Capacity Enhancement of ABB 11N1 with Thermal Energy Storage

by

Sanjeev Jolly, P.E.
Joseph Nitzken, P.E.
Donald Shepherd
Caldwell Energy & Environmental, Inc.Louisville, Kentucky, USA
Douglas T. Eberlein, P.E.
Wisconsin Electric, Milwaukee, Wisconsin, USA

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ABSTRACT

Many electrical utilities are experiencing increased seasonal demand due to air conditioning use. Current air quality rules and regulatory changes in the industry favor the installation of combustion turbines (CT) for new generation capacity. Ironically, CT output declines as ambient temperature increases. As the output of a combustion turbine degrades during hot weather conditions, warm ambient air to the compressor inlet can be chilled to restore lost capacity. For a thermal energy storage (TES) system, a refrigeration system is utilized to build a cold energy reserve and this reserve is used to cool inlet air during peak hours to enhance turbine capacity. Typically, a TES system can be installed at half the cost of installing a new turbine. Two such systems were installed at Wisconsin Electric's (WE) Concord and Paris Generating Stations, each having four ABB 11N1 combustion turbines running in simple cycle mode. This paper discusses the feasibility study and design for a TES based inlet chilling system to provide peaking capacity enhancement for these ABB 11N1 machines. The system design is based on cooling the inlet air for 4 hours a day, 5 days a week. Each of these systems includes a 3,400 ton refrigerant plant, two 2 1/4 million gallon tanks, and 2,600 tons of ice-making capacity. The instantaneous cooling load for each set of 4 turbines is 16,200 tons. With this Combustion Turbine Inlet Air Cooling System (CTIAC), a total gain of more than 112 MW (approximately 17%) was expected for the two plants, making it the world's largest inlet air cooling system. Finally, the system performance is discussed during summer operation.

INTRODUCTION

The output of a combustion turbine decreases as the ambient temperature increases (see Figure 1). This is because the mass flow rate of air decreases as the air becomes less dense at higher ambient temperatures. Since the output of a CT is proportional to the airflow, a reduction in the airflow would cause a reduction in the output. In addition to output, the heat rate degrades with an increase in the ambient temperature. As a rule of thumb, for simple cycle operation, every 10°F rise in the ambient temperature decreases the output by 3.2% and increases the heat rate by 0.9%. Therefore, significant capacity enhancement is possible with inlet air cooling. As far as the heat rate is concerned, the net increment depends upon whether the turbine is running in simple/combined cycle mode, type of cooling method used, and the cooling load. There are two basic classifications of inlet air cooling – evaporative cooling, where the air can be cooled to the wet bulb temperature, and refrigerated cooling, where the air is cooled to below the dew point temperature [1]. The main advantage of evaporative cooling is simplicity of operation and low cost, but the capacity enhancement is limited and humidity dependent [2] whereas significant capacity enhancement is possible through refrigerated cooling since the air can be cooled to temperatures much lower than the dew point temperature.

Refrigerated cooling can be used to provide continuous cooling throughout the day or on-peak cooling using thermal energy storage (TES). With continuous cooling, the inlet air can be cooled instantaneously whenever needed. There are several ways of providing continuous cooling – direct refrigeration, absorption chillers, electrical chillers, etc. [3]. This type of cooling is most suitable for base load plants or whenever cooling is required for more than 6-8 hours a day.

In thermal energy storage, a cold reserve is built during the off-peak hours and this cold reserve is used during the on-peak hours to cool the inlet air. The cold storage medium could be ice, chilled water, eutectic, etc. Since the cold reserve is built during the off-peak hours, the auxiliary power consumption during the peak hours is minimal, resulting in maximum peak capacity enhancement.

