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AN EVALUATION OF THERMAL STORAGE OPTIONS FOR COMBUSTION TURBINE INLET AIR COOLING

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ABSTRACT

A transient simulation model was written to describe the performance of combustion turbine inlet air cooling systems based on a combination of chilled water and ice as thermal storage media. The transient simulation model was used in conjunction with simpler computer models to design inlet air cooling systems for four different power plant daily load profiles based on three alternative storage options: chilled water alone, ice alone, and an optimized combination of the two media. Alternative storage options were evaluated on the bases of first cost and a life cycle analysis of each system.

Both the capacity enhancement cost and the incremental electrical energy production cost were found to be lowest for cooling systems based on chilled water storage alone. Systems based either partially or entirely on ice storage lead to greater power plant generating capacity increases than do systems based on chilled water alone. Systems based on the optimized combination of thermal storage media are significantly less expensive than are systems based purely on ice storage.

NOMENCLATURE

ADB	Ambient dry bulb temperature
AWB	Ambient wet bulb temperature
CEC	Capacity enhancement cost
C_F	Cost of fuel
C_{OP}	Cost of off-peak electrical energy
C_P	Cost of incremental electrical energy produced with inlet air cooling
C_{sys}	Installed cost of cooling system
$C_{sys;w}$	Installed cost of cooling system based on chilled water storage alone
d	Discount rate
ΔE	Incremental electrical energy produced with inlet air cooling
E_{OP}	Annual electrical energy consumption of cooling system
ΔF	Incremental fuel consumed due to inlet air cooling
LDB	Leaving dry bulb temperature
i	Inflation rate
MCEC	Marginal capacity enhancement cost
N_P	Discounted payback period
ΔP_{sys}	Net power plant generating capacity increase
$\Delta P_{sys;w}$	Net power plant generating capacity increase due to system based on chilled water storage alone
PWF	Present worth factor
S_{ann}	Average annual savings due to use of inlet air cooling system
WMFR	Water mass flow rate

I. INTRODUCTION

Both the generating capacity and conversion efficiency of combustion turbines tend to decrease as the ambient dry bulb temperature increases. Since the volumetric flow rate at the compressor stage inlet is approximately constant over a wide range of inlet temperatures, an increase in the ambient dry bulb temperature reduces the air mass flow rate and thus reduces the generating capacity. A smaller air mass flow rate also decreases the pressure ratio across the turbine stage, leading to a reduction in the conversion efficiency (Kitchen 1994). Since many combustion turbine power plants are used most intensively during the summer when ambient dry bulb temperatures are high, utilities have sought ways to cool the compressor inlet air in order to maintain generating capacity and (to a lesser extent) to maintain efficiency.

