UNION ELECTRIC COMPANY'S COMBUSTION TURBINE INLET AIR COOLING STUDY

Charles R. Henderson
Engineer
Union Electric Company
St. Louis, Missouri

Jerry A. Ebeling
Project Manager
Burns & McDonnell
Kansas City, Missouri

Richard C. Smith
Supervising Engineer
Union Electric Company
St. Louis, Missouri

ABSTRACT

Union Electric Company is a summer peaking utility, experiencing peak electrical load demands during the hot summer months. Combustion turbine generators are often used to meet the summer peak demands. However, the generating capability of a combustion turbine decreases as the ambient air temperature increases. When system peak demands are at their highest levels on the hottest days of the year, the generating capacity of the combustion turbines are at their lowest values. This lost generating capacity can be recovered by cooling the air entering the combustion turbines.

Various combustion turbine inlet air cooling technologies were investigated for a General Electric Model 7B combustion turbine. The cooling technologies evaluated in the study were evaporative cooling, thermal energy storage (ice), on-line mechanical chiller, direct absorption chiller, steam absorption chiller with heat recovery steam generator (HRSG), and once-through cooling using well water. Conceptual designs, performance estimates, installation and operating costs were developed for each alternative.

INTRODUCTION

Combustion turbine generators are mass flow machines, with the generating capabilities related to the mass flow of air through the unit. The mass air flow depends on the air density which is a function of the inlet air temperature. Cooling the ambient air during warm weather before it enters the combustion turbine increases the air density and allows the unit to operate at a higher level of output than otherwise possible. For every one degree Fahrenheit reduction in inlet air temperature, output can be increased by approximately 0.38 percent. Output improvements of 25 percent can be attained, depending on site conditions and design parameters. Figure 1 shows how output, heat rate, air flow, and turbine exhaust temperature vary with compressor inlet air temperature for a General Electric Model 7B machine on base load output.

Figure 1. Effect of Compressor Inlet Temperature on Base Load Operation.
generation on a utility system. It may also provide for better utilization of existing peaking capacity and extend the life of certain machine components.

Several combustion turbine inlet air cooling approaches are presented in this paper. These include evaporative cooling, thermal energy storage, mechanical chiller, absorption chiller, and once through well water cooling. The paper discusses conceptual designs, performance estimates, capital costs, and O&M costs that were developed for each of the cooling alternatives. The cooling systems were tailored for application on a typical G.E. 7B combustion turbine.

The turbine evaluated in the study is a MS 7000 B industrial model placed into initial operation in 1974. The machine is located at Union Electric's Meramec Power Plant and is fueled by No. 2 fuel oil. The compressor section has 17 stages and the expansion turbine has 3 stages. The turbine and generator are direct coupled and rotate at 3,600 rpm. The generator is a G. E. air cooled, open-ventilated, synchronous unit.

The ISO ratings for the machine are as follows:

- **Base Load Rating**: 59,000 kW
- **Peak Load Rating**: 65,200 kW
- **Base Load Heat Rate (LHV)**: 11,120 Btu/kWh
- **Peak Load Heat Rate (LHV)**: 11,010 Btu/kWh
- **Base ISO Airflow**: 1,896,000 lb/hr
- **Peak ISO Airflow**: 1,896,000 lb/hr
- **Base Load Generator Rating**: 68,889 kVA
- **Peak Load Generator Rating**: 75,889 kVA

The design conditions that were used in developing and evaluating each air cooling alternative were:

- **Site Elevation**: 418 feet msl
- **Ambient Air Temperature**: 100 °F, Dry Bulb, 76 °F, Wet Bulb
- **Combustion Turbine Operating Cycle**: 15 hrs/wk

### DESIGN PARAMETERS

As part of the study work, it was necessary to develop appropriate and realistic design parameters for use in the conceptual design of each air cooling alternative. The use of consistent design parameters would allow the comparison of system costs and performances, as well as provide a basis for an economic evaluation of the systems. The design parameters selected in study were as follows:

