MODELING OF HYBRID COMBUSTION TURBINE INLET AIR COOLING SYSTEMS

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ABSTRACT

Thermal energy storage has been proposed as a viable cost-effective capacity enhancement method when applied to cool inlet air for combustion turbines. Emphasis has been placed on the use of ice-only thermal energy storage systems for inlet air cooling. In this paper, the efficacy and cost-effectiveness of ice storage compared with chilled water storage, hybrid (ice/chilled water) storage, and evaporative cooling are explored.

Detailed mechanistic models of a combustion gas turbine, ice harvester, chiller, and associated storage components are developed and calibrated using manufacturer's data. The performance of different systems with a series of four power plant load profiles is determined by simulation. Appropriate storage systems are sized to provide inlet air cooling for a combustion turbine dispatched to meet the load profiles. Capacity enhancement costs are determined for chilled water storage, ice storage, and the hybrid storage systems providing inlet air cooling for the combustion turbine.

INTRODUCTION

Many utilities in the United States establish their peak demand during the cooling season in the summer months. These "summer peaking" utilities typically reach their peak demand during mid to late afternoon (a time corresponding to peak building cooling demands). A dilemma that arises in dispatching combustion turbines to meet this peak power demand centers around the fact that the performance of the turbine decreases as the inlet air temperature (ambient temperature) increases. Figure 1 illustrates the variation in production capacity as a function of the air inlet temperature (Cross 1994) for a specific stationary combustion turbine. Note that other turbines, e.g., aeroderivatives, will have a different performance characteristic.

It is clear from Figure 1 that when the peak loads are highest and power production is needed most, the performance of the combustion turbine is at its worst. Several methods for cooling the combustion turbine's inlet air temperature have been employed to compensate for the degradation in combustion turbine performance at high ambient conditions. These include evaporative cooling, direct cooling with electric chillers, direct cooling with absorption chillers fired using exhaust heat, and thermal energy storage. Evaporative coolers typically do not significantly reduce the inlet air temperature since the process is limited by the local wet-bulb temperature and provides only a marginal increase in turbine capacity. The use of direct electric chillers is not attractive because of the large parasitic power requirement during on-peak operation. The use of indirect-fired absorption technology is possible, but it is usually cost prohibitive. The use of thermal energy storage, charged during off-peak periods, has emerged as a strong candidate to yield the greatest increase in capacity at a very competitive marginal cost.

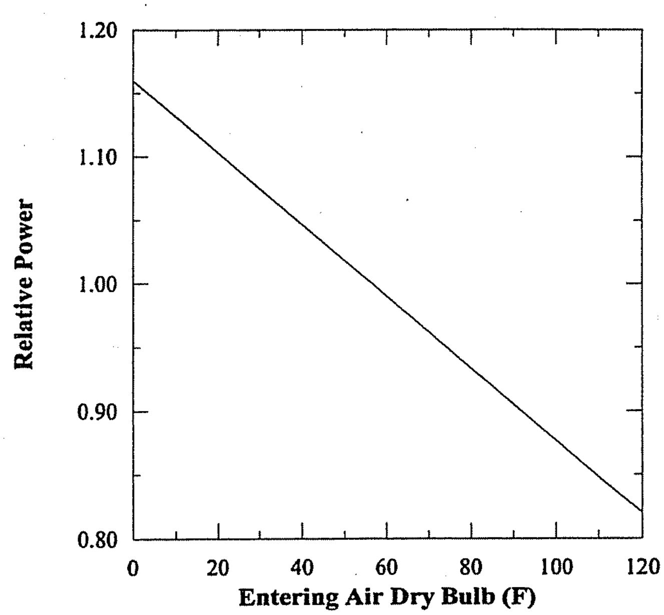

Figure 1 *Typical stationary combustion turbine performance over a range of inlet (ambient) dry-bulb temperatures. (Figure does not apply to derivative combustion turbines.)*

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A utility scale application of thermal energy storage for inlet air cooling was first demonstrated in 1991 at the Rokeby Power Station in Lincoln, Nebraska (EPRI 1993). The Rokeby Power Station is a single combustion turbine. With 100°F inlet air temperature, the capacity of the turbine is derated to 53.1 MW. Inlet cooling is provided by a tube ice thermal storage system operating on a weekly cycle. The dynamic ice harvesting system has approximately 550 tons of ice-making capacity. The inlet cooling system was designed to provide 40°F inlet air temperature, which results in a 26% increase in turbine capacity to 67.1 MW (Ebeling et al. 1992).