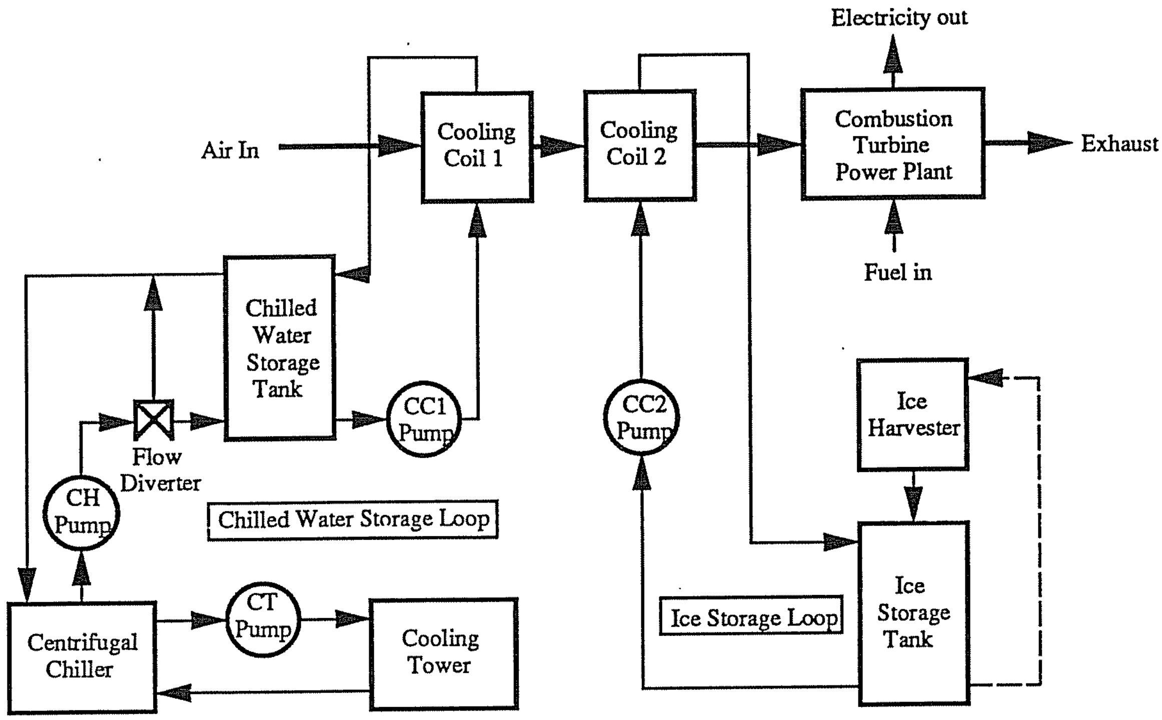

FIGURE 1: COMBUSTION TURBINE INLET AIR COOLING SYSTEM SCHEMATIC

Approaches taken to date include the installation of evaporative coolers, on-line chillers, and ice generating equipment to charge a thermal storage tank during off-peak hours in order to cool air before it enters the compressor stage. Although evaporative coolers are relatively inexpensive to install and operate, they are limited by a finite approach to the ambient wet bulb temperature. Evaporative coolers thus function particularly poorly in humid areas. On-line chillers can provide significantly lower inlet air temperatures than can evaporative coolers, but cost between $400 and $600 per additional kilowatt of power plant generating capacity at design weather conditions. These costs are only marginally lower than the unit cost for the combustion turbine itself. The required refrigeration capacity of ice generating equipment is far lower than that for an on-line chiller designed to meet the same air cooling load, since ice can be produced and stored overnight and over the weekend, and subsequently melted by circulating water exposed directly or indirectly to the inlet air flow stream when the combustion turbine is in use. The unit costs for power plant capacity enhancement associated with ice storage are thus significantly lower than those for on-line chillers (Ebeling 1994a).

J. Andrepont (1994) suggested the use of chilled water as a thermal storage medium for combustion turbine inlet air cooling. Chillers are much less expensive than ice harvesters for a given refrigeration capacity, and therefore are capable of reducing inlet air cooling system costs below those achievable with ice storage. The main disadvantage associated with the use of chilled water thermal storage is that for ambient dry bulb temperatures of 32° - 38° C (90° - 100° F), the lower limit for the combustion turbine entering dry bulb temperature is approximately 8° C (46° F). This lower limit is higher than the desired inlet air temperature for many combustion turbines. The capacity increase for such turbines would thus be lower with a chilled water storage based cooling system than with an ice storage based cooling system, which can easily provide inlet air temperatures of 4.4° C (40° F).

A third thermal storage option is to use a combination of chilled water and ice to meet the inlet air cooling load. A cooling system based on both chilled water and ice storage will generally be less expensive than one based on ice storage alone, yet can provide the same compressor stage inlet dry bulb temperature and power plant capacity enhancement as the ice based system. The purpose of this paper is to compare the three thermal storage options discussed above for use in inlet air cooling systems designed for a single shaft combustion turbine power plant operated in the upper mid-Western United States. The cooling systems are compared on the bases of both first costs and life-cycle benefit to the owner of the power plant.