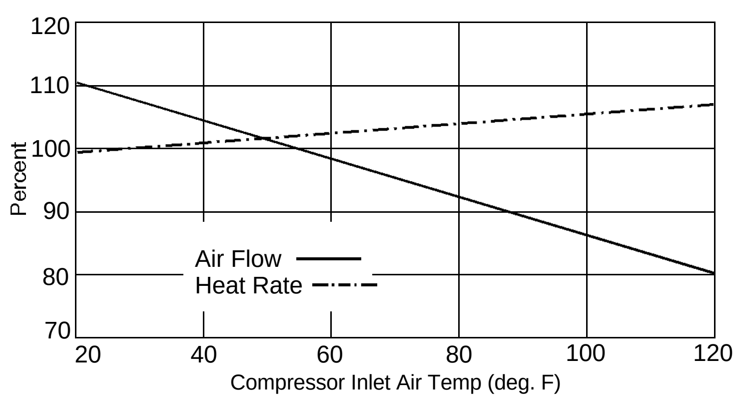

Figure 1: Effect of Compressor Inlet Air Temp on CT Performance

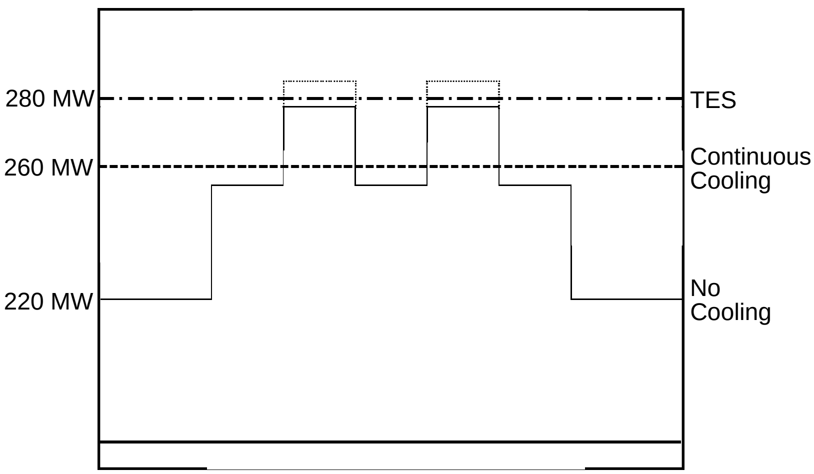

Figure 2: Load profile of a power plant where TES is highly applicable.

Figure 2 shows a good application of a TES system [4]. In this case, maximum possible capacity enhancement is needed for 6 hours a day – 3 hours in the morning and 3 hours in the afternoon, while the demand decreases significantly during the off-peak hours. Some of the excess turbine capacity not being utilized during the off-peak hours can be used to build an ice reserve in a storage tank. This ice reserve can then be used during the peak hours to cool inlet air and provide capacity enhancement for 6 hours a day when demand for power is highest.

The main objective of this paper is to discuss an actual inlet air cooling system utilizing TES application to enhance peak turbine capacity. The TES system was installed at the Wisconsin Electric Power Company's (WE) combustion turbines located in Concord and Paris.

FEASABILITY STUDY

Wisconsin Electric operates ABB 11 N1 combustion turbines at the Concord and Paris generating stations in Wisconsin. Each station has four CTs running in simple cycle mode. The combined rated capacity of the two stations is 618.4 MW. WE was interested in evaluating various inlet air cooling options to optimize the capacity increment for their CTs.

The inlet housing of the ABB 11 N1 turbine is three-sided as shown in Figure 3. Air enters horizontally through the three sides to the inlet duct which consists of the inlet silencers and the elbow. The inlet plenum is connected to the bottom of the inlet duct through which air is directed horizontally to the compressor.

Fern Engineering in conjunction with Electric Power Research Institute (EPRI) performed the feasibility study. The main options evaluated were direct spray system and inlet chilling.

The direct spray system proposed consisted of arrays of spray nozzles with large droplet eliminator (LDE). The system would be installed downstream of final filters. The arrays would be arranged in a manner that would ensure uniform distribution of injected flow in the air stream. See Figure 4 on page 3 for location of the spray system. The water delivery system consists of all stainless steel components and pump skid to handle demineralized water. The water transfer system consisting of arrays, nozzles and interconnecting piping would deliver water at a pressure of 3,000 psi. The high pressure water was selected to minimize the droplet size. Carryover of water droplets to the compressor inlet is a concern and is to be avoided by all means. Small droplet sizes are desirable because they improve evaporation rate, reduce droplet agglomeration, and thereby, reduce the possibility of water carryover to the compressor intake. The total capacity increase with the direct spray was estimated to be 60.1 MW for the two plants.

The other option proposed was inlet air chilling through thermal energy storage. The system consisted of a chiller to

