Technical and Economic Assessment of the Use of Ammonia Expanders for Energy Recovery in Air-Cooled Power Plants

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July 1982

**Prepared for the U.S. Department of Energy
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SUMMARY

Binary cycle power plants have been the subject of much discussion among engineers and scientists for nearly 100 years. Current economic and environmental concerns have stimulated new interest and research. Ammonia has been recommended by other studies as the leading contender for use as simply the heat rejection medium in an air-cooled power plant.

This study investigates the technical feasibility and economic potential of including an expander in the heat rejection system of an air-cooled power plant. The expander would be used during certain parts of the year to increase the total output of the power plant. Five different plant locations (Miami, San Francisco, Bakersfield, Chicago, Anchorage) were investigated to show the effect which climate has on the economic potential of this ammonia bottoming cycle.

The study shows that the expected energy costs for the bottoming cycle only will be less than 50 mills/kWh for any of the five plant locations. This cost assumes that an ammonia phase-change heat rejection system is already a part of the existing plant. The colder climates of Chicago and Anchorage demonstrate an even smaller energy cost of less than 15 mills/kWh. Further investigation of the concept is merited to substantiate these costs and determine the needed technology.

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TECHNICAL AND ECONOMIC ASSESSMENT OF THE USE OF AMMONIA EXPANDERS FOR ENERGY RECOVERY IN AIR-COOLED POWER PLANTS

1.0 INTRODUCTION

Rising environmental concerns together with rising fuel costs in recent years have created new incentives for considering dry or dry/wet cooling systems that will result in combining the highest possible power generation efficiency with low water consumption. One of the many ideas to receive growing attention is the use of ammonia as a heat transport medium between the condensing steam at the plant outlet and an all dry or dry/wet air-cooled tower. This concept has been shown to be potentially cost-effective as well as environmentally favorable through the reduced consumption of water (Allemann et al. 1976).

The Dry-Cooling Enhancement Program at the Pacific Northwest Laboratory (PNL) with the support of the U.S. Department of Energy and the Electric Power Research Institute has included investigations into many aspects of this concept (Johnson 1981). The work is now culminating in the operation of a demonstration facility approximately equivalent to cooling a 15-MWe plant adjacent to the Kern power plant in Bakersfield, California. This plant uses ammonia as the heat transport fluid and has the capability of either all dry or dry/wet cooling.

The ammonia fluid can operate as both a heat transport medium and a thermodynamic working fluid at low ambient temperatures. Unlike steam, which cannot easily be used in a steam-Rankine cycle below 100°F because of the large specific volume, ammonia vapor has a relatively small specific volume at these lower temperatures (e.g., saturated ammonia vapor at 100°F: 1.42 ft³/lb; saturated steam at 100°F: 350 ft³/lb). Since ammonia can be used in this low temperature range, power may be obtained by including an expander between the outlet of the steam turbine (which may be as low as 100°F) and the cooling tower (which is at ambient temperature). When the ambient temperature drops below the design point of the simple dry (or dry/wet) heat rejection system,

the ammonia may be expanded and the total output of the power plant may rise considerably above the rated capacity of the base plant with no additional fuel requirement.

The purpose of this study is to explore the technical and economic feasibility of including an expander in the ammonia cooling system of an air-cooled power plant and to explore the effect of meteorology at various sites on the cost effectiveness of the expander addition. As discussed in this report the expander and its associated support equipment would not alter the design of the existing power plant or cooling system and would only operate under ambient conditions that would increase the total power output from the plant. In studies by Electricité de France (EDF) (e.g., Fleury 1981), alternative designs are considered, including the possible replacement of the low-pressure steam turbine by an ammonia turbine of much smaller size (and much higher pressure).

Before discussion of the ammonia expander system analyzed in this report, previous studies assessing the potential of binary cycles applied to thermal power plants are reviewed and summarized. Following this background discussion, the meteorology at the five sites in the United States selected for the cost comparison are described. Next, various technical aspects of the plant, including the cooling system optimization and ammonia expander operation, are discussed. This is followed by a general description of the expander system design and a discussion of the performance analysis. The potential for increasing the total power output of a power plant by redesigning the low-pressure steam turbine is also explored, along with the identification of other power optimization options that appear to warrant further study. Finally, the cost of obtaining energy from the addition of the ammonia expander is approximated.

2.0 BACKGROUND

The concept of a binary cycle has been discussed for many years. As early as 1848 some engine designs employed ether as a secondary fluid in combination with steam. In the 1920s, the General Electric Company developed and built binary plants with mercury vapor used in a topping cycle with a standard steam Rankine cycle. Engineers are continuing to look for one or more fluids that can be used in conjunction with or in place of steam to improve the overall conversion efficiency of power plants. Under special applications, such as in geothermal power plants, binary cycles using other working fluids in addition to steam are being studied. However, no binary Rankine cycle system identified to date would economically augment or supplant the steam cycle in conventional thermal power plants.

Ammonia is promising as a fluid that can be used in a cycle at temperatures below those of conventional commercial power plants because of its thermodynamic properties, relative safety, and the familiarity industry has with its technology. Considerable work in exploring the potential of ammonia has taken place in Europe. This work, as well as recent work in the United States, will be briefly discussed in the following sections.

2.1 PREVIOUS STUDIES

Heller (1965) reported the work he and his colleagues had done in determining the technical and economic feasibility of a steam-ammonia binary cycle power plant. In a rather extensive initial analysis they rated several working fluids by considering all requirements of a good working fluid. Ammonia was found to be the clear first choice.

The Heller analysis of the binary cycle was primarily theoretical. Some economic factors were also discussed but usually in a qualitative sense only. The conclusion of the study was that the development of a steam-ammonia binary cycle with regenerative heating in both the steam and ammonia parts of the cycle was very promising. Not only did the power output of the cycle increase at the ambient air temperatures, but an improvement in the overall conversion

efficiency was expected along with lower capital costs. The need for additional heat exchangers was expected to increase the total capital costs, but these increases were outweighed by the decrease in cost provided by much smaller equipment in the low-temperature part of the cycle. (The specific volume of ammonia is much smaller than steam, so the equipment can be smaller.) Although many questions were left unanswered by this report, it was an excellent first step toward assessing the potential of steam-ammonia binary cycles.

Heller's study stimulated interest in England, and the results of a study performed there were reported at the 1968 World Power Conference (Eaves and Hadrill 1968) by members of the Central Electric Generating Board of the United Kingdom. This study was more practically oriented than Heller's and included a discussion of using either ammonia or freon as a secondary working fluid. These systems were compared to systems using gas turbines in a topping cycle. The conclusions, contrary to the Heller study, were that the cost of the heat exchangers would be significant and that although the total capital costs of a binary system might be less, the overall efficiency would also be less than a conventional steam power plant. The net effect indicated that there were no incentives for further developing either the steam-ammonia or the steam-freon binary cycle plants. The study also showed that the use of gas turbine topping cycles was promising for certain applications. In general, the study illustrated the complexities involved when determining the economic feasibility of such plants. Such seemingly insignificant factors as the operational role of the plant become very important in the final analysis when determining the cost of power.

