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AN EVALUATION OF THERMAL ENERGY STORAGE OPTIONS FOR PRECOOLING GAS TURBINE INLET AIR

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SUMMARY

Most utilities in the United States experience the greatest demand for electricity during periods of high ambient temperature. Unfortunately, the performance of many power plants decreases with increasing ambient temperature. This is especially true of gas turbines that use a constant volume of incoming ambient air for the working fluid. As ambient temperature increases, the density of the inlet air and the generating capacity of the gas turbine decrease.

Several approaches have been used to reduce the temperature of gas turbine inlet air. One of the most successful uses off-peak electric power to drive vapor-compression-cycle ice makers. The ice is stored until the next time high ambient temperature is encountered, when the ice is used in a heat exchanger to cool the gas turbine inlet air. An alternative concept would use seasonal thermal energy storage to store winter chill for inlet air cooling. The objective of this study was to compare the performance and economics of seasonal thermal energy storage in aquifers (aquifer thermal energy storage) with diurnal ice thermal energy storage for gas turbine inlet air cooling. The investigation consisted of developing computer codes to model the performance of a gas turbine, energy storage system, heat exchangers, and ancillary equipment. The performance models were combined with cost models to calculate unit capital costs and levelized energy costs for each concept. The levelized energy cost was calculated for three technologies (an oversized gas turbine, a diurnal ice thermal energy storage system, and an aquifer seasonal thermal energy storage system) operating at two annual capacity factors (0.05 and 0.2) in two locations (Minneapolis, Minnesota and Birmingham, Alabama).

Precooling gas turbine inlet air with cold water supplied by an aquifer thermal energy storage system provided lower cost electricity than simply increasing the size of the turbine for meteorological and geological conditions existing in the Minneapolis vicinity. A 15 to 20% cost reduction resulted for both 0.05 and 0.2 annual operating factors. In contrast, ice storage precooling was found to be between 5 and 20% more expensive than larger gas turbines for the Minneapolis location.

In Birmingham, aquifer thermal energy storage precooling was preferred at the higher capacity factor and ice storage precooling was the best option at the lower capacity factor. In both cases, the levelized cost was reduced by approximately 5% when compared to larger gas turbines.

These preliminary results indicate that aquifer thermal energy storage systems should be given serious consideration as a option for increasing peak generating capacity with combustion turbines. Like all aquifer thermal energy storage applications, the concept's cost effectiveness will vary significantly with site-specific geologic conditions.

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1.0 INTRODUCTION

Gas turbines are increasingly becoming the preferred power generating option for many power producers. It is projected that gas turbines will represent up to 50% of the estimated 140,000 MWe of new capacity that must be added in the next decade (Power Engineering 1990). While gas turbines have many attractive features, the degradation of gas turbine generating capacity during time periods with high ambient temperature can be a significant problem for summer peaking utilities.

A simple cycle gas turbine operates with a constant volume flow rate of air, which serves as the working fluid for the gas turbine. As ambient temperature increases, the density of the inlet air decreases, reducing the mass flow rate through the gas turbine. The generating capacity of the gas turbine is determined by the mass flow rate. Therefore, the generating capacity of the gas turbine decreases with ambient temperature (Kerrebrock 1977; Wilson 1984). Approximately one-third of a typical gas turbine's power output is lost as the ambient temperature climbs from -6.6°C to 35°C (20°F to 95°F) (Ondryas 1991). The overall effect on performance is specific to a given turbine design and is well characterized (Smith undated; Allen undated; BPA 1988).

A review of data on existing gas turbine installations shows that the summer capacity of a typical gas turbine is between 15 and 25% lower than the winter capacity of the same gas turbine. Nationally, this represents an 8000 MWe reduction in summer peak generating capacity (DOE 1991). With projections for an additional 5000 MWe to 10,000 MWe of new gas turbine capacity to be added each year for the next 10 years (Power Engineering 1990), the corresponding deficit in gas turbine summer generating capacity could be as large as 20,000 MWe.

Traditionally evaporative cooling has been used to cool gas turbine inlet air. Water is evaporated by the air entering the compressor, lowering the air temperature. However, evaporative cooling can only reduce the temperature of the inlet air to the wet bulb temperature. Actual cooling will be determined by the wet bulb and dry bulb temperatures of the incoming air and the efficiency of the evaporative cooler. As an example, evaporative coolers will reduce the temperature of inlet air at 37.8° (100°F) and a relative humidity of 45% to between 23.8°C and 26.6°C (75°F and 80°F). Further reductions in inlet air temperature will require additional cooling (Ondryas 1991).

Absorption cycle chillers, driven by the thermal energy in the exhaust of the gas turbine and electrically driven mechanical chillers, have been considered for gas turbine inlet air cooling but both approaches have significant shortcomings (Ondryas 1991). The most successful approach to providing gas turbine inlet cooling uses off-peak electric power to drive a vapor compression cycle ice maker, producing ice that is then stored. During the next peak in electricity demand, the stored ice is used to cool the gas turbine inlet air to approximately 4.4°C (40°F). A diurnal ice thermal energy storage (TES) system has been installed at Lincoln Electric System's Rokeby Station and has increased the summer peaking capacity of a gas turbine from 53 MWe to 64 MWe. The capital cost to increase the capacity of the gas turbine was approximately $165/kWe (Power 1992).