- **Air Cooling Heat Exchanger**: Indirect Contact Coil
- **Air Cooling Heat Exchanger**
- **Design Air Flow Velocity**: 350 to 400 fpm
- **Coil Terminal Temperature Difference (TTD)**
  - **Thermal Energy Storage**: 7°F
  - **Mechanical Chiller**: -4°F
  - **Absorption Chiller**: -4°F
  - **Absorption Chiller/HRSG**: -4°F
  - **Well Water Cooling**: -4°F
- **Chilled Water Temperature**
  - **Thermal Energy Storage**: 33°F
  - **Mechanical Chiller**: 45°F
  - **Absorption Chiller**: 45°F
  - **Absorption Chiller/HRSG**: 45°F
  - **Well Water Cooling**: 55°F
- **Cooled Inlet Air Flow Temperature**
  - **Thermal Energy Storage**: 33°F
  - **Mechanical Chiller**: 45°F
  - **Absorption Chiller**: 45°F
  - **Absorption Chiller/HRSG**: 45°F
  - **Well Water Cooling**: 59°F
- **Cooling System Refrigerants**
  - **Thermal Energy Storage**: Ammonia
  - **Mechanical Chiller**: HCFC-22
  - **Absorption Chiller**: Water/Lithium Bromide
  - **Absorption Chiller/HRSG**: Water/Lithium Bromide
- **Coefficients of Performance**
  - **Thermal Energy Storage**: 4.89
  - **Mechanical Chiller**: 5.49
  - **Absorption Chiller**: 1
  - **Absorption Chiller/HRSG**: 1.12

For the G.E. Model 7B combustion turbine, the practical minimum inlet air temperature was calculated to be 40 °F to avoid ice formation. As the mass flow of air converges into the inlet ductwork of a combustion turbine, the velocity increases and static pressure decreases. The inlet air temperature drops with the decrease in static air pressure according to the ideal gas law, \( PV = nRT \). The air becomes saturated with moisture as it is cooled and dehumidified below the ambient wet bulb temperature. This situation would occur for all of the inlet air cooling alternatives except the evaporative cooler system.

If the air is cooled below a theoretical target temperature, the air temperature at the combustion turbine bell mouth could reach 32°F or lower. Ice could form on the bell mouth and inlet ductwork under this condition. The ice could be ingested by the compressor section, resulting in possible damage.

Based on pressure drop estimates for the air filters and inlet air duct for the G.E. 7B combustion turbine,
calculations showed that the temperature at the bell mouth would be approximately 35°F with a 40°F air temperature leaving the cooling heat exchanger.

**DISCUSSION OF INLET AIR COOLING SYSTEMS**

**Evaporative Cooling System**

Evaporative coolers operate on the differential between the ambient air wet and dry bulb temperatures. By directly exposing the air to water at a temperature equal to the wet bulb temperature of the dry air, the sensible heat in the air vaporizes the liquid water. The adiabatic heat transfer from the air to water provides sensible cooling of the air, lowering the dry bulb temperature. The sensible heat used to vaporize the water enters the air as latent heat in the water vapor.

No external energy is required except pumping power for the evaporative cooling process. The wet bulb temperature of the air remains constant, but the dew point temperature, relative humidity, specific humidity, and enthalpy of the air increase. The effectiveness of an evaporative cooler depends on its design and the relative humidity of the air. The cooled air dry bulb temperature can approach, but never actually reach, the ambient wet bulb temperature (100% efficiency). The best available commercial designs provide efficiencies of up to 90%.

The advantages of evaporative cooling include relatively low capital and operating costs, small space requirements, simple design and operation, and reduction of dust loading on the inlet filtration system. The main disadvantages include a limited increase in combustion turbine output, and reduced effectiveness in humid climates.

**Figure 2. Evaporative Cooling Conceptual Design.**

Figure 2 shows the conceptual design for the G.E. 7B evaporative cooler system, assuming a cooler efficiency of 90% and an air velocity of 400 fpm. The system includes two evaporative cooler compartments with two heat transfer media packs per compartment (four total). The total heat transfer media would be approximately 918 square feet. Based on the design ambient air conditions, the inlet air temperature would be cooled to 78°F dry bulb and 76°F wet bulb. Water would be circulated by four 25 percent capacity circulating water pumps from the basin to the top of the cooling media. The total circulating water flow rate would be 1,200 gpm. Make-up water flow to the basin would be 23 gpm to account for evaporative and blowdown losses. Figure 3 illustrates the evaporative cooling system process on a psychrometric chart.