Slusarek (1968) at the Franklin Institute, followed by Steigelmann and Seth (1972), performed a similar study in the United States. The study culminated in a 1972 report that compared the economic feasibility of several types of dry-cooled power plants. The report concluded that an all-steam power plant with an ammonia heat rejection system was the most favorable. The steam-ammonia binary cycle was a very close second and was probably equal within the uncertainties contained in the analysis. Although the results of this study may have been reasonably accurate, the report gave very little specific information to support these conclusions. For instance, plant

efficiencies were given for each different system at several design points, but no explanation was given to support the efficiencies used. Additionally, a very important consideration that was left out entirely was the effect of meteorology on the plant design and consequently on plant costs and power costs. Although unclear, the design of the binary-cycle system seemed to be determined through a completely theoretical analysis with no consideration given to cost tradeoffs. As in Heller's study, the design included regenerative heating and superheating. The optimum crossover temperature between the steam and ammonia cycles was given as 224°F.

In general, the conflicting results of previous studies have shown that there is no obvious answer to the question of the economic attractiveness of steam-ammonia binary cycle power plants. The advantages and disadvantages are identifiable but very complex and difficult to characterize clearly.

2.2 CURRENT STUDIES

The most complete study to date is a part of an ongoing project sponsored by Electricité de France (EDF) (Blanc-Feraud and Fleury 1978; Fleury 1981) in France. EDF is presently engaged in the initial phases of constructing a 20-MWe demonstration plant adjacent to the Gennevilliers power station near Paris, France, to test the technology. Their work has addressed the economic attractiveness of the steam-ammonia binary cycle and has included the effects of meteorology on the system design and economics. Interestingly, the conclusion of the French study was that the optimum system would contain no regenerative heating and no superheating. This is a significantly different conclusion from that reached in the other studies mentioned.

In summary, previous studies do not clearly indicate an economic advantage for binary-cycle power plants, although the work being done in France is yielding very optimistic predictions of low costs. It is apparent that determining the optimum binary power plant configuration is highly sensitive to both technical and economic assumptions and considerations. Finding this optimum is a very complex task.

2.3 SCOPE OF THE PRESENT STUDY

The scope of this project was limited to a simple, semi-quantitative analysis of the suitability of using an ammonia cycle in combination with a standard steam cycle for producing commercial power. The main purpose is to decide whether or not further investigation in this area is merited and, if so, to help set a direction for further studies. The objective is to determine the feasibility, both technical and economic, of adding an ammonia expander and necessary support equipment in the coolant loop to recover the excess power available at low ambient air temperatures. To help make the project objectives attainable, it was assumed that a steam power plant with an ammonia cooling system was already designed and optimized. This report discusses some of the technical operational aspects and problems as well as the system costs. A very detailed program would be required to design and estimate the cost of a completely optimized binary-cycle system and therefore was not undertaken in this study.

In this report the steam power plant with an optimized ammonia heat rejection loop are referred to as the "base plant". The ammonia expander and support equipment added to the heat rejection system are referred to as the "expander addition".

3.0 DESIGN BASIS

In order to limit the scope of this study, the final operation of the base plant with the expander addition must be based on several design assumptions. These assumptions relate to the location of the plant, the optimization of base plant cooling system, and the design considerations involved in using an ammonia expander. The effect of each on the overall system design and specifically on the results of this study will be briefly discussed in this section.

3.1 METEOROLOGY - PLANT LOCATIONS

The performance and economic attractiveness of the energy recovery addition is significantly dependent on site meteorology. This dependence can be primarily attributed to two factors:

- 1) The design of the base plant cooling system is determined by site meteorology and, as will be shown later, this design significantly affects the expander addition.
- 2) The output of the expander addition is directly affected by site meteorology. As the ambient air temperature decreases, the ammonia expander output increases.

Assuming that there is a market for the power generated by the bottoming cycle, site meteorology becomes the most significant variable in determining the economic incentives for installing an expander. In an attempt to quantify the effect of site meteorology, five different geographic locations representative of climatic conditions in different parts of the U.S. were investigated. These sites represent the wide extremes in U.S. climate, from cold in Anchorage to hot and humid in Miami and hot and dry in Bakersfield. In addition, the intermediate climates of Chicago and San Francisco were chosen. Although these sites do not represent all the significant variations in duration and time distribution of ambient temperatures, investigation of other sites could lead to results that are not simply intermediate to those presented here. It is possible that these sites do provide a broad representation.

Psychrometric data for each site were obtained from the National Climatic Center in Ashville, North Carolina. It was desired to predict an average operating year based on the meteorological history of each site. Usable data was available for the following years:

<u>Site</u>	<u>Dates of Available Data</u>
Anchorage	May 1941 - Oct 1953; Apr 1956 - Oct 1968
Chicago	Oct 1946 - June 1965
Miami	Jan 1948 - June 1970
Bakersfield	Jul 1961 - Jan 1972
San Francisco	Jan 1948 - June 1965

Both wet bulb and dry bulb temperatures were used to design the cooling system since the evaporative cooling operation depends on the air humidity as well as the temperature. However, only the dry bulb temperature was used to establish operating temperatures throughout the cooling system and ultimately the enthalpy drop across the ammonia expander.

The psychrometric data, recorded at two-degree intervals, gave the number of operating hours for each interval for the total years the data were available. These total hours were averaged to determine the number of hours at each interval for an average year.

Figure 3.1 is a bar graph depicting the actual hours per year for each two-degree dry bulb temperature increment. The graph displays distinct differences between the climates. Temperatures at San Francisco and Miami reside in a narrow band because of the tempering effect of the ocean. The temperature distributions at the other sites are much broader with Bakersfield peaking at warm temperatures and Anchorage peaking at the coldest. Chicago is unique with both a warm (summer) peak and a cold (winter) peak. The area under each curve is the same and equal to the total number of hours per year (8760). Figure 3.2 shows the same data plotted with temperature as a function of cumulative hours for Anchorage. From this graph and similar ones for other sites, one can easily determine the number of hours above or below a certain temperature.

FIGURE 3.1. Annual Temperature Profiles

FIGURE 3.2. Cumulative Temperature Profile for Anchorage, Alaska

Also plotted in Figure 3.2 are the corresponding wet bulb temperatures. Actually, there will be many different wet bulb temperatures for each dry bulb temperature; however, for design purposes the wet bulb temperatures were taken from graphs like the one shown in Figure 3.2 and correspond to the dry bulb temperature with the same number of cumulative hours.