II. DESCRIPTION OF INLET AIR COOLING SYSTEM MODELS

A computer model of the combustion turbine inlet air cooling system shown in Figure 1 was written using the TRNSYS simulation program (Klein et al. 1994). The system consists of a chilled water storage loop and an ice storage loop. Air drawn into the combustion turbine passes through two separate heat exchangers: a cooling coil fed by water being circulated through the chilled water storage tank, and a cooling coil fed by water circulating through the ice storage tank. Either storage loop can be disabled by making minor changes to the simulation deck, making it possible to model cooling systems based on chilled water storage alone or ice storage alone, in addition to the combination of storage media.

The combustion turbine operates between four and eight hours per day, five days per week. Four different power plant load profiles are considered, as described in the following section. The volumetric flow rate at the compressor stage inlet is 246 m^3/sec (520,300 ft^3/min). The minimum allowable inlet dry bulb temperature is 4.4° C (40° F). Inlet temperatures lower than this value can lead to ice formation at the compressor inlet, which can damage the compressor blades. For the inlet configurations and operating conditions considered, the generating capacity increases nearly linearly from 78.6 MW for an entering dry bulb temperature of 35° C (95° F) to approximately 92.6 MW for an entering dry bulb temperature of 4.4° C. The TRNSYS model simulates the performance of the combustion turbine both with and without inlet air cooling simultaneously for purposes of comparison.

The chilled water storage loop operates on the basis of a daily full storage strategy. The chiller and cooling tower operate up to fifteen hours per day, five days per week, while the combustion turbine is not in use. When completely charged, the temperature of the water in the storage tank is 4.4° C. The ice storage loop operates on the basis of a weekly full storage strategy. The ice harvester, which rejects heat to the environment via an evaporative condenser unit included in the ice harvester model, operates up to fifteen hours per weekday and 24 hours per day on the weekend. As long as the ice storage tank is filled to at least 20% of its full capacity, return water from the cooling coil entering at the top of the tank will exit at a temperature of 0° C (32° F) (Stewart 1994). Both the chiller and ice harvester use ammonia as the refrigerant.

The air velocity at the first cooling coil face is 2.03 m/sec (400 ft/min). The maximum water velocity through the cooling coil tubes is 3.05 m/sec (10 ft/sec). The water flow rate through both cooling coils can be modulated by a controller (not shown in Figure 1) to meet the varying air cooling load. Both cooling coils have 0.330 mm (0.013 in) thick straight rectangular fins spaced 3.18 mm (0.125 in) apart. The outer tube diameter is 10.2 mm (0.402 in); the tube wall thickness is 0.89 mm (0.035 in). The tubes are made of stainless steel; the fins are made of aluminum The use of copper was avoided to minimize the corrosion hazard associated with an ammonia leak.

The TRNSYS model generates detailed output files for use in system design and analysis. Key water and air temperatures and mass flow rates are recorded at ten minute intervals. Internal energy changes, refrigeration equipment energy consumption, power plant fuel consumption, and power plant energy production are totaled at the end of the simulation. The TRNSYS model also calculates cooling system component group costs based on the capacities and dimensions of the components comprising each group. Equations relating the cost of the chiller, the cooling tower, the storage tanks, the pumps, and the pipes to their respective sizes are derived from information in the 1992 *Means Facilities Cost Data* Catalog (Waier et al. 1992). Equations for the cost of custom ordered cooling coils and the ice harvester are based on information provided by J. Ebeling (1994b) and one of the co-authors of this paper, who was formerly involved in the development of ice harvesting technology at the Paul Mueller Company. Component group costs are recorded at the end of the simulation together with two additional economic parameters discussed in section IV.

Five additional computer models were written to provide "guess values" for component sizes in the TRNSYS simulation using the simultaneous equation solver EES (Klein and Alvarado 1994). The EES models are based on a restricted set of design conditions that do not fully reflect the time dependent nature of the air cooling loads and the power plant load profiles. One of the EES models describes the performance of the combustion turbine, two describe the performance of the chilled water storage loop for alternative storage options, and two describe the performance of the ice storage loop for alternative storage options. The EES and TRNSYS models were used interactively in the cooling system design process to ensure that the desired cooling coil loads and power plant outputs were met at each five minute TRNSYS time step for a "design week". Design weather conditions and power plant load profiles are discussed in the following section. Further details concerning the computer models used are provided in Cross (1994) and Cross et al. (1994).