A second utility scale application using ice storage for inlet air cooling was installed and became operational in 1993 at the Butler Warner Generation Station in Fayetteville, NC. The Butler Warner site has eight combustion turbines (three of which are equipped with heat recovery steam generators). Similar to the Lincoln site, the Butler Warner site uses cooling coils to reduce the combustion turbine inlet air temperature to 40°F. The storage system uses plate-type ice harvesters operating on a weekly cycle. Inlet cooling provided an increase in combustion turbine capacity in excess of 26% (from 100°F inlet conditions) at the Butler Warner site (Ebeling et al. 1994a).

Although the two utility scale applications that have been installed to date use ice as a thermal storage medium, Andrepont (1994) analyzed the use of chilled water as a storage medium for inlet air cooling. An attractive feature of the chilled water storage system is the relatively low capital cost for the chilling equipment compared to the cost of the dynamic ice harvesting equipment. The primary disadvantage of using chilled water storage for inlet air cooling is that for ambient dry-bulb temperatures of 80°- to 90°F, the practically feasible lower limit of inlet air temperature is approximately 46°F. The net result is that turbine inlet air cooling systems based on chilled water storage will not yield as great a capacity enhancement as ice-based storage systems.

The authors propose the use of a hybrid (ice and chilled water) thermal storage system in an attempt to optimize the performance of a combustion turbine inlet cooling system. The benefit of a hybrid system is that the lower marginal cost system (chilled water) can be sized to meet the larger share of the inlet air cooling load while a smaller ice storage system allows 40°F compressor stage inlet air to be achieved yielding the greatest turbine capacity enhancement.

This paper attempts to quantify the benefit, if any, to pursuing inlet air cooling system designs based on hybrid thermal storage strategies. The hybrid system will be compared to both chilled water-alone and ice storage-alone storage systems. In addition to exploring the performance of ice and chilled water storage systems for inlet air cooling, this paper will investigate the impact that evaporative coolers have on inlet air cooling strategies for field retrofit applications.

COMBUSTION TURBINE PERFORMANCE

The performance (capacity and heat rate) of a combustion turbine depends on the amount of mass that passes through the turbine. As the combustion turbine inlet air temperature increases, air density decreases, the mass flow rate through the turbine decreases, and the capacity of the turbine decreases, as illustrated in Figure 1. A lower limit of air temperature leaving the coil (entering the turbine) established by many manufacturers is 36°F, with 40°F being the design limit used at both the Rokeby and Butler Warner stations. The minimum inlet air temperature is established to avoid icing as the flow accelerates into the turbine bell mouth.

In addition to capacity derating at elevated ambient temperatures, the combustion turbine's heat rate (ratio of fuel energy input to power produced) increases as well. Figure 2 shows the variation in turbine efficiency (inverse of heat rate) as a function of inlet air temperature for a range of turbine part-load ratios (PLR).

There are three other parameters that influence the combustion turbine capacity and efficiency—inlet pressure drop, exhaust pressure drop, and injected water mass flow rate. The addition of cooling coils, evaporative condensers, etc., creates inlet pressure drops that decrease both the turbine capacity and efficiency, as shown in Figure 3. The effect of outlet pressure drops on performance is similar to that at the inlet.

Water is often injected into combustion turbines and serves two purposes—emission (NO_x) control and power augmentation. The impact of water injection on the combustion turbine capacity and efficiency is given in Figure 4.