3.2 COOLING SYSTEM OPTIMIZATION

The purpose of the study is to evaluate the addition of an ammonia expander and support equipment to the already optimized cooling system of a power plant, not to determine an optimum binary-cycle power plant. Therefore, it is necessary to discuss the determination of an optimum cooling system design for each site before discussing the ammonia expander addition.

A computer code developed at PNL, BNW-II, was used to arrive at the optimum cooling system design for each site. BNW-II finds the design that gives the lowest cost of bus-bar power from the plant, reflecting both operational costs and capital costs. Table 3.1 is included to show the various input parameters, both those externally fixed and those internally optimized by the code. Values typical of those used in earlier parametric studies of dry/wet cooling systems were used as input to the code. These basic values are listed in Table 3.2.

The design of the cooling system is very important to the results of this study. The power obtained from the ammonia expander is dependent on the enthalpy potential available. This enthalpy drop is exclusively dependent on two temperatures: the vapor temperature at the condenser/reboiler (or the steam turbine outlet temperature) and the vapor temperature at the cooling tower. Of course, the temperature at the cooling tower is dependent both on the tower design and the ambient air conditions. Once the cooling system is optimized and the design parameters fixed, the output of the ammonia expander becomes a function only of the plant operation (i.e., the steam turbine outlet temperature), the ambient air conditions, and the expander efficiency.

TABLE 3.1. Cooling System Optimization – Major Parameters and Options

	Internally Optimized Independent	Internally Optimized Dependent	Externally Optimized Independent	Externally Fixed
I. Heat Exchanger	<ol style="list-style-type: none"> 1. Tube Length(a) 2. Number Tubes In-Depth(a) 3. Face Velocity(a) 	<ol style="list-style-type: none"> 1. Frontal Area 2. Width/Length Ratio 3. Total Tubes (Bundle Number or Overall Size) 	<ol style="list-style-type: none"> 1. Unit Geometry (Tube OD, Tube Pitch, Fin Height, Staggered vs In-Line, etc.) 2. Tube and Fin Material 3. Fin Type 4. Spacer Material and Type 5. Header Type 	
II. Fan System		<ol style="list-style-type: none"> 1. Number 2. Blade Diameter(a) 3. Blade Angle(a) 4. Number of Blades(a) 5. Fan Power 	<ol style="list-style-type: none"> 1. Velocity Recovery vs No Velocity Recovery 	
III. Tower		<ol style="list-style-type: none"> 1. Number(a) (Polygonal) 2. Diameter (Polygonal) 		
IV. Piping and Pump		<ol style="list-style-type: none"> 1. Pump Power 2. Pipe Lengths 3. Pipe Diameter 	<ol style="list-style-type: none"> 1. Design Velocity 2. Pipe Wall Thickness 3. Piping Temperature Drop 	
V. Condenser/Reboiler (NH ₃)	<ol style="list-style-type: none"> 1. Terminal Temperature Difference (NH₃)(a) 	<ol style="list-style-type: none"> 1. Tube Number 	<ol style="list-style-type: none"> 1. Tube Length 2. Tube Material 3. Tube Diameter 4. Tube Type 5. Exit Quality 	
VI. Turbine	<ol style="list-style-type: none"> 1. Design Turbine Exhaust Temperature(a) 		<ol style="list-style-type: none"> 1. Turbine Type 	
VII. Common to Two or More Components		<ol style="list-style-type: none"> 1. Range of Air 2. Tower Initial Temperature Difference (ITD) 	<ol style="list-style-type: none"> 1. Ratio of Tower Roof Area to Fan Swept Area 2. Design Ambient Temperature 3. Water Consumption 	
VIII. Base Plant				<ol style="list-style-type: none"> 1. Fuel Cost 2. Plant Cost 3. Heat Rate 4. Capacity Factor 5. Fixed Charge Rate 6. Size
IX. Capacity Leveling				<ol style="list-style-type: none"> 1. Energy Penalty Charges 2. Capacity Penalty Charges
X. Site				<ol style="list-style-type: none"> 1. Meteorology

(a) Optional treatment as external fixed parameter.

TABLE 3.2. Cooling System Optimization - Input Variables

Fuel Cost	- 66.9 ¢/MBtu
Plant Cost	- 571 \$/kW
Capacity Factor	- 0.70
Fixed Charge Rate	- 18%
Plant Size	- 1000 MW
Energy Penalty Charges	- 15 mills/kWh
Capacity Penalty Charges	- 300,000 \$/MW
Water Consumption	- 2000 acre-ft/yr.

It should be noted that BNW-II incorporates dry/wet heat exchanger surfaces in its tower design. This was chosen over an all-dry design because it is believed to be more typical of the type of cooling that will be used in near-future air-cooled power plants. Since the use of partial evaporative cooling essentially provides additional cooling during the high-temperature peaks occurring in the summer, the optimum driving force or ITD (inlet temperature difference) for the cooling tower is larger than it would be for an all-dry design which allows for smaller (less expensive) cooling towers. At the same ambient conditions, the available enthalpy drop across the ammonia expander is greater for the all-dry design (since the vapor temperature at the cooling tower is lower), and the total plant output increases. The optimum design would be a complex tradeoff between increased power and increased cooling system costs. In the present study the size of the cooling tower was fixed as the optimum dry/wet system specified from the BNW-II code.

As indicated in Table 3.2, the dry/wet cooling towers were allowed to consume twenty percent of the water required for complete evaporative cooling, which for a 1000-MWe plant is equivalent to 2000 acre-feet of water each year. Unless the cost of acquiring water and disposing of the blowdown is very high, this figure is set by the technical and societal assessment of water availability.

Parameters describing the final optimum cooling system design for each site are given in Tables A.1 through A.4 in the appendix. Pertinent information such as steam turbine outlet temperature, steam turbine backpressure, heat

exchanger temperatures, etc. are given as a function of ambient dry bulb temperature. Since the optimum cooling system design is a function of the meteorology, each design is quite different. For example the ITD of the cooling tower at the design point for Anchorage is about 63°F, whereas for Miami it is about 42°F. This means that the economics at Miami allows for a larger cooling tower than at Anchorage, which also means a higher cost.

To assure that each system was compared on a common basis, the design point was chosen to be the point at which the backpressure of the turbine is 2.2 in. Hg. The design point in BNW-II was defined as the point at which the net power output from the plant was 1000 MW. For certain sites the design point occurred at turbine backpressures below 2.2 in. Hg. Although changing this definition would affect the results of the cost comparisons done with BNW-II, it does not affect the results of this study.