**Thermal Energy Storage**

For the past several years, large scale commercial thermal energy storage, in the form of ice storage, has been used to level large cooling loads in the produce industry, and in building air conditioning systems. Using ice storage for combustion turbine inlet air cooling is a relatively new application. In the process, ice is produced using low cost, off-peak electricity and stored in a large tank. When utility system peaks occur, the stored ice provides a relatively constant temperature (32 °F) heat sink to cool the combustion turbine inlet air flow.

Ice was selected for the storage media since the volume required to store chilled water would be on the order of 7 times greater than ice (ice latent heat of fusion - 144 Btu/lb).

**Figure 3. Evaporative Temperature (F)**

**Figure 3. Evaporative Cooling Process.**

**Figure 4. Conceptual Design for Air Cooling Portion of the Thermal Energy Storage System for the G.E. 7B.** Chilled water would be taken from the
bottom of the ice storage tank at 33 °F and circulated by two (2) 50 percent capacity chilled water pumps through the air cooling coils and back to the top of the tank. The cooled air temperature would be maintained at 40 °F by chilled water flow control valves at the inlet of the air cooling coils. The chilled water flow rate would be approximately 6,575 gpm at maximum turbine output.

![Air Cooling Coils](image)

Figure 4. Thermal Energy Storage Conceptual Design.

With a inlet air flow of 1,949,095 pounds per hour, the calculated thermal load would be 47.3 MMBtu/hr based on ambient air conditions of 100 °F dry bulb and 76 °F wet bulb. The rate of condensation from the air cooling coils would be approximately 34 gpm at 40 °F.

Twenty-two air cooling coils would be required to give the necessary 1,229 square feet of coil face area calculated from the system cooling thermal load. Each coil would have a maximum overall length of 15 feet and height of 4 feet. The coils would be eight tube rows deep, and be made of copper tubes with aluminum fins. The air flow velocity through the coils would be approximately 346 feet per minute, for a total pressure drop in the wet condition of approximately 0.65 inches of water. The water side pressure drop through the coils is estimated at 33 feet of water.

The required ice storage tank would be approximately 760,000 gallons based on a weekly levelized thermal load. The ice storage volume includes the required ice storage of 690,000 gallons plus a ten percent free board volume of 70,000 gallons for tank storage inefficiency. The maximum ice storage volume would include a base inventory of 625,000 gallons (25,000 ton-hours) plus reserve inventory of 68,000 gallons (2,700 ton-hours) to insure a constant 33 °F chilled water temperature out of the tank.

The ice storage tank design week is based on three consecutive days of five hours of generation each (15 hours total). The maximum stored ice volume would occur at the beginning of the generation week and would provide approximately eight hours of continuous cooling. The minimum ice volume would occur at the end of the generation week and would be the reserve inventory of 2,700 ton-hours, or less than one hour of cooling.

Figure 5 shows the conceptual design of the ice production system for the thermal energy storage system. Based on the weekly generation cycle of 15 hours per week and assuming 153 hours per week of ice production, the levelized net thermal load for the ice production system would be 4.3 MMBtu/hr or 355 tons of cooling. Based on a 89 percent system efficiency, the refrigeration system would be rated at 400 tons.

![Conceptual Design Ice Production System](image)

Figure 5. Conceptual Design Ice Production System.

The air cooling process used in the thermal energy storage system involves sensible cooling and dehumidification. Sensible cooling occurs until the dry bulb temperature of the air reaches the dew point temperature and the air becomes saturated with water vapor. Additional cooling provides both sensible cooling and water vapor removal. Sensible cooling with dehumidification has an advantage over the evaporative cooling process. Water vapor is less dense than air so the total air/vapor mixture density increases as relative humidity decreases. Therefore, the net effect of increasing the air density by lowering its dry bulb temperature is enhanced by dehumidification. Also, since the specific heat of water vapor is higher than the
specific heat of air, more energy is required from the combustion process to the heat the water vapor. So, both the combustion turbine efficiency and turbine output increase. Figure 6 illustrates the thermal energy storage system air cooling process on a psychrometric chart.