The model for the combustion turbine is based on performance data for full and part-load operating characteristics (Figures 1 and 2). Curve fits were developed to yield the turbine capacity and efficiency. These were then corrected for the following: inlet air temperature, inlet air pressure drop, outlet air pressure drop, and water injection flow rate. The pressure

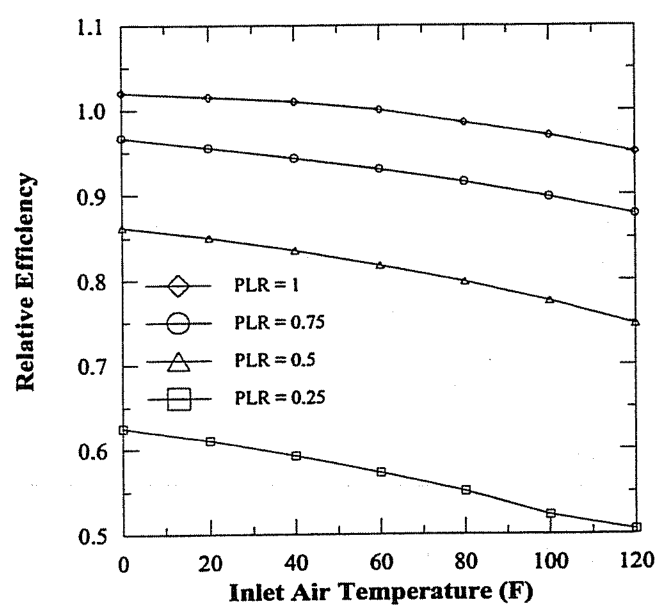

Figure 2 Combustion turbine efficiency variation as a function of ambient temperature and turbine part-load ratio.

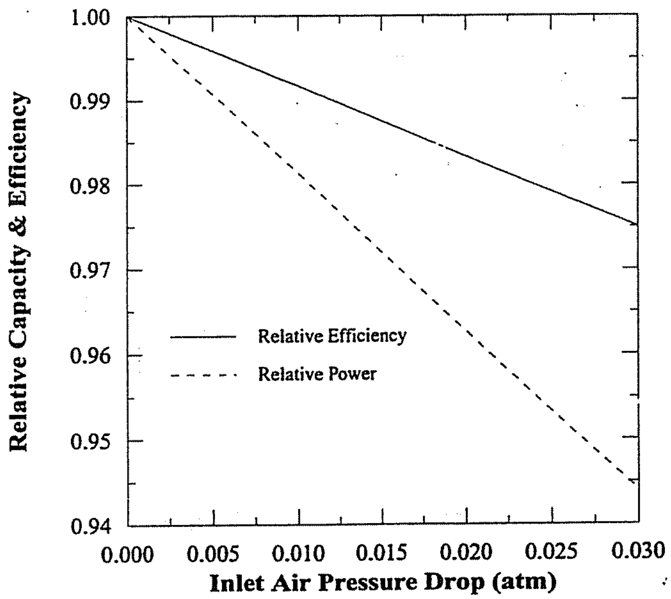

Figure 3 *Combustion turbine capacity and efficiency changes as a function of inlet air pressure drop.*

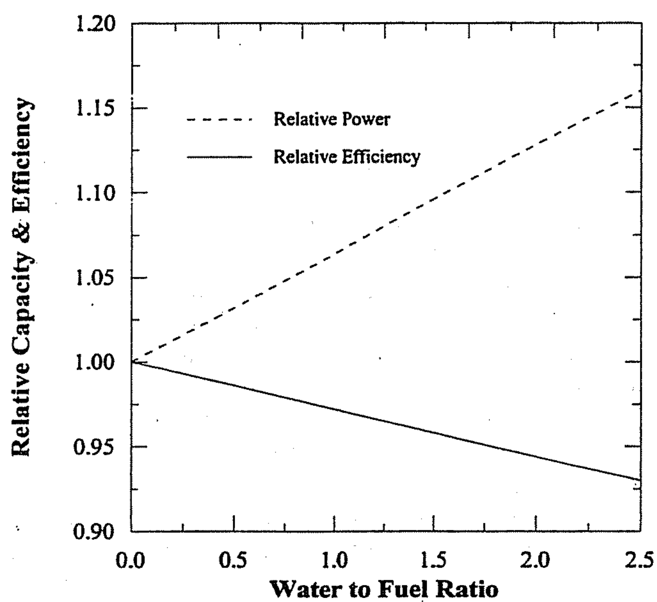

Figure 4 *Combustion turbine capacity and efficiency as a function of water flow rate.*

