	NA	12/8	NA	65	145	78	54	44
ce3	NA	12/12	NA	77	236	78	54	60
ce4	NA	24/4	NA	65	82	61	34	119
ce5	NA	24/8	NA	65	145	78	54	44
ce6	NA	24/12	NA	77	1200	78	54	46
ce7	NA	24/24	NA	77	1200	78	54	46
cr1	CC-4	12/4	NA	392	434	368	241	302
cr2	CC-4	12/8	NA	392	514	422	304	203
cr3	CC-4	12/4	WDLS	233	258	237	167	262
cr4	CC-4	12/4	WDLL	218	246	222	148	247
cr5	CC-4	12/8	WDLS	578	716	617	491	337
cr6	CC-4	12/8	WDLL	374	522	414	299	192
cr7	CC-3	12/4	IDLS	269	294	273	205	246
cr8	CC-3	12/4	IDLL	244	271	248	177	221
cr9	CC-3	12/8	IDLS	836	992	877	727	477
cr10	CC-3	12/8	IDLL	501	660	544	410	244
cr11	LiBr-2	12/4	NA	432	489	399	243	381
cr12	LiBr-2	12/8	NA	432	581	467	311	198
cr13	LiBr-2	12/4	WDLS	250	279	250	171	257
cr14	LiBr-2	12/4	WDLL	215	244	217	144	228
cr15	LiBr-2	12/8	WDLS	663	823	670	519	336
cr16	LiBr-2	12/8	WDLL	358	489	386	276	176
cr17	VC-2	12/4	NA	245	264	234	169	204
cr18	VC-2	12/8	NA	245	296	259	203	148
cr19	VC-1	12/4	IWLS	127	139	117	100	130
cr20	VC-1	12/4	IWLL	123	136	114	96	127
cr21	VC-1	12/4	IDLS	148	159	149	113	149
cr22	VC-1	12/4	IDLL	140	151	141	105	142
cr23	VC-1	12/8	IWLS	202	247	193	188	130
cr24	VC-1	12/8	IWLL	180	223	172	167	115
cr25	VC-1	12/8	IDLS	267	317	282	237	169

Table C.2. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
cr26	VC-1	12/8	IDLL	219	265	233	192	137
cr27	VC-2	12/4	WWLS	175	198	159	132	185
cr28	VC-2	12/4	WWLL	169	191	154	126	179
cr29	VC-2	12/4	WDLS	159	174	160	114	189
cr30	VC-2	12/4	WDLL	152	166	153	108	180
cr31	VC-2	12/8	WWLS	271	346	256	250	155
cr32	VC-2	12/8	WWLL	242	313	229	221	138
cr33	VC-2	12/8	WDLS	252	311	270	219	163
cr34	VC-2	12/8	WDLL	213	269	230	183	135
cr35	AA-2	12/4	NA	1176	1294	1105	745	800
cr36	AA-2	12/8	NA	1176	1482	1249	912	578
cr37	AA-2	12/4	WDLS	540	607	551	378	513
cr38	AA-2	12/4	WDLL	439	502	450	297	422
cr39	AA-2	12/8	WDLS	1817	2265	1913	1487	898
cr40	AA-2	12/8	WDLL	872	1183	952	686	392
cr41	AA-1	12/4	IDLS	602	663	612	451	485
cr42	AA-1	12/4	IDLL	491	552	503	355	391
cr43	AA-1	12/8	IDLS	2177	2591	2262	1853	1136
cr44	AA-1	12/8	IDLL	1075	1391	1155	875	489
cr45	CC-4/CC-5	12/4	NA	403	429	387	296	334
cr46	CC-4/CC-5	12/8	NA	403	471	421	343	270
cr47	AA-2/AA-3	12/4	NA	1351	1450	1288	976	967
cr48	AA-2/AA-3	12/8	NA	1351	1577	1405	1116	818
cr49	VC-2/VC-3	12/4	NA	287	300	279	229	257
cr50	VC-2/VC-3	12/8	NA	287	321	297	256	218
cr51	CC-3/CC-6	12/4	IDLS	291	305	293	251	280
cr52	CC-3/CC-6	12/8	IDLS	759	839	781	695	548
cr53	AA-1/AA-3	12/4	IDLS	673	710	675	567	587
cr54	AA-1/AA-3	12/8	IDLS	1610	1799	1635	1408	935
cr55	VC-1/VC-3	12/4	IWLS	202	212	195	175	216
cr56	VC-1/VC-3	12/8	IWLS	252	284	247	241	203
cr57	VC-1/VC-3	12/4	IWLL	201	211	194	173	215
cr58	VC-1/VC-3	12/8	IWLL	240	271	235	229	195

Table C.2. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
cr59	VC-1/VC-3	12/4	IDLS	216	225	216	184	226
cr60	VC-1/VC-3	12/8	IDLS	296	329	306	273	227
cr61	VC-1/VC-3	12/4	IDLL	213	222	213	181	223
cr62	VC-1/VC-3	12/8	IDLL	267	299	277	246	209
cr63	CC-3/CC-6	12/4	IDLL	285	301	288	242	279
cr64	CC-3/CC-6	12/8	IDLL	589	669	613	537	446
cr65	AA-1/AA-3	12/4	IDLL	815	875	817	651	724
cr66	AA-1/AA-3	12/8	IDLL	1269	1508	1326	1081	745
cr67	CC-2	12/4	NA	266	286	254	181	229
cr68	CC-2	12/8	NA	266	325	282	220	161
cr69	CC-2	12/4	WDLS	221	243	224	159	253
cr70	CC-2	12/4	WDLL	195	216	197	136	226
cr71	CC-2	12/8	WDLS	519	641	554	443	310
cr72	CC-2	12/8	WDLL	295	386	321	245	170
cr73	CC-1	12/4	IDLS	254	276	258	194	236
cr74	CC-1	12/4	IDLL	214	235	218	159	202
cr75	CC-1	12/8	IDLS	766	908	805	668	444
cr76	CC-1	12/8	IDLL	387	482	415	330	214
cr77	CC-2/CC-5	12/4	NA	314	329	305	245	290
cr78	CC-2/CC-5	12/8	NA	314	355	325	279	237
cr79	CC-1/CC-6	12/4	IDLS	282	295	284	244	276
cr80	CC-1/CC-6	12/8	IDLS	718	791	738	659	530
cr81	CC-1/CC-6	12/4	IDLL	227	238	229	193	219
cr82	CC-1/CC-6	12/8	IDLL	339	383	352	307	240
cr83	VC-2/CC-5	12/4	NA	299	313	291	235	276
cr84	VC-2/CC-5	12/8	NA	299	337	310	266	227
cr85	VC-1/CC-6	12/4	IWLL	213	222	208	189	238
cr86	VC-2/CC-6	12/8	IWLL	380	418	379	371	357

Table C.3. Combined-Cycle $/kW; Uncooled Base; 12 hr/day plant; 12 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
dr1	CC-4	12/12	NA	435	566	422	304	251
dr2	LiBr-2	12/12	NA	520	693	491	317	233
dr3	AA-2	12/12	NA	1295	1599	1249	912	704
dr4	VC-2	12/12	NA	265	316	259	203	175
dr5	VC-1	12/12	IWLS	309	388	281	278	209
dr6	VC-1	12/12	IWLL	246	316	223	219	165
dr7	VC-1	12/12	IDLS	465	555	462	393	307
dr8	VC-1	12/12	IDLL	310	382	308	256	200
dr9	VC-2	12/12	WWLS	396	525	355	350	236
dr10	VC-2	12/12	WWLL	320	434	286	278	189
dr11	VC-2	12/12	WDLS	386	483	384	315	260
dr12	VC-2	12/12	WDLL	274	356	273	219	180
dr13	CC-4/CC-5	12/12	NA	430	495	421	343	304
dr14	AA-2/AA-3	12/12	NA	1450	1650	1406	1116	918
dr15	VC-2/VC-3	12/12	NA	301	332	297	256	238
dr16	VC-1/VC-3	12/12	IWLL	285	331	270	264	229
dr17	CC-2	12/12	NA	287	348	282	220	193
dr18	CC-2/CC-5	12/12	NA	330	370	326	279	262
dr19	VC-2/CC-5	12/12	NA	314	350	310	266	251
dr20	VC-2/CC-5	12/12	IWLL	297	348	280	274	241

Table C.4. Combined-Cycle $/kW; Uncooled Base; 24 hr/day plant; 4 or 8 or 12 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er1	CC-4	24/4	NA	392	434	368	241	302
er2	CC-4	24/8	NA	392	514	422	304	203
er3	CC-4	24/12	NA	435	723	422	304	223
er4	CC-4	24/4	WDLS	167	182	168	120	197
er5	CC-4	24/4	WDLL	166	183	168	116	197
er6	CC-4	24/8	WDLS	268	332	288	233	172
er7	CC-4	24/8	WDLL	249	322	271	210	150
er8	CC-4	24/12	WDLS	417	584	415	341	239
er9	CC-4	24/12	WDLL	345	553	342	265	178
er10	CC-3	24/4	IDLS	153	166	154	117	153
er11	CC-3	24/4	IDLL	152	165	153	113	151
er12	CC-3	24/8	IDLS	278	331	294	246	173
er13	CC-3	24/8	IDLL	252	312	270	218	148
er14	CC-3	24/12	IDLS	484	631	481	408	282
er15	CC-3	24/12	IDLL	381	554	378	304	200
er16	LiBr-2	24/4	NA	432	489	399	243	381
er17	LiBr-2	24/8	NA	432	581	467	311	198
er18	LiBr-2	24/12	NA	520	932	491	317	200
er19	LiBr-2	24/4	WDLS	168	187	170	119	193
er20	LiBr-2	24/4	WDLL	162	181	164	113	187
er21	LiBr-2	24/8	WDLS	274	352	293	235	169
er22	LiBr-2	24/8	WDLL	240	317	260	201	143
er23	LiBr-2	24/12	WDLS	428	638	429	344	237
er24	LiBr-2	24/12	WDLL	324	515	321	247	167
er25	VC-2	24/4	NA	245	264	234	169	204
er26	VC-2	24/8	NA	245	296	259	203	148
er27	VC-2	24/12	NA	265	368	259	203	159
er28	VC-1	24/4	IWLS	128	141	119	101	131
er29	VC-1	24/4	IWLL	123	136	114	96	127
er30	VC-1	24/4	IDLS	148	159	149	113	149
er31	VC-1	24/4	IDLL	140	151	141	105	142
er32	VC-1	24/8	IWLS	213	260	202	198	135
er33	VC-1	24/8	IWLL	180	223	172	167	115

Table C.4. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er34	VC-1	24/8	IDLS	267	317	282	237	169
er35	VC-1	24/8	IDLL	219	265	233	192	137
er36	VC-1	24/12	IWLS	462	620	422	403	256
er37	VC-1	24/12	IWLL	325	458	295	279	178
er38	VC-1	24/12	IDLS	465	601	462	393	277
er39	VC-1	24/12	IDLL	310	422	308	256	179
er40	VC-2	24/4	WWLS	172	194	157	130	182
er41	VC-2	24/4	WWLL	169	191	154	126	179
er42	VC-2	24/4	WDLS	159	174	160	114	189
er43	VC-2	24/4	WDLL	152	166	153	108	180
er44	VC-2	24/8	WWLS	263	335	248	241	150
er45	VC-2	24/8	WWLL	242	313	229	221	138
er46	VC-2	24/8	WDLS	252	311	270	219	163
er47	VC-2	24/8	WDLL	213	269	230	183	135
er48	VC-2	24/12	WWLS	487	708	441	416	226
er49	VC-2	24/12	WWLL	411	629	369	345	188
er50	VC-2	24/12	WDLS	386	540	384	315	223
er51	VC-2	24/12	WDLL	274	407	273	219	155
er52	AA-2	24/4	NA	1176	1294	1105	745	800
er53	AA-2	24/8	NA	1176	1482	1249	912	578
er54	AA-2	24/12	NA	1295	1944	1249	912	642
er55	AA-2	24/4	WDLS	286	320	292	204	297
er56	AA-2	24/4	WDLL	268	302	274	188	280
er57	AA-2	24/8	WDLS	568	719	610	482	311
er58	AA-2	24/8	WDLL	466	607	505	386	245
er59	AA-2	24/12	WDLS	1029	1470	1022	820	500
er60	AA-2	24/12	WDLL	699	1083	690	531	314
er61	AA-1	24/4	IDLS	283	312	288	215	247
er62	AA-1	24/4	IDLL	266	295	271	198	232
er63	AA-1	24/8	IDLS	604	731	638	528	335
er64	AA-1	24/8	IDLL	497	619	530	424	262
er65	AA-1	24/12	IDLS	1144	1514	1137	954	600
er66	AA-1	24/12	IDLL	784	1119	775	622	376
er67	CC-4/CC-5	24/4	NA	403	429	387	296	334

Table C.4. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er68	CC-4/CC-5	24/8	NA	403	471	421	343	270
er69	CC-4/CC-5	24/12	NA	430	562	421	343	284
er70	AA-2/AA-3	24/4	NA	1351	1450	1288	976	967
er71	AA-2/AA-3	24/8	NA	1351	1577	1405	1116	818
er72	AA-2/AA-3	24/12	NA	1450	1872	1406	1116	872
er73	VC-2/VC-3	24/4	NA	287	300	279	229	257
er74	VC-2/VC-3	24/8	NA	287	321	297	256	218
er75	VC-2/VC-3	24/12	NA	301	362	297	256	226
er76	CC-3	24/4	IWLS	132	146	122	104	134
er77	CC-3	24/4	IWLL	130	144	120	101	132
er78	CC-3	24/8	IWLS	220	269	208	204	138
er79	CC-3	24/8	IWLL	198	249	187	182	121
er80	CC-3	24/12	IWLS	479	645	435	418	261
er81	CC-3	24/12	IWLL	390	588	347	327	194
er82	AA-1	24/4	IWLS	215	243	193	171	190
er83	AA-1	24/4	IWLL	201	229	180	158	180
er84	AA-1	24/8	IWLS	424	531	387	393	234
er85	AA-1	24/8	IWLL	344	438	313	315	187
er86	AA-1	24/12	IWLS	1072	1467	953	928	520
er87	AA-1	24/12	IWLL	739	1100	643	618	332
er88	CC-3/CC-6	24/4	IWLS	641	655	635	583	785
er89	CC-3/CC-6	24/8	IWLS	1244	1330	1260	1233	1290
er90	CC-3/CC-6	24/12	IWLS	2647	2973	2645	2583	2667
er91	CC-3/CC-6	24/4	IWLL	662	679	654	599	804
er92	CC-3/CC-6	24/8	IWLL	1323	1432	1332	1306	1340
er93	CC-3/CC-6	24/12	IWLL	3061	3578	3013	2935	2895
er94	CC-3/CC-6	24/4	IDLS	218	226	219	191	233
er95	CC-3/CC-6	24/8	IDLS	405	441	417	387	358
er96	CC-3/CC-6	24/12	IDLS	646	737	646	600	543
er97	CC-3/CC-6	24/4	IDLL	223	232	224	194	238
er98	CC-3/CC-6	24/8	IDLL	415	457	428	393	363
er99	CC-3/CC-6	24/12	IDLL	644	757	643	590	537
er100	AA-1/AA-3	24/4	IWLS	433	457	414	385	445
er101	AA-1/AA-3	24/8	IWLS	556	629	531	534	430