**Mechanical Chiller**

Mechanical chillers would be considered an on-line system utilizing a refrigeration cycle to provide chilled water for cooling the air. The chilled water portion of the system would be a closed-loop system utilizing a head tank for system expansion, chilled water evaporator, water pumps, and air cooling coils. A mechanical chiller system would use electricity produced by the combustion turbine to power an electrical refrigeration motor/compressor. Therefore, the net capacity increase from a mechanical chiller system is limited due to the amount of energy required to operate the refrigeration system.

The range of chilled water temperatures that can be realistically achieved with a mechanical chiller is 40 to 45 °F. Manufacturer's rate their equipment at a chilled water temperature of 45 °F and 100 percent rated load. To achieve temperatures below 45 °F, the mechanical chiller must be derated in proportion to the desired temperature. For example, to achieve a water temperature of 43.5 °F would require the unit to operate at 70 percent of its rated capacity. Although it is desirable to have as low an inlet air temperature as possible to maximize the incremental capacity gained, it is not economical to purchase and operate mechanical chillers much below 100 percent of their rated capacity. A chilled water temperature of 45 °F was used in the study.

Figure 7 shows the conceptual design of the air cooling portion of the mechanical chiller system. Figure 8 shows the conceptual design for the condenser cooling portion. Chilled water would be generated by the evaporator section of the two mechanical chillers at 45 °F and circulated by two fifty percent capacity chilled water pumps through the air cooling coils and back to a 1,000 gallon head tank. The chilled water flow rate would be approximately 9,540 gpm.

**Figure 6. Thermal Energy Storage Process.**

**Figure 7. Mechanical Chiller Conceptual Design**

With an inlet flow of 1,918,144 lb/hr, the calculated air cooling coil thermal load would be 37.8 MMBtu/hr. Two mechanical chillers would be required each rated at 1,600 tons capacity. The cooled air temperature would be 49 °F. The rate of condensation from the air cooling coils would be approximately 25 gpm at 49 °F.

**Figure 8. Condenser Cooling Conceptual Design.**

As with the thermal energy storage system, the air cooling process used in the mechanical chiller system involves sensible cooling and dehumidification. The wet and dry bulb temperatures as well as the specific humidity and enthalpy of the air decrease during the
cooling process. Figure 9 illustrates the mechanical chiller system air cooling process on a psychometric chart.

![Psychometric Chart](https://example.com/psychometric_chart.png)

**Figure 9.** Mechanical Chiller, Absorption Chiller, and Absorption/HRSG Chiller Processes.

**Absorption Chiller**

An absorption chiller system, like a mechanical chiller system, is an on-line system. A refrigeration cycle is utilized to provide chilled water to cool the air while the gas turbine is operating. An absorption chiller is different from a mechanical chiller because it utilizes waste heat directly from the combustion turbine exhaust gas as the driving energy source for the system. There are no large electric drive motors required. Because less electrical energy is required, the operating cost of this system would be less than either the thermal energy storage or mechanical chiller systems.

The range of chilled water temperatures that can be achieved with a lithium bromide cycle, typically used in absorption chiller systems, is 40 to 45 °F. Manufacturer's rate their equipment at a chilled water temperature of 45 °F much like an mechanical chiller. A chilled water temperature of 45 °F was used in the study.

Figure 10 shows the conceptual design for the inlet air cooling portion of the absorption chiller system while Figure 11 shows the condenser cooling portion. Chilled water would be generated by the evaporator section of three absorption chillers at 45 °F and circulated by two (2) - 50 percent capacity pumps through the air cooling coils and back to a 1,000 gallon head tank. The chilled water flow rate would be approximately 9,540 gpm.