From the cooling system optimizations done by BNW-II the ambient dry bulb temperatures corresponding to the design points are:

Anchorage	31°F
Chicago	43°F
Bakersfield	48°F
San Francisco	45°F
Miami	53°F.

Figure 3.3 shows the temperatures at various locations in the cooling system as a function of ambient temperature for the Anchorage design. The design point is indicated. As the ambient temperature drops below the design point, the cooling tower inlet temperature decreases, thereby creating a temperature difference between the tower and the condenser/reboiler which could be used to drive the expander. This is the triangular region shaded in Figure 3.3.

At ambient temperatures above the design point, the turbine outlet pressure is forced to increase to maintain the cooling tower ITD required for that particular design. The backpressure increases up to a maximum pressure optimized by BNW-II. At ambient temperatures above this point, wet cooling is employed.

FIGURE 3.3. Cooling System Temperatures

3.3 AMMONIA EXPANDER OPERATION

The expander is an ammonia turbine that produces shaft work at the expense of a drop in pressure and temperature in the ammonia stream. As discussed above, to the left of the design point (Figure 3.3) or at low ambient temperatures, a pressure and temperature drop exists which is suitable for power extraction via the expander. Proper operation and design of the expander must be identified to make the best use of the energy available.

In general, operational characteristics of expanders are well defined. This is especially true in gas processing where they are used to provide

refrigeration (Bergmann and Mafi 1979)^(a). The two most important factors relative to expander design are the inlet-to-outlet isentropic enthalpy drop and the volumetric discharge rate. Enthalpy drop determines the rotor tip speed (subject to stress limits), and the discharge volume flow determines the expander wheel flow area. Specifically, the physical structure of the expander is a function of its specific speed, which is defined as:

$$N_s = \frac{N \sqrt{Q}}{(H)^{.75}}$$

where N_s = specific speed

N = shaft speed, rpm

Q = discharge flow, ft³/sec

H = head, ft-lb/lb (isentropic expansion).

All of the above conditions are at the expander design point, which yields the maximum expander efficiency. The value of N_s not only identifies the maximum efficiency attainable, but also determines many of the design parameters, including the following:

- ratio of energy between the nozzles and wheel
- ratio between the outside and eye diameter of the wheel
- blade angle
- blade height.

Thus, all of the above parameters are also dependent on the selected design point. In this study the expander design point must be carefully chosen because the expander operates under off-design conditions most of the time, and the choice of the design point will influence the average efficiency over all the operating hours of the expander. The expander design points for each of the sites were optimized by the Mafi-Trench Company. Conversations with the

(a) S. Mafi also made available two of his papers - "Turboexpanders: Performance, Application, and Operation" and "Waste Heat Power Recovery Systems." Mafi-Trench Corporation, Westlake Village, California 91361.

Mafi-Trench Company also indicated that the expander efficiency is largely a function of the inlet/outlet pressure ratio and is much less dependent on changes in the magnitude of the inlet and outlet pressure provided the pressure ratio remains constant.

4.0 RESULTS

4.1 SYSTEM DESIGN

4.1.1 General Description

The entire coolant loop, including the energy recovery addition, is shown in Figure 4.1. The condenser/reboiler and the cooling tower are exactly the same as given in the base plant design. Other components, although found in the base plant, are substantially changed with the addition of the expander. The moisture content of the vapor leaving the phase separator is much more critical with the addition of the expander. Essentially no moisture is allowed to enter the expander. This will probably necessitate the use of a vane-type phase separator design rather than the less expensive centrifugal design being used in the Advanced Concept Test (ACT) demonstration at Bakersfield (Johnson et al. 1981). The pumps will also be changed to accommodate much larger and variable head requirements. At the lowest ambient temperatures the pressure difference between the condenser/reboiler and the cooling tower is maximum. The pumps must be designed to provide this maximum pressure

FIGURE 4.1. Energy Recovery System

difference as well as all lower pressure requirements. The piping will be generally modified to accommodate the range of pressure and temperatures that will occur during the year.

Other modifications include the following. A feed heater must be added to the system to provide heat to overcome the subcooling which occurs when power is extracted from the system. (The degree of liquid ammonia subcooling increases as the ambient temperature drops.) Generators are sized to match the output and shaft speed of the expanders. Other electrical equipment, such as voltage regulators and switching gear, are required to supply the generated electrical energy to the existing utility grid.

4.1.2 Analysis

The scope of this preconceptual study did not include a detailed analysis of the expander addition. A simplified analysis was done in conjunction with the Mafi-Trench Company to provide reasonable designs for the expanders and estimates for the amount of energy recovered. The following assumptions were necessary to complete this analysis:

- 1) The cooling tower operation is fixed by the design discussed in Section 3.
- 2) The heat rejected from the power plant is fixed and is used entirely for vaporizing ammonia. (Subcooling is neglected.)
- 3) The ammonia vapor flow rate leaving the condenser/reboiler is constant.
- 4) No energy losses occur between the condenser/reboiler and the ammonia expander.
- 5) Piping losses are exactly the same as calculated in the cooling system optimization code and occur primarily between the ammonia expander and the cooling tower.

Since the output of the expander increases with decreasing ambient temperature and the number of operating hours decreases at the same time, the

proper design point for the expander must be determined to recover the maximum amount of energy. These tradeoffs were identified for each site, and the appropriate designs were determined. Pertinent parameters are given in Table 4.1. As expected, a large amount of energy is recovered at both Anchorage and Chicago, while very little is gained at San Francisco or Miami.

In each case, the expanders were chosen to be radial-flow, single-stage expanders. The manufacturing and design technology necessary to build these expanders is well established. For Miami and San Francisco the flow rate is about the same as for the other sites, but the enthalpy drop is very small. As described in Section 3, this yields a high specific speed. The configuration of these expanders is more like a propeller than a turbine in order to accommodate the low enthalpy drop. There may be an advantage in these cases to change the design to an axial-flow, multistage configuration. Axial-flow turbines tend to be more efficient for cases of high flow rate and low enthalpy drop. For Miami, San Francisco, and Bakersfield the high flow rates necessitate using four expanders in parallel rather than the two used for Anchorage and Chicago.

Once the design point was determined, it was very important to establish the off-design operation. Efficiency curves for the expanders used at each of the five sites are given in Figure 4.2 as a function of ambient air temperature. The curves are limited to a certain temperature range straddling the design point. The expanders can operate outside of this range, but the efficiencies drop so fast that they become extremely difficult to estimate.