Table C.4. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er102	AA-1/AA-3	24/12	IWLS	556	629	531	534	430
er103	AA-1/AA-3	24/4	IWLL	558	597	531	475	576
er104	AA-1/AA-3	24/8	IWLL	677	789	653	636	514
er105	AA-1/AA-3	24/12	IWLL	677	789	653	636	514
er106	AA-1/AA-3	24/4	IDLS	477	500	479	417	476
er107	AA-1/AA-3	24/8	IDLS	665	746	685	618	491
er108	AA-1/AA-3	24/12	IDLS	981	1161	976	868	627
er109	AA-1/AA-3	24/4	IDLL	620	658	621	515	618
er110	AA-1/AA-3	24/8	IDLL	811	941	844	732	577
er111	AA-1/AA-3	24/12	IDLL	1048	1329	1037	885	657
er112	VC-1/VC-3	24/4	IWLS	203	213	196	176	217
er113	VC-1/VC-3	24/8	IWLS	259	293	253	247	206
er114	VC-1/VC-3	24/12	IWLS	427	517	399	384	281
er115	VC-1/VC-3	24/4	IWLL	201	211	194	173	215
er116	VC-1/VC-3	24/8	IWLL	240	271	235	229	195
er117	VC-1/VC-3	24/12	IWLL	337	414	317	303	234
er118	VC-1/VC-3	24/4	IDLS	216	225	216	184	226
er119	VC-1/VC-3	24/8	IDLS	296	329	306	273	227
er120	VC-1/VC-3	24/12	IDLS	429	506	426	377	294
er121	VC-1/VC-3	24/4	IDLL	213	222	213	181	223
er122	VC-1/VC-3	24/8	IDLL	267	299	277	246	209
er123	VC-1/VC-3	24/12	IDLL	328	394	326	289	236
er124	CC-2	24/4	NA	266	286	254	181	229
er125	CC-2	24/8	NA	266	325	282	220	161
er126	CC-2	24/12	NA	287	409	282	220	174
er127	CC-2	24/4	WDLS	162	177	164	117	194
er128	CC-2	24/4	WDLL	157	172	158	111	189
er129	CC-2	24/8	WDLS	256	316	274	223	167
er130	CC-2	24/8	WDLL	223	282	241	191	142
er131	CC-2	24/12	WDLS	389	545	388	319	228
er132	CC-2	24/12	WDLL	291	437	290	231	164
er133	CC-1	24/4	IDLS	148	159	149	113	149
er134	CC-1	24/4	IDLL	142	154	143	107	144
er135	CC-1	24/8	IDLS	263	312	278	233	166

Table C.4. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er136	CC-1	24/8	IDLL	222	269	236	194	138
er137	CC-1	24/12	IDLS	451	586	448	381	268
er138	CC-1	24/12	IDLL	315	433	313	259	181
er139	CC-2/CC-5	24/4	NA	314	329	305	245	290
er140	CC-2/CC-5	24/8	NA	314	355	325	279	237
er141	CC-2/CC-5	24/12	NA	330	407	326	279	246
er142	CC-1	24/4	IWLS	129	142	120	102	132
er143	CC-1	24/4	IWLL	125	137	115	97	129
er144	CC-1	24/8	IWLS	211	258	200	196	134
er145	CC-1	24/8	IWLL	183	226	174	169	117
er146	CC-1	24/12	IWLS	449	605	410	393	249
er147	CC-1	24/12	IWLL	331	471	300	283	179
er148	CC-1/CC-6	24/4	IWLS	639	653	634	581	784
er149	CC-1/CC-6	24/8	IWLS	1239	1324	1255	1228	1288
er150	CC-1/CC-6	24/12	IWLS	2630	2952	2630	2568	2660
er151	CC-1/CC-6	24/4	IWLL	652	667	645	591	798
er152	CC-1/CC-6	24/8	IWLL	1284	1381	1298	1271	1322
er153	CC-1/CC-6	24/12	IWLL	2859	3281	2841	2768	2816
er154	CC-1/CC-6	24/4	IDLS	215	222	216	189	231
er155	CC-1/CC-6	24/8	IDLS	396	431	408	379	354
er156	CC-1/CC-6	24/12	IDLS	627	714	627	584	535
Er157	CC-1/CC-6	24/4	IDLL	216	224	217	189	234
Er158	CC-1/CC-6	24/8	IDLL	390	426	402	373	353
Er159	CC-1/CC-6	24/12	IDLL	587	680	588	547	516
er160	VC-2/CC-5	24/4	NA	299	313	291	235	276
er161	VC-2/CC-5	24/8	NA	299	337	310	266	227
er162	VC-2/CC-5	24/12	NA	314	384	310	266	235
er163	VC-1/CC-5	24/4	IWLL	207	215	202	184	229
er164	VC-1/CC-5	24/8	IWLL	364	398	362	355	340
er165	VC-1/CC-5	24/12	IWLL	588	688	571	555	510

Table C.5. Combined-Cycle $/kW; Uncooled Base; 24 hr/day plant; 24 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
fr1	CC-4	24/24	NA	435	723	422	304	223
fr2	LiBr-2	24/24	NA	520	932	491	317	200
fr3	AA-2	24/24	NA	1295	1944	1249	912	642
fr4	VC-2	24/24	NA	265	368	259	203	159
fr5	CC-4/CC-5	24/24	NA	430	563	421	343	285
fr6	AA-2/AA-3	24/24	NA	1451	1873	1406	1116	874
fr7	VC-2/VC-3	24/24	NA	301	362	297	256	226
fr8	CC-2	24/24	NA	287	409	282	220	174
fr9	CC-2/CC-5	24/24	NA	330	407	326	279	246
fr10	VC-2/CC-5	24/24	NA	314	385	310	266	235

Table C.6. Industrial Turbine $/kW; Uncooled Base

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
ie1	NA	4/4	NA	68	85	64	36	124
ie2	NA	8/8	NA	68	151	81	56	46
ir1	AA-2	4/4	NA	1075	1151	1029	758	843
ir2	AA-2	8/8	NA	1075	1268	1125	893	628
ir3	CC-2	4/4	NA	353	375	341	257	327
ir4	CC-2	8/8	NA	353	414	371	304	236
ir5	LiBr-1	4/4	NA	397	432	376	255	407
ir6	LiBr-1	8/8	NA	397	493	422	315	227
ir7	VC-2	4/4	NA	307	331	293	209	263
ir8	VC-2	8/8	NA	307	374	325	253	186
ir9	VC-1	4/4	IWLS	158	174	147	125	169
ir10	VC-1	4/4	IWLL	155	170	144	122	165
ir11	VC-1	4/4	IDLS	182	196	184	140	191
ir12	VC-1	4/4	IDLL	175	189	176	132	184
ir13	VC-1	8/8	IWLS	249	302	237	232	164
ir14	VC-1	8/8	IWLL	225	277	215	208	147
ir15	VC-1	8/8	IDLS	323	382	341	287	209
ir16	VC-1	8/8	IDLL	271	328	288	238	172
ir17	VC-2	4/4	WWLS	221	248	202	167	243
ir18	VC-2	4/4	WWLL	214	242	195	161	237
ir19	VC-2	4/4	WDLS	200	218	201	144	247
ir20	VC-2	4/4	WDLL	192	210	194	137	238
ir21	VC-2	8/8	WWLS	339	431	321	313	199
ir22	VC-2	8/8	WWLL	305	394	289	279	178
ir23	VC-2	8/8	WDLS	311	384	334	272	206
ir24	VC-2	8/8	WDLL	268	339	290	231	174
ir25	CC-2/CC-5	4/4	NA	413	425	407	352	425
ir26	CC-2/CC-5	8/8	NA	413	448	424	387	362
ir27	AA-2/AA-3	4/4	NA	1126	1172	1096	919	963
ir28	AA-2/AA-3	8/8	NA	1126	1234	1154	1008	842
ir29	VC-2/VC-3	4/4	NA	367	385	356	289	330
ir30	VC-2/VC-3	8/8	NA	367	413	380	325	276
ir31	VC-1/VC-3	4/4	IWLS	254	266	246	219	276

Table C.6. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
ir32	VC-1/VC-3	8/8	IWLS	316	357	310	301	256
ir33	VC-1/VC-3	4/4	IWLL	253	266	245	218	275
ir34	VC-1/VC-3	8/8	IWLL	303	344	298	289	247
ir35	VC-1/VC-3	4/4	IDLS	270	282	271	229	288
ir36	VC-1/VC-3	8/8	IDLS	368	411	381	339	284
ir37	VC-1/VC-3	4/4	IDLL	268	280	269	226	286
ir38	VC-1/VC-3	8/8	IDLL	337	380	350	311	264
ir39	VC-2/CC-5	4/4	NA	410	426	401	339	402
ir40	VC-2/CC-5	8/8	NA	410	451	422	376	341
ir41	VC-1/CC-5	4/4	IWLL	309	320	301	274	352
ir42	VC-1/CC-5	8/8	IWLL	352	390	349	341	314

Table C.7. Aeroderivative Turbine NPV ($M); Uncooled Base

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
ae1	NA	4/4	NA	2.91	2.31	3.09	6.41	1.47
ae2	NA	8/8	NA	3.00	1.36	2.51	4.45	4.22
ar1	AA-2	4/4	NA	-0.69	-1.52	-0.33	5.31	0.80
ar2	AA-2	8/8	NA	-0.43	-2.31	-1.17	3.49	3.51
ar3	CC-2	4/4	NA	7.66	7.40	7.59	12.02	4.28
ar4	CC-2	8/8	NA	7.91	6.62	7.11	10.21	7.44
ar5	LiBr-1	4/4	NA	5.60	5.24	5.62	9.89	2.26
ar6	LiBr-1	8/8	NA	5.79	4.39	5.02	7.98	5.58
ar7	VC-2	4/4	NA	7.19	6.82	7.23	11.65	4.44
ar8	VC-2	8/8	NA	7.36	5.92	6.67	9.77	7.57
ar9	VC-1	4/4	IWLS	10.41	10.18	10.33	14.30	5.66
ar10	VC-1	4/4	IWLL	10.21	9.93	10.17	14.10	5.60
ar11	VC-1	4/4	IDLS	10.12	9.90	9.88	14.10	5.50
ar12	VC-1	4/4	IDLL	9.84	9.58	9.61	13.83	5.41
ar13	VC-1	8/8	IWLS	9.49	8.07	9.01	11.44	8.49
ar14	VC-1	8/8	IWLL	9.34	7.85	8.92	11.26	8.49
ar15	VC-1	8/8	IDLS	8.61	7.20	7.83	10.79	8.02
ar16	VC-1	8/8	IDLL	8.57	7.09	7.82	10.70	8.12
ar17	VC-2	4/4	WWLS	7.66	7.33	7.66	11.32	3.22
ar18	VC-2	4/4	WWLL	7.59	7.23	7.61	11.24	3.22
ar19	VC-2	4/4	WDLS	7.85	7.61	7.65	11.57	3.19
ar20	VC-2	4/4	WDLL	7.70	7.42	7.49	11.42	3.16
ar21	VC-2	8/8	WWLS	6.65	5.11	6.27	8.35	6.08
ar22	VC-2	8/8	WWLL	6.75	5.19	6.39	8.43	6.18
ar23	VC-2	8/8	WDLS	6.89	5.49	6.12	8.72	6.00
ar24	VC-2	8/8	WDLL	6.91	5.46	6.15	8.71	6.11

Table C.8. Combined-Cycle NPV ($M); Uncooled Base; 12 hr/day plant; 4 or 8 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
ce1	NA	12/4	NA	4.40	3.36	4.71	10.28	1.98
ce2	NA	12/8	NA	4.90	2.21	4.12	8.21	6.62
ce3	NA	12/12	NA	5.95	1.89	5.77	10.09	7.38
ce4	NA	24/4	NA	3.71	2.69	4.03	9.61	1.31
ce5	NA	24/8	NA	4.21	1.54	3.44	7.54	5.94
ce6	NA	24/12	NA	5.89	-0.53	5.62	10.39	9.47
ce7	NA	24/24	NA	6.17	-0.24	5.84	11.55	9.69
cr1	CC-4	12/4	NA	2.20	0.72	2.89	10.49	2.95
cr2	CC-4	12/8	NA	2.62	-0.74	1.86	8.43	7.72
cr3	CC-4	12/4	WDLS	8.97	7.79	8.66	15.49	3.63
cr4	CC-4	12/4	WDLL	8.08	6.75	7.78	14.60	3.37
cr5	CC-4	12/8	WDLS	-2.12	-6.16	-3.35	3.94	4.21
cr6	CC-4	12/8	WDLL	3.26	-0.46	2.25	7.98	6.95
cr7	CC-3	12/4	IDLS	9.85	8.41	9.55	17.06	6.12
cr8	CC-3	12/4	IDLL	9.09	7.52	8.83	16.18	5.90
cr9	CC-3	12/8	IDLS	-12.61	-17.73	-13.62	-4.25	1.11
cr10	CC-3	12/8	IDLL	0.42	-3.80	-0.34	6.31	7.49
cr11	LiBr-2	12/4	NA	1.36	-0.28	2.05	10.22	1.15
cr12	LiBr-2	12/8	NA	2.10	-1.29	0.88	8.27	6.76
cr13	LiBr-2	12/4	WDLS	8.55	7.30	8.25	15.53	3.74
cr14	LiBr-2	12/4	WDLL	8.87	7.62	8.49	15.59	3.87
cr15	LiBr-2	12/8	WDLS	-5.04	-9.27	-5.28	3.14	4.17
cr16	LiBr-2	12/8	WDLL	4.50	0.77	3.30	9.80	7.86
cr17	VC-2	12/4	NA	9.46	8.45	9.57	17.07	6.60
cr18	VC-2	12/8	NA	10.82	8.17	9.34	15.88	12.14
cr19	VC-1	12/4	IWLS	16.61	15.84	16.50	22.79	9.36
cr20	VC-1	12/4	IWLL	16.36	15.53	16.30	22.54	9.30
cr21	VC-1	12/4	IDLS	15.74	15.00	15.22	22.17	8.89
cr22	VC-1	12/4	IDLL	15.45	14.66	14.94	21.89	8.84
cr23	VC-1	12/8	IWLS	15.17	12.46	14.24	19.03	14.12
cr24	VC-1	12/8	IWLL	15.28	12.48	14.41	19.10	14.30
cr25	VC-1	12/8	IDLS	12.53	9.83	10.80	17.08	12.75