![Absorption Chiller Conceptual Design](https://example.com/absorption_chiller_conceptual_design.png)

**Figure 10.** Absorption Chiller Conceptual Design.

With an air flow of 1,917,679 lb/hr, the calculated air cooling coil thermal load would be 37.8 MMBtu/hr. Two absorption chillers would be rated at 1,000 tons each and a third chiller would be rate at 1,200 tons. The cooled air temperature would be 49 °F. The rate of condensation from the air cooling coils would be approximately 25 gpm at 49 °F.

![Condenser Cooling Conceptual Design](https://example.com/condenser_cooling_conceptual_design.png)

**Figure 11.** Condenser Cooling Conceptual Design.

Figure 12 shows the conceptual design for the heat supply portion of the absorption chiller system. Exhaust gas from the combustion turbine would be routed in ducts directly to the generator sections of the three absorption chillers. The required exhaust gas flow to the generators would be 240,000 lbs/hr at a temperature of 947 °F. The exhaust gas would exit the absorption chiller at approximately 350 °F. Three induced draft fans would take the exhaust gas from each absorption chiller and force it back into the combustion turbine exhaust duct.
As with the thermal energy storage and mechanical chiller systems, the air cooling process used in the absorption chiller system involves sensible cooling and dehumidification. The wet and dry bulb temperatures as well as the specific humidity and enthalpy of the air decrease during the cooling process. Figure 9 illustrates the absorption chiller system air cooling process on a psychometric chart.

**Absorption/HRSG Chiller**

An absorption/HRSG chiller system is very similar to an absorption chiller system. However, the absorption/HRSG chiller system utilizes steam from a waste heat recovery steam generator (HRSG) as the driving energy source for the system. Exhaust gas from the combustion turbine is used as the heat source for the HRSG. Other than the steam production system, the process for producing chilled water and cooling the combustion turbine inlet air is similar to the absorption chiller system.

Figure 13 shows the steam supply portion of the absorption/HRSG chiller system while Figure 14 shows the conceptual design for the heat supply portion. Exhaust gas from the combustion turbine would be routed through a duct directly to the heat recovery steam generator. The required exhaust gas flow would be 210,000 pounds per hour at a temperature of 947 °F. The exhaust gas would exit the HRSG at approximately 350 °F. The gas side pressure drop through the unit is estimated to be six inches of water. Two 50 percent capacity induced draft fans would take the exhaust gas from the HRSG and force it back into the combustion turbine exhaust duct. Dampers on the outlet of the induced draft fans would be used to control the flow of exhaust gas.

**Once Through Well Water Cooling**

The well water cooling system utilizes the cooler temperature of ground water to cool the air to the combustion turbine. Well water is pumped through air cooling coils located in the turbine air inlet. Energy for well water pumping is the only external energy required by the system.

The well water cooling system has an advantage over the evaporative cooling, because evaporative cooling humidifies or adds less dense water vapor to the air, the resulting capacity and efficiency increase for evaporative cooling would not be as great as a process involving sensible cooling and dehumidification. For a given dry
bulb temperature, the total air/vapor mixture density decreases as the relative humidity increases. Therefore the net effect of increasing the air density by lowering its dry bulb temperature is offset by the higher percent of water vapor in the air/water mixture. For well water cooling compared to evaporative cooling, the specific heat of the air is lower than that of water vapor. So, less energy is required from the combustion process to heat the drier air.

The advantage of well water cooling like evaporative cooling includes relatively low capital and operating costs. The disadvantage of well cooling involves the flow rate of well water required for the cooling process, extraction and disposal. The flow of water needed for effective cooling of the inlet air flow ranges from 6,000 to 10,000 gpm. This flow rate would require the construction of several wells and a substantial piping network. Discharge or disposal of the heated well water leaving the air cooling coils may also be a problem. Another disadvantage is the potential of the well water to be corrosive or to cause deposits on piping and equipment.

Figure 15 shows the conceptual design for the well water cooling system. Well water would be pumped to the air cooling coils by nine wells located at the combustion turbine site. The required well water flow, based on the calculated air cooling thermal load of 25.5 MMBtu/hr, would be 7,050 gpm. The air outlet conditions from the coils would be 59 °F and 100% relative humidity based on 55 °F well water and a coil TTD of 4 °F. Figure 16 illustrates the sensible cooling and dehumidification process for the well water cooling system on a psychometric chart.