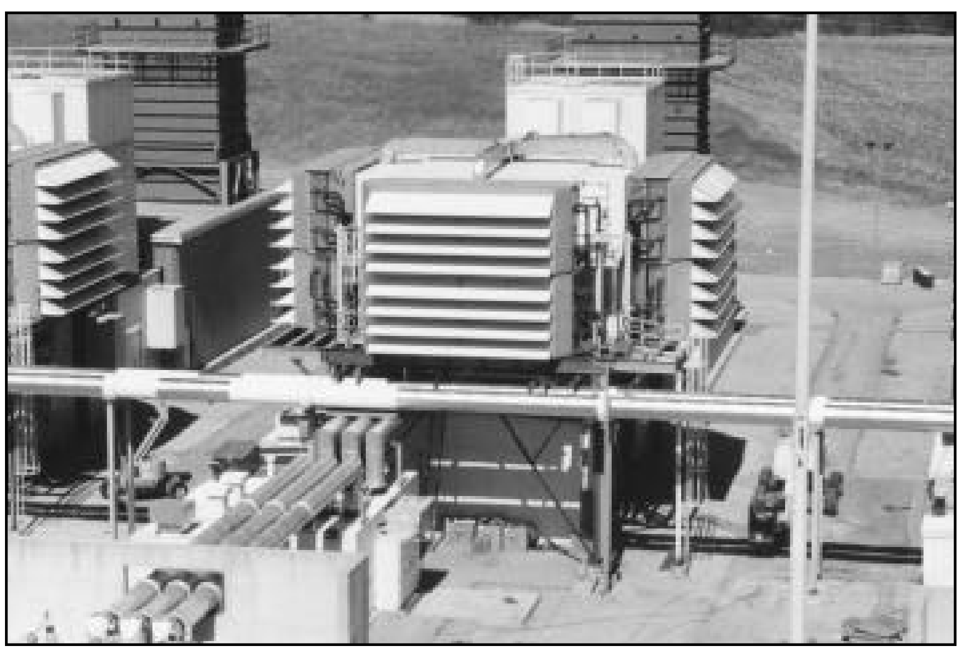

Figure 3: Three-sided filter housing of ABB 11N1.

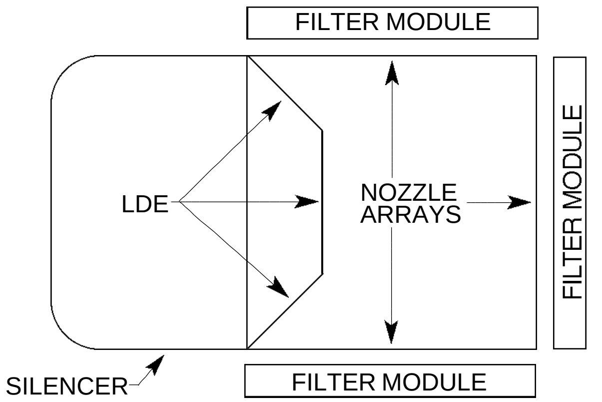

Figure 4: Proposed location of the nozzle arrays and large droplet mist eliminator for fogging turbine inlet air.

produce chilled water, a tank for storing chilled water and a chilled water distribution system to circulate chilled water through the cooling coils and back to the storage tank. The system was capable of cooling the inlet air from the design conditions to 50ºF for 4 hours each day, 5 days a week. The total capacity increase with inlet chilling was estimated to be 132.5 MW.

Although the power gain with direct spray system was more cost effective on a unit basis, inlet chilling was selected since WE was interested in greater power increases than possible with a direct spray system. The initial study focused on using chilled water as the cold storage media which cooled inlet air to 50ºF. Even greater increases in capacity enhancement are possible if ice is used for storage since chilled water at 32ºF can cool the inlet air to 40-42ºF. Subsequent discussions with WE led to the selection of a TES system using ice for turbine capacity enhancement.

THERMAL ENERGY STORAGE (TES) SYSTEM

A typical schematic of a TES system using ice is shown in Figure 5. The system consists of a vapor compression cycle to build ice, a chilled water loop to provide inlet air cooling, and controls and instrumentation for proper system operation. During the off-peak hours, ice is made with the ice harvesters located at the top of the tank. The tank stores the ice. The ice making system consists of screw compressor, condenser, low pressure (LP) receiver, expansion valve, high pressure (HP) receiver, refrigerant pump and ice harvestors. The chilled water system consists of storage tank, cooling coils, chilled water circulation pump and interconnecting piping. Not all the water is converted into ice in the tank. Some water remains in the tank for initial circulation. During peak hours, water from the bottom of the tank is circulated through the coils with the help of chilled water pumps. Since there is ice in the tank, chilled water being circulated