drops in the inlet airstream due to cooling coils and evaporative coolers are based on manufacturer's catalog data. For situations where an evaporative cooler is present with a coil for inlet air cooling, the penalty in turbine capacity and efficiency will be greater than that without the inlet section disturbances.

HYBRID INLET AIR COOLING SYSTEM

The proposed hybrid inlet air cooling system consists of two distinct systems—a stratified chilled water storage system and an ice storage-based system. A schematic of the system designed is shown in Figure 5.

The airflow into the turbine first passes through an optional evaporative cooler. The first coil provides cooling delivered by a chilled water storage system. In the second coil, cooling is provided by water circulating through an ice storage tank. Both the chiller and ice harvester operate only during the utility off-peak periods to charge storage.

Stratified Chilled Water Storage System

The stratified chilled water storage system is designed as a daily shift system. The chilled water storage tank is charged by a water-cooled centrifugal chiller with a nominal COP of 5.29. The chiller operates for a maximum of 15 hours per day, delivering 40°F chilled water to charge the storage tank. During the period when the combustion turbine is dispatched, a circulating pump provides the 40°F chilled water to an inlet air cooling coil (the actual temperature of chilled water at the coil inlet is higher than 40°F due to the tank performance characteristic during discharge, piping heat gains, and pump energy input) and 53°F water is returned to the storage tank. Further details on the stratified chilled water storage tank are given by Cross (1994).

The chiller is modeled based on curve fits from manufacturer's performance data (Cross 1994). The models provide the full and part-load performance of the chiller over the range of conditions encountered during the simulation period.

Ice Storage System

The ice storage system consists of a plate ice-type dynamic ice maker, a storage tank, pumps, piping, and a cooling coil downstream of the chilled water coil. Note that Figure 5 shows two separate coils being served from the chilled water storage system and ice storage system, respectively. The coils were "split" for modeling purposes. An actual design would, most likely, have a single coil with multiple circuits. The storage system is sized to operate on a weekly cycle.

The ice harvester is an evaporatively cooled ammonia-based plate ice machine. The model for the ice harvester is based on Knebel (1991) for a sequentially defrosted system configured in a pumped overfeed arrangement. Ice building and defrost timing cycles can be varied in the model. During a charge cycle, water is pumped from the bottom of the tank over the plates. During discharge, warm water returns from the coil and enters a spray distribution system in the top of the storage tank to yield an even ice melt-out.

An effectiveness model is used to characterize the performance of the ice tank during melt-out (Jekel 1991). The tank effectiveness is given by:

$$\varepsilon = \frac{T_{EW} - T_{LW}}{T_{EW} - 32} \tag{1}$$

where T_{EW} is the tank entering water temperature and T_{LW} is the tank leaving water temperature.