Table C.8. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
cr26	VC-1	12/8	IDLL	13.37	10.59	11.73	17.74	13.36
cr27	VC-2	12/4	WWLS	11.78	10.89	11.80	17.72	5.12
cr28	VC-2	12/4	WWLL	11.73	10.81	11.78	17.67	5.15
cr29	VC-2	12/4	WDLS	12.15	11.47	11.67	18.25	5.02
cr30	VC-2	12/4	WDLL	11.99	11.27	11.51	18.10	5.02
cr31	VC-2	12/8	WWLS	10.10	7.26	9.34	13.67	9.73
cr32	VC-2	12/8	WWLL	10.58	7.72	9.80	14.09	10.03
cr33	VC-2	12/8	WDLS	10.42	7.90	8.70	14.36	9.44
cr34	VC-2	12/8	WDLL	10.93	8.35	9.25	14.76	9.87
cr35	AA-2	12/4	NA	-20.11	-23.10	-18.33	-7.19	-6.23
cr36	AA-2	12/8	NA	-19.28	-24.12	-20.14	-8.94	-2.52
cr37	AA-2	12/4	WDLS	-1.09	-3.56	-1.66	7.30	-0.36
cr38	AA-2	12/4	WDLL	2.08	-0.04	1.54	9.78	1.01
cr39	AA-2	12/8	WDLS	-43.27	-51.70	-43.99	-28.56	-11.10
cr40	AA-2	12/8	WDLL	-8.82	-13.90	-10.11	-1.41	2.50
cr41	AA-1	12/4	IDLS	-3.99	-6.91	-4.51	5.33	0.21
cr42	AA-1	12/4	IDLL	0.57	-1.94	0.09	9.02	2.32
cr43	AA-1	12/8	IDLS	-68.93	-79.01	-69.06	-50.76	-22.72
cr44	AA-1	12/8	IDLL	-16.68	-22.55	-17.65	-7.73	0.37
cr45	CC-4/CC-5	12/4	NA	2.78	1.05	3.60	12.23	4.55
cr46	CC-4/CC-5	12/8	NA	3.14	-0.53	2.50	10.28	9.20
cr47	AA-2/AA-3	12/4	NA	-40.11	-44.23	-37.51	-24.19	-20.34
cr48	AA-2/AA-3	12/8	NA	-39.21	-45.36	-39.48	-25.63	-16.87
cr49	VC-2/VC-3	12/4	NA	13.30	11.98	13.57	22.38	11.85
cr50	VC-2/VC-3	12/8	NA	15.66	12.61	14.29	22.34	18.38
cr51	CC-3/CC-6	12/4	IDLS	17.13	15.46	16.68	25.77	13.40
cr52	CC-3/CC-6	12/8	IDLS	-15.7	-21.4	-16.9	-5.43	-1.22

Table C.8. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
				9	7	5		
cr53	AA-1/AA-3	12/4	IDLS	-13.45	-17.17	-13.58	-1.51	-6.29
cr54	AA-1/AA-3	12/8	IDLS	-83.09	-94.47	-82.42	-61.06	-30.64
cr55	VC-1/VC-3	12/4	IWLS	21.03	19.96	21.04	28.54	14.81
cr56	VC-1/VC-3	12/8	IWLS	20.42	17.29	19.64	25.79	20.54
cr57	VC-1/VC-3	12/4	IWLL	21.41	20.31	21.39	28.90	15.31
cr58	VC-1/VC-3	12/8	IWLL	21.72	18.51	20.85	27.03	21.75
cr59	VC-1/VC-3	12/4	IDLS	20.10	19.07	19.68	27.88	14.31
cr60	VC-1/VC-3	12/8	IDLS	17.60	14.49	15.96	23.71	19.09
cr61	VC-1/VC-3	12/4	IDLL	20.43	19.37	19.91	28.19	14.80
cr62	VC-1/VC-3	12/8	IDLL	19.60	16.39	17.94	25.48	20.74
cr63	CC-3/CC-6	12/4	IDLL	15.52	13.71	15.14	23.99	12.42
cr64	CC-3/CC-6	12/8	IDLL	-2.38	-7.11	-3.17	5.20	5.04
cr65	AA-1/AA-3	12/4	IDLL	-16.69	-20.28	-16.65	-5.69	-10.31
cr66	AA-1/AA-3	12/8	IDLL	-35.31	-42.64	-35.52	-23.33	-12.62
cr67	CC-2	12/4	NA	7.72	6.66	7.95	15.31	5.37
cr68	CC-2	12/8	NA	8.74	5.98	7.48	13.78	10.65
cr69	CC-2	12/4	WDLS	9.78	8.79	9.35	16.19	3.91
cr70	CC-2	12/4	WDLL	9.72	8.69	9.31	16.07	3.99
cr71	CC-2	12/8	WDLS	0.79	-2.75	-0.68	6.35	5.24
cr72	CC-2	12/8	WDLL	6.76	3.65	5.39	11.07	8.17
cr73	CC-1	12/4	IDLS	11.09	9.87	10.63	18.12	6.63
cr74	CC-1	12/4	IDLL	11.48	10.27	11.06	18.33	6.91
cr75	CC-1	12/8	IDLS	-8.26	-12.79	-9.66	-0.62	2.93
cr76	CC-1	12/8	IDLL	5.52	2.03	4.26	10.83	9.64
cr77	CC-2/CC-5	12/4	NA	8.79	7.52	9.11	17.48	7.18
cr78	CC-2/CC-5	12/8	NA	9.80	6.78	8.62	16.11	12.40
cr79	CC-1/CC-6	12/4	IDLS	18.66	17.29	17.98	27.09	14.05
cr80	CC-1/CC-6	12/8	IDLS	-10.69	-15.63	-12.39	-1.16	0.92

Table C.8. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
cr81	CC-1/CC-6	12/4	IDLL	21.04	19.66	20.45	29.19	16.49
cr82	CC-1/CC-6	12/8	IDLL	14.80	11.08	13.36	21.53	19.24
cr83	VC-2/CC-5	12/4	NA	10.56	9.34	10.76	19.28	8.40
cr84	VC-2/CC-5	12/8	NA	11.95	9.05	10.53	18.27	13.89
cr85	VC-1/CC-6	12/4	IWLL	23.09	22.04	22.96	30.59	15.85
cr86	VC-2/CC-6	12/8	IWLL	13.37	10.12	12.51	18.73	12.16

Table C.9. Combined-Cycle NPV ($M); Uncooled Base; 12 hr/day plant; 12 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
dr1	CC-4	12/12	NA	4.45	-0.56	5.74	11.63	9.58
dr2	LiBr-2	12/12	NA	3.62	-0.84	4.58	11.80	9.39
dr3	AA-2	12/12	NA	-17.28	-22.39	-15.11	-4.49	-0.48
dr4	VC-2	12/12	NA	15.81	11.99	15.74	21.90	16.17
dr5	VC-1	12/12	IWLS	18.31	13.28	18.98	22.71	17.16
dr6	VC-1	12/12	IWLL	18.98	13.90	19.70	23.25	17.76
dr7	VC-1	12/12	IDLS	12.10	7.33	12.07	18.10	14.06
dr8	VC-1	12/12	IDLL	15.69	10.94	15.61	21.05	16.22
dr9	VC-2	12/12	WWLS	12.44	7.33	13.19	16.41	12.10
dr10	VC-2	12/12	WWLL	13.68	8.62	14.36	17.51	12.81
dr11	VC-2	12/12	WDLS	12.23	7.80	11.91	17.01	11.39
dr12	VC-2	12/12	WDLL	13.95	9.50	13.63	18.40	12.54
dr13	CC-4/CC-5	12/12	NA	4.48	-1.12	7.16	13.29	10.39
dr14	AA-2/AA-3	12/12	NA	-37.66	-44.14	-33.43	-20.69	-14.55
dr15	VC-2/VC-3	12/12	NA	23.73	19.48	24.39	31.88	25.89
dr16	VC-1/VC-3	12/12	IWLL	27.89	22.35	29.24	34.06	28.35
dr17	CC-2	12/12	NA	12.90	8.68	13.23	18.86	13.90
dr18	CC-2/CC-5	12/12	NA	13.66	8.93	15.30	21.16	15.09
dr19	VC-2/CC-5	12/12	NA	16.71	12.40	17.92	24.35	17.45
dr20	VC-2/CC-5	12/12	IWLL	19.83	14.00	22.02	25.50	18.98

Table C.10. Combined-Cycle NPV ($M); Uncooled Base; 24 hr/day plant; 4 or 8 or 12 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er1	CC-4	24/4	NA	2.90	1.87	2.84	12.05	3.36
er2	CC-4	24/8	NA	4.26	1.80	2.04	11.71	8.84
er3	CC-4	24/12	NA	3.59	-4.83	5.14	11.10	11.55
er4	CC-4	24/4	WDLS	11.51	11.05	10.53	18.51	4.68
er5	CC-4	24/4	WDLL	10.72	10.15	9.72	17.77	4.38
er6	CC-4	24/8	WDLS	9.34	7.22	6.67	15.06	9.09
er7	CC-4	24/8	WDLL	8.63	6.34	6.03	14.34	8.96
er8	CC-4	24/12	WDLS	8.26	0.08	9.16	13.37	12.91
er9	CC-4	24/12	WDLL	7.47	-0.78	8.46	12.55	12.91
er10	CC-3	24/4	IDLS	15.30	14.77	14.32	22.73	8.79
er11	CC-3	24/4	IDLL	14.18	13.53	13.19	21.65	8.33
er12	CC-3	24/8	IDLS	11.88	9.53	9.25	18.36	12.89
er13	CC-3	24/8	IDLL	11.07	8.54	8.57	17.48	12.68
er14	CC-3	24/12	IDLS	6.71	-2.48	8.42	13.28	14.03
er15	CC-3	24/12	IDLL	6.85	-2.26	8.62	13.05	14.36
er16	LiBr-2	24/4	NA	1.60	0.42	1.59	11.30	1.13
er17	LiBr-2	24/8	NA	2.85	0.33	0.22	10.63	7.03
er18	LiBr-2	24/12	NA	3.34	-4.73	4.36	11.81	12.05
er19	LiBr-2	24/4	WDLS	11.71	11.26	10.60	18.77	4.80
er20	LiBr-2	24/4	WDLL	11.39	10.90	10.27	18.46	4.69
er21	LiBr-2	24/8	WDLS	9.61	7.26	6.69	15.43	9.26
er22	LiBr-2	24/8	WDLL	9.80	7.45	6.93	15.57	9.52
er23	LiBr-2	24/12	WDLS	9.03	0.62	9.20	14.52	13.29
er24	LiBr-2	24/12	WDLL	10.25	2.32	10.56	15.53	14.17
er25	VC-2	24/4	NA	10.15	9.60	9.52	18.63	7.00
er26	VC-2	24/8	NA	12.46	10.71	9.52	19.16	13.26
er27	VC-2	24/12	NA	15.78	9.28	15.68	22.29	18.94
er28	VC-1	24/4	IWLS	17.14	16.78	16.33	24.20	9.70
er29	VC-1	24/4	IWLL	16.95	16.53	16.19	24.00	9.66
er30	VC-1	24/4	IDLS	16.33	16.02	15.11	23.64	9.26
er31	VC-1	24/4	IDLL	16.05	15.68	14.83	23.36	9.20
er32	VC-1	24/8	IWLS	16.17	14.22	13.92	21.75	14.97
er33	VC-1	24/8	IWLL	16.68	14.70	14.45	22.19	15.33

Table C.10. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er34	VC-1	24/8	IDLS	13.97	12.09	10.85	20.19	13.80
er35	VC-1	24/8	IDLL	14.83	12.91	11.79	20.87	14.42
er36	VC-1	24/12	IWLS	11.78	3.84	13.30	17.63	17.42
er37	VC-1	24/12	IWLL	15.08	7.39	16.32	20.41	19.25
er38	VC-1	24/12	IDLS	11.48	4.24	11.67	17.92	16.62
er39	VC-1	24/12	IDLL	15.14	7.98	15.23	20.91	18.89
er40	VC-2	24/4	WWLS	12.04	11.58	11.36	18.82	5.12
er41	VC-2	24/4	WWLL	11.92	11.42	11.28	18.70	5.11
er42	VC-2	24/4	WDLS	12.34	12.07	11.16	19.28	4.98
er43	VC-2	24/4	WDLL	12.18	11.88	11.01	19.13	4.99
er44	VC-2	24/8	WWLS	10.93	8.93	8.79	16.11	10.07
er45	VC-2	24/8	WWLL	11.19	9.14	9.07	16.32	10.26
er46	VC-2	24/8	WDLS	11.05	9.35	7.98	16.61	9.68
er47	VC-2	24/8	WDLL	11.56	9.82	8.53	17.02	10.11
er48	VC-2	24/12	WWLS	9.30	1.75	10.44	14.31	14.35
er49	VC-2	24/12	WWLL	10.61	3.13	11.67	15.40	15.01
er50	VC-2	24/12	WDLS	11.93	5.19	11.70	17.03	14.28
er51	VC-2	24/12	WDLL	13.67	6.96	13.43	18.46	15.43
er52	AA-2	24/4	NA	-19.41	-21.95	-18.38	-5.64	-5.83
er53	AA-2	24/8	NA	-17.64	-21.58	-19.96	-5.65	-1.39
er54	AA-2	24/12	NA	-17.80	-27.29	-15.49	-4.64	1.11
er55	AA-2	24/4	WDLS	7.71	6.74	6.54	15.37	3.13
er56	AA-2	24/4	WDLL	7.89	6.93	6.72	15.47	3.24
er57	AA-2	24/8	WDLS	-0.32	-3.76	-3.24	7.22	5.39
er58	AA-2	24/8	WDLL	2.91	-0.10	0.06	9.82	6.81
er59	AA-2	24/12	WDLS	-10.76	-20.88	-9.46	-1.39	5.87
er60	AA-2	24/12	WDLL	-0.34	-9.05	0.64	6.86	10.15
er61	AA-1	24/4	IDLS	10.03	8.89	8.86	18.17	6.53
er62	AA-1	24/4	IDLL	10.13	9.00	8.96	18.18	6.61
er63	AA-1	24/8	IDLS	-1.45	-5.38	-4.29	7.03	7.19
er64	AA-1	24/8	IDLL	2.91	-0.54	0.18	10.65	9.25
er65	AA-1	24/12	IDLS	-19.49	-31.09	-17.33	-8.26	2.93
er66	AA-1	24/12	IDLL	-4.51	-14.32	-2.72	4.07	9.76
er67	CC-4/CC-5	24/4	NA	4.92	3.69	4.88	15.31	6.34

Table C.10. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er68	CC-4/CC-5	24/8	NA	8.22	5.51	5.88	17.14	13.63
er69	CC-4/CC-5	24/12	NA	1.34	-8.05	4.80	10.61	9.83
er70	AA-2/AA-3	24/4	NA	-37.97	-41.59	-36.23	-21.12	-18.55
er71	AA-2/AA-3	24/8	NA	-34.13	-39.32	-36.10	-18.77	-12.44
er72	AA-2/AA-3	24/12	NA	-40.00	-51.89	-35.14	-22.53	-15.15
er73	VC-2/VC-3	24/4	NA	15.44	14.62	14.85	25.45	13.64
er74	VC-2/VC-3	24/8	NA	20.73	18.65	17.67	29.20	22.81
er75	VC-2/VC-3	24/12	NA	23.10	15.75	23.91	31.74	27.95
er76	CC-3	24/4	IWLS	16.20	15.63	15.66	23.36	9.27
er77	CC-3	24/4	IWLL	15.46	14.76	15.05	22.61	8.98
er78	CC-3	24/8	IWLS	14.32	11.93	12.64	20.10	14.15
er79	CC-3	24/8	IWLL	13.67	11.09	12.22	19.41	13.96
er80	CC-3	24/12	IWLS	7.13	-2.69	10.36	13.07	14.87
er81	CC-3	24/12	IWLL	7.22	-2.61	10.51	12.80	15.15
er82	AA-1	24/4	IWLS	12.94	11.80	12.86	20.35	7.96
er83	AA-1	24/4	IWLL	12.99	11.84	12.92	20.35	8.00
er84	AA-1	24/8	IWLS	6.12	2.36	5.72	12.67	10.88
er85	AA-1	24/8	IWLL	8.78	5.40	8.06	15.02	12.05
er86	AA-1	24/12	IWLS	-16.30	-29.18	-9.79	-7.01	5.96
er87	AA-1	24/12	IWLL	-2.78	-13.74	1.81	4.52	11.61
er88	CC-3/CC-6	24/4	IWLS	-8.87	-10.17	-9.40	0.35	-16.84
er89	CC-3/CC-6	24/8	IWLS	-50.99	-54.87	-52.52	-42.15	-53.06
er90	CC-3/CC-6	24/12	IWLS	-152.30	-165.10	-147.81	-141.25	-147.73
er91	CC-3/CC-6	24/4	IWLL	-10.08	-11.57	-10.42	-0.87	-17.48
er92	CC-3/CC-6	24/8	IWLL	-52.49	-56.67	-53.64	-43.69	-53.88
er93	CC-3/CC-6	24/12	IWLL	-154.56	-167.76	-149.70	-144.03	-149.45
er94	CC-3/CC-6	24/4	IDLS	24.81	24.15	23.57	33.71	18.06
er95	CC-3/CC-6	24/8	IDLS	15.70	13.04	12.67	24.05	16.41
er96	CC-3/CC-6	24/12	IDLS	3.54	-6.13	5.39	12.38	10.99
er97	CC-3/CC-6	24/4	IDLL	23.21	22.40	21.95	32.11	17.17
er98	CC-3/CC-6	24/8	IDLL	14.09	11.19	11.23	22.29	15.49
er99	CC-3/CC-6	24/12	IDLL	2.04	-7.58	3.97	10.36	9.73