While the diurnal ice TES system installed at the Rokeby Station is very attractive, other thermal energy storage options may also be of interest. Aquifer thermal energy storage (ATES) includes a range of technologies that can economically store thermal energy for long periods of time. An ATES system can collect and store winter chill, available during cold winter months, for gas turbine inlet air cooling when required.

This report documents the technical and economic feasibility of using ATES for gas turbine inlet air cooling. Section 2.0 discusses the various options for cooling. Section 3.0 describes the approach used for this study and Section 4.0 discusses the design and performance evaluation effort. The economic evaluation is discussed in Section 5.0 and overall conclusions from this study are detailed in Section 6.0. Reference information is included in Section 7.0.

2.0 OPTIONS FOR GAS TURBINE PRECOOLING INLET AIR

While there are a number of options for improving the summer peak capacity of gas turbines, the research documented in this report focuses on a comparison of seasonal TES with diurnal TES. Three options will be discussed: evaporative cooling, diurnal TES using ice storage, and seasonal thermal energy storage using aquifers as the storage media [aquifer thermal energy storage (ATES)].

2.1 EVAPORATIVE COOLING

Evaporative cooling represents current practice for most gas turbine installations. Typical systems pass incoming gas turbine inlet air through a honeycomb medium saturated with water to achieve gas turbine inlet temperatures of approximately 26°C (80°F), with a pressure drop of approximately 2 KPa (0.6 in. H_2O). Care must be taken when applying evaporative cooling to ensure that condensation or carryover of water into the compressor will not result in fouling and ultimately lead to errosion of turbine blades (Allen undated). Further reductions in inlet air temperature will require additional inlet air cooling (Ondryas 1991).

2.2 DIURNAL ICE TES SYSTEM

One diurnal ice TES system has been constructed for gas turbine inlet air cooling. The system, installed at Lincoln Electric System's Rokeby Station, consists of a refrigeration unit, ice/water storage tank, and an air cooling heat exchanger. During off-peak periods, cold water from the bottom of the storage tank is pumped to three ice-making units located on the top of the tank. The cold water is frozen and the ice is stored in the tank. Off-peak electric power is used to drive the ice makers.

During periods with high ambient temperature, cold water is pumped from the tank and is used to cool gas turbine inlet air. The inlet air passes through the shell side of a heat exchanger while chilled water passes through the tube side. Indirect heat transfer is used to protect the gas turbine from impurities in the chilled water (Power 1992; Ebeling et al. 1992). The heat exchanger is sized to reduce the temperature of the inlet air from 38.3°C to 4.4°C (101°F to 40°F) with a pressure drop of 1.67 KPa (0.5 in. H_2O).

Reducing inlet air temperature to 4.4°C (40°F) increases the peak capacity of the Rokeby gas turbine from 53 MWe to 64 MWe. The 20% increase in gas turbine capacity is obtained at a cost of approximately $165/kWe. The Electric Power Research Institute (EPRI) estimates that new gas turbines cost at least $387/kWe (EPRI 1989). This represents a 58% reduction in the cost of adding peaking capacity rather than installing new gas turbines.

2.3 SEASONAL THERMAL ENERGY STORAGE

STES technology includes a number of techniques that can efficiently and economically store thermal energy for long periods of time. Current research suggests that aquifer thermal energy storage is the most economical concept and is nearest to commercialization. Therefore, this study focused on evaluating ATES as a representative STES system; but, other systems are available and may be attractive for specific applications.

Aquifer thermal energy storage is a technology that allows relatively low-grade thermal energy to be stored and retrieved for future use on either a seasonal or diurnal basis. Water pumped from a set of supply wells is either heated or cooled and then injected into a set of storage wells. Later, the storage wells are pumped and the warm or cool water can be used to meet a thermal load (see Figure 1). The principal advantages of ATES are the use of an existing aquifer formation as both the media and physical containment component of the storage system, the use of water as the heat transfer medium, and the concept's ability to store energy on either a seasonal or diurnal basis. The concept is limited, however, to locations where the energy source, energy application, and a suitable aquifer are in close proximity to each other.

For gas turbine inlet air cooling applications, water withdrawn from supply wells is chilled by cold winter air in a conventional cooling tower prior to injection into storage wells. The chilled water is stored until periods with high ambient temperature, when chilled water is withdrawn from the aquifer and used in a heat exchanger to cool the gas turbine inlet air.