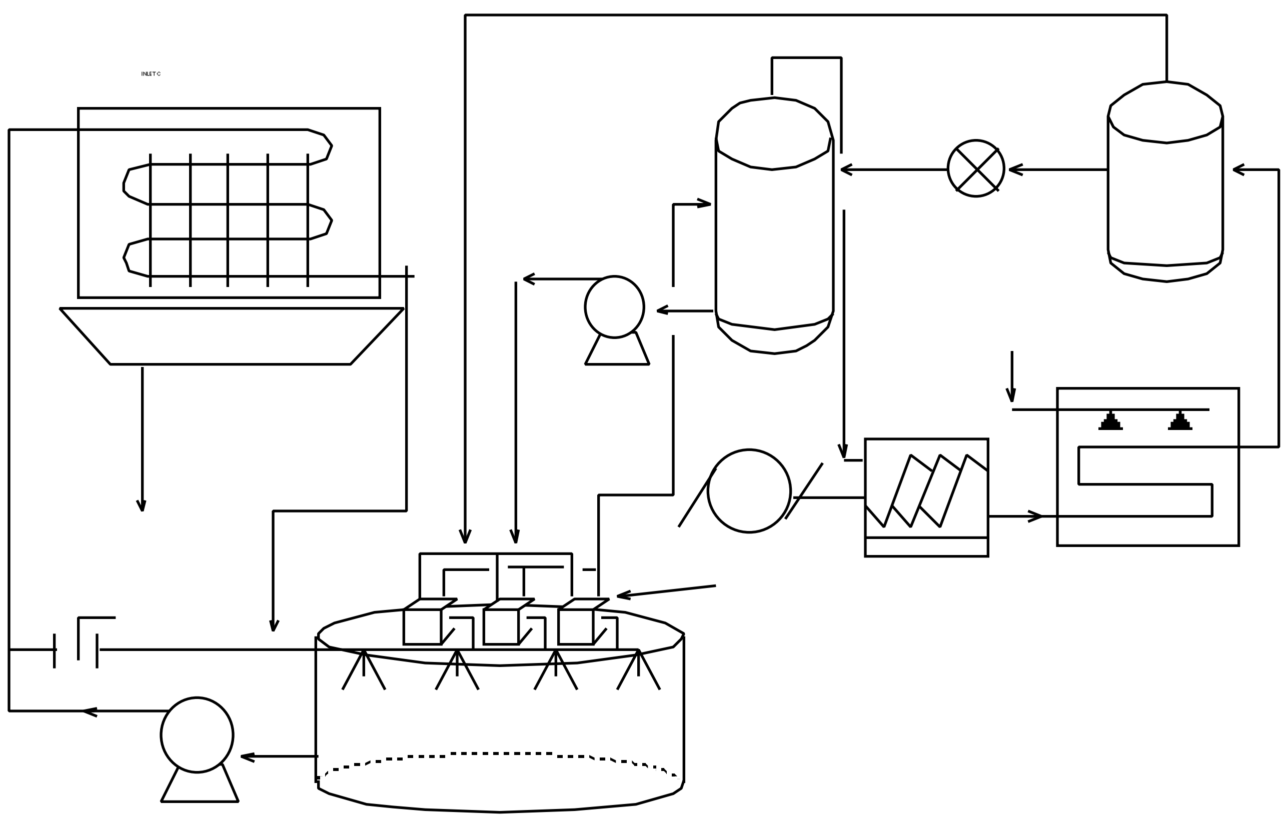

Figure 5: Schematic of a typical turbine inlet air cooling system with Thermal Energy Storage.

through the coils is at or near the freezing temperature. Because of low chilled water temperatures, it is possible to cool the inlet air to the 40-42ºF range. To prevent formation of ice at the compressor inlet, it is not recommended to cool the inlet air below this temperature.

Warm water from the coil outlet is sent back to the tank through a well distributed spray system at the tank top. Even distribution of return water is important to help uniform ice melting and to prevent short circuiting of warm water to the suction line. If warm water is allowed to circulate instead of chilled water, it degrades the heat transfer characteristics and the ability to provide the required cooling of air, thus reducing the capacity enhancement.

Since the air is cooled below the dew point temperature, water vapor in the air condenses. The condensate has sufficient cooling capacity. It can be recovered and used as make-up water for the cooling tower or for some other useful cooling purpose.

DESIGN INFORMATION

The preferred duty cycle was 4 hours a day, 5 days a week. If the TES system were used, the system would operate on a weekly cycle of 20 hours a week with the recharge of storage tank during the night and on the weekends.

T dry bulb = 95ºF
T wet bulb = 74ºF

T inlet air = 42ºF
Volumetric flow rate = 538,200 cfm @ ISO conditions of 59ºF, 60% RH and sea level.