Figure 15. Well Water Cooling Conceptual Design.

Figure 16. Once Through Well Water Cooling Process.

REVIEW OF EXISTING MACHINE AUXILIARY SYSTEMS

A review of the G.E. 7B auxiliary systems was performed to determine if there would be any operational limitations when operating with inlet air cooling. It was determined from this work that the rated capacity of the generator was less than the predicted capacity for the thermal energy storage and chiller system alternatives. Two possible solutions to allow operation with the existing generator would be either to operate the generator at the higher loads with a corresponding reduction in anticipated life or to add generator cooling to operate at a lower point on the respective load/temperature curve.

The designs for the affected inlet air cooling alternatives have included the additional cooling requirements for generator cooling. Other machine auxiliary systems, including unit transformers, lube oil and fuel oil, appeared to be adequate for operation at the higher generating capacity.

PREDICTED PERFORMANCE

Based on the conceptual designs and study parameters, the performance for each inlet air cooling alternative was predicted along with existing performance of the G.E. 7B gas turbine. The predicted base and peak load, total and incremental net generating capacities of each air cooling alternative are shown on Table 1. The predicted capacities of the existing combustion turbine operating at design ambient conditions are shown for comparison.
Table 1. Predicted Net Capacity.

<table>
<thead>
<tr>
<th>System</th>
<th>Base Total (kW)</th>
<th>Base Incremental (kW)</th>
<th>Peak Total (kW)</th>
<th>Peak Incremental (kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Existing Unit</td>
<td>47,530</td>
<td>0</td>
<td>53,385</td>
<td>0</td>
</tr>
<tr>
<td>Evaporative Cooling</td>
<td>51,711</td>
<td>4,181</td>
<td>57,790</td>
<td>4,405</td>
</tr>
<tr>
<td>Thermal Energy Storage</td>
<td>60,101</td>
<td>12,571</td>
<td>66,681</td>
<td>13,296</td>
</tr>
<tr>
<td>Mechanical Chiller</td>
<td>55,475</td>
<td>7,945</td>
<td>61,947</td>
<td>8,562</td>
</tr>
<tr>
<td>Absorption Chiller</td>
<td>57,064</td>
<td>9,534</td>
<td>63,534</td>
<td>10,149</td>
</tr>
<tr>
<td>Absorption/HRSG Chiller</td>
<td>57,184</td>
<td>9,654</td>
<td>63,654</td>
<td>10,269</td>
</tr>
<tr>
<td>Well Water Cooling</td>
<td>56,028</td>
<td>8,498</td>
<td>63,243</td>
<td>9,858</td>
</tr>
</tbody>
</table>

The thermal energy storage system would provide the greatest capacity improvement of approximately 25 percent over the existing unit. The evaporative cooling system would provide the lowest capacity improvement of approximately 8 percent.

The predicted net heat rate on base load operation for each air cooling alternative is shown on Table 2. The predicted heat rate of the existing combustion turbine operating at design ambient conditions is shown for comparison.

Table 2. Predicted Base Load Heat Rate.

<table>
<thead>
<tr>
<th>System</th>
<th>Base Load Heat Rate (Btu/kW-hr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Existing Unit</td>
<td>12,872</td>
</tr>
<tr>
<td>Evaporative Cooling</td>
<td>12,479</td>
</tr>
<tr>
<td>Thermal Energy Storage</td>
<td>13,298</td>
</tr>
<tr>
<td>Mechanical Chiller</td>
<td>12,664</td>
</tr>
<tr>
<td>Absorption Chiller</td>
<td>12,310</td>
</tr>
<tr>
<td>Absorption/HRSG Chiller</td>
<td>12,285</td>
</tr>
<tr>
<td>Well Water Cooling</td>
<td>12,247</td>
</tr>
</tbody>
</table>

The well water cooling system provides the lowest predicted heat rate of 4.9 percent less than the existing combustion turbine. The low heat rate is the combined result of the improved efficiency of the combustion turbine and low auxiliary power requirements.