Table C.10. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er100	AA-1/AA-3	24/4	IWLS	7.07	4.93	7.51	16.50	4.08
er101	AA-1/AA-3	24/8	IWLS	2.42	-2.61	2.57	11.29	9.65
er102	AA-1/AA-3	24/12	IWLS	10.43	3.18	13.20	16.97	17.66
er103	AA-1/AA-3	24/4	IWLL	-1.05	-3.39	-0.38	8.43	-2.17
er104	AA-1/AA-3	24/8	IWLL	-3.13	-7.99	-3.12	5.60	4.26
er105	AA-1/AA-3	24/12	IWLL	-0.08	-7.39	2.50	6.37	8.59
er106	AA-1/AA-3	24/4	IDLS	3.63	1.61	2.50	13.84	2.16
er107	AA-1/AA-3	24/8	IDLS	-6.01	-11.33	-8.86	4.90	5.29
er108	AA-1/AA-3	24/12	IDLS	-27.73	-42.09	-23.64	-13.43	-1.16
er109	AA-1/AA-3	24/4	IDLL	-4.45	-6.63	-5.38	5.78	-4.03
er110	AA-1/AA-3	24/8	IDLL	-9.73	-14.71	-12.30	0.60	0.97
er111	AA-1/AA-3	24/12	IDLL	-24.68	-37.25	-20.55	-12.62	-4.55
er112	VC-1/VC-3	24/4	IWLS	22.98	22.37	22.19	31.45	16.53
er113	VC-1/VC-3	24/8	IWLS	24.81	22.51	22.49	32.05	24.69
er114	VC-1/VC-3	24/12	IWLS	18.98	10.08	21.52	26.87	26.44
er115	VC-1/VC-3	24/4	IWLL	23.26	22.60	22.45	31.72	16.90
er116	VC-1/VC-3	24/8	IWLL	26.15	23.82	23.75	33.32	25.75
er117	VC-1/VC-3	24/12	IWLL	22.40	13.75	24.66	29.74	28.33
er118	VC-1/VC-3	24/4	IDLS	22.12	21.56	20.89	30.86	16.06
er119	VC-1/VC-3	24/8	IDLS	22.46	20.23	19.21	30.38	23.43
er120	VC-1/VC-3	24/12	IDLS	18.64	10.48	19.75	27.17	25.58
er121	VC-1/VC-3	24/4	IDLL	22.30	21.70	20.98	31.02	16.41
er122	VC-1/VC-3	24/8	IDLL	24.19	21.91	20.92	31.92	24.83
er123	VC-1/VC-3	24/12	IDLL	22.44	14.36	23.45	30.27	27.93
er124	CC-2	24/4	NA	8.42	7.81	7.90	16.87	5.77
er125	CC-2	24/8	NA	10.38	8.52	7.66	17.07	11.78
er126	CC-2	24/12	NA	12.52	5.48	12.94	18.84	16.36
er127	CC-2	24/4	WDLS	11.99	11.67	10.90	18.95	4.82
er128	CC-2	24/4	WDLL	11.66	11.29	10.55	18.62	4.71
er129	CC-2	24/8	WDLS	10.43	8.62	7.52	16.01	9.41
er130	CC-2	24/8	WDLL	10.58	8.72	7.71	16.07	9.62
er131	CC-2	24/12	WDLS	10.69	3.44	10.91	15.63	13.72
er132	CC-2	24/12	WDLL	11.64	4.47	11.86	16.33	14.39
er133	CC-1	24/4	IDLS	16.05	15.69	14.91	23.38	9.10

Table C.10. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er134	CC-1	24/4	IDLL	15.57	15.16	14.42	22.90	8.91
er135	CC-1	24/8	IDLS	13.56	11.58	10.60	19.80	13.58
er136	CC-1	24/8	IDLL	13.95	11.95	11.07	20.03	13.89
er137	CC-1	24/12	IDLS	10.54	2.68	11.23	16.79	15.97
er138	CC-1	24/12	IDLL	13.08	5.45	13.72	18.74	17.39
er139	CC-2/CC-5	24/4	NA	10.93	10.16	10.39	20.55	8.97
er140	CC-2/CC-5	24/8	NA	14.88	12.82	12.00	22.96	16.83
er141	CC-2/CC-5	24/12	NA	11.05	3.16	13.28	19.03	15.07
er142	CC-1	24/4	IWLS	16.83	16.43	16.09	23.92	9.53
er143	CC-1	24/4	IWLL	16.51	16.05	15.84	23.58	9.41
er144	CC-1	24/8	IWLS	15.69	13.66	13.59	21.31	14.71
er145	CC-1	24/8	IWLL	15.86	13.80	13.82	21.41	14.86
er146	CC-1	24/12	IWLS	10.77	2.32	12.83	16.49	16.68
er147	CC-1	24/12	IWLL	13.02	4.81	14.82	18.25	17.84
er148	CC-1/CC-6	24/4	IWLS	-8.15	-9.25	-8.90	0.99	-16.54
er149	CC-1/CC-6	24/8	IWLS	-49.41	-52.88	-51.42	-40.77	-52.41
er150	CC-1/CC-6	24/12	IWLS	-148.14	-159.38	-145.01	-137.35	-145.64
er151	CC-1/CC-6	24/4	IWLL	-8.94	-10.16	-9.56	0.19	-17.01
er152	CC-1/CC-6	24/8	IWLL	-50.10	-53.73	-51.90	-41.52	-52.90
er153	CC-1/CC-6	24/12	IWLL	-148.26	-159.70	-145.01	-138.10	-146.54
er154	CC-1/CC-6	24/4	IDLS	25.63	25.15	24.21	34.41	18.40
er155	CC-1/CC-6	24/8	IDLS	17.53	15.27	14.14	25.61	17.18
er156	CC-1/CC-6	24/12	IDLS	7.72	-0.51	8.45	16.20	13.10
er157	CC-1/CC-6	24/4	IDLL	24.71	24.17	23.30	33.47	17.81
er158	CC-1/CC-6	24/8	IDLL	17.22	14.89	13.94	25.06	16.81
er159	CC-1/CC-6	24/12	IDLL	8.81	0.80	9.51	16.54	13.03
er160	VC-2/CC-5	24/4	NA	12.70	11.98	12.04	22.35	10.19
er161	VC-2/CC-5	24/8	NA	17.02	15.09	13.91	25.13	18.32
er162	VC-2/CC-5	24/12	NA	14.48	7.18	16.15	22.67	17.75
er163	VC-1/CC-5	24/4	IWLL	26.00	25.45	24.99	34.46	18.40
er164	VC-1/CC-5	24/8	IWLL	20.00	17.70	17.42	27.23	18.17
er165	VC-1/CC-5	24/12	IWLL	10.84	2.77	12.18	18.22	14.87

Table C.11. Combined-Cycle NPV ($M); Uncooled Base; 24 hr/day plant; 24 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
fr1	CC-4	24/24	NA	1.18	-9.09	3.86	9.85	10.55
fr2	LiBr-2	24/24	NA	2.71	-6.15	4.12	11.95	12.04
fr3	AA-2	24/24	NA	-19.22	-29.94	-16.20	-4.97	0.64
fr4	VC-2	24/24	NA	15.78	8.93	15.79	23.30	19.23
fr5	CC-4/CC-5	24/24	NA	-7.92	-20.30	-1.50	2.66	1.31
fr6	AA-2/AA-3	24/24	NA	-46.51	-61.29	-39.20	-27.64	-20.66
fr7	VC-2/VC-3	24/24	NA	22.00	13.40	23.62	31.88	27.32
fr8	CC-2	24/24	NA	11.49	3.54	12.44	18.85	16.03
fr9	CC-2/CC-5	24/24	NA	3.29	-6.57	7.83	12.43	7.27
fr10	VC-2/CC-5	24/24	NA	7.83	-0.83	11.36	17.15	10.63

Table C.12. Industrial Turbine NPV ($M); Uncooled Base

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
ie1	NA	4/4	NA	2.86	2.22	3.04	6.43	1.35
ie2	NA	8/8	NA	3.10	1.44	2.59	5.02	4.17
ir1	AA-2	4/4	NA	-14.03	-15.84	-13.04	-5.78	-5.16
ir2	AA-2	8/8	NA	-13.38	-16.19	-14.24	-6.84	-3.01
ir3	CC-2	4/4	NA	3.75	3.13	3.81	8.54	2.24
ir4	CC-2	8/8	NA	4.36	2.79	3.39	7.47	5.35
ir5	LiBr-1	4/4	NA	2.05	1.30	2.25	7.05	0.65
ir6	LiBr-1	8/8	NA	2.53	0.80	1.59	5.80	4.02
ir7	VC-2	4/4	NA	4.07	3.34	4.27	8.90	2.95
ir8	VC-2	8/8	NA	4.51	2.71	3.75	7.70	6.06
ir9	VC-1	4/4	IWLS	9.09	8.52	9.13	12.90	4.89
ir10	VC-1	4/4	IWLL	8.92	8.30	9.00	12.73	4.85
ir11	VC-1	4/4	IDLS	8.48	7.94	8.23	12.47	4.56
ir12	VC-1	4/4	IDLL	8.28	7.71	8.04	12.28	4.52
ir13	VC-1	8/8	IWLS	7.49	5.64	7.13	9.85	7.41
ir14	VC-1	8/8	IWLL	7.59	5.68	7.27	9.92	7.55
ir15	VC-1	8/8	IDLS	5.64	3.80	4.70	8.48	6.45
ir16	VC-1	8/8	IDLL	6.26	4.38	5.38	8.98	6.90
ir17	VC-2	4/4	WWLS	6.25	5.60	6.39	9.93	2.53
ir18	VC-2	4/4	WWLL	6.24	5.56	6.39	9.91	2.56
ir19	VC-2	4/4	WDLS	6.56	6.06	6.32	10.35	2.46
ir20	VC-2	4/4	WDLL	6.46	5.93	6.22	10.26	2.47
ir21	VC-2	8/8	WWLS	4.60	2.65	4.37	6.79	5.12
ir22	VC-2	8/8	WWLL	4.99	3.04	4.74	7.14	5.35
ir23	VC-2	8/8	WDLS	4.91	3.21	3.96	7.36	4.93
ir24	VC-2	8/8	WDLL	5.32	3.59	4.40	7.69	5.26
ir25	CC-2/CC-5	4/4	NA	4.48	3.72	4.46	10.06	2.51
ir26	CC-2/CC-5	8/8	NA	5.78	4.06	4.65	9.76	6.25
ir27	AA-2/AA-3	4/4	NA	-27.70	-30.13	-26.47	-17.56	-16.53
ir28	AA-2/AA-3	8/8	NA	-26.3	-29.7	-27.2	-17.85	-14.04

Table C.12. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
				5	7	5		
ir29	VC-2/VC-3	4/4	NA	4.20	3.21	4.54	9.94	4.05
ir30	VC-2/VC-3	8/8	NA	4.50	2.35	3.89	8.75	7.02
ir31	VC-1/VC-3	4/4	IWLS	9.60	8.78	9.77	14.25	6.12
ir32	VC-1/VC-3	8/8	IWLS	7.76	5.55	7.57	11.09	8.49
ir33	VC-1/VC-3	4/4	IWLL	9.91	9.08	10.05	14.54	6.50
ir34	VC-1/VC-3	8/8	IWLL	8.76	6.52	8.49	12.04	9.41
ir35	VC-1/VC-3	4/4	IDLS	8.95	8.17	8.81	13.79	5.77
ir36	VC-1/VC-3	8/8	IDLS	5.79	3.60	4.98	9.64	7.47
ir37	VC-1/VC-3	4/4	IDLL	9.22	8.43	9.01	14.05	6.15
ir38	VC-1/VC-3	8/8	IDLL	7.29	5.05	6.45	10.98	8.71
ir39	VC-2/CC-5	4/4	NA	4.08	3.14	4.26	9.86	3.12
ir40	VC-2/CC-5	8/8	NA	5.10	3.09	4.23	9.34	6.74
ir41	VC-1/CC-5	4/4	IWLL	9.27	8.46	9.32	13.96	5.13
ir42	VC-1/CC-5	8/8	IWLL	8.42	6.29	8.01	11.72	8.32

Table C.13. Aeroderivative Turbine $/kW; Evaporatively-Cooled Base

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
ar1	AA-2	4/4	NA	1153	1157	1133	1039	894
ar2	AA-2	8/8	NA	1153	1161	1161	1043	899
ar3	CC-2	4/4	NA	411	407	408	385	386
ar4	CC-2	8/8	NA	411	410	414	388	376
ar5	LiBr-1	4/4	NA	500	497	493	449	575
ar6	LiBr-1	8/8	NA	500	502	506	454	483
ar7	VC-2	4/4	NA	374	372	369	340	327
ar8	VC-2	8/8	NA	374	375	378	343	318
ar9	VC-1	4/4	IWLS	206	210	198	207	224
ar10	VC-1	4/4	IWLL	203	208	196	205	222
ar11	VC-1	4/4	IDLS	231	232	238	226	247
ar12	VC-1	4/4	IDLL	224	226	232	219	241
ar13	VC-1	8/8	IWLS	308	320	286	313	277
ar14	VC-1	8/8	IWLL	284	296	263	288	256
ar15	VC-1	8/8	IDLS	385	391	392	373	338
ar16	VC-1	8/8	IDLL	333	340	340	322	292
ar17	VC-2	4/4	WWLS	314	324	299	323	384
ar18	VC-2	4/4	WWLL	307	317	292	315	377
ar19	VC-2	4/4	WDLS	284	284	296	280	387
ar20	VC-2	4/4	WDLL	277	277	289	273	379
ar21	VC-2	8/8	WWLS	468	488	426	483	441
ar22	VC-2	8/8	WWLL	427	448	388	441	405
ar23	VC-2	8/8	WDLS	424	429	434	416	450
ar24	VC-2	8/8	WDLL	376	382	386	368	399