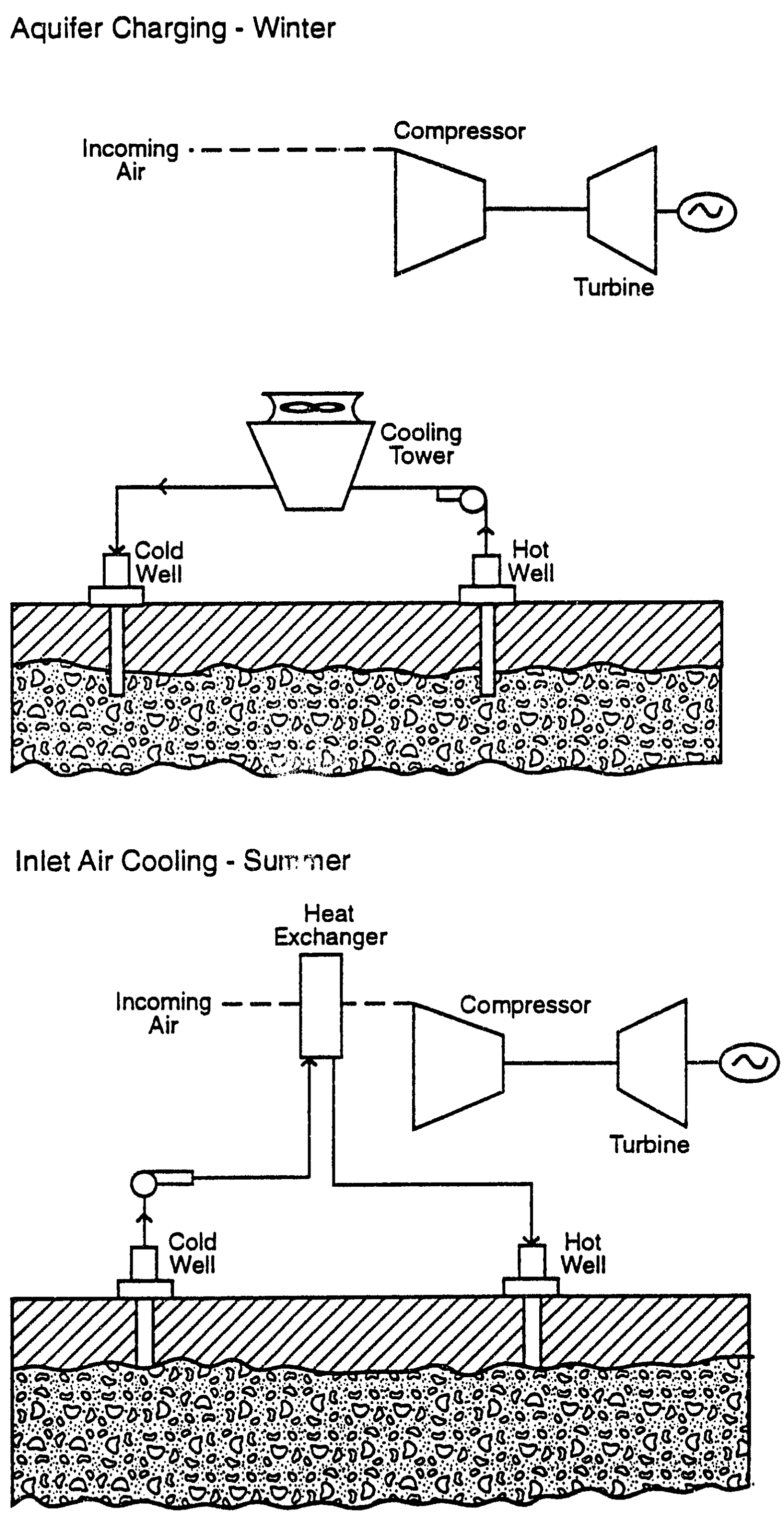

FIGURE 1. Gas Turbine Inlet Air Cooling Using Aquifer Thermal Energy Storage

The heated water is then returned to the aquifer and stored until the next winter, when it is again cooled using winter chill. ATES eliminates the need to use off-peak electric power for charging storage, and reduces the cost of charging storage while minimizing fuel consumption and environmental impacts associated with the generation of off-peak electric power.

3.0 APPROACH

The study consisted of comparing the levelized cost of power for three technologies proposed to increase the capacity of a gas turbine during periods of high ambient temperature. The three options include:

- Install a Larger Gas Turbine - The first approach consisted of installing a larger gas turbine with a capacity sufficiently large to compensate for the reduced generating capacity during periods with high ambient temperature.
- Diurnal Ice Storage - Diurnal ice storage uses off-peak power to drive an ice maker that generates and stores the ice until high ambient temperatures are encountered. This approach is similar to the system installed by Lincoln Electric and was described in Section 2.2.
- Seasonal Thermal Energy Storage - ATES used for storing winter chill was selected as the representative seasonal TES concept. The system collects winter chill that is stored until needed to provide gas turbine inlet air precooling.

The marginal cost of power ($/kWh) was calculated for the three technologies operating at two annual capacity factors (0.05 and 0.2) in two locations (Minneapolis, Minnesota and Birmingham, Alabama). Thus, a total of 12 cases were evaluated. The reference technology was presumed to be a gas-fired combustion turbine with performance characteristics representative of a General Electric model MS7001EA. System designs were based on precooling the inlet air from the summer design temperature (37.8°C or 100°F at both locations) to 4.4°C (40°F) with ice storage, 7.2°C (45°F) with ATES in Minneapolis, and 11.1°C (52°F) with ATES in Birmingham. These temperatures were selected based on previous experience (Ebeling et al. 1992; Spurr 1986; Midkiff and Brett 1991). As a result, each of the inlet air precooling systems results in a different amount of incremental power relative to operating without precooling. The size of the larger gas turbines was arbitrarily set to provide the same incremental generating capacity at the summer design temperature (37.8°C or 100°F) as the two ATES precooled systems. Larger gas turbines matching the incremental power produced by ice storage precooling were not evaluated. Turbine performance was calculated using average meteorological conditions for the locations included in the comparison and for the two capacity factors used in the analysis.

The performance evaluation and the results of component sizing calculations are presented in Section 4.0. Results of the economic evaluation are presented in Section 5.0.

4.0 DESIGN AND PERFORMANCE EVALUATION

Design characteristics and projected system performance was estimated for the reference gas turbine and each of the three options for improving gas turbine peak performance. The reference gas turbine is discussed in Section 4.1. The larger gas turbine is described in Section 4.2 while the ATES system is presented in Section 4.3. Finally, the diurnal ice TES system is described in Section 4.4.