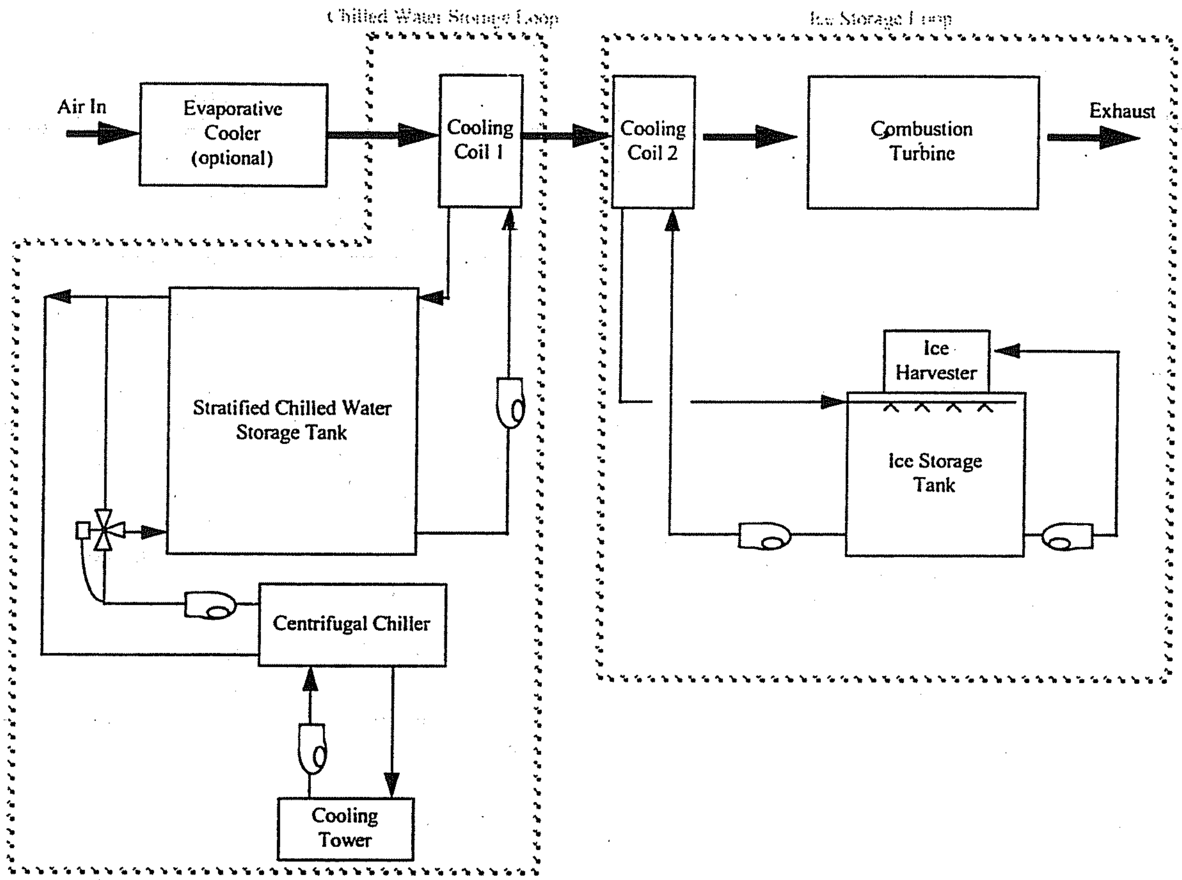

Figure 5 *Hybrid combustion turbine inlet cooling system.*

If the instantaneous ice inventory exceeds 20% of the total storage capacity (discharge fraction of 0.8 or less), the water leaving the tank is at 32°F. As the ice inventory drops below 20% of the total storage capacity (discharge fractions greater than 0.8), the tank leaving water temperature approaches the tank entering water temperature (Stewart 1994). On this basis, the effectiveness is assumed to be unity for discharge fractions less than 0.8. For discharge fractions between 0.8 and 1.0, the effectiveness is assumed to drop linearly from unity to zero.

RESULTS

Simulations were performed with the system simulation program TRNSYS (Klein 1994). The program was also used to size the system components (e.g., chiller, ice harvester, and storage capacities).

In the hybrid system, a greater portion of the inlet air cooling load is shifted to the more cost-effective stratified chilled water storage system. If a coil design condition of 92°F dry-bulb and 80°F wet-bulb is selected and it is further assumed that the chilled water loop cooling coil can achieve a 46°F air outlet temperature, then 90% of the total cooling duty is performed by the chilled water coil/loop. The remaining 10% of the cooling load can be handled by a much smaller ice system to achieve maximum combustion turbine capacity en-hancement by delivering approximately 40°F inlet air.