III. DESIGN CONDITIONS FOR COOLING SYSTEM AND POWER PLANT OPERATION

Four different power plant daily load profiles are considered as represented in Figure 2. These include four, six, and eight hours of full capacity electric power generation with a 4.4° C inlet dry bulb temperature and an eight hour symmetrically peaked profile. The symmetrically peaked profile increases linearly from the power plant capacity at design weather conditions to the full power plant capacity with 4.4° C inlet air, and then decreases linearly back to the original value. The peaked load profile results in approximately the same amount of incremental electrical energy produced with inlet air cooling as the four hour step profile. All load profiles are centered around 5:00 p.m. daylight saving time, the hour at which the ambient dry bulb temperature attains its maximum value. Cooling systems based on the three storage options discussed above were designed for each power plant load profile. Since systems based on chilled water storage alone cannot achieve inlet air temperatures of 4.4° C, the corresponding normalized daily power plant load profiles are "clipped" as shown in Figure 2.

Each of the twelve inlet cooling systems considered was designed on the basis of a "design week", which is composed of seven days characterized by the dry and wet bulb temperature profiles, ADB and AWB respectively, shown in Figure 3. The maximum dry bulb temperature is 35° C (95° F); the coincident wet bulb temperature is 24.4° C (77° F). The ambient pressure is assumed constant at 99.3 kPa (14.4 lb/in^2). The relative humidity corresponding to these atmospheric conditions is 43%. The maximum dry bulb temperature occurs at 5:00 p.m., which is also the time of maximum electrical demand for the symmetrically peaked power plant load profile. As indicated in Figure 2, the cooling coils operate up to eight hours per day, between 1:00 and 9:00 p.m. For systems that lower the dry and wet bulb temperatures of the inlet air stream to 4.4° C, the cooling load varies by up to 6% from its average value of 55.4 kJ/kg (23.8 Btu/lb). The TRNSYS model adjusts the cooling coil water mass flow rates to meet the cooling coil load required to maintain the power plant electric output at its desired value at each simulation time step.