4.1 REFERENCE GAS TURBINE PERFORMANCE AND OPERATING CONDITIONS

The reference gas turbine was assumed to produce 83.50 MWe at International Standards Organization (ISO) standard atmospheric conditions (15°C or 59°F and 60% relative humidity). Capacity at other operating temperatures was calculated based on the relationship expressed by Equation (1).

$$\text{Generating Capacity, MW} = 83.5 * \{1 + [4.39 * 10^{-3} * (59 - T)]\} \quad (1)$$

where T is turbine inlet temperature, °F.

Thus, at the summer design temperature of 37.8°C (100°F), generating capacity is reduced to 68.47 MW. As noted in Section 3.0, turbine inlet air could likely be cooled to 7.2°C (45°F) in Minneapolis and 11.1°C (52°F) in Birmingham using chilled water provided by ATES systems and 4.4°C (40°F) using chilled water provided by ice storage systems. These inlet temperatures result in generating capacities of 88.63 MW and 86.07 MW for Minneapolis and Birmingham ATES systems, respectively, and 90.47 MW for ice storage systems. This corresponds to peak capacity increases of 20.16 MW, 17.60 MW, and 22.00 MW. Using a larger gas turbine to obtain peak increases in generating capacity equivalent to the two ATES systems, the required increase in ISO capacity would be 24.59 MW and 21.46 MW. Turbine capacity data are summarized in Table 1.

TABLE 1. Gas Turbine Generating Capacities

Operating Condition	Power Output, MW
ISO conditions (15°C/59°F)	83.50
Summer design (37.8°C/100°F)	68.47
Minneapolis ATES precool (7.2°C/45°F)	88.63
Birmingham ATES precool (11.2°C/52°F)	86.07
Ice storage precool (4.4°C/40°F)	90.47
Minneapolis ATES peak increase	20.16
Birmingham ATES peak increase	17.60
Ice storage peak increase	22.00
Minneapolis ISO increase	24.59
Birmingham ISO increase	21.46

As noted in Section 3.0, the gas turbines were evaluated at annual operating capacity factors of 0.05 and 0.2, corresponding to 438 and 1752 operating hours per year. Reference turbine and larger gas turbine performance was based on the average temperature occurring during the operating period, which was presumed to be the warmest hours of the year. Hourly weather data compiled by the U.S. Department of Commerce (1963) was used to calculate average operating temperatures, which are summarized in Table 2 along with the corresponding capacity relative to the ISO temperature [15°C (59°F)].

The reference gas turbine, larger gas turbine, and ATES-precooled gas turbines have the flexibility to match various operating schedules, including actually operating during the warmest hours of the year if desirable; ice storage systems do not have this flexibility. Design of the ice storage systems required specification of the operating schedules (hours/day, days/week, weeks/year) yielding the presumed annual capacity factors of 0.05 and 0.2. An attempt was made to select a reasonable combination that would be representative of a typical peaking cycle with a concentration of operation during the warmest hours of the year. The operating schedules evaluated are summarized in Table 3.

TABLE 2. Average Ambient Air Temperatures

Location	Capacity Factor	Temperature, °C (°F)		Relative Capacity
Birmingham	0.05	33.1	(91.6)	0.857
Birmingham	0.2	29.3	(84.7)	0.887
Minneapolis	0.05	29.6	(85.2)	0.885
Minneapolis	0.2	24.8	(76.6)	0.923

TABLE 3. Ice Storage System Operating Schedules

Capacity Factor	Days/Week	Hours/Day	Weeks/Year[a]
Reference Case			
0.05	7	6	10
0.2	7	12	21
Sensitivity Case			
0.05	5	6	15
0.2	5	12	29

(a) Multiplying days/week times hours/day times weeks/year does not exactly yield presumed annual operating hours because of rounding.

It is inevitable that a fixed operating schedule will preclude some of the warmest hours of the year. Thus, the average ambient temperature would certainly be lower than shown in Table 2 for systems designed to operate in this fashion. The impact of a lower average ambient air temperature is to reduce the annual incremental kWh produced by inlet air precooling, but increase the annual incremental kWh produced by a larger gas turbine operating on the same schedule. In general, a lower average ambient air temperature (if all other factors are equal) will favor building a larger turbine instead of precooling the inlet air. Determining the actual annual average ambient air temperature for a fixed operating schedule would require hour-by-hour weather modeling, which was beyond the scope of this preliminary evaluation. Therefore, the ice storage precooling systems may actually be at greater disadvantage when compared to the ATES-precooling systems than indicated in this analysis.

4.2 LARGER GAS TURBINE DESIGN CHARACTERISTICS AND PERFORMANCE

Incremental annual energy production for larger gas turbines was calculated by multiplying incremental ISO capacity by the "relative capacity" at the average operating temperature and the annual operating hours. The resultant energy production data are presented in Table 4.

Incremental fuel costs for larger gas turbines were calculated based on the incremental annual energy production and incremental heat rate. The latter was calculated based on the average operating temperatures presented in Table 2 and the heat rate relationship defined by Equation (2).

$$\text{Heat Rate (HHV)} = 11{,}588 * [0.941 + (3.66 * 10^{-4} * T) + (1.22 * 10^{-5} * T^2)] \quad (2)$$

where T is turbine inlet temperature, °F

Incremental heat rates and annual fuel input for larger gas turbines are summarized in Table 5.