Table C.14. Combined-Cycle $/kW; Evaporatively-Cooled Base; 12 hr/day plant; 4 or 8 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
cr1	CC-4	12/4	NA	629	631	610	522	420
cr2	CC-4	12/8	NA	629	636	667	528	498
cr3	CC-4	12/4	WDLS	324	331	340	310	373
cr4	CC-4	12/4	WDLL	325	336	346	306	377
cr5	CC-4	12/8	WDLS	803	823	842	754	825
cr6	CC-4	12/8	WDLL	627	661	689	562	639
cr7	CC-3	12/4	IDLS	346	356	359	329	304
cr8	CC-3	12/4	IDLL	332	345	349	312	283
cr9	CC-3	12/8	IDLS	1074	1102	1110	1005	861
cr10	CC-3	12/8	IDLL	751	793	806	672	533
cr11	LiBr-2	12/4	NA	698	710	669	549	627
cr12	LiBr-2	12/8	NA	698	719	742	558	642
cr13	LiBr-2	12/4	WDLS	347	358	358	316	365
cr14	LiBr-2	12/4	WDLL	311	324	326	284	338
cr15	LiBr-2	12/8	WDLS	920	945	914	797	823
cr16	LiBr-2	12/8	WDLL	555	595	589	476	510
cr17	VC-2	12/4	NA	336	337	320	296	232
cr18	VC-2	12/8	NA	336	339	339	299	257
cr19	VC-1	12/4	IWLS	163	169	151	160	145
cr20	VC-1	12/4	IWLL	159	165	148	157	142
cr21	VC-1	12/4	IDLS	190	193	192	181	167
cr22	VC-1	12/4	IDLL	182	185	184	173	159
cr23	VC-1	12/8	IWLS	260	274	239	260	212
cr24	VC-1	12/8	IWLL	235	249	215	235	191
cr25	VC-1	12/8	IDLS	343	352	350	327	276
cr26	VC-1	12/8	IDLL	289	298	295	273	228
cr27	VC-2	12/4	WWLS	243	254	222	244	221
cr28	VC-2	12/4	WWLL	236	247	215	237	214
cr29	VC-2	12/4	WDLS	221	223	223	211	226
cr30	VC-2	12/4	WDLL	213	216	216	203	217
cr31	VC-2	12/8	WWLS	376	398	340	384	319
cr32	VC-2	12/8	WWLL	341	362	306	346	287
cr33	VC-2	12/8	WDLS	349	358	358	336	334

Table C.14. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
cr34	VC-2	12/8	WDLL	304	314	313	290	287
cr35	AA-2	12/4	NA	1724	1745	1669	1440	1061
cr36	AA-2	12/8	NA	1724	1754	1801	1448	1256
cr37	AA-2	12/4	WDLS	749	780	789	701	730
cr38	AA-2	12/4	WDLL	640	673	684	594	629
cr39	AA-2	12/8	WDLS	2522	2603	2610	2285	2201
cr40	AA-2	12/8	WDLL	1367	1447	1472	1203	1169
cr41	AA-1	12/4	IDLS	773	804	804	726	599
cr42	AA-1	12/4	IDLL	661	694	696	614	496
cr43	AA-1	12/8	IDLS	2796	2878	2861	2563	2050
cr44	AA-1	12/8	IDLL	1547	1635	1638	1370	1014
cr45	CC-4/CC-5	12/4	NA	517	522	506	459	385
cr46	CC-4/CC-5	12/8	NA	517	525	532	462	415
cr47	AA-2/AA-3	12/4	NA	1700	1736	1646	1460	1098
cr48	AA-2/AA-3	12/8	NA	1700	1742	1741	1465	1203
cr49	VC-2/VC-3	12/4	NA	342	344	332	315	272
cr50	VC-2/VC-3	12/8	NA	342	346	344	317	285
cr51	CC-3/CC-6	12/4	IDLS	331	337	337	321	304
cr52	CC-3/CC-6	12/8	IDLS	865	886	880	820	709
cr53	AA-1/AA-3	12/4	IDLS	766	786	778	730	638
cr54	AA-1/AA-3	12/8	IDLS	1834	1900	1845	1662	1209
cr55	VC-1/VC-3	12/4	IWLS	235	238	227	234	228
cr56	VC-1/VC-3	12/8	IWLS	294	303	281	291	261
cr57	VC-1/VC-3	12/4	IWLL	234	237	226	233	227
cr58	VC-1/VC-3	12/8	IWLL	281	290	269	279	252
cr59	VC-1/VC-3	12/4	IDLS	251	253	252	245	238
cr60	VC-1/VC-3	12/8	IDLS	344	351	348	330	292
cr61	VC-1/VC-3	12/4	IDLL	249	250	249	243	236
cr62	VC-1/VC-3	12/8	IDLL	314	320	317	302	271
cr63	CC-3/CC-6	12/4	IDLL	331	337	338	320	305
cr64	CC-3/CC-6	12/8	IDLL	702	721	722	667	606
cr65	AA-1/AA-3	12/4	IDLL	988	1018	1008	928	813
cr66	AA-1/AA-3	12/8	IDLL	1584	1661	1633	1412	1083

Table C.14. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
cr67	CC-2	12/4	NA	374	373	368	332	299
cr68	CC-2	12/8	NA	374	376	390	336	335
cr69	CC-2	12/4	WDLS	307	313	321	294	360
cr70	CC-2	12/4	WDLL	281	287	296	267	334
cr71	CC-2	12/8	WDLS	721	737	756	681	759
cr72	CC-2	12/8	WDLL	448	464	481	417	488
cr73	CC-1	12/4	IDLS	326	335	339	311	292
cr74	CC-1	12/4	IDLL	284	292	296	269	254
cr75	CC-1	12/8	IDLS	984	1009	1018	924	801
cr76	CC-1	12/8	IDLL	534	555	564	495	426
cr77	CC-2/CC-5	12/4	NA	385	386	381	357	329
cr78	CC-2/CC-5	12/8	NA	385	387	394	359	348
cr79	CC-1/CC-6	12/4	IDLS	321	326	327	312	299
cr80	CC-1/CC-6	12/8	IDLS	818	836	832	779	685
cr81	CC-1/CC-6	12/4	IDLL	261	266	267	252	239
cr82	CC-1/CC-6	12/8	IDLL	397	409	408	374	321
cr83	VC-2/CC-5	12/4	NA	363	364	359	336	312
cr84	VC-2/CC-5	12/8	NA	363	366	366	338	310
cr85	VC-1/CC-6	12/4	IWLL	244	246	240	245	259
cr86	VC-2/CC-6	12/8	IWLL	438	443	427	444	451

Table C.15. Combined-Cycle $/kW; Evaporatively-Cooled Base; 12 hr/day plant; 12 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
dr1	CC-4	12/12	NA	656	640	638	528	444
dr2	LiBr-2	12/12	NA	810	792	766	578	474
dr3	AA-2	12/12	NA	1797	1760	1740	1448	1152
dr4	VC-2	12/12	NA	345	341	339	299	268
dr5	VC-1	12/12	IWLS	381	411	348	384	302
dr6	VC-1	12/12	IWLL	310	338	281	311	244
dr7	VC-1	12/12	IDLS	573	588	572	543	445
dr8	VC-1	12/12	IDLL	395	410	394	369	299
dr9	VC-2	12/12	WWLS	521	567	471	537	411
dr10	VC-2	12/12	WWLL	430	474	385	441	334
dr11	VC-2	12/12	WDLS	507	523	509	485	452
dr12	VC-2	12/12	WDLL	374	390	376	352	325
dr13	CC-4/CC-5	12/12	NA	533	527	522	462	401
dr14	AA-2/AA-3	12/12	NA	1766	1746	1711	1465	1175
dr15	VC-2/VC-3	12/12	NA	349	347	344	317	291
dr16	VC-1/VC-3	12/12	IWLL	327	344	309	322	278
dr17	CC-2	12/12	NA	384	378	378	336	307
dr18	CC-2/CC-5	12/12	NA	392	389	388	359	335
dr19	VC-2/CC-5	12/12	NA	370	367	366	338	316
dr20	VC-2/CC-5	12/12	IWLL	344	363	325	343	300

Table C.16. Combined-Cycle $/kW; Evaporatively-Cooled Base; 24 hr/day plant; 4 or 8 or 12 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er1	CC-4	24/4	NA	629	631	610	522	420
er2	CC-4	24/8	NA	629	636	638	528	418
er3	CC-4	24/12	NA	656	669	638	528	436
er4	CC-4	24/4	WDLS	231	234	241	222	280
er5	CC-4	24/4	WDLL	238	242	249	226	289
er6	CC-4	24/8	WDLS	372	381	381	357	354
er7	CC-4	24/8	WDLL	371	384	384	350	339
er8	CC-4	24/12	WDLS	548	558	550	524	452
er9	CC-4	24/12	WDLL	503	517	504	464	377
er10	CC-3	24/4	IDLS	197	201	203	188	189
er11	CC-3	24/4	IDLL	200	205	208	189	189
er12	CC-3	24/8	IDLS	357	368	364	340	282
er13	CC-3	24/8	IDLL	343	357	352	322	257
er14	CC-3	24/12	IDLS	596	609	595	564	440
er15	CC-3	24/12	IDLL	511	527	509	469	341
er16	LiBr-2	24/4	NA	698	710	669	549	627
er17	LiBr-2	24/8	NA	698	719	709	558	495
er18	LiBr-2	24/12	NA	810	857	766	578	445
er19	LiBr-2	24/4	WDLS	233	240	243	221	274
er20	LiBr-2	24/4	WDLL	229	236	240	217	271
er21	LiBr-2	24/8	WDLS	380	405	389	361	347
er22	LiBr-2	24/8	WDLL	349	373	359	325	310
er23	LiBr-2	24/12	WDLS	563	609	569	529	448
er24	LiBr-2	24/12	WDLL	453	486	453	410	338
er25	VC-2	24/4	NA	336	337	329	296	260
er26	VC-2	24/8	NA	336	339	339	299	257
er27	VC-2	24/12	NA	345	352	339	299	264
er28	VC-1	24/4	IWLS	165	171	156	163	162
er29	VC-1	24/4	IWLL	159	165	151	157	157
er30	VC-1	24/4	IDLS	190	193	196	181	184
er31	VC-1	24/4	IDLL	182	185	188	173	176
er32	VC-1	24/8	IWLS	274	289	250	274	221
er33	VC-1	24/8	IWLL	235	249	215	235	191
er34	VC-1	24/8	IDLS	343	352	350	327	276

Table C.16. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er35	VC-1	24/8	IDLL	289	298	295	273	228
er36	VC-1	24/12	IWLS	570	600	523	558	399
er37	VC-1	24/12	IWLL	412	440	375	401	284
er38	VC-1	24/12	IDLS	573	581	572	543	432
er39	VC-1	24/12	IDLL	395	406	394	369	289
er40	VC-2	24/4	WWLS	239	250	225	240	259
er41	VC-2	24/4	WWLL	236	247	222	237	256
er42	VC-2	24/4	WDLS	221	223	230	211	268
er43	VC-2	24/4	WDLL	213	216	222	203	259
er44	VC-2	24/8	WWLS	365	386	329	371	308
er45	VC-2	24/8	WWLL	341	362	306	346	287
er46	VC-2	24/8	WDLS	349	358	358	336	334
er47	VC-2	24/8	WDLL	304	314	313	290	287
er48	VC-2	24/12	WWLS	641	676	584	640	428
er49	VC-2	24/12	WWLL	558	596	502	554	363
er50	VC-2	24/12	WDLS	507	516	509	485	422
er51	VC-2	24/12	WDLL	374	386	376	352	304
er52	AA-2	24/4	NA	1724	1745	1669	1440	1061
er53	AA-2	24/8	NA	1724	1754	1740	1448	1088
er54	AA-2	24/12	NA	1797	1834	1740	1448	1150
er55	AA-2	24/4	WDLS	397	412	418	378	422
er56	AA-2	24/4	WDLL	381	396	404	360	407
er57	AA-2	24/8	WDLS	789	827	808	741	639
er58	AA-2	24/8	WDLL	680	716	701	629	537
er59	AA-2	24/12	WDLS	1354	1404	1355	1260	945
er60	AA-2	24/12	WDLL	984	1020	981	891	642
er61	AA-1	24/4	IDLS	364	378	379	346	305
er62	AA-1	24/4	IDLL	348	363	365	329	290
er63	AA-1	24/8	IDLS	775	812	790	730	547
er64	AA-1	24/8	IDLL	668	704	684	618	448
er65	AA-1	24/12	IDLS	1411	1463	1408	1319	934
er66	AA-1	24/12	IDLL	1030	1068	1022	934	626
er67	CC-4/CC-5	24/4	NA	517	522	506	459	385
er68	CC-4/CC-5	24/8	NA	517	525	521	462	388
er69	CC-4/CC-5	24/12	NA	533	543	522	462	400

Table C.16. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er70	AA-2/AA-3	24/4	NA	1700	1736	1646	1460	1098
er71	AA-2/AA-3	24/8	NA	1700	1742	1709	1465	1133
er72	AA-2/AA-3	24/12	NA	1766	1814	1711	1465	1187
er73	VC-2/VC-3	24/4	NA	342	344	337	315	285
er74	VC-2/VC-3	24/8	NA	342	346	344	317	285
er75	VC-2/VC-3	24/12	NA	349	354	344	317	290
er76	CC-3	24/4	IWLS	170	177	161	168	165
er77	CC-3	24/4	IWLL	170	177	160	167	165
er78	CC-3	24/8	IWLS	282	299	257	283	225
er79	CC-3	24/8	IWLL	263	281	237	263	205
er80	CC-3	24/12	IWLS	590	624	539	578	407
er81	CC-3	24/12	IWLL	517	560	457	499	324
er82	AA-1	24/4	IWLS	276	295	253	274	235
er83	AA-1	24/4	IWLL	261	280	239	259	223
er84	AA-1	24/8	IWLS	545	590	480	543	381
er85	AA-1	24/8	IWLL	453	494	396	450	314
er86	AA-1	24/12	IWLS	1322	1418	1180	1284	810
er87	AA-1	24/12	IWLL	963	1051	836	922	545
er88	CC-3/CC-6	24/4	IWLS	729	723	730	746	851
er89	CC-3/CC-6	24/8	IWLS	1414	1403	1404	1452	1599
er90	CC-3/CC-6	24/12	IWLS	2949	2926	2950	3044	3262
er91	CC-3/CC-6	24/4	IWLL	757	753	756	775	875
er92	CC-3/CC-6	24/8	IWLL	1522	1519	1498	1560	1682
er93	CC-3/CC-6	24/12	IWLL	3492	3507	3433	3574	3627
er94	CC-3/CC-6	24/4	IDLS	248	249	252	245	253
er95	CC-3/CC-6	24/8	IDLS	461	465	465	455	444
er96	CC-3/CC-6	24/12	IDLS	720	725	720	707	664
er97	CC-3/CC-6	24/4	IDLL	256	258	260	252	259
er98	CC-3/CC-6	24/8	IDLL	481	487	486	473	459
er99	CC-3/CC-6	24/12	IDLL	736	742	736	720	676
er100	AA-1/AA-3	24/4	IWLS	493	506	477	495	484
er101	AA-1/AA-3	24/8	IWLS	634	664	593	631	536
er102	AA-1/AA-3	24/12	IWLS	627	657	593	631	520
er103	AA-1/AA-3	24/4	IWLL	667	686	642	662	643
er104	AA-1/AA-3	24/8	IWLL	814	853	763	801	687