For design purposes, it is assumed that the combustion turbine will be available for a period ranging from four and eight hours per day, five days per week. A total of four different plant load profiles were considered in this analysis. Three of the four load profiles are "stepped" profiles of four, six, and eight hours duration. The fourth load profile is an eight-hour symmetrically peaked profile. The eight-hour peaked profile has the same electrical energy delivered as the four-hour step profile. All power plant load profiles are shown in Figure 6, which also shows the available combustion turbine generation with both 46°F inlet air (chilled water storage only) and 40°F inlet air (ice-only or hybrid system). Regardless of the load profile, the turbine is assumed to be dispatched for a total of 32 hours per year over consecutive days.

The goal of the study was to quantify the benefit, if any, to pursuing inlet air cooling system designs based on hybrid thermal storage strategies. The hybrid system will be compared with both chilled water-alone and ice storage-alone storage systems on a performance and first-cost basis. In addition to exploring the performance of ice and chilled water storage systems for inlet air cooling, this paper will investigate the impact that evaporative coolers have on inlet air cooling strategies for field retrofit applications.

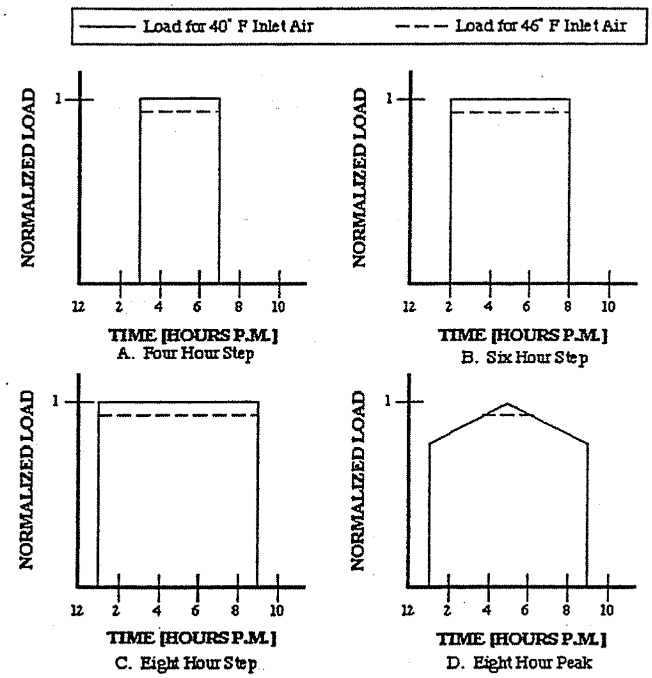

Figure 6 Normalized combustion turbine load profiles.

The cost figures presented in this section include the installed costs for all components in the system considered expressed as a capacity enhancement cost, *CEC*.

$$CEC = \frac{C_{system}}{\Delta P_{system}} \qquad (2)$$

where C_{system} is the total cost (U.S. dollars) of the inlet air cooling system and ΔP_{system} is the net power plant generating capacity increase at design weather conditions (kW). The cost data were determined from 1992 Means (Waier et al. 1992) or actual bids from previous installations (Ebeling 1994b).

For inlet air cooling systems using ice, an additional cost figure is reported, marginal capacity enhancement cost, *MCEC*. The marginal capacity enhancement costs are computed relative to the chilled water-only system by the following:

$$MCEC = \frac{C_{system} - C_{system,\,chw}}{\Delta P_{system} - \Delta P_{system,\,chw}} \qquad (3)$$

where:

C_{system} = current inlet cooling system cost,
$C_{system,chw}$ = chilled water-only system cost,
ΔP_{system} = current system capacity increase, and
$\Delta P_{system,chw}$ = capacity enhancement provided by a chilled water-only system.