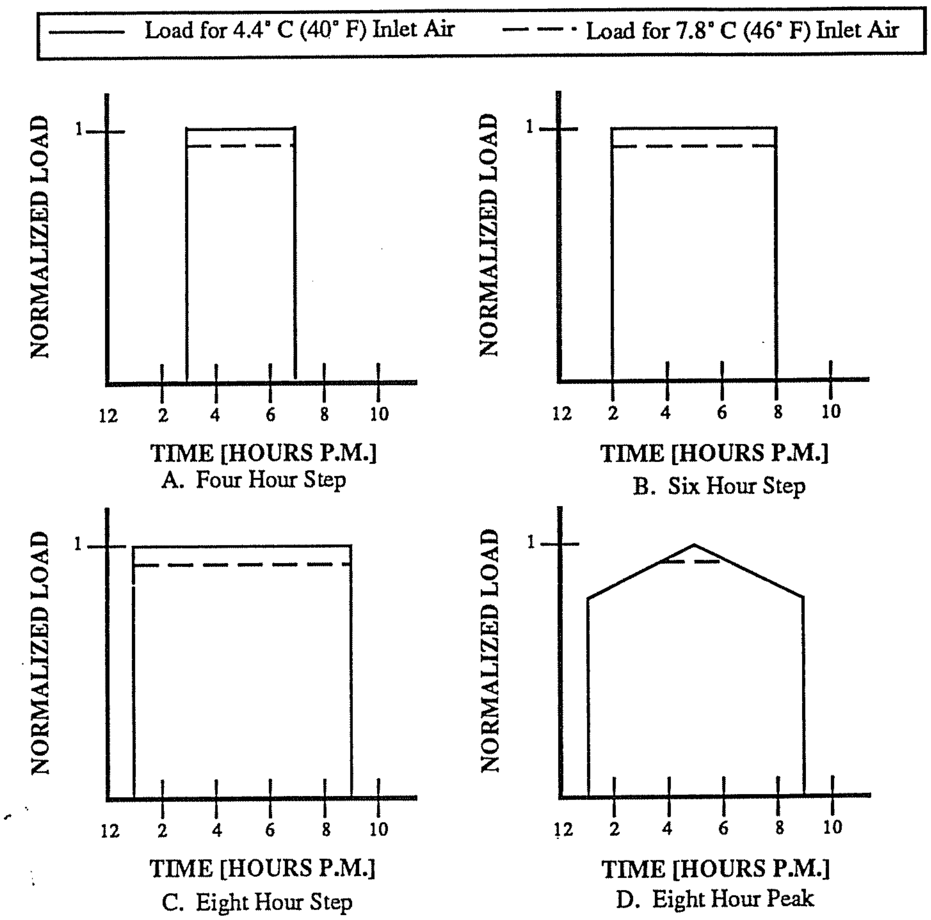

FIGURE 2: NORMALIZED DAILY POWER PLANT LOAD PROFILES

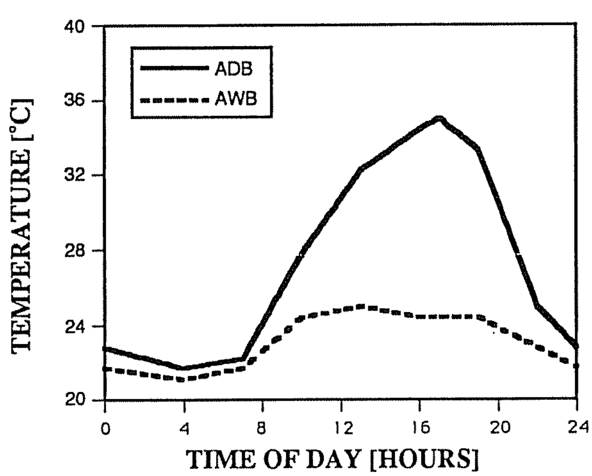

FIGURE 3: DESIGN DAY AMBIENT DRY AND WET BULB TEMPERATURE PROFILES

IV. MEASURES FOR THE COMPARISON OF ALTERNATIVE STORAGE OPTIONS

Inlet air cooling systems based on each of the three alternative storage options are compared on the bases of first cost and the anticipated life cycle benefit to the owner. Two measures are used to compare first cost: the capacity enhancement cost and the marginal capacity enhancement cost. The anticipated life cycle benefit is measured by the cost of the incremental electrical energy produced due to inlet air cooling based on a 20 year cooling system payback period.

The capacity enhancement cost, CEC, is defined as the installed cost of the cooling system in U.S. dollars, C_{sys}, divided by the resulting net power plant generating capacity increase at design weather conditions in kilowatts, ΔP_{sys}:

$$CEC = \frac{C_{sys}}{\Delta P_{sys}} \tag{1}$$

The marginal capacity enhancement cost is used to compare systems based either partially or entirely on ice storage to systems

TABLE 1: COOLING COIL DESIGNS USED TO DETERMINE OPTIMUM CAPACITY SPLIT FOR HYBRID COOLING SYSTEMS

First Cooling Coil			Second Cooling Coil			System
Number of Rows	WMFR [kg/sec]	LDB [° C]	Number of Rows	WMFR [kg/sec]	LDB [° C]	CEC [$/kW]
6	426	11.6	4	353	4.4	222
7	544	10.1	3	530	4.4	223
8	461	9.2	3	306	4.4	215
9	**447**	**8.4**	**3**	**185**	**4.4**	**211**
10	455	7.8	2	706	4.4	217

based on chilled water storage alone. The marginal capacity enhancement cost, MCEC, is defined in terms of C_{sys}, ΔP_{sys}, the installed cost of the system designed on the basis of chilled water storage alone for the same power plant load profile, $C_{sys;w}$, and the power plant capacity increase in kilowatts for the latter system, $\Delta P_{sys;w}$ as:

$$MCEC = \frac{C_{sys} - C_{sys;w}}{\Delta P_{sys} - \Delta P_{sys;w}} \qquad (2)$$