Table 4.1 gives the estimated values for the ideal annual energy recovery and the actual annual energy recovery for each site. The ideal energy recovery assumes that the expander is 100% efficient over the entire operating range. The actual energy recovery is computed using the efficiencies shown in Figure 4.2. The ideal operating hours refer to the time the ambient temperature is below the minimum backpressure temperature. The actual operating hours refer to the time the ambient temperature is within the limits of expander operation (shown in Figure 4.2).

TABLE 4.1. Expander Addition Characteristics

Site Location	Anchorage	Chicago	Bakersfield	San Francisco	Miami
Number of Expanders	2	2	4	4	4
Gas	NH ₃	NH ₃	NH ₃	NH ₃	NH ₃
Flow Rate, lb/hr	9,553,025	9,290,905	9,220,974	9,292,881	9,113,064
Inlet Pressure, psia	205.3	205.3	208.6	205.3	205.3
Inlet Temperature, °F	98	98	99	98	98
Outlet Pressure, psia	122	129	172	185	185
Outlet Temperature, °F	67	70	87	92	92
Shaft Speed, rpm	3600	3600	3600	1800	1800
Estimated Efficiency, %	86	86	84	84	84
Shaft Power, kW (Per Exp)	33,700	29,300	5100	1700	2250
Expander Wheel Diameter, in.	52.8	49.9	29.9	34.6	39.9
Inlet Flange Diameter, in.	46	46	42	54	54
Outlet Flange Diameter, in.	58	58	46	54	54
Ideal Yearly Recovery, kWh	118x10 ⁶	91.9x10 ⁶	23.3x10 ⁶	1.4x10 ⁶	.8x10 ⁶
Actual Yearly Recovery, kWh	69x10 ⁶	53.5x10 ⁶	14.9x10 ⁶ /60.4	.2x10 ⁶	.6x10 ⁶ /28.4
Ideal Operating Hours, hr	3200	2700	1350	350	150
Actual Operating Hours, hr	1450	1550	900	50	120
Skid Size, LxWxH, ft	30x12x15	30x12x15	28x12x12	28x12x12	28x12x12
Estimated Cost, each, \$	1.8x10 ⁶	1.74x10 ⁶	0.66x10 ⁶	0.66x10 ⁶	0.72x10 ⁶

FIGURE 4.2. Estimated Expander Efficiencies

4.1.3 Additional Power

Initially it was assumed that the expander additions did not change the operation of the base plant. That is, the backpressure of the steam turbine as a function of ambient air temperature remained unchanged. The values given in Table 4.1 reflect these assumptions. Further analysis revealed that additional power could be recovered with the same equipment by raising the steam turbine backpressure. If the backpressure of the steam turbine is allowed to rise, the corresponding rise in the inlet conditions to the ammonia expander can, under certain conditions, increase the total plant output by increasing the power output from the ammonia expander more than the corresponding losses in the steam turbine power output. Differences in the thermodynamic properties of steam and ammonia in this temperature range provide for the increase in total power output.

The expander addition designed for Miami will be used to illustrate this potential for added power recovery. Figures 4.3 and 4.4 show the temperatures

FIGURE 4.3. System Temperatures for Base Case

FIGURE 4.4. System Temperatures with Enhanced Energy Recovery

at various locations in the system as a function of the ambient air temperature. Figure 4.3 is for the base case without an expander in the system. At about 60°F, the limited tower capability causes the steam turbine backpressure to rise. This is similar to the discussion of Figure 3.3 for Anchorage.

Figure 4.4 shows the alternative method of operation where the steam turbine backpressure is allowed to rise at the higher ambient temperatures, thus transferring some of the power generating function from the steam turbine to the ammonia expander. For this study the additional energy was assumed to have a maximum when the expander is operating at its design point over the entire year. To calculate this additional energy both the expander gains and the steam turbine penalties must be accounted for.

The gains from the expander were assumed to be constant at constant pressure ratio and constant expander efficiency (for Miami the efficiency was 84%). The penalties from the steam turbine, however, are variable depending on the backpressure. These penalties were estimated from a standard nondimensional heat rate curve which, in terms of backpressure (BP), is:

$$HR = 0.9988 - 0.0154(BP) + 0.0056(BP)^2 - 0.003(BP)^3.$$

This is illustrated in Figures 4.5 and 4.6. If the normal operation of the plant has the steam turbine backpressure at 2.2 in. Hg (104°F) and the output of the expander at zero, the potential gains from allowing the steam turbine backpressure to rise are shown in Figure 4.5. The net gain is maximum at a backpressure of about 4 in. Hg and decreases slowly above that until at about 8 in. Hg, where the losses from the steam turbine equal the gains from the expander. In comparison, if the plant is normally operating at a steam turbine backpressure of 3.8 in. Hg. (as is the case for Miami with ambient temperatures above 74°F--see Figure 4.3) and the output of the expander is zero, the potential gains from allowing the steam turbine backpressure to rise are much smaller, as shown in Figure 4.6. In summary, maximum gains are made in allowing the backpressure to increase to about 4 in. Hg.

FIGURE 4.5. Steam Turbine and Ammonia Expander Power Tradeoff at the Miami Design Point (Ambient Air Temperature 53°F)

FIGURE 4.6. Steam Turbine and Ammonia Expander Power Tradeoff at Miami (Ambient Air Temperatures Above 74°F)

The following simplistic assessment of each site was carried out to approximate the gains from allowing the steam turbine backpressure to float.

- The ammonia expander was assumed to continue operation at ambient temperatures above the designed cut-in point. The temperature range across the ammonia expander was kept constant at the design value (Table 4.1) as the ambient temperature increased by allowing the steam condenser reboiler pressure and temperature to increase correspondingly.
- For each ambient temperature-time increment, the amount of power generated by the ammonia expander and the corresponding decrease in power generated by the steam turbine was calculated assuming the ammonia expander operated at the design point efficiency.
- When the ambient temperature increased to the point that the steam condenser was operating at 5.5 in. Hg, the steam condenser pressure was fixed and the temperature range across the ammonia expander and its output was reduced in the succeeding temperature-time increments.

Table 4.2 shows the increase in power calculated for each site using this simplistic approach. For example, the energy gained at Miami through the addition of an ammonia expander is 0.6×10^6 kWh annually if minimum steam turbine backpressure is maintained as in Figure 4.3. If the steam turbine backpressure is allowed to rise and the ammonia expander is maintained at the inlet/outlet design temperature difference (as in Figure 4.4), the amount of energy recovered can be increased to at least 30×10^6 kWh annually. Table 4.2 gives the results of a similar assessment at each of the five sites. As evident from the table, the net increase in total plant power is significant, ranging from a factor of 3 at Anchorage to a factor of 300 at San Francisco.