TABLE 4. Incremental Energy Production for Larger Turbines

Location	Capacity Factor	Incremental Energy, MWh/yr
Birmingham	0.05	8,055
Birmingham	0.2	33,349
Minneapolis	0.05	9,532
Minneapolis	0.2	39,764

TABLE 5. Incremental Heat Rates and Fuel Consumption for Larger Turbines

Location	Capacity Factor	Heat Rate, Btu/kWh	Fuel Input MWh/yr (10^6 Btu/yr)
Birmingham	0.05	12,480	29,500 (100,500)
Birmingham	0.2	12,280	119,980 (409,500)
Minneapolis	0.05	12,290	34,310 (117,100)
Minneapolis	0.2	12,060	140,520 (479,600)

4.3 ATES SYSTEM DESIGN CHARACTERISTICS AND PERFORMANCE

For the ATES systems, inlet air at 37.8°C (100°F) and 40% relative humidity is cooled to saturated conditions at 11.1°C (52°F) in Birmingham and 7.2°C (45°F) in Minneapolis. Inlet air flow is 1,028,000 kg/hr (2,254,700 lb/hr) at ISO conditions. At the design temperatures noted above, the air flow rates increase to 1,036,000 kg/hr (2,285,500 lb/hr) and 1,051,100 kg/hr (2,317,200 lb/hr), respectively. The resultant peak cooling loads for the two design conditions are 13.4 MWht ($45.71 * 10^6$ Btu) in Birmingham and 16.76 MWht ($57.24 * 10^6$ Btu) in Minnesota. Annual cooling loads were calculated based on the average ambient temperatures and relative humidities for the four location and operational capacity factor combinations. Cooling load and capacity factor data are summarized in Table 6.

Aquifer-related design characteristics can vary significantly depending on location. The ATES system characteristics presented in Table 7 were taken directly or derived from data presented in Spurr (1986) for Minneapolis and Midkiff and Brett (1991) for Birmingham. Spurr (1986) describes a feasibility study conducted for a proposed ATES cooling system in Minneapolis-St. Paul. Midkiff and Brett (1991) describes the design and operating performance for an ATES cooling system operating at the University of Alabama.

TABLE 6. Peak and Annual Cooling Loads/ATES Systems

Location	Operational Capacity Factor	Peak Load, MWht (10^6 Btu/hr)	Annual Load, MWht (10^6 Btu)	Cooling Capacity Factor
Birmingham	0.05	13.4 (45.7)	4,156 (14,200)	0.035
Birmingham	0.2	13.4 (45.7)	14,700 (50,100)	0.125
Minneapolis	0.05	16.8 (57.2)	5,300 (18,100)	0.036
Minneapolis	0.2	16.8 (57.2)	17,140 (58,500)	0.117

TABLE 7. ATES System Design Data

Parameter	Minneapolis	Birmingham
Average injection temperature	2.5°C (36.5°F)	5.8°C (42.5°F)
Average recovery temperature	3.3°C (38.0°F)	7.2°C (45.0°F)
Cooling water return temperature	15.5°C (60.0°F)	21.1°C (70.0°F)
Storage charging period	120 days	60 days
Flow rate per well	0.126 m^3/sec (2000 gpm)	0.00095 m^3/sec (150 gpm)
Well depth	122 m (400 ft)	30.5 m (100 ft)
Well spacing	152 m (500 ft)	45.7 m (150 ft)
Transmission piping length	305 m (1000 ft)	152 m (500 ft)

Although the ATES precooled and larger gas turbine systems (at a single location) were sized to produce the same incremental power production at the summer design temperature, power production is different at all other temperatures. The ATES system was designed to produce a constant power output regardless of ambient air temperature; the inlet air is always cooled to the same temperature, either 7.2°C (45°F) at Minneapolis or 11.1°C (52°F) at Birmingham. On the other hand, power production increases for the non-cooled turbines as the ambient air temperature drops from the summer design point. Therefore, incremental annual energy production for the precooled systems is less than the systems with larger turbines.

Incremental energy production for the ATES system was calculated by comparing energy production with precooling to the energy production with the reference 83.5 MW gas turbine operating at the average ambient temperature. Energy production data for the ATES system is summarized in Table 8.

TABLE 8. Incremental Energy Production in ATES Precooled Systems

Location	Operational Capacity Factor	Energy Production, MWh/yr
Birmingham	0.05	6,360
Birmingham	0.2	21,006
Minneapolis	0.05	6,452
Minneapolis	0.2	20,288

Incremental fuel requirements for the ATES system were also calculated by evaluating the difference between operating the 83.5 MW (ISO) turbine with and without precooling. Precooling lowers the inlet air temperature and average heat rate, but a greater increase in energy production results in an overall increase in fuel consumption, as shown in Table 9.

The relative size (and cost) of the inlet air heat exchanger was calculated based on a comparison of thermal duties and log-mean temperature differences (LMTD) with the size and cost specified for the Lincoln Electric System plant. Size (heat transfer area) was presumed to be proportional to thermal duty and inversely proportional to LMTD. The overall heat transfer coefficient was assumed to be constant. Relative thermal duties, LMTDs, and sizes are summarized in Table 10.