Both the capacity enhancement cost and the marginal capacity enhancement cost should be compared to the unit cost for installing a second combustion turbine to decide whether the cooling system is a better investment than another combustion turbine.

The cost of the incremental electric power produced due to inlet air cooling is derived from a life cycle analysis of system performance and the discounted payback period concept discussed by Duffie and Beckman (1991). The discounted payback period, Np, is defined as the number of years required for the discounted sum of the annual savings associated with a given project to equal the initial investment. The discounted sum of the annual savings is determined by multiplying the average annual saving by the "present worth factor", PWF, which depends on the discount rate, d, the inflation rate, i, and the payback period as follows:

$$PWF = \left(\frac{1}{d - i}\right)*\left[1 - \left(\frac{1+i}{1+d}\right)^{Np}\right] \qquad (i \neq d) \qquad (3a)$$

$$PWF = \frac{Np}{1 - i} \qquad (i = d) \qquad (3b)$$

The defining equation for Np can thus be written as:

$$C_{sys} = PWF*S_{ann} \qquad (4)$$

where S_{ann} is the average annual saving resulting from use of the inlet air cooling system. Based on the 1995 forecasted use of the combustion turbine during the cooling season and the assumption of 2.5% annual load growth over the next 20 years, the average annual number of hours during which inlet air cooling will be required is 32. This cooling requirement is the same for all power plant daily load profiles. Assuming further that a demand exists for all the incremental electrical energy in kilowatt-hours, ΔE, that is produced due to inlet air cooling, and that the power plant owner always has the option of purchasing this amount of electrical energy from another utility at a peak wholesale price, Cp, in dollars per kilowatt-hour, the average annual saving associated with the use of the inlet air cooling system is:

$$S_{ann} = Cp*\Delta E - C_F*\Delta F - C_{OP}*E_{OP} \qquad (5)$$

Here ΔF is the average annual incremental amount of fuel in kilograms used by the combustion turbine due to inlet air cooling, C_F is the cost of the fuel in dollars per kilogram, E_{OP} is the average annual amount of off-peak electrical energy in kilowatt-hours required by the cooling system, and C_{OP} is the cost of off-peak electrical energy in dollars per kilowatt-hour.

Solving Equations (4) and (5) for Cp, which is identified with the cost of the incremental electrical energy produced with inlet air cooling based on the discounted payback period Np, yields:

$$Cp = \frac{1}{\Delta E}*\left(\frac{C_{sys}}{PWF} + C_F*\Delta F + C_{OP}*E_{OP}\right) \qquad (6)$$

The payback period is specified to be 20 years. Other economic parameters include a discount rate of 10.17%, a fuel inflation rate of 5.50%, a fuel cost of $0.128/kg ($0.058/lb), and an off-peak electricity cost of $0.0124/kW-hr. The present worth factor is 12.41. The quantities ΔE, ΔF, and E_{OP} result from an annual simulation for each inlet air cooling system. Since the combustion turbine operates for a relatively short period during the average cooling season, design temperature profiles were used for the annual simulations. Although the use of design weather conditions slightly over predicts the life cycle benefit associated with inlet air cooling, it enables comparisons between different storage options to be made on an equal basis.