The maximum steam turbine backpressure was assumed to be 5.5 in. Hg. according to General Electric's suggested turbine operating procedures, 5.5 in. Hg is the maximum backpressure allowed for continuous operation. This limit was used to obtain a conservative estimate of the additional power

TABLE 4.2. Estimated Energy Recovered

<u>Site</u>	<u>Energy Recovered, kWh/yr (Preoptimized Cooling Loop)</u>	<u>Energy Recovered, kWh/yr (Higher Steam Turbine Backpressure)</u>
Miami	0.60×10^6	30.0×10^6
San Francisco	0.20×10^6	66.5×10^6
Bakersfield	14.9×10^6	60.4×10^6
Chicago	53.5×10^6	214×10^6
Anchorage	69.0×10^6	225×10^6

available. However, the turbine can operate at higher backpressures for limited amounts of time. Optimizing allowable periods of even higher backpressures with ambient conditions would yield additional energy.

Though the scenario used in this study shows the potential for generating additional power, other promising schemes are possible if a complete reoptimization of the cooling loop with the expander system was allowed. For example, the design of the expander could be reoptimized with a high-backpressure turbine, allowing the expander to recover a greater amount of energy at higher ambient air temperatures. Another option would be to always maintain the turbine backpressure at the higher value, e.g., 5.5 in. Hg, thereby allowing the ammonia expander to displace a greater fraction of the steam turbine output at lower ambient temperatures. A third option might be to investigate the economic effects of discharging from the steam turbine at atmospheric pressure, thus obviating the sealing requirements intrinsic in operating in a partial vacuum. However, as these options would require a reoptimization of the cooling loop and a redesigning of the steam turbines, they would require a much more detailed analysis of the system and were deemed outside of the scope of this study.

4.2 SYSTEM COST

The objective in estimating costs for the system was only to present rough estimates of the costs involved. Costs were estimated for all additions and modifications to the existing cooling system. Capital costs were broken down into the categories listed in Table 4.3. Mafi-Trench Corporation provided the

TABLE 4.3. System Costs

Site	Anchorage	Chicago	Bakersfield	San Francisco	Miami
Ammonia Expander Cost (x 10 ⁶ \$)	3.60	3.48	2.64	2.64	2.88
Generator (x 10 ⁶ \$)	2.76	2.52	0.54	0.23	0.31
Feedheater (x 10 ⁶ \$)	0.24	0.23	0.20	0.15	0.12
Pump/Driver (x 10 ⁶ \$)	0.53	0.48	0.34	0.19	0.13
Miscellaneous (x 10 ⁶ \$)	0.71	0.67	0.37	0.32	0.34
Total Cost (x 10 ⁶ \$)	16.5	15.5	8.59	7.42	7.95
Annual Cost with O&M (x 10 ⁶ \$)	3.14	2.95	1.64	1.41	1.51
Total Energy (MWh)	69,000	53,500	14,900	200	600
Total Energy Enhanced (MWh)	225,000	214,000	60,400	66,500	30,000
Incremental Cost (\$/kWh)	0.046	0.055	0.11	7.05	2.52
Incremental Cost, Enhanced (\$/kWh)	0.014	0.014	0.027	0.021	0.050

capital costs for the ammonia expanders, and Westinghouse Electric and the Ideal Electric Co. provided the capital costs for the generators. Pump, pump driver, and feed heater costs were approximated from Guthrie (1974) and conversations with manufacturers. Piping, valves, controls, and other costs were lumped into a single estimate.

Shipping and installation was assumed to be 40% of the total capital costs. Indirect costs were an additional 20% and contingency, interest, and escalation were an additional 25%.

CAPITAL COSTS

X 1.4 (shipping, installation)

X 1.2 (indirect costs)

X 1.25 (interest, contingency, escalation)

TOTAL COST = 2.1 X CAPITAL COSTS

Annual operation and maintenance (O&M) costs were assumed to be 2% of the capital costs, and a fixed charge rate of 18% was used to determine the final incremental cost of energy recovered.

Major contributors to the performance of the expanders and their ultimate cost effectiveness are a function of two temperatures: the ambient air temperature and the turbine outlet temperature. These parameters include the number of operating hours, the enthalpy drop available to the expander, and the expander efficiency curve. Since the steam turbine backpressure is relatively constant when the expander system examined in this study is operating, the ambient air temperature becomes the dominant variable.

To observe the effect of ambient air temperature on the cost per unit energy generated by the ammonia expander, a comparison was made by plotting energy cost at a site against annual degree-hours, Figure 4.7. A degree-hour is analogous to the degree-day used in building space conditioning calculations. The design point temperature of the cooling tower at each site is the reference for the degree-hour calculation. When the ambient temperature falls below this cooling tower design temperature, the temperature difference is computed each hour. The sum of these numbers over the entire year is the number of annual degree-hours.

Figure 4.7 shows a somewhat proportional relationship between the number of degree-hours at each location and the resultant cost of energy. The cost of energy at San Francisco is the highest, notably higher than predicted by the approximate curve. This arises because, as shown in Table 4.1, the number of actual operating hours of the expander at San Francisco is only 14% of the number of ideal operating hours. This compares with 46% to 80% for the other sites. These actual operating hours were determined through the expander design optimization performed by the Mafi-Trench Co.

FIGURE 4.7. Energy Costs vs. Degree Hours

A second set of points in Figure 4.7 shows the cost of energy when the steam turbine backpressure is allowed to rise to a higher value. Though the relationship between the energy cost and degree-hours is less clear, it is evident that the cost of energy decreases significantly at all sites, ranging from 50 mills/kWh at Miami to less than 15 mills/kWh at Chicago and Anchorage. This dramatic improvement is due to both an increase in the number of operating hours and the efficiency advantage of extracting power through the ammonia expander rather than the steam turbine in this temperature range. It should be

emphasized that these energy costs are only approximations and a more detailed treatment of the system would no doubt yield different results. However, as earlier discussion demonstrated, these costs are not the most optimistic since other scenarios of operation could possibly yield even lower costs/kWhr.

5.0 CONCLUSIONS

As anticipated, the addition of the energy recovery system to a previously optimized ammonia dry/wet cooling loop is notably more cost effective in colder climates than in warmer ones. Employing the expander in these colder climates has the added advantage of matching the winter peaks in power demands with the expander's maximum power output. Preliminary cost analysis indicates that energy from the expander system may be provided at less than 50 to 60 mills/kWh at Anchorage and Chicago, assuming the power plant already includes an ammonia heat rejection system. Though the cost estimates were not intended to be precise, the figures used were selected to be conservative.