4.4 DIURNAL ICE STORAGE DESIGN CHARACTERISTICS AND PERFORMANCE

For the diurnal ice storage system, inlet air at 37.8°C (100°F) and 40% relative humidity is cooled to saturated conditions at 4.4°C (40°F) for ice storage systems in both Minneapolis and Birmingham. Inlet air flow is 1,028,000 kg/hr (2,254,700 lb/hr) at ISO conditions. At the design

TABLE 9. Incremental Fuel Consumption in ATES Precooled Systems

Location	Operational Capacity Factor	Fuel Consumption, MWht/yr (MMBtu/yr)
Birmingham	0.05	13,600 (42,700)
Birmingham	0.2	45,100 (141,600)
Minneapolis	0.05	14,000 (43,900)
Minneapolis	0.2	44,400 (139,400)

TABLE 10. Precooling Heat Exchanger Sizing

System	Thermal Duty, MWt (MMBtu/hr)	LMTD, °C (°F)	Relative Size
Lincoln Electric	14.7 (46.25)	14.3 (25.8)	1.00
Minneapolis/ATES	18.2 (57.24)	10.5 (18.9)	1.69
Birmingham/ATES	14.5 (45.71)	8.8 (15.8)	1.61

temperatures noted above, the air flow rate increases to 1,061,400 kg/hr (2,340,000 lb/hr). The resultant peak cooling load is 2.03 MWt (63.66 * 10^6 Btu/hr). Annual cooling loads were calculated based on the average ambient temperatures and relative humidities for the four location and operational capacity factor combinations. Cooling load and capacity factor data are summarized in Table 11.

Diurnal ice storage system design and performance was based on the characteristics of the Lincoln Electric system. Key elements of the ice storage system are refrigeration, storage tank, air cooling coils (common to ice and ATES systems), and circulating water pumps and piping. Ancillary equipment includes controls, electrical, and water treatment.

The refrigeration unit uses ice harvesting technology. Ice is generated by alternating freezing and shedding cycles that last about 8 minutes and 45 seconds, respectively. The refrigerant is ammonia. The overall system coefficient of performance is 2.7 (i.e., 2.7 kWh of cooling capacity is charged to storage per 1.0 kWh of electrical input). This figure includes electricity consumed by motors driving the compressor, evaporative condenser fan, water circulation pumps, and the evaporative condenser cooling water supply pump. Refrigeration system performance is enhanced by the use of 12.7°C (55°F) well water for cooling in the evaporative condenser.

The ice/water mixture is contained in a cylindrical tank constructed from cast-in-place concrete. The tank is sized for a 50/50 mixture of ice and water when fully charged, with a 5% reserve ice capacity at the end of each discharge cycle. An additional 10% freeboard volume is included at the top of the tank. The tank is externally insulated (R value = 10) to provide a thermal storage efficiency of 98%.

TABLE 11. Peak and Annual Cooling Loads/Ice Storage Systems

Location	Operational Capacity Factor	Peak Load, MWt (10^6 Btu/hr)	Annual Load, MWht/yr (10^6 Btu)	Cooling Capacity Factor
Birmingham	0.05	20.3 (63.7)	6,890 (21,940)	0.039
Birmingham	0.2	20.3 (63.7)	25,700 (80,780)	0.145
Minneapolis	0.05	20.3 (63.7)	6,620 (20,810)	0.037
Minneapolis	0.2	20.3 (63.7)	19,800 (62,300)	0.124

The diurnal ice TES system is designed to produce a constant power output regardless of ambient air temperature. However, power production increases for the non-cooled turbines as the ambient air temperature drops from the summer design point. Therefore, incremental annual energy production for the diurnal ice TES systems is less than systems with larger turbines. Incremental energy production for the diurnal ice TES system was calculated by comparing energy production with precooling to energy production with the reference 83.5 MW gas turbine operating at the average ambient temperature. Energy production data for precooled systems are summarized in Table 12.

Incremental fuel requirements for the diurnal ice TES system was also calculated by evaluating the difference between operating the 83.5 MW (ISO) turbine with and without precooling. Precooling lowers the inlet air temperature and average heat rate, but a greater increase in energy production results in an overall increase in fuel consumption, as shown in Table 13.

The relative size (and cost) of the inlet air precooling heat exchanger was calculated based on a comparison of thermal duties and log-mean temperature differences (LMTD) with the relative size and cost specified for

TABLE 12. Incremental Energy Production in Ice Storage Precooled Systems

Location	Operational Capacity Factor	Energy Production, MWh/yr
Birmingham	0.05	8,285
Birmingham	0.2	28,707
Minneapolis	0.05	7,257
Minneapolis	0.2	23,505

TABLE 13. Incremental Fuel Consumption in Ice Storage Precooled Systems

Location	Operational Capacity Factor	Fuel Consumption, MWh/yr (MMBtu/yr)
Birmingham	0.05	18,000 (56,700)
Birmingham	0.2	62,900 (197,500)
Minneapolis	0.05	15,900 (49,900)
Minneapolis	0.2	51,900 (163,200)

the Lincoln Electric System plant. Size (heat transfer area) was presumed to be proportional to thermal duty and inversely proportional to LMTD. The overall heat transfer coefficient was assumed to be constant. Relative thermal duties, LMTDs, and sizes are summarized in Table 14.

TABLE 14. Precooling Heat Exchanger Sizing

System	Thermal Duty, MWt (MMBtu/hr)	LMTD, °C (°F)	Relative Size
Lincoln Electric	13.6 (46.25)	14.3 (25.8)	1.00
Ice storage	18.7 (63.66)	14.1 (25.4)	1.40

5.0 ECONOMIC EVALUATION

The economic evaluation was conducted by calculating and comparing the levelized cost of electricity produced by the concepts being considered. Levelized cost analysis combines initial capital cost, annually recurring cost, and system performance characteristics with financial parameters to produce a single figure-of-merit (the levelized cost) that is economically correct and can be used to compare the projected energy costs of alternative peak power production concepts.