OPTIMAL CAPACITY SPLIT

A major variable in the design of the hybrid system is the division in total cooling load between the chilled water storage system and the ice storage system. As the leaving air temperature from the first coil (chilled water system) rises, the total load imposed on the second coil (or ice system) increases as well as the system cost. To determine the optimum leaving air temperature from the first coil, the load on the first coil was varied by changing the number of rows in the first coil as well as the chilled water mass flow rate. During the process, the second ice-based cooling coil continually maintained a 40°F air outlet temperature (combustion turbine inlet air temperature). The optimum capacity split, as determined by the leaving air dry-bulb temperature from the first coil, is determined by minimizing the overall ice and chilled water storage system costs (on a cost per kW capacity enhancement basis). The results for the four-hour square load profile are shown in Table 1.

Table 1 shows that the system's capacity enhancement cost, *CEC*, is minimized with a leaving air dry-bulb temperature from the chilled water supplied cooling coil of 47.2°F. The capacity enhancement cost is not a strong function of the capacity split for the four-hour load plant load profile.

Another option for configuring an inlet cooling system is to use an existing evaporative cooler with the storage-based cooling system (an optional evaporative cooler is shown in Figure 5). An analysis identical to that given earlier was performed for a system with a 89% effective evaporative cooler upstream of the cooling coils. Table 2 shows the variation in leaving air dry-bulb (storage system capacity split) and system cost (storage system costs only, no evaporative cooler).

TABLE 1
Optimum Chilled Water/ice Capacity Split

Chilled Water-Based Cooling Coil			Ice-Based Cooling Coil			Cost
Coil Rows	Chilled Water (GPM)	LADB (°F)	Coil Rows	Ice Water (GPM)	LADB (°F)	*CEC* ($/kW)
6	6,749	52.8	4	5,596	40.0	222
7	8,630	50.1	3	8,394	40.0	223
8	7,302	48.6	3	4,846	40.0	215
9	7,086	47.2	3	2,935	40.0	211
10	7,214	46.0	1	11,192	40.0	217

TABLE 2
Optimum Chilled Water/ice Capacity Split with Evaporative Cooler

Chilled Water-Based Cooling Coil			Ice-Based Cooling Coil			Cost
Coil Rows	Chilled Water (GPM)	LADB (°F)	Coil Rows	Ice Water (GPM)	LADB (°F)	*CEC* ($/kW)
7	7,469	49.0	3	6,770	40.0	312.7
8	7,655	47.5	3	3,385	40.0	310.6
9	7,658	46.3	3	2,279	40.0	310.8
10	7,674	45.3	2	5,527	40.0	311.2

First, the chilled water coil requires fewer rows when an evaporative cooler is placed upstream of the coil. This results from an increase in coil effectiveness since a greater portion of the cooling coil will be wet with nearly saturated coil inlet air conditions. Also, the total load on the cooling system is greater since the enthalpy of the air out of the evaporative cooler is increased slightly due to the addition of water into the airstream.

STORAGE SYSTEM CAPACITY ENHANCEMENT COSTS

Three system designs are compared for the four plant load profiles given in Figure 6—chilled water storage only, ice storage only, and the optimized hybrid ice/chilled water storage system. The results of the 12 simulations for the design period are given in Table 3. Table 3 shows the predicted capacity enhancement, ΔP_{system}, capacity enhancement cost, *CEC*, and the marginal capacity enhancement cost, *MCEC*.

The ice storage-only systems have the highest combustion turbine capacity enhancement cost and are followed by the hybrid system and then the chilled water storage system. The combustion turbine capacity enhancement for the hybrid system includes the additional parasitic energy attributed to the pumps for both the chilled water storage and ice storage cooling system loops. Although the ice-based cooling system yields the greatest increase in turbine capacity, it is at a significant marginal cost. With the exception of the eight-hour peaked profile, the marginal capacity enhancement cost of the ice-based systems equal or exceed the cost of purchasing additional combustion turbine capacity (~ $500/kW). As the total energy of the stepped load profile increases, the storage costs increase due to the larger storage volume and compressor capacity required in the full storage strategy.