Table C.16. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er105	AA-1/AA-3	24/12	IWLL	801	840	763	801	659
er106	AA-1/AA-3	24/4	IDLS	543	553	552	536	518
er107	AA-1/AA-3	24/8	IDLS	758	788	765	730	611
er108	AA-1/AA-3	24/12	IDLS	1094	1142	1090	1025	768
er109	AA-1/AA-3	24/4	IDLL	743	757	755	722	692
er110	AA-1/AA-3	24/8	IDLL	983	1022	996	928	777
er111	AA-1/AA-3	24/12	IDLL	1247	1293	1235	1137	874
er112	VC-1/VC-3	24/4	IWLS	236	239	231	235	239
er113	VC-1/VC-3	24/8	IWLS	302	312	288	299	265
er114	VC-1/VC-3	24/12	IWLS	485	508	454	465	355
er115	VC-1/VC-3	24/4	IWLL	234	237	229	233	237
er116	VC-1/VC-3	24/8	IWLL	281	290	269	279	252
er117	VC-1/VC-3	24/12	IWLL	387	406	364	373	298
er118	VC-1/VC-3	24/4	IDLS	251	253	255	245	249
er119	VC-1/VC-3	24/8	IDLS	344	351	348	330	292
er120	VC-1/VC-3	24/12	IDLS	487	496	484	457	372
er121	VC-1/VC-3	24/4	IDLL	249	250	252	243	247
er122	VC-1/VC-3	24/8	IDLL	314	320	317	302	271
er123	VC-1/VC-3	24/12	IDLL	377	386	375	355	301
er124	CC-2	24/4	NA	374	373	368	332	299
er125	CC-2	24/8	NA	374	376	378	336	293
er126	CC-2	24/12	NA	384	389	378	336	301
er127	CC-2	24/4	WDLS	225	228	235	216	276
er128	CC-2	24/4	WDLL	222	225	232	212	273
er129	CC-2	24/8	WDLS	355	363	364	342	343
er130	CC-2	24/8	WDLL	322	331	331	308	308
er131	CC-2	24/12	WDLS	512	520	514	491	431
er132	CC-2	24/12	WDLL	404	413	405	381	331
er133	CC-1	24/4	IDLS	190	193	196	181	184
er134	CC-1	24/4	IDLL	185	188	191	176	180
er135	CC-1	24/8	IDLS	337	347	344	322	271
er136	CC-1	24/8	IDLL	294	304	301	279	233
er137	CC-1	24/12	IDLS	556	567	555	527	417
er138	CC-1	24/12	IDLL	406	415	405	379	296
er139	CC-2/CC-5	24/4	NA	385	386	381	357	329

Table C.16. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er140	CC-2/CC-5	24/8	NA	385	387	388	359	328
er141	CC-2/CC-5	24/12	NA	392	396	388	359	333
er142	CC-1	24/4	IWLS	166	172	157	163	163
er143	CC-1	24/4	IWLL	162	168	153	159	160
er144	CC-1	24/8	IWLS	271	286	248	271	219
er145	CC-1	24/8	IWLL	239	253	219	239	194
er146	CC-1	24/12	IWLS	554	584	508	543	389
er147	CC-1	24/12	IWLL	423	451	384	412	290
er148	CC-1/CC-6	24/4	IWLS	727	720	728	744	850
er149	CC-1/CC-6	24/8	IWLS	1408	1396	1399	1446	1597
er150	CC-1/CC-6	24/12	IWLS	2930	2905	2934	3026	3254
er151	CC-1/CC-6	24/4	IWLL	744	738	744	762	867
er152	CC-1/CC-6	24/8	IWLL	1471	1462	1456	1510	1655
er153	CC-1/CC-6	24/12	IWLL	3233	3221	3213	3332	3506
er154	CC-1/CC-6	24/4	IDLS	244	245	248	241	251
er155	CC-1/CC-6	24/8	IDLS	451	454	454	446	440
er156	CC-1/CC-6	24/12	IDLS	698	702	699	688	654
er157	CC-1/CC-6	24/4	IDLL	247	248	251	244	254
er158	CC-1/CC-6	24/8	IDLL	449	452	453	445	444
er159	CC-1/CC-6	24/12	IDLL	665	667	667	659	643
er160	VC-2/CC-5	24/4	NA	363	364	359	336	312
er161	VC-2/CC-5	24/8	NA	363	366	366	338	310
er162	VC-2/CC-5	24/12	NA	370	375	366	338	314
er163	VC-1/CC-5	24/4	IWLL	236	238	233	237	249
er164	VC-1/CC-5	24/8	IWLL	417	422	406	422	426
er165	VC-1/CC-5	24/12	IWLL	663	676	644	666	633

Table C.17. Combined-Cycle $/kW; Evaporatively-Cooled Base; 24 hr/day plant; 24 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
fr1	CC-4	24/24	NA	656	669	638	528	436
fr2	LiBr-2	24/24	NA	810	857	766	578	445
fr3	AA-2	24/24	NA	1797	1834	1740	1448	1150
fr4	VC-2	24/24	NA	345	352	339	299	264
fr5	CC-4/CC-5	24/24	NA	533	544	522	462	401
fr6	AA-2/AA-3	24/24	NA	1767	1815	1711	1466	1189
fr7	VC-2/VC-3	24/24	NA	349	354	344	317	291
fr8	CC-2	24/24	NA	384	389	378	336	301
fr9	CC-2/CC-5	24/24	NA	392	397	388	359	333
fr10	VC-2/CC-5	24/24	NA	370	375	366	339	315

Table C.18. Industrial Turbine $/kW; Evaporatively-Cooled Base

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
ir1	AA-2	4/4	NA	1400	1407	1374	1252	1050
ir2	AA-2	8/8	NA	1400	1412	1409	1258	1062
ir3	CC-2	4/4	NA	462	460	457	426	408
ir4	CC-2	8/8	NA	462	462	465	429	401
ir5	LiBr-1	4/4	NA	564	565	553	492	597
ir6	LiBr-1	8/8	NA	564	571	571	500	499
ir7	VC-2	4/4	NA	424	425	416	374	334
ir8	VC-2	8/8	NA	424	428	429	378	329
ir9	VC-1	4/4	IWLS	203	210	193	202	207
ir10	VC-1	4/4	IWLL	200	207	191	198	204
ir11	VC-1	4/4	IDLS	234	237	242	225	235
ir12	VC-1	4/4	IDLL	227	231	235	217	227
ir13	VC-1	8/8	IWLS	319	334	293	321	269
ir14	VC-1	8/8	IWLL	293	310	269	294	245
ir15	VC-1	8/8	IDLS	413	423	422	396	343
ir16	VC-1	8/8	IDLL	357	368	365	340	291
ir17	VC-2	4/4	WWLS	305	317	287	308	342
ir18	VC-2	4/4	WWLL	298	311	281	301	334
ir19	VC-2	4/4	WDLS	276	278	287	267	347
ir20	VC-2	4/4	WDLL	269	272	281	259	338
ir21	VC-2	8/8	WWLS	468	492	423	478	410
ir22	VC-2	8/8	WWLL	428	454	385	437	372
ir23	VC-2	8/8	WDLS	430	438	440	416	425
ir24	VC-2	8/8	WDLL	382	393	393	366	372
ir25	CC-2/CC-5	4/4	NA	474	472	472	459	462
ir26	CC-2/CC-5	8/8	NA	474	474	476	460	457
ir27	AA-2/AA-3	4/4	NA	1289	1300	1270	1195	1047
ir28	AA-2/AA-3	8/8	NA	1289	1303	1293	1198	1061
ir29	VC-2/VC-3	4/4	NA	441	444	435	406	367
ir30	VC-2/VC-3	8/8	NA	441	446	444	408	367
ir31	VC-1/VC-3	4/4	IWLS	297	300	291	296	305
ir32	VC-1/VC-3	8/8	IWLS	369	380	354	368	333
ir33	VC-1/VC-3	4/4	IWLL	297	301	291	296	304
ir34	VC-1/VC-3	8/8	IWLL	357	368	341	355	323

Table C.18. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
ir35	VC-1/VC-3	4/4	IDLS	316	318	321	310	318
ir36	VC-1/VC-3	8/8	IDLS	430	437	435	414	370
ir37	VC-1/VC-3	4/4	IDLL	315	317	320	308	316
ir38	VC-1/VC-3	8/8	IDLL	399	407	404	384	347
ir39	VC-2/CC-5	4/4	NA	478	479	474	453	439
ir40	VC-2/CC-5	8/8	NA	478	480	480	454	436
ir41	VC-1/CC-5	4/4	IWLL	353	355	349	356	382
ir42	VC-1/CC-5	8/8	IWLL	404	412	391	406	395

Table C.19. Aeroderivative Turbine NPV ($M); Evaporatively-Cooled Base

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
ar1	AA-2	4/4	NA	-3.63	-3.85	-3.44	-1.12	-0.67
ar2	AA-2	8/8	NA	-3.46	-3.68	-3.71	-0.97	-0.73
ar3	CC-2	4/4	NA	4.73	5.07	4.47	5.59	2.80
ar4	CC-2	8/8	NA	4.88	5.24	4.58	5.74	3.20
ar5	LiBr-1	4/4	NA	2.67	2.91	2.50	3.46	0.79
ar6	LiBr-1	8/8	NA	2.76	3.01	2.49	3.52	1.34
ar7	VC-2	4/4	NA	4.26	4.49	4.12	5.22	2.97
ar8	VC-2	8/8	NA	4.34	4.54	4.13	5.31	3.33
ar9	VC-1	4/4	IWLS	7.47	7.85	7.21	7.88	4.19
ar10	VC-1	4/4	IWLL	7.27	7.60	7.05	7.67	4.13
ar11	VC-1	4/4	IDLS	7.18	7.57	6.77	7.68	4.02
ar12	VC-1	4/4	IDLL	6.90	7.25	6.50	7.40	3.94
ar13	VC-1	8/8	IWLS	6.46	6.69	6.48	6.98	4.25
ar14	VC-1	8/8	IWLL	6.31	6.47	6.38	6.79	4.26
ar15	VC-1	8/8	IDLS	5.58	5.82	5.29	6.33	3.79
ar16	VC-1	8/8	IDLL	5.54	5.72	5.29	6.24	3.88
ar17	VC-2	4/4	WWLS	4.72	4.99	4.55	4.89	1.75
ar18	VC-2	4/4	WWLL	4.65	4.90	4.50	4.81	1.74
ar19	VC-2	4/4	WDLS	4.91	5.28	4.53	5.14	1.72
ar20	VC-2	4/4	WDLL	4.76	5.09	4.38	4.99	1.69
ar21	VC-2	8/8	WWLS	3.62	3.73	3.74	3.89	1.84
ar22	VC-2	8/8	WWLL	3.72	3.81	3.86	3.96	1.94
ar23	VC-2	8/8	WDLS	3.86	4.12	3.58	4.26	1.76
ar24	VC-2	8/8	WDLL	3.88	4.08	3.62	4.25	1.87

Table C.20. Combined-Cycle NPV ($M); Evaporatively-Cooled Base; 12 hr/day plant; 4 or 8 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
cr1	CC-4	12/4	NA	-2.49	-2.93	-2.11	-0.07	0.70
cr2	CC-4	12/8	NA	-2.06	-2.75	-2.37	1.34	0.19
cr3	CC-4	12/4	WDLS	4.29	4.15	3.66	4.93	1.39
cr4	CC-4	12/4	WDLL	3.40	3.11	2.78	4.04	1.12
cr5	CC-4	12/8	WDLS	-6.80	-8.17	-7.58	-3.15	-3.32
cr6	CC-4	12/8	WDLL	-1.42	-2.47	-1.98	0.89	-0.58
cr7	CC-3	12/4	IDLS	5.17	4.76	4.55	6.50	3.87
cr8	CC-3	12/4	IDLL	4.41	3.88	3.83	5.62	3.65
cr9	CC-3	12/8	IDLS	-17.29	-19.74	-17.85	-11.33	-6.42
cr10	CC-3	12/8	IDLL	-4.27	-5.81	-4.56	-0.78	-0.04
cr11	LiBr-2	12/4	NA	-3.33	-3.92	-2.95	-0.35	-1.09
cr12	LiBr-2	12/8	NA	-2.58	-3.30	-3.35	1.18	-0.76
cr13	LiBr-2	12/4	WDLS	3.87	3.66	3.25	4.96	1.49
cr14	LiBr-2	12/4	WDLL	4.19	3.97	3.49	5.03	1.62
cr15	LiBr-2	12/8	WDLS	-9.73	-11.28	-9.51	-3.95	-3.36
cr16	LiBr-2	12/8	WDLL	-0.19	-1.24	-0.93	2.71	0.33
cr17	VC-2	12/4	NA	4.27	4.32	4.38	5.09	4.97
cr18	VC-2	12/8	NA	5.63	5.68	4.93	7.37	5.24
cr19	VC-1	12/4	IWLS	11.42	11.71	11.31	10.81	7.74
cr20	VC-1	12/4	IWLL	11.17	11.40	11.11	10.56	7.68
cr21	VC-1	12/4	IDLS	10.55	10.87	10.03	10.19	7.27
cr22	VC-1	12/4	IDLL	10.26	10.53	9.75	9.91	7.22
cr23	VC-1	12/8	IWLS	9.98	9.96	9.82	10.53	7.22
cr24	VC-1	12/8	IWLL	10.09	9.98	9.99	10.60	7.40
cr25	VC-1	12/8	IDLS	7.34	7.34	6.38	8.57	5.85
cr26	VC-1	12/8	IDLL	8.18	8.10	7.31	9.23	6.46
cr27	VC-2	12/4	WWLS	6.59	6.76	6.61	5.74	3.50
cr28	VC-2	12/4	WWLL	6.54	6.69	6.59	5.69	3.53
cr29	VC-2	12/4	WDLS	6.96	7.34	6.48	6.27	3.40
cr30	VC-2	12/4	WDLL	6.80	7.14	6.32	6.12	3.40
cr31	VC-2	12/8	WWLS	4.91	4.76	4.92	5.16	2.83
cr32	VC-2	12/8	WWLL	5.39	5.22	5.38	5.59	3.13
cr33	VC-2	12/8	WDLS	5.23	5.40	4.29	5.86	2.54