Further analysis indicated that optimizing the entire ammonia cooling system for operation with an expander yields a significant amount of additional power and substantially reduces the cost of energy produced. For instance, by simply raising the steam turbine backpressure slightly above the normal operating value, the cost of power from the ammonia system for Anchorage and Chicago is reduced to 10 to 20 mills/kWh. (The increase in power from the ammonia expander more than offsets the loss in power from the steam turbine, although the effective cost of power from the steam turbine would increase slightly.)

Other more radical changes, such as reducing the size of the low-pressure steam turbine and replacing it with an ammonia expander, may yield binary-cycle designs that are cost effective at any site. Such changes are not untenable in light of the current demonstration project being conducted by Electricité de France (EDF) in France. The binary-cycle plant under construction will have a crossover temperature from the steam cycle to the ammonia cycle of approximately 165°F. In addition to demonstrating the technical and economic feasibility of the binary-cycle concept, the EDF project will provide operational experience and understanding of environmental concerns. The results are expected to support the preliminary findings of the PNL study.

The promising potential for economic power generation through the addition of an ammonia expander system indicates that a more comprehensive analysis

of the various system options is warranted. Building on the R&D efforts of the French, further investigations in the U.S. should explore:

- 1) the effect of optimizing the entire low-pressure part of the steam plant (including the heat rejection system) to obtain the most cost-effective design
- 2) the acceptability of bottoming cycles to the utility industry.

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APPENDIX

COOLING SYSTEM DESIGN PARAMETERS

TABLE A.1. Cooling System Design Parameters: Anchorage

Ambient Temp Deg F	Turbine Outlet Temp Deg F	Back Pressure In. Hg	Heat Rate Factor	Range Deg F	TTDP Deg F	HX LMTD Deg F	HX ITD Deg F	Power Generated Megawatts	Percent of Hx Deluged	Percent of Heat Reject By Wet Hx	Water Evaporated Acre Feet	PCT Time Ambient Temp Percent
81.0	113.9	2.91	.9938	17.686	6.502	12.36	23.24	995.481	96.5926	98.7436	10.5761	.080
73.0	113.9	2.91	.9938	24.002	6.502	16.41	31.24	995.481	80.1786	90.0817	108.4436	1.000
67.0	113.9	2.91	.9938	28.586	6.502	19.59	37.24	995.481	64.8904	79.0764	171.5854	1.930
63.0	113.9	2.91	.9938	31.565	6.502	21.77	41.24	995.481	57.0119	71.7111	292.4499	3.840
60.0	113.9	2.91	.9938	33.797	6.502	23.41	44.24	995.481	49.7285	64.5792	205.0330	3.100
58.0	113.9	2.91	.9938	35.363	6.502	24.44	46.24	995.481	44.6130	59.1669	227.1038	3.830
56.0	113.9	2.91	.9938	36.847	6.502	25.53	48.24	995.481	39.6867	53.6693	247.9101	4.710
54.0	113.9	2.91	.9938	38.367	6.502	26.60	50.24	995.481	33.9536	47.1726	219.0278	4.810
52.0	113.9	2.91	.9938	39.872	6.502	27.68	52.24	995.481	28.5502	40.6080	177.8706	4.600
50.0	113.9	2.91	.9938	41.279	6.502	28.84	54.24	995.481	23.3980	34.0794	138.6345	4.340
48.0	113.9	2.91	.9938	42.776	6.502	29.92	56.24	995.481	18.1290	26.9893	95.2295	3.820
46.0	113.9	2.91	.9938	44.309	6.502	30.98	58.24	995.481	12.7432	19.3972	63.4538	3.600
43.0	113.9	2.91	.9938	46.587	6.502	32.58	61.24	995.481	4.3607	7.1130	38.0590	6.050
39.0	111.7	2.74	.9923	47.912	6.494	33.49	62.97	997.016	.0000	.0000	.0000	6.230
36.0	108.9	2.52	.9906	47.852	6.486	33.45	62.90	998.704	.0000	.0000	.0000	3.660
33.0	106.0	2.33	.9894	47.807	6.480	33.43	62.85	999.983	.0000	.0000	.0000	44.400

TABLE A.2. Cooling System Design Parameters: Chicago

Ambient Temp Deg F	Turbine Outlet Temp Deg F	Back Pressure In. Hg	Heat Rate Factor	Range Deg F	TTDP Deg F	HX LMTD Deg F	HX ITD Deg F	Power Generated Megawatts	Percent of Hx Deluged	Percent of Heat Reject By Wet Hx	Water Evaporated Acre Feet	PCT Time Ambient Temp Percent
98.0	118.4	3.30	.9980	9.443	6.250	6.23	12.10	989.160	89.8345	97.6231	16.9437	.110
90.0	118.4	3.30	.9980	15.549	6.250	10.47	20.10	989.160	75.3386	90.5048	262.9964	2.080
84.0	118.4	3.30	.9980	20.132	6.250	13.64	26.10	989.160	61.3281	80.7222	115.0211	1.100
82.0	118.4	3.30	.9980	21.696	6.250	14.67	28.10	989.160	57.1660	76.9881	200.7580	2.070
77.0	118.4	3.30	.9980	25.472	6.250	17.36	33.10	989.160	46.0242	65.9557	499.3493	6.400
72.0	118.4	3.30	.9980	29.231	6.250	20.05	38.10	989.160	33.8891	52.1486	431.6334	7.340
67.0	118.4	3.30	.9980	32.958	6.250	22.78	43.10	989.160	22.2718	36.5677	331.4382	8.420
62.0	118.4	3.30	.9980	36.779	6.250	25.43	48.10	989.160	9.7959	17.8519	140.8151	7.590
57.0	118.1	3.27	.9977	40.337	6.249	27.93	52.82	989.447	.0000	.0000	.0000	7.140
52.0	113.1	2.85	.9933	40.244	6.228	27.83	52.64	993.866	.0000	.0000	.0000	6.640
47.0	108.3	2.48	.9904	40.155	6.215	27.77	52.53	996.836	.0000	.0000	.0000	6.720
42.0	103.5	2.16	.9885	40.100	6.206	27.73	52.45	998.681	.0000	.0000	.0000	6.620
37.0	98.8	1.88	.9876	40.070	6.202	27.71	52.41	999.670	.0000	.0000	.0000	8.460
33.0	95.0	1.68	.9872	40.061	6.200	27.71	52.40	999.998	.0000	.0000	.0000	29.310