5.1 METHODOLOGY AND ASSUMPTIONS

The methodology used was that defined in Brown et al. (1987), which is consistent with the "required revenue" approach suggested by the Electric Power Research Institute (EPRI) in their Technical Assessment Guide. The specific financial assumptions used to calculate the levelized energy cost are listed in Table 15. These assumptions, taken from EPRI's Technical Assessment Guide (EPRI 1989), are intended to be representative of the electric utility industry.

TABLE 15. Financial Assumptions

Parameter	Assumption
System economic life	30 years
System depreciable life	20 years
Nominal after-tax discount rate	9.82%
General inflation rate	5.0%
Combined federal/state income tax rate	38%
Property tax and insurance rate	2%
System construction period	1 year
Price year	1990
First year of system operation	1995

Marginal capital and annual operating and maintenance (O&M) costs for the turbine were also estimated based on data presented in EPRI's Technical Assessment Guide (EPRI 1989). Incremental capital costs were estimated by subtracting the cost of the reference turbine from the cost of a turbine sized to include the incremental ISO capacity shown in Table 1. Incremental O&M costs for the turbine were based on the total ISO capacity, the incremental capital cost, and the incremental annual energy production shown in Table 4. ATES system capital and annual O&M costs were estimated using the AQUACOOL model described in Brown, Hattrup, and Watts (1991). Ice storage system capital costs, including the precooling heat exchanger common to ATES and ice storage systems, were scaled-up from estimates presented in Ebeling et al. (1992) based on the relative size of system components. Annual O&M for the precooling heat exchanger and other ice storage system components, including operating overheads, was assumed to be 10% of initial capital. Incremental natural gas requirements shown in Tables 5, 9, and 13 were assumed to cost $2.33/MMBtu and escalate at 4.3%/year in excess of general inflation (Energy Information Administration 1992). The cost of off-peak electricity used to charge the ice storage system was assumed to be $0.03/kWh, based on the average off-peak energy charges for the industrial sector presented in Brown, Garrett, and Sedgwick (1991). Electricity costs were presumed to escalate at 0.55%/year above general inflation based on forecasts prepared by the Energy Information Administration (1992).

Startup and working capital costs were estimated based on information presented in EPRI (1989). Startup costs include operator training, equipment checkout, minor changes in equipment, extra maintenance, and fuel consumption incurred after the plant is constructed, but prior to regular operation. Working capital represents a "revolving account" used to pay for current expenses and an investment in spare parts. Startup and working capital were estimated by the relationships described in Equations (3) and (4).

$$\begin{aligned}\text{Startup capital cost} = {} & 0.02 * \text{total system construction cost} + \\ & 1/12 * \text{total annual O\&M} + \\ & 1/52 * \text{total annual fuel}\end{aligned} \tag{3}$$

$$\text{Working capital cost} = 0.005 * \text{total system construction cost} + 1/6 * \text{total annual O\&M} + 1/6 * \text{total annual fuel} \quad (4)$$

5.2 RESULTS

Incremental initial capital costs, annually recurring costs, and annual energy production are shown in Tables 16, 17, and 18 for larger gas turbines, ATES precooled systems, and ice storage precooled systems. Marginal electricity costs are shown in Table 20 for all cases. The levelized cost results show that some form of precooling is likely to result in lower cost power production than simply installing a larger gas turbine for each of the four operating capacity factors and location combinations investigated. Precooling with an ATES system offered a significant advantage compared to larger turbines or ice storage precooling in Minneapolis, while in Birmingham, the preferred precooling system depended on the operating capacity factor.

Capital costs for larger gas turbine and ATES precooling systems are either constant or increase very little at higher capacity factors. This is in contrast to the ice storage precooling systems, which increase significantly in cost between 0.05 and 0.2 capacity factors. The impact is a greater percentage reduction in levelized energy cost for larger turbine and ATES precooling systems than for ice storage precooling systems when comparing 0.05 and 0.2 capacity factor results.

The larger gas turbines also benefit from the lower average operating temperature at the Minneapolis site, which increases the annual electricity production, thus lowering the levelized cost. The opposite effect occurs for the precooling systems, although this is masked in the results for the ATES precooling systems by the difference in inlet air precooling achieved in Birmingham [11.1°C (52°F)] and Minneapolis [7.2°C (45°F)].

ATES precooling system costs are sensitive to site-specific aquifer characteristics. The principal difference between the two locations is the substantial increase in productivity per well in Minneapolis [0.126 m^3/sec (2000 gpm)] compared to Birmingham [0.00095 m^3/sec (150 gpm)]. ATES

TABLE 16. Larger Turbine Cost Summary

(1000s of 1990 dollars, except for annual energy production)

Element	Birmingham Operating Capacity Factor 0.05	Birmingham Operating Capacity Factor 0.2	Minneapolis Operating Capacity Factor 0.05	Minneapolis Operating Capacity Factor 0.2
System capital	6,000	6,000	6,851	6,851
Fixed O&M	12	12	14	14
Variable O&M	56	232	66	273
Fuel	234	954	273	1,117
Startup capital	130	159	148	181
Working capital	80	230	91	266
Annual energy, MWh	8,055	33,349	9,532	39,764

TABLE 17. ATES Precooling Cost Summary

(1000s of 1990 dollars, except for annual energy production)

Element	Birmingham Operating Capacity Factor 0.05	Birmingham Operating Capacity Factor 0.2	Minneapolis Operating Capacity Factor 0.05	Minneapolis Operating Capacity Factor 0.2
System capital	4,050	4,511	2,464	3,002
Total O&M	445	497	259	339
Fuel	100	330	102	325
Startup capital	120	138	73	95
Working capital	111	160	73	126
Annual energy, MWh[a]	6,193	20,534	6,092	19,328

(a) On-peak pumping power subtracted from gross energy output shown in Table 8.