Table C.20. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
cr34	VC-2	12/8	WDLL	5.74	5.85	4.84	6.25	2.97
cr35	AA-2	12/4	NA	-24.79	-26.75	-23.33	-17.76	-8.48
cr36	AA-2	12/8	NA	-23.97	-26.13	-24.37	-16.03	-10.04
cr37	AA-2	12/4	WDLS	-5.78	-7.21	-6.66	-3.27	-2.61
cr38	AA-2	12/4	WDLL	-2.61	-3.69	-3.46	-0.79	-1.23
cr39	AA-2	12/8	WDLS	-47.96	-53.71	-48.22	-35.65	-18.63
cr40	AA-2	12/8	WDLL	-13.51	-15.91	-14.34	-8.50	-5.03
cr41	AA-1	12/4	IDLS	-8.68	-10.55	-9.51	-5.23	-2.03
cr42	AA-1	12/4	IDLL	-4.11	-5.58	-4.91	-1.54	0.07
cr43	AA-1	12/8	IDLS	-73.61	-81.02	-73.29	-57.85	-30.25
cr44	AA-1	12/8	IDLL	-21.36	-24.56	-21.88	-14.82	-7.15
cr45	CC-4/CC-5	12/4	NA	-1.93	-2.62	-1.42	1.64	2.28
cr46	CC-4/CC-5	12/8	NA	-1.56	-2.57	-1.75	3.16	1.65
cr47	AA-2/AA-3	12/4	NA	-44.82	-47.90	-42.54	-34.78	-22.61
cr48	AA-2/AA-3	12/8	NA	-43.92	-47.40	-43.74	-32.74	-24.42
cr49	VC-2/VC-3	12/4	NA	8.09	7.83	8.35	10.37	10.21
cr50	VC-2/VC-3	12/8	NA	10.45	10.09	9.85	13.81	11.45
cr51	CC-3/CC-6	12/4	IDLS	12.42	11.79	11.65	15.18	11.13
cr52	CC-3/CC-6	12/8	IDLS	-20.50	-23.50	-21.20	-12.54	-8.77
cr53	AA-1/AA-3	12/4	IDLS	-18.15	-20.84	-18.61	-12.10	-8.56
cr54	AA-1/AA-3	12/8	IDLS	-87.80	-96.50	-86.67	-68.17	-38.19
cr55	VC-1/VC-3	12/4	IWLS	15.81	15.81	15.83	16.53	13.17
cr56	VC-1/VC-3	12/8	IWLS	15.21	14.78	15.20	17.26	13.62
cr57	VC-1/VC-3	12/4	IWLL	16.20	16.16	16.17	16.89	13.66
cr58	VC-1/VC-3	12/8	IWLL	16.50	16.00	16.41	18.50	14.82
cr59	VC-1/VC-3	12/4	IDLS	14.88	14.92	14.47	15.87	12.67
cr60	VC-1/VC-3	12/8	IDLS	12.39	11.97	11.52	15.18	12.16
cr61	VC-1/VC-3	12/4	IDLL	15.21	15.22	14.70	16.18	13.16
cr62	VC-1/VC-3	12/8	IDLL	14.39	13.87	13.50	16.95	13.81
cr63	CC-3/CC-6	12/4	IDLL	10.82	10.04	10.11	13.40	10.15
cr64	CC-3/CC-6	12/8	IDLL	-7.09	-9.14	-7.43	-1.92	-2.51
cr65	AA-1/AA-3	12/4	IDLL	-21.40	-23.95	-21.67	-16.28	-12.58
cr66	AA-1/AA-3	12/8	IDLL	-40.02	-44.67	-39.77	-30.45	-20.17
cr67	CC-2	12/4	NA	3.04	3.02	2.95	4.75	3.12
cr68	CC-2	12/8	NA	4.06	3.97	3.25	6.70	3.13

Table C.20. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
cr69	CC-2	12/4	WDLS	5.09	5.14	4.35	5.62	1.66
cr70	CC-2	12/4	WDLL	5.04	5.05	4.31	5.50	1.74
cr71	CC-2	12/8	WDLS	-3.89	-4.76	-4.91	-0.74	-2.28
cr72	CC-2	12/8	WDLL	2.07	1.65	1.17	3.98	0.64
cr73	CC-1	12/4	IDLS	6.41	6.23	5.63	7.55	4.38
cr74	CC-1	12/4	IDLL	6.80	6.62	6.06	7.77	4.67
cr75	CC-1	12/8	IDLS	-12.95	-14.80	-13.89	-7.71	-4.60
cr76	CC-1	12/8	IDLL	0.84	0.02	0.03	3.74	2.11
cr77	CC-2/CC-5	12/4	NA	4.08	3.85	4.09	6.89	4.91
cr78	CC-2/CC-5	12/8	NA	5.10	4.75	4.37	8.99	4.84
cr79	CC-1/CC-6	12/4	IDLS	13.95	13.62	12.95	16.50	11.78
cr80	CC-1/CC-6	12/8	IDLS	-15.40	-17.67	-16.64	-8.27	-6.63
cr81	CC-1/CC-6	12/4	IDLL	16.33	15.99	15.42	18.60	14.22
cr82	CC-1/CC-6	12/8	IDLL	10.09	9.04	9.11	14.42	11.69
cr83	VC-2/CC-5	12/4	NA	5.85	5.67	5.74	8.69	6.13
cr84	VC-2/CC-5	12/8	NA	7.24	7.01	6.60	11.15	7.33
cr85	VC-1/CC-6	12/4	IWLL	18.39	18.37	17.93	20.00	13.58
cr86	VC-2/CC-6	12/8	IWLL	8.66	8.09	8.58	11.62	5.59

Table C.20. (contd)

Table C.21. Combined-Cycle NPV ($M); Evaporatively-Cooled Base; 12 hr/day plant; 12 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
dr1	CC-4	12/12	NA	-1.66	-2.59	-0.18	1.37	2.07
dr2	LiBr-2	12/12	NA	-2.49	-2.86	-1.34	1.54	1.88
dr3	AA-2	12/12	NA	-23.39	-24.41	-21.03	-14.75	-7.99
dr4	VC-2	12/12	NA	9.70	9.96	9.82	11.65	8.66
dr5	VC-1	12/12	IWLS	12.20	11.26	13.07	12.45	9.65
dr6	VC-1	12/12	IWLL	12.88	11.87	13.78	13.00	10.25
dr7	VC-1	12/12	IDLS	5.99	5.31	6.15	7.85	6.55
dr8	VC-1	12/12	IDLL	9.59	8.91	9.70	10.80	8.71
dr9	VC-2	12/12	WWLS	6.33	5.31	7.28	6.15	4.58
dr10	VC-2	12/12	WWLL	7.58	6.60	8.44	7.25	5.30
dr11	VC-2	12/12	WDLS	6.13	5.78	5.99	6.75	3.88
dr12	VC-2	12/12	WDLL	7.85	7.48	7.71	8.15	5.03
dr13	CC-4/CC-5	12/12	NA	-1.66	-3.18	1.20	2.99	2.85
dr14	AA-2/AA-3	12/12	NA	-43.81	-46.20	-39.39	-30.98	-22.10
dr15	VC-2/VC-3	12/12	NA	17.59	17.41	18.43	21.59	18.34
dr16	VC-1/VC-3	12/12	IWLL	21.74	20.29	23.28	23.76	20.80
dr17	CC-2	12/12	NA	6.79	6.65	7.31	8.60	6.39
dr18	CC-2/CC-5	12/12	NA	7.52	6.86	9.35	10.86	7.55
dr19	VC-2/CC-5	12/12	NA	10.57	10.34	11.96	14.05	9.90
dr20	VC-2/CC-5	12/12	IWLL	13.69	11.94	16.06	15.21	11.43

Table C.20. (contd)

Table C.22. Combined-Cycle NPV ($M); Evaporatively-Cooled Base; 24 hr/day plant; 4 or 8 or 12 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er1	CC-4	24/4	NA	-1.10	-1.10	-1.48	2.16	1.79
er2	CC-4	24/8	NA	-0.24	-0.02	-1.69	3.88	2.62
er3	CC-4	24/12	NA	-2.47	-4.45	-0.63	0.52	1.93
er4	CC-4	24/4	WDLS	7.51	8.07	6.22	8.62	3.11
er5	CC-4	24/4	WDLL	6.72	7.17	5.40	7.88	2.80
er6	CC-4	24/8	WDLS	4.83	5.40	2.94	7.22	2.86
er7	CC-4	24/8	WDLL	4.12	4.52	2.29	6.50	2.74
er8	CC-4	24/12	WDLS	2.20	0.47	3.39	2.79	3.29
er9	CC-4	24/12	WDLL	1.41	-0.39	2.68	1.97	3.28
er10	CC-3	24/4	IDLS	11.30	11.80	10.00	12.84	7.22
er11	CC-3	24/4	IDLL	10.18	10.56	8.87	11.76	6.76
er12	CC-3	24/8	IDLS	7.38	7.71	5.52	10.53	6.66
er13	CC-3	24/8	IDLL	6.57	6.72	4.84	9.64	6.45
er14	CC-3	24/12	IDLS	0.65	-2.09	2.65	2.70	4.40
er15	CC-3	24/12	IDLL	0.79	-1.87	2.85	2.47	4.74
er16	LiBr-2	24/4	NA	-2.40	-2.55	-2.73	1.41	-0.44
er17	LiBr-2	24/8	NA	-1.66	-1.49	-3.51	2.79	0.80
er18	LiBr-2	24/12	NA	-2.72	-4.34	-1.41	1.23	2.43
er19	LiBr-2	24/4	WDLS	7.71	8.28	6.29	8.88	3.23
er20	LiBr-2	24/4	WDLL	7.39	7.92	5.95	8.57	3.11
er21	LiBr-2	24/8	WDLS	5.10	5.44	2.96	7.59	3.04
er22	LiBr-2	24/8	WDLL	5.30	5.63	3.20	7.73	3.29
er23	LiBr-2	24/12	WDLS	2.97	1.01	3.43	3.94	3.67
er24	LiBr-2	24/12	WDLL	4.20	2.71	4.79	4.95	4.55
er25	VC-2	24/4	NA	6.15	6.63	5.20	8.73	5.43
er26	VC-2	24/8	NA	7.96	8.89	5.79	11.33	7.04
er27	VC-2	24/12	NA	9.72	9.67	9.91	11.71	9.32
er28	VC-1	24/4	IWLS	13.14	13.80	12.01	14.31	8.13
er29	VC-1	24/4	IWLL	12.95	13.56	11.87	14.11	8.09
er30	VC-1	24/4	IDLS	12.33	13.04	10.79	13.75	7.69
er31	VC-1	24/4	IDLL	12.05	12.71	10.51	13.46	7.63
er32	VC-1	24/8	IWLS	11.67	12.40	10.19	13.92	8.75

Table C.20. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er33	VC-1	24/8	IWLL	12.18	12.88	10.72	14.36	9.11
er34	VC-1	24/8	IDLS	9.47	10.27	7.12	12.36	7.57
er35	VC-1	24/8	IDLL	10.33	11.09	8.05	13.04	8.19
er36	VC-1	24/12	IWLS	5.73	4.23	7.53	7.05	7.80
er37	VC-1	24/12	IWLL	9.03	7.77	10.55	9.83	9.63
er38	VC-1	24/12	IDLS	5.42	4.63	5.90	7.34	6.99
er39	VC-1	24/12	IDLL	9.08	8.37	9.46	10.33	9.26
er40	VC-2	24/4	WWLS	8.04	8.61	7.04	8.93	3.55
er41	VC-2	24/4	WWLL	7.92	8.45	6.96	8.81	3.54
er42	VC-2	24/4	WDLS	8.34	9.10	6.85	9.39	3.41
er43	VC-2	24/4	WDLL	8.18	8.91	6.69	9.24	3.42
er44	VC-2	24/8	WWLS	6.42	7.11	5.06	8.28	3.85
er45	VC-2	24/8	WWLL	6.68	7.32	5.33	8.49	4.04
er46	VC-2	24/8	WDLS	6.54	7.52	4.24	8.78	3.45
er47	VC-2	24/8	WDLL	7.06	8.00	4.80	9.19	3.88
er48	VC-2	24/12	WWLS	3.24	2.14	4.67	3.73	4.73
er49	VC-2	24/12	WWLL	4.55	3.52	5.90	4.82	5.39
er50	VC-2	24/12	WDLS	5.87	5.57	5.93	6.45	4.66
er51	VC-2	24/12	WDLL	7.62	7.35	7.66	7.88	5.81
er52	AA-2	24/4	NA	-23.41	-24.92	-22.70	-15.53	-7.40
er53	AA-2	24/8	NA	-22.15	-23.40	-23.69	-13.49	-7.62
er54	AA-2	24/12	NA	-23.86	-26.90	-21.26	-15.22	-8.51
er55	AA-2	24/4	WDLS	3.71	3.76	2.22	5.47	1.56
er56	AA-2	24/4	WDLL	3.89	3.96	2.40	5.58	1.67
er57	AA-2	24/8	WDLS	-4.82	-5.58	-6.97	-0.61	-0.84
er58	AA-2	24/8	WDLL	-1.59	-1.92	-3.67	1.99	0.59
er59	AA-2	24/12	WDLS	-16.82	-20.50	-15.23	-11.97	-3.76
er60	AA-2	24/12	WDLL	-6.39	-8.66	-5.13	-3.72	0.53
er61	AA-1	24/4	IDLS	6.03	5.92	4.55	8.28	4.96
er62	AA-1	24/4	IDLL	6.13	6.03	4.65	8.28	5.04
er63	AA-1	24/8	IDLS	-5.96	-7.21	-8.03	-0.80	0.96
er64	AA-1	24/8	IDLL	-1.60	-2.36	-3.56	2.82	3.03
er65	AA-1	24/12	IDLS	-25.55	-30.71	-23.10	-18.84	-6.69
er66	AA-1	24/12	IDLL	-10.57	-13.93	-8.49	-6.51	0.14
er67	CC-4/CC-5	24/4	NA	0.89	0.69	0.54	5.39	4.74

Table C.20. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er68	CC-4/CC-5	24/8	NA	3.69	3.66	2.13	9.28	7.38
er69	CC-4/CC-5	24/12	NA	-4.76	-7.70	-1.01	-0.01	0.17
er70	AA-2/AA-3	24/4	NA	-41.99	-44.59	-40.57	-31.03	-20.14
er71	AA-2/AA-3	24/8	NA	-38.67	-41.17	-39.86	-26.63	-18.69
er72	AA-2/AA-3	24/12	NA	-46.09	-51.54	-40.95	-33.15	-24.82
er73	VC-2/VC-3	24/4	NA	11.42	11.62	10.50	15.53	12.05
er74	VC-2/VC-3	24/8	NA	16.20	16.80	13.91	21.34	16.56
er75	VC-2/VC-3	24/12	NA	17.01	16.10	18.10	21.12	18.29
er76	CC-3	24/4	IWLS	12.20	12.65	11.34	13.47	7.70
er77	CC-3	24/4	IWLL	11.46	11.78	10.73	12.72	7.41
er78	CC-3	24/8	IWLS	9.81	10.10	8.90	12.27	7.93
er79	CC-3	24/8	IWLL	9.17	9.27	8.48	11.58	7.74
er80	CC-3	24/12	IWLS	1.07	-2.30	4.59	2.49	5.25
er81	CC-3	24/12	IWLL	1.16	-2.22	4.74	2.22	5.53
er82	AA-1	24/4	IWLS	8.94	8.82	8.54	10.46	6.38
er83	AA-1	24/4	IWLL	8.99	8.86	8.60	10.45	6.43
er84	AA-1	24/8	IWLS	1.62	0.54	1.99	4.83	4.66
er85	AA-1	24/8	IWLL	4.28	3.57	4.32	7.18	5.83
er86	AA-1	24/12	IWLS	-22.35	-28.79	-15.56	-17.59	-3.67
er87	AA-1	24/12	IWLL	-8.83	-13.35	-3.96	-6.06	1.99
er88	CC-3/CC-6	24/4	IWLS	-12.90	-13.17	-13.74	-9.57	-18.43
er89	CC-3/CC-6	24/8	IWLS	-55.52	-56.71	-56.28	-50.01	-59.31
er90	CC-3/CC-6	24/12	IWLS	-158.39	-164.75	-153.62	-151.87	-157.39
er91	CC-3/CC-6	24/4	IWLL	-14.11	-14.57	-14.77	-10.79	-19.07
er92	CC-3/CC-6	24/8	IWLL	-57.02	-58.52	-57.40	-51.55	-60.13
er93	CC-3/CC-6	24/12	IWLL	-160.66	-167.41	-155.51	-154.65	-159.11
er94	CC-3/CC-6	24/4	IDLS	20.79	21.15	19.23	23.79	16.47
er95	CC-3/CC-6	24/8	IDLS	11.17	11.19	8.92	16.19	10.16
er96	CC-3/CC-6	24/12	IDLS	-2.55	-5.78	-0.42	1.76	1.33
er97	CC-3/CC-6	24/4	IDLL	19.18	19.40	17.61	22.19	15.57
er98	CC-3/CC-6	24/8	IDLL	9.56	9.35	7.47	14.43	9.24
er99	CC-3/CC-6	24/12	IDLL	-4.06	-7.23	-1.84	-0.26	0.07
er100	AA-1/AA-3	24/4	IWLS	3.05	1.93	3.17	6.58	2.49