The thermal energy storage system provides the highest predicted heat rate of approximately 3.3 percent more than the existing unit. This is due to the energy required for ice production during off-peak periods.

Table 3. Estimated Installed Capital Costs.

<table>
<thead>
<tr>
<th>System</th>
<th>Total Est. Cost ($)</th>
<th>Incremental Unit Cost ($/kW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>75 MW CTG</td>
<td>---</td>
<td>400 - 500</td>
</tr>
<tr>
<td>Evaporative Cooling</td>
<td>$600,000</td>
<td>115 - 170</td>
</tr>
<tr>
<td>Thermal Energy Storage</td>
<td>$2,950,000</td>
<td>190 - 280</td>
</tr>
<tr>
<td>Mechanical Chiller</td>
<td>$3,700,000</td>
<td>375 - 560</td>
</tr>
<tr>
<td>Absorption Chiller</td>
<td>$6,120,000</td>
<td>515 - 770</td>
</tr>
<tr>
<td>Absorption/HRSG Chiller</td>
<td>$5,700,000</td>
<td>470 - 710</td>
</tr>
<tr>
<td>Well Water Cooling</td>
<td>$2,190,000</td>
<td>205 - 310</td>
</tr>
</tbody>
</table>

Note 1: Estimate ± 20 percent

Table 4. O&M Cost Estimate - Life Cycle Analysis.

<table>
<thead>
<tr>
<th>System</th>
<th>Fixed Cost ($/kW-yr)</th>
<th>Variable Cost (mils/kW-hr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Existing Unit</td>
<td>4.06 - 6.10</td>
<td>2.4 - 3.6</td>
</tr>
<tr>
<td>Evaporative Cooling</td>
<td>4.86 - 7.30</td>
<td>2.4 - 3.6</td>
</tr>
<tr>
<td>Thermal Energy Storage</td>
<td>7.93 - 11.89</td>
<td>4.8 - 7.2</td>
</tr>
<tr>
<td>Mechanical Chiller</td>
<td>9.44 - 14.16</td>
<td>3.2 - 4.8</td>
</tr>
<tr>
<td>Absorption Chiller</td>
<td>13.44 - 20.16</td>
<td>3.2 - 4.8</td>
</tr>
<tr>
<td>Absorption/HRSG Chiller</td>
<td>12.75 - 19.13</td>
<td>3.2 - 4.8</td>
</tr>
<tr>
<td>Well Water Cooling</td>
<td>7.2 - 10.80</td>
<td>3.2 - 4.8</td>
</tr>
</tbody>
</table>

The evaporative cooler has the lowest estimated capital cost at $600,000 while the absorption chiller system has the highest cost at $6,120,000.

Table 3 also shows the estimated unit cost of incremental capacity gained for each alternative. The cost of new combustion turbine capacity is shown for comparison purposes.

The well water cooling system provides the lowest predicted heat rate of 4.9 percent less than the existing combustion turbine. The low heat rate is the combined result of the improved efficiency of the combustion turbine and low auxiliary power requirements.

Operating Costs

A ten-year life cycle cost analysis was performed for each air cooling alternative. Table 4 shows the average estimated operating and maintenance (O&M) costs for each inlet air cooling alternative. The estimated O&M cost for the existing combustion turbine is shown for comparison.
CONCLUSIONS

Based on the inlet air cooling study for the G.E. 7B gas turbine, the following conclusions are presented:

- The evaporative cooler system is the least expensive capital cost alternative. However, capacity improvement is limited to approximately 4.1 MW due to thermodynamics.

- The thermal energy storage system provides the greatest incremental capacity improvement of about 12.6 MW.

- The mechanical chiller and absorption chiller systems are estimated to high very high installed capital costs.

- The well water cooling system provides limited but economical incremental capacity improvement. However, the once through system requires large well water flow rates and may present a environmental disposal problem.

Combustion turbine power augmentation through air cooling may be an attractive alternative for utilities to boost the summer power output from existing gas turbines. Air cooling may be an economical attractive alternative to delay the installation of new peaking power generation. Capacity improvements of up to 25 percent can be realized, depending on particular site conditions and design parameters.

REFERENCES