TABLE A.3. Cooling System Design Parameters: Bakersfield

Ambient Temp Deg F	Turbine Outlet Temp Deg F	Back Pressure In. Hg	Heat Rate Factor	Range Deg F	TTDP Deg F	HX LMTD Deg F	HX ITD Deg F	Power Generated Megawatts	Percent of Hx Deluged	Percent of Heat Reject By Wet Hx	Water Evaporated Acre Feet	PCT Time Ambient Temp Percent
109.0	123.6	3.80	1.0045	5.708	5.024	3.90	7.43	982.731	58.0289	93.4763	19.8706	.120
104.0	123.6	3.80	1.0045	9.654	5.024	6.44	12.43	982.731	49.8772	86.8222	256.2969	1.760
98.0	123.6	3.80	1.0045	14.172	5.024	9.67	18.43	982.731	42.3849	77.7640	202.5527	1.660
94.0	123.6	3.80	1.0045	17.195	5.024	11.82	22.43	982.731	37.3651	70.6701	277.5374	2.610
88.0	123.6	3.80	1.0045	21.803	5.024	14.97	28.43	982.731	30.0714	58.4794	557.9108	6.780
82.0	123.6	3.80	1.0045	26.351	5.024	18.18	34.43	982.731	21.4955	43.6648	328.5369	5.700
77.0	123.6	3.80	1.0045	29.994	5.024	20.97	39.43	982.731	14.4731	30.1397	242.8151	6.450
72.0	123.6	3.80	1.0045	33.796	5.024	23.63	44.43	982.731	6.0817	13.5608	116.9394	7.280
67.0	122.4	3.68	1.0028	36.676	5.018	25.64	48.21	984.376	.0000	.0000	.0000	8.220
62.0	117.4	3.20	.9969	36.517	4.996	25.51	47.99	990.244	.0000	.0000	.0000	9.100
57.0	112.5	2.79	.9928	36.405	4.981	25.45	47.85	994.382	.0000	.0000	.0000	10.330
52.0	107.7	2.44	.9901	36.329	4.971	25.40	47.75	997.161	.0000	.0000	.0000	11.140
47.0	102.9	2.13	.9884	36.283	4.964	25.37	47.69	998.873	.0000	.0000	.0000	10.210
42.0	98.3	1.85	.9875	36.258	4.961	25.35	47.66	999.774	.0000	.0000	.0000	8.360
37.0	93.7	1.62	.9872	36.250	4.960	25.34	47.65	1000.077	.0000	.0000	.0000	6.110
33.0	90.1	1.45	.9873	36.252	4.960	25.35	47.66	1000.003	.0000	.0000	.0000	4.170

TABLE A.4. Cooling System Design Parameters: San Francisco

Ambient Temp Deg F	Turbine Outlet Temp Deg F	Back Pressure In. Hg	Heat Rate Factor	Range Deg F	TTDP Deg F	HX LMTD Deg F	HX ITD Deg F	Power Generated Megawatts	Percent of Hx Deluged	Percent of Heat Reject By Wet Hx	Water Evaporated Acre Feet	PCT Time Ambient Temp Percent
93.0	113.2	2.85	.9933	8.994	6.228	6.08	11.65	993.863	81.8731	95.9101	9.4708	.060
86.0	113.2	2.85	.9933	14.295	6.228	9.83	18.65	993.863	72.3840	90.0975	40.1222	.300
79.0	113.2	2.85	.9933	19.584	6.228	13.58	25.65	993.863	63.1043	81.8740	117.2016	1.080
74.0	113.2	2.85	.9933	23.319	6.228	16.30	30.65	993.863	54.2268	73.2251	59.2207	.660
72.0	113.2	2.85	.9933	24.877	6.228	17.33	32.65	993.863	49.9426	68.7619	82.6686	1.010
67.0	113.2	2.85	.9933	28.621	6.228	20.04	37.65	993.863	39.9603	56.8941	550.1487	8.790
62.0	113.2	2.85	.9933	32.355	6.228	22.76	42.65	993.863	27.3658	41.0479	606.7043	14.340
57.0	113.2	2.85	.9933	36.072	6.228	25.49	47.65	993.863	13.3401	21.5851	543.4006	26.060
52.0	113.2	2.85	.9933	39.864	6.228	28.16	52.65	993.863	.0000	.0000	.0000	26.650
47.0	108.3	2.48	.9904	39.776	6.215	28.10	52.53	996.834	.0000	.0000	.0000	14.000
42.0	103.5	2.16	.9885	39.721	6.206	28.06	52.46	998.680	.0000	.0000	.0000	5.370
37.0	98.8	1.88	.9876	39.692	6.202	28.04	52.42	999.670	.0000	.0000	.0000	1.430
33.0	95.0	1.68	.9872	39.682	6.200	28.04	52.41	999.998	.0000	.0000	.0000	.250

TABLE A.5. Cooling System Design Parameters: Miami

Ambient Temp Deg F	Turbine Outlet Temp Deg F	Back Pressure In. Hg	Heat Rate Factor	Range Deg F	TTDP Deg F	HX LMTD Deg F	HX ITD Deg F	Power Generated Megawatts	Percent of Hx Deluged	Percent of Heat Reject By Wet Hx	Water Evaporated Acre Feet	PCT Time Ambient Temp Percent
96.0	124.0	3.84	1.0051	15.302	6.321	10.65	20.07	982.542	47.0999	74.9165	5.3956	.050
91.0	124.0	3.84	1.0051	19.122	6.321	13.30	25.07	982.542	37.8516	63.1743	194.5004	2.300
88.0	124.0	3.84	1.0051	21.411	6.321	14.89	28.07	982.542	30.9681	54.1991	133.9560	1.910
87.0	124.0	3.84	1.0051	22.139	6.321	15.45	29.07	982.542	28.5401	50.9759	125.0367	1.910
82.0	124.0	3.84	1.0051	25.871	6.321	18.17	34.07	982.542	19.4385	35.4174	1064.8255	25.420
77.0	124.0	3.84	1.0051	29.629	6.321	20.87	39.07	982.542	7.1713	14.7733	480.1266	28.410
72.0	122.4	3.68	1.0029	32.219	6.311	22.66	42.47	984.759	.0000	.0000	.0000	19.620
67.0	117.4	3.20	.9969	32.078	6.283	22.57	42.28	990.667	.0000	.0000	.0000	9.600
62.0	112.4	2.79	.9928	31.979	6.264	22.49	42.15	994.855	.0000	.0000	.0000	5.000
57.0	107.5	2.43	.9900	31.912	6.251	22.45	42.06	997.650	.0000	.0000	.0000	3.100
52.0	102.7	2.11	.9883	31.871	6.243	22.42	42.01	999.359	.0000	.0000	.0000	1.590
47.0	98.0	1.84	.9875	31.850	6.239	22.41	41.99	1000.242	.0000	.0000	.0000	.750
42.0	93.3	1.60	.9872	31.844	6.238	22.40	41.98	1000.516	.0000	.0000	.0000	.280
37.0	88.7	1.39	.9874	31.848	6.238	22.40	41.98	1000.353	.0000	.0000	.0000	.060

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