V. COOLING SYSTEM DESIGN AND SIMULATION RESULTS

The optimum capacity split between chilled water and ice storage for cooling systems based on both storage media is expressed as the leaving dry bulb temperature from the first cooling coil that results in the lowest capacity enhancement cost. The leaving dry bulb temperature from the second cooling coil is always 4.4° C, the minimum allowable compressor stage inlet air temperature. The optimum capacity split for the weather conditions considered is 8.4° C (47.2° F), based on varying the number of cooling coil rows and water mass flow rates, WMFR, for the four hour step power plant load profile as shown in Table 1. Table 1 indicates that this optimum is not a strong function of

TABLE 2: COOLING SYSTEM COSTS

Storage Option	Load Profile	C_{sys} [$]	ΔP_{sys} [kW]	ΔE [kW-hr/yr]	CEC [$/kW]	MCEC [$/kW]	Cp [$/kW-hr]
Water	4 hr step	2,165,000	12,578	380,000	172	---	0.49
Ice	4 hr step	3,655,000	14,051	438,000	260	1,012	0.70
Water/Ice	4 hr step	2,948,000	13,964	435,000	211	565	0.58
Water	6 hr step	3,204,000	12,578	383,000	255	---	0.70
Ice	6 hr step	5,030,000	14,051	429,000	358	1,240	0.97
Water/Ice	6 hr step	4,178,000	13,964	428,000	299	703	0.81
Water	8 hr step	4,848,000	12,578	371,000	385	---	1.08
Ice	8 hr step	6,386,000	14,051	418,000	454	1,044	1.26
Water/Ice	8 hr step	5,973,000	13,964	415,000	428	812	1.19
Water	8 hr peaked	2,652,000	12,578	197,000	211	---	1.12
Ice	8 hr peaked	4,958,000	14,051	201,000	353	1,566	2.02
Water/Ice	8 hr peaked	3,160,000	13,964	201,000	226	366	1.30

the number of rows in the first cooling coil. The optimum cooling coil design is highlighted, and features a total of twelve rows. The optimum number of rows for systems based on chilled water storage alone and ice storage alone were not determined in a similar manner, but were rather taken from papers by J. Ebeling and his co-workers (1994) and J. Andrepont (1994). The number of cooling coil rows for these storage options is ten in each case.

Component group cost data for the three alternative storage options are shown in Figure 4 for the four hour step power plant load profile. The total installed cost for the cooling system based on chilled water storage alone is less than the installed cost for the cooling systems based either partially or entirely on ice storage. This cost differential results primarily from the substantially higher unit refrigeration equipment costs for the second two storage options. However, the second two storage options result in lower compressor stage inlet air temperatures and hence greater power plant capacity increases. Pipe and pump costs represent approximately 8% of the total installed cost for the system based on both chilled water and ice storage. The pipe runs between the storage tanks and the cooling coils are 91 meters (300 ft) long. Increasing the length of these pipe runs would penalize the hybrid system more heavily than systems based on only one storage medium for the plumbing lay-out shown in Figure 1.

The three alternative thermal storage options are compared on the basis of first costs for the four daily power plant load profiles in the sixth and seventh columns of Table 2. Although the compressor stage inlet dry bulb temperature is the same for systems based either partially or entirely on ice storage, the net generating capacity increase is somewhat lower for hybrid storage systems due to the slightly higher parasitic cooling coil pump power requirement. Cooling systems based on chilled water storage alone yield the lowest capacity enhancement costs for all power plant load profiles; cooling systems based on ice storage alone yield the highest capacity enhancement costs. The marginal capacity enhancement costs for systems based on ice storage alone are higher than the unit cost of approximately $700/kW (Brown et al. 1994) for purchasing a second combustion turbine. However, the marginal capacity enhancement costs for systems based on both storage media are roughly equivalent to or lower than combustion turbine unit costs, depending on the power plant load profile. The marginal capacity enhancement cost for the hybrid storage based system is particularly low for the eight hour peaked power plant load profile, because the cooling coil fed by the ice storage tank only needs to be activated for about one hour per day in that case. If the marginal capacity enhancement associated with ice storage is required, the hybrid storage option is more attractive from an economic perspective than the pure ice storage option for all power plant load profiles.