TABLE 18. Ice Storage Precooling Cost Summary
(1000s of 1990 dollars, except for annual energy production)

Element	Birmingham Operating Capacity Factor 0.05	Birmingham Operating Capacity Factor 0.2	Minneapolis Operating Capacity Factor 0.05	Minneapolis Operating Capacity Factor 0.2
System capital	3,666	6,394	3,666	6,394
Total O&M	367	639	366	639
Fuel	132	460	116	380
Electricity	71	263	68	226
Startup capital	108	195	107	193
Working capital	113	259	110	239
Annual Energy, MWh[a]	8,150	28,211	7,129	23,079

(a) On-peak pumping power subtracted from gross energy output shown in Table 12.

TABLE 19. Marginal Levelized Energy Costs

System Type	Birmingham Operating Capacity Factor 0.05	Birmingham Operating Capacity Factor 0.2	Minneapolis Operating Capacity Factor 0.05	Minneapolis Operating Capacity Factor 0.2
Larger turbine	0.142	0.086	0.138	0.084
ATES precooling	0.170	0.080	0.118	0.069
Ice storage precooling	0.133	0.090	0.147	0.101

TABLE 20. Ice Storage Precooling Levelized Cost Results

System Type	Birmingham Operating Capacity Factor 0.05	Birmingham Operating Capacity Factor 0.2	Minneapolis Operating Capacity Factor 0.05	Minneapolis Operating Capacity Factor 0.2
5 days/week (reference)	0.133	0.090	0.147	0.101
7 days/week (sensitivity)	0.127	0.083	0.139	0.092

precooling would be preferred at the lower capacity factor if well productivity at a site was about 0.0315 m^3 (500 gpm) or higher. ATES precooling system capital costs are too high in Birmingham to be cost effective at the lower capacity factor.

The larger turbine systems all have higher capital costs than either of the two precooling systems for corresponding cases. However, marginal energy production is greater for the larger turbine systems because the marginal power production is greater for all ambient temperatures less than the 37.8°C (100°F) summer design point. The difference in energy production is greatest at an annual operating factor of 0.2, because the average ambient temperature is lower for the longer operating period.

For both Minneapolis and Birmingham, the ATES system is most attractive at the higher capacity factor. This result can be attributed to the annually recurring costs, which are mostly variable for the larger gas turbine, but primarily fixed for the ATES system. While fuel costs are certainly variable with production and experience has demonstrated that the majority of gas turbine O&M costs are variable with production, further experience with ATES systems is required to determine fixed and variable O&M portions.

The results described above for ice storage precooling were based on charging and discharging every day of the week (diurnal mode) per the reference case operating schedule detailed in Table 3. System designs based on charging 7 days/week, but discharging only 5 days per week (diurnal/weekly mode) were evaluated as a sensitivity case. The results (see Table 20) indicate a cost advantage of 5 to 8% for the diurnal/weekly mode, which would not change the preferred technology for each city and capacity factor combination. In addition, the sensitivity results do not reflect the reduction in average air temperature that would occur in the diurnal/weekly mode, which would reduce or eliminate the cost advantage.

6.0 CONCLUSIONS

Precooling gas turbine inlet air with cold water supplied by an ATES system provided lower cost electricity (levelized $/kWh) than simply increasing the size of the turbine for meteorological and geological conditions existing in the Minneapolis vicinity. A significant (15 to 20%) cost advantage resulted for both 0.05 and 0.2 annual operating factors. In contrast, ice storage precooling was found to be significantly (5 to 20%) more expensive than larger gas turbines in Minneapolis.

In Birmingham, ATES precooling was preferred at the higher capacity factor and ice storage precooling was the best option at the lower capacity factor. In both cases the levelized cost advantage compared to larger gas turbines was estimated to be about 5%. The levelized energy cost advantage shown for ATES at the Minneapolis site was reduced or eliminated (depending on capacity factor) at the Birmingham site principally because of much lower well flow rates in the Birmingham vicinity. The ice storage precooling system benefitted from higher average annual operating temperatures in Birmingham, which resulted in an increase in incremental annual power production for this site not experienced by the other systems.

Incremental capital costs were 25 to 65% lower per peak kW for the ATES precooling systems and 15 to 50% lower per peak kW for the ice storage precooling systems, but incremental energy production was generally greater for the larger turbine systems.

These preliminary results indicate that ATES and ice storage precooling systems should be considered as options for increasing peak generating capacity of combustion turbines. The preferred system will depend on site-specific conditions and operating requirements. Like all ATES applications, its cost effectiveness will vary significantly with site-specific geologic conditions. Generally, ice storage systems will look most attractive in low capacity factor applications, while ATES systems will look best at higher capacity factors.

Further experience with ATES and ice storage precooling systems will be needed to reduce the cost uncertainty associated with these systems and allow more conclusive comparative assessments. In addition, other inlet air precooling options should be considered, such as surface-engineered seasonal chill storage systems using ice or artificial snow.

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