TABLE 3
Inlet Cooling System Related Combustion Turbine Capacity Enhancement Costs

Storage Option	Load Profile	ΔP_{system} (kW)	*CEC* ($/kW)	*MCEC* ($/kW)
Water	4 hr step	12,578	172	—
Ice	4 hr step	14,051	260	1,012
Hybrid	4 hr step	13,964	211	565
Water	6 hr step	12,578	255	—
Ice	6 hr step	14,051	358	1,240
Hybrid	6 hr step	13,964	299	703
Water	8 hr step	12,578	385	—
Ice	8 hr step	14,051	454	1,044
Hybrid	8 hr step	13,964	428	812
Water	8 hr peaked	12,578	211	—
Ice	8 hr peaked	14,051	353	1,566
Hybrid	8 hr peaked	13,964	226	366

TABLE 4
Inlet Cooling System Related Combustion Turbine Capacity Enhancement Costs with an Upstream Evaporative Cooler

Storage Option	Load Profile	ΔP_{system} (kW)	*CEC* ($/kW)	*MCEC* ($/kW)
Water	4 hr step	8,015	260	—
Ice	4 hr step	9,508	380	1,025
Hybrid	4 hr step	9,458	311	590
Water	6 hr step	8,015	384	—
Ice	6 hr step	9,508	525	1,283
Hybrid	6 hr step	9,458	438	736
Water	8 hr step	8,015	581	—
Ice	8 hr step	9,508	668	1,135
Hybrid	8 hr step	9,458	620	837
Water	8 hr peaked	8,015	332	—
Ice	8 hr peaked	9,508	554	1,745
Hybrid	8 hr peaked	9,458	357	498

Table 4 shows the results of capacity enhancement costs when a storage-based inlet air cooling system is applied to a combustion turbine with an existing evaporative cooler. The system costs reported do not include the cost of the evaporative cooler.

The results in Table 4 show that an existing evaporative inlet air cooler upstream of the coils decreases the capacity enhancement associated with the storage-based cooling systems significantly. The storage-based cooling system cost is slightly less in all cases with the evaporative cooler than without it; however, the cost on a capacity enhancement basis is considerably higher with an existing evaporative cooler. It is not economically judicious to install both an evaporative cooler with an augmenting inlet air cooling system. This would not be true if the evaporative cooler fill material could be used in conjunction with chilled water to provide a direct-contact heat exchanger. If the evaporative cooler fill material could be used as a direct-contact heat exchanger using chilled water, significant first-cost savings could be achieved by eliminating the cooling coils.

CONCLUSIONS

The use of thermal energy storage offers a cost-competitive alternative to enhancing the performance of combustion turbines for peaking power production. Chilled water system configurations had the lowest capacity enhancement cost compared to ice and hybrid storage systems; however, the actual capacity enhancement is not as great as with ice-only or hybrid systems. Systems that use ice as a storage medium (ice-only and hybrid) can yield an 11% greater increase in plant capacity when compared with chilled water-based inlet air cooling systems.

The use of evaporative coolers in conjunction with storage based inlet air cooling systems is not economical for the load profiles considered here.

The hybrid inlet air cooling system presented yields the same performance benefits of ice-only inlet air cooling systems at a 6% to 36% cost savings compared with the ice-only systems. The combustion turbine load profile can have a significant impact on the sizing and cost of a storage system to provide inlet air cooling. The eight-hour peaked load profile had a higher capacity enhancement cost than the four-hour step profile (each having the same incremental delivered kWh due to inlet air cooling). Interestingly, the hybrid system was the least sensitive to the load profile having only a 7% increase in capacity enhancement cost in moving from the four-hour step to the eight-hour peak while the chilled water and ice storage systems had 23% and 36% cost increases, respectively.

An operational issue not specifically addressed in this paper is the impact of long inactive periods of the thermal storage system. A daily cycle has been assumed for the chilled water storage system. A problem with stratified chilled water storage that results over a period of inactivity is that ambient gains and conduction within the tank lead to destratification, thereby reducing or eliminating the stored energy of the tank. A method of addressing this concern is to charge the stratified chilled water storage only on days prior to when a high probability exists of dispatching the combustion turbine.

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