Table C.20. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er101	AA-1/AA-3	24/8	IWLS	-2.11	-4.46	-1.19	3.44	3.40
er102	AA-1/AA-3	24/12	IWLS	3.77	0.42	7.39	6.35	8.74
er103	AA-1/AA-3	24/4	IWLL	-5.07	-6.39	-4.72	-1.49	-3.77
er104	AA-1/AA-3	24/8	IWLL	-7.66	-9.83	-6.88	-2.26	-1.98
er105	AA-1/AA-3	24/12	IWLL	-6.74	-10.15	-3.31	-4.25	-0.34
er106	AA-1/AA-3	24/4	IDLS	-0.40	-1.39	-1.84	3.92	0.56
er107	AA-1/AA-3	24/8	IDLS	-10.54	-13.17	-12.62	-2.95	-0.95
er108	AA-1/AA-3	24/12	IDLS	-33.83	-41.74	-29.45	-24.05	-10.82
er109	AA-1/AA-3	24/4	IDLL	-8.47	-9.63	-9.72	-4.14	-5.63
er110	AA-1/AA-3	24/8	IDLL	-14.26	-16.56	-16.06	-7.26	-5.28
er111	AA-1/AA-3	24/12	IDLL	-30.78	-36.90	-26.35	-23.24	-14.21
er112	VC-1/VC-3	24/4	IWLS	18.96	19.37	17.85	21.53	14.94
er113	VC-1/VC-3	24/8	IWLS	20.28	20.66	18.73	24.19	18.44
er114	VC-1/VC-3	24/12	IWLS	12.88	10.43	15.71	16.25	16.78
er115	VC-1/VC-3	24/4	IWLL	19.24	19.61	18.11	21.80	15.31
er116	VC-1/VC-3	24/8	IWLL	21.62	21.97	19.99	25.46	19.50
er117	VC-1/VC-3	24/12	IWLL	16.31	14.10	18.86	19.12	18.67
er118	VC-1/VC-3	24/4	IDLS	18.10	18.56	16.55	20.94	14.46
er119	VC-1/VC-3	24/8	IDLS	17.93	18.39	15.45	22.52	17.18
er120	VC-1/VC-3	24/12	IDLS	12.55	10.83	13.94	16.55	15.92
er121	VC-1/VC-3	24/4	IDLL	18.27	18.70	16.64	21.10	14.82
er122	VC-1/VC-3	24/8	IDLL	19.66	20.06	17.16	24.06	18.58
er123	VC-1/VC-3	24/12	IDLL	16.34	14.71	17.64	19.65	18.27
er124	CC-2	24/4	NA	4.42	4.84	3.58	6.97	4.20
er125	CC-2	24/8	NA	6.38	7.18	4.44	10.65	5.91
er126	CC-2	24/12	NA	6.46	5.87	7.17	8.26	6.74
er127	CC-2	24/4	WDLS	7.99	8.70	6.58	9.05	3.25
er128	CC-2	24/4	WDLL	7.66	8.31	6.23	8.73	3.14
er129	CC-2	24/8	WDLS	6.43	7.28	4.29	9.59	3.55
er130	CC-2	24/8	WDLL	6.58	7.39	4.49	9.65	3.76
er131	CC-2	24/12	WDLS	4.63	3.83	5.14	5.06	4.09
er132	CC-2	24/12	WDLL	5.58	4.86	6.08	5.75	4.77
er133	CC-1	24/4	IDLS	12.05	12.72	10.59	13.48	7.53
er134	CC-1	24/4	IDLL	11.57	12.19	10.11	13.00	7.34
er135	CC-1	24/8	IDLS	9.56	10.24	7.37	13.38	7.72

Table C.20. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
er136	CC-1	24/8	IDLL	9.95	10.61	7.84	13.61	8.03
er137	CC-1	24/12	IDLS	4.49	3.07	5.46	6.21	6.35
er138	CC-1	24/12	IDLL	7.02	5.84	7.95	8.16	7.77
er139	CC-2/CC-5	24/4	NA	6.91	7.16	6.05	10.63	7.38
er140	CC-2/CC-5	24/8	NA	10.85	11.46	8.75	16.52	10.94
er141	CC-2/CC-5	24/12	NA	4.96	3.51	7.47	8.41	5.41
er142	CC-1	24/4	IWLS	12.83	13.46	11.78	14.02	7.95
er143	CC-1	24/4	IWLL	12.51	13.08	11.52	13.69	7.84
er144	CC-1	24/8	IWLS	11.69	12.32	10.37	14.89	8.85
er145	CC-1	24/8	IWLL	11.86	12.46	10.59	14.99	9.00
er146	CC-1	24/12	IWLS	4.72	2.70	7.06	5.91	7.05
er147	CC-1	24/12	IWLL	6.96	5.20	9.05	7.67	8.21
er148	CC-1/CC-6	24/4	IWLS	-12.17	-12.25	-13.24	-8.93	-18.13
er149	CC-1/CC-6	24/8	IWLS	-53.43	-54.24	-54.67	-47.21	-58.29
er150	CC-1/CC-6	24/12	IWLS	-154.24	-159.03	-150.82	-147.97	-155.30
er151	CC-1/CC-6	24/4	IWLL	-12.96	-13.16	-13.91	-9.73	-18.60
er152	CC-1/CC-6	24/8	IWLL	-54.13	-55.09	-55.15	-47.97	-58.79
er153	CC-1/CC-6	24/12	IWLL	-154.36	-159.35	-150.82	-148.72	-156.20
er154	CC-1/CC-6	24/4	IDLS	21.60	22.15	19.87	24.49	16.80
er155	CC-1/CC-6	24/8	IDLS	13.51	13.90	10.89	19.16	11.29
er156	CC-1/CC-6	24/12	IDLS	1.62	-0.16	2.64	5.58	3.44
er157	CC-1/CC-6	24/4	IDLL	20.69	21.18	18.95	23.55	16.21
er158	CC-1/CC-6	24/8	IDLL	13.19	13.53	10.69	18.62	10.92
er159	CC-1/CC-6	24/12	IDLL	2.71	1.15	3.70	5.93	3.37
er160	VC-2/CC-5	24/4	NA	8.68	8.98	7.70	12.43	8.60
er161	VC-2/CC-5	24/8	NA	13.00	13.73	10.66	18.68	12.43
er162	VC-2/CC-5	24/12	NA	8.38	7.53	10.34	12.05	8.09
er163	VC-1/CC-5	24/4	IWLL	21.98	22.46	20.65	24.54	16.81
er164	VC-1/CC-5	24/8	IWLL	15.97	16.34	14.17	20.79	12.28
er165	VC-1/CC-5	24/12	IWLL	4.75	3.12	6.38	7.60	5.21

Table C.20. (contd)

Table C.23. Combined-Cycle NPV ($M); Evaporatively-Cooled Base; 24 hr/day plant; 24 hr/day cooling

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
fr1	CC-4	24/24	NA	-5.17	-9.02	-2.15	-1.94	0.68
fr2	LiBr-2	24/24	NA	-3.64	-6.09	-1.89	0.16	2.17
fr3	AA-2	24/24	NA	-25.58	-29.88	-22.21	-16.76	-9.23
fr4	VC-2	24/24	NA	9.42	8.99	9.78	11.50	9.36
fr5	CC-4/CC-5	24/24	NA	-14.31	-20.27	-7.55	-9.18	-8.60
fr6	AA-2/AA-3	24/24	NA	-52.90	-61.27	-45.24	-39.48	-30.57
fr7	VC-2/VC-3	24/24	NA	15.61	13.43	17.57	20.04	17.41
fr8	CC-2	24/24	NA	5.14	3.61	6.43	7.05	6.16
fr9	CC-2/CC-5	24/24	NA	-3.10	-6.55	1.78	0.60	-2.64
fr10	VC-2/CC-5	24/24	NA	1.43	-0.81	5.31	5.31	0.72

Table C.20. (contd)

Table C.24. Industrial Turbine NPV ($M); Evaporatively-Cooled Base

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
ir1	AA-2	4/4	NA	-17.03	-18.19	-16.22	-12.33	-6.62
ir2	AA-2	8/8	NA	-16.62	-17.76	-16.97	-12.00	33.30
ir3	CC-2	4/4	NA	0.76	0.78	0.64	1.98	0.78
ir4	CC-2	8/8	NA	1.13	1.22	0.66	2.32	41.66
ir5	LiBr-1	4/4	NA	-0.95	-1.05	-0.93	0.49	-0.81
ir6	LiBr-1	8/8	NA	-0.71	-0.77	-1.14	0.64	40.33
ir7	VC-2	4/4	NA	1.08	0.99	1.10	2.34	1.49
ir8	VC-2	8/8	NA	1.27	1.15	1.03	2.55	42.37
ir9	VC-1	4/4	IWLS	6.10	6.17	5.96	6.35	3.43
ir10	VC-1	4/4	IWLL	5.92	5.95	5.82	6.17	3.39
ir11	VC-1	4/4	IDLS	5.48	5.59	5.05	5.91	3.10
ir12	VC-1	4/4	IDLL	5.29	5.35	4.86	5.72	3.06
ir13	VC-1	8/8	IWLS	4.25	4.07	4.41	4.69	43.73
ir14	VC-1	8/8	IWLL	4.35	4.11	4.55	4.76	43.86
ir15	VC-1	8/8	IDLS	2.40	2.24	1.97	3.32	42.76
ir16	VC-1	8/8	IDLL	3.02	2.81	2.66	3.82	43.21
ir17	VC-2	4/4	WWLS	3.26	3.25	3.22	3.37	1.07
ir18	VC-2	4/4	WWLL	3.24	3.21	3.21	3.35	1.10
ir19	VC-2	4/4	WDLS	3.56	3.71	3.14	3.79	1.00
ir20	VC-2	4/4	WDLL	3.47	3.58	3.05	3.70	1.01
ir21	VC-2	8/8	WWLS	1.36	1.08	1.64	1.63	41.43
ir22	VC-2	8/8	WWLL	1.75	1.47	2.02	1.98	41.66
ir23	VC-2	8/8	WDLS	1.68	1.64	1.23	2.21	41.24
ir24	VC-2	8/8	WDLL	2.09	2.02	1.68	2.54	41.57
ir25	CC-2/CC-5	4/4	NA	1.48	1.37	1.28	3.51	1.05
ir26	CC-2/CC-5	8/8	NA	2.54	2.49	1.93	4.61	42.56
ir27	AA-2/AA-3	4/4	NA	-30.70	-32.48	-29.64	-24.11	-17.99
ir28	AA-2/AA-3	8/8	NA	-29.59	-31.34	-29.97	-23.00	22.27
ir29	VC-2/VC-3	4/4	NA	1.20	0.86	1.36	3.38	2.59

Table C.20. (contd)

Case	Chiller	Ophrs	Storage	Nash.	Hous.	Minn.	Phoe.	San Fran.
ir30	VC-2/VC-3	8/8	NA	1.26	0.78	1.16	3.59	43.33
ir31	VC-1/VC-3	4/4	IWLS	6.60	6.43	6.60	7.69	4.66
ir32	VC-1/VC-3	8/8	IWLS	4.52	3.99	4.85	5.94	44.80
ir33	VC-1/VC-3	4/4	IWLL	6.92	6.73	6.87	7.98	5.04
ir34	VC-1/VC-3	8/8	IWLL	5.52	4.95	5.77	6.89	45.72
ir35	VC-1/VC-3	4/4	IDLS	5.96	5.82	5.64	7.24	4.31
ir36	VC-1/VC-3	8/8	IDLS	2.55	2.03	2.26	4.49	43.78
ir37	VC-1/VC-3	4/4	IDLL	6.23	6.08	5.84	7.49	4.69
ir38	VC-1/VC-3	8/8	IDLL	4.05	3.48	3.73	5.82	45.02
ir39	VC-2/CC-5	4/4	NA	1.08	0.79	1.08	3.30	1.66
ir40	VC-2/CC-5	8/8	NA	1.87	1.53	1.51	4.18	43.05
ir41	VC-1/CC-5	4/4	IWLL	6.27	6.11	6.14	7.41	3.67
ir42	VC-1/CC-5	8/8	IWLL	5.18	4.72	5.29	6.56	44.63

Table C.20. (contd)

Distribution

No. of
Copies

OFFSITE

2 DOE/Office of Scientific and Technical Information

G. Albrecht
Westinghouse Electric Corporation
Energy Center
4400 Alafaya Trail, MC 550
Orlando, FL 32826

C. Dorgan
Thermal Energy Storage Applications Research Center
University of Wisconsin
150 East Gilman
Room 2200
Madison, WI 53703-1441

R. Dracker
Bechtel
50 Beale Street
San Francisco, CA 94105-1895

J. Ebeling
Burns and McDonnell Engineering Company
4800 East 63rd Street
Kansas City, MO 64130

A. F. Ellis
Southern Company Services, Inc.
P.O. Box 2625
Birmingham, AL 35202

D. Erickson
Energy Concepts Company
627 Ridgely Avenue
Annapolis, MD 21401

Table C.20. (contd)

No. of
Copies

K. Gainer
Business Communications Company, Inc.
25 Van Zant Street
Suite No. 13
Norwalk, CT 06855-1713

J. Kimball
Gas Research Institute
8600 West Bryn Manor Avenue
Chicago, IL 60631-3562

D. Knebel
12830 Link Drive
Shakapee, MN 55379

C. Kohlenberger
Kohlenberger Associates
Consulting Engineers, Inc.
611 South Euclid Street
Fullerton, CA 92635

C. Landry
Turbo Refrigerating
1815 Shady Oaks Drive
Denton, TX 76205

J. McNamara
Resorcon, Inc.
701 Rodi Road
Suite 102
Pittsburgh, PA 15235

I. Ondryas
Fluor Daniel, Inc.
3333 Michelson Drive
Irvine, CA 92730

T. Ramisch
Pacific Power
920 SW 6th Avenue
Portland, OR 97204

Table C.20. (contd)

No. of Copies	
	P. Sarkisian Rocky Research P.O. Box 61800 Boulder City, NV 89006-1800
	B. Silvetti Calmac Manufacturing Corp. 101 W. Sheffield Ave. Englewood, NJ 07631-4804
	D. Sopocy Sargent & Lundy 20th Floor 55 East Monroe Street Chicago, IL 60603
5	U.S. Department of Energy Office of Energy Efficiency and Renewable Energy 1000 Independence Avenue, S.W. Washington, D.C. 20585 ATTN: S. Blazewecz EE-221 R. Eaton EE-142 D. Knowles EE-221 B. Parks EE-221 N. Rossmeissl EE-142

Table C.20. (contd)

No. of
Copies

G. Weidinger
U.S. Generating Company
7500 Old Georgetown Road
Bethesda, MD 20814

P. Winters
CBI - Walker
1245 Corporate Blvd
Suite 102
Aurora, IL 60504

ONSITE

DOE Richland Operations Office

50 **Pacific Northwest National Laboratory**

S. J. Arey (20)	K5-20
D. R. Brown (20)	K8-17
S. Katipamula	K5-16
J. H. Konynenbelt	P7-41
S. Somasundaram	K5-20
Information Release Office (7)	