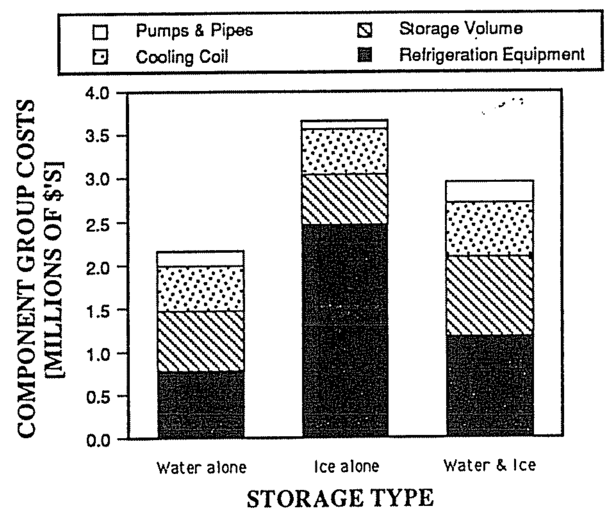

FIGURE 4: COMPONENT GROUP COSTS FOR FOUR HOUR STEP LOAD PROFILE

Annual simulation results for the twelve inlet air cooling systems are also presented in Table 2. The cost of the incremental electrical energy produced with inlet air cooling based on a 20 year system payback period and the other economic parameters discussed in section IV is shown in the eighth column. As in the case of the capacity enhancement cost, Cp is lowest for cooling systems based on chilled water storage alone and highest for systems based on ice storage alone for all power plant load profiles considered. On-peak wholesale electricity prices for the upper mid-Western United States range between $0.075 and $0.150 per kilowatt hour. Since incremental production costs range between $0.49 and $2.02 per kilowatt-hour, investment in an inlet air cooling system does not appear to be justified as long as power is always available from another utility in the price range quoted, regardless of storage option. The cost of the incremental energy produced with inlet cooling decreases linearly as the required duration of cooling system use increases. Cooling system use would thus have to increase by a factor of 3.2 to 7.5 (to 100 to 240 hours annually) in order to reduce Cp to $0.15/kilowatt-hour for the most cost effective storage option.

VI. CONCLUSIONS

For the combustion turbine, design weather conditions, and operating strategies considered, inlet air cooling systems based on chilled water storage alone offer lower capacity enhancement costs and incremental energy production costs than systems based on either ice storage or a combination of the two media. Inclusion of an ice storage component offers an 11% greater generating capacity increase than is available with chilled water storage alone, however. The cost of this marginal capacity enhancement is significantly lower than the marginal capacity enhancement cost provided by cooling systems based on ice storage alone. Depending on the power plant daily load profile, the marginal enhancement capacity cost associated with hybrid thermal storage is 23% to 78% of the marginal capacity enhancement cost associated with pure ice thermal storage.

Before making the decision to install any inlet air cooling system, it is important to perform a life cycle analysis to ensure that the required investment is justified. The concept of an incremental energy production cost based on a specified system payback period is a useful measure for evaluating the potential life cycle benefit for a given cooling system. The least expensive system considered here would have to operate at least 100 hours per year with the combustion turbine in order to generate on-peak electricity more cost effectively than electricity could be purchased from another utility.

Enough variation exists between different combustion turbines, sites, and usage patterns that different conclusions regarding the suitability of the three alternative storage options may be reached in different cases. For example, despite the fact that systems based on ice storage are more expensive than systems based on both storage media, they require somewhat less storage volume (32% less for the step power plant load profiles considered) than do hybrid systems. For extremely space limited sites, pure ice storage might therefore be the most attractive option. Other factors that could influence the choice of storage option include: the minimum allowable compressor stage inlet air temperature, the slope of the combustion turbine generating capacity as a function of inlet air temperature, design weather conditions, anticipated maintenance costs, required pipe lengths, and costs associated with installing another combustion turbine. Each decision concerning the optimum storage option will have to be made on a case by case basis. The models and concepts developed here are useful tools for making such a decision.

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