Elevation = 850 ft. (Concord)
= 780 ft. (Paris)

Table 1 shows the properties of various refrigerants [8]

Refrigerant	R-717	R-22	R-502
Latent Heat (Btu/lb)	565.0	93.2	68.9
Mass flow rate reqd. to produce 1 ton of refrigeration (lb/min-ton)	0.422	2.33	4.33
Displacement (CFM/ton)	3.41	3.55	3.61
COP at standard conditions	4.76	4.66	4.37
Theoretical power reqd. (BHP/ton)	0.989	1.011	1.079
Cost per lb.($)	0.59	3.20	5.24

SYSTEM DESCRIPTION

Based on the above design conditions, the CTIAC system for the Concord and Paris Generating Stations consist of the following: two (2) 301,592 ft^3 (2,256,000 gallon), above ground all steel ice storage tanks; each tank structurally supports four (4) 325 ton capacity ice harvesters on the roof, a 2,600 ton refrigeration plant utilizing four (4) twin screw compressors with 850 ton capacity each for the ice harvesters; six (6) cooling coils arranged around the inlet structure of each turbine and four (4) evaporative condensers with heat rejection capacity of 1,060 tons.

Refrigeration System

The 2,600 ton refrigeration plant is an ammonia-based liquid overfeed system divided into two subsystems. Each subsystem consists of two twin screw compressors, four ice harvesters mounted on the roof of each tank, a two-cell evaporative condenser, one economizer section, one high and one low pressure receiver, two recirculation pumps and one storage tank. The liquid overfeed system supplies more refrigerant to the ice harvesters than is evaporated allowing a better heat transfer between the refrigerant and water. A ratio of 3.5:1 is used for this application.

Selection of Refrigerant

The main factors that influence the selection of a refrigerant are performance, cost, safety, reliability and environmental acceptability [6]. Refrigeration capacity and efficiency determine the performance of the refrigerant. The refrigeration capacity is defined as the cooling effect provided by a given volumetric flow rate of refrigerant. The higher the refrigeration capacity, the less is the refrigerant required to provide the cooling.

Chlorofluorocarbons (CFCs) and hydrochlorofluorocarbons (HCFCs) are being phased out because of their impact on depleting the ozone layer. Hydroflourocarbons (HFCs) are being used as near- and intermediate range refrigerant but would ultimately be unsuitable because of their high Global Warming Potential (GWP) [7]. Because of the environmental issues, there is an increase in interest toward natural refrigerants like ammonia, carbon dioxide, hydrocarbons, etc.

Ammonia was selected as the refrigerant for the refrigeration cycle because of its low cost, higher efficiency, non-polluting properties, and ease of use. Ammonia is considered highly efficient for reciprocating and rotary screw compressors in the medium to low temperature ranges because of small compressor displacement per ton of refrigeration produced, high latent heat and low BHP/ton. Ammonia vessels and piping are smaller due to their high latent heat/low mass flow requirements. In addition, ammonia is hard to ignite, has limited explosive potential and is relatively easy to detect even when present in small quantities due to its distinctive odor.

Ammonia also has physical properties that demand respect. It is an irritant to human tissue which can cause caustic burns to the skin or damage to the mucous membranes of the respiratory tract or eyes. Since it boils at –28ºF under atmospheric conditions, liquid ammonia in contact with the skin can cause frostbite. Special procedures, equipment, and permits are necessary for facilities using ammonia in quantities over the 10,000 pound threshold.

The United States Environmental Protection Agency (EPA) Clean Air Act of 1990 Section 112(r)(7) requires a Risk Management Plan (RMP) to be developed and submitted in accordance with 40CFR Part 68. The RMP primarily deals with the offsite response to an ammonia leak at the facility and is intended to protect the general public. The Occupational Safety and Health Administration (OSHA) also requires a Process Safety Management (PSM) program to be developed for facilities that use ammonia. The PSM requirements are comprehensive and primarily intended to protect workers. Wisconsin Electric contracted with a firm specializing in this work to develop both the RMP and PSM programs.

The Company met with local emergency responders and the county emergency planning coordinators to formulate response plans to be implemented in case of a significant ammonia leak. Since large agricultural and commercial users of ammonia were present in these areas, the officials were already familiar with such response plans. Information was also provided to the general public and facility neighbors during an open house public information meeting held before the project began construction. Internally, procedures were developed and employees trained on the special operating, maintenance, and leak response measures needed for ammonia systems. Prior experience as an energy utility in handling typical power plant chemicals helped to implement these measures.

Compressors

The compressor skid consists of the compressor, an electric motor to drive the unit, an oil separator, a liquid injection oil pump, and the control system. All the major components of the system are mounted on a structural steel base. The base of the skid is bolted directly to a concrete pad on the floor.

There are four compressors per site with two compressors per ice tank. The compressors are twin screws 1250 horsepower with 850 tons at 12ºF saturation suction temperature (SST) & 96ºF saturation condensing temperature (SCT) each. The compressor takes suction from the low pressure receiver (LP) at approximately 25.5 psig and compresses the ammonia vapor to 185 psig. The ammonia vapor is then condensed in the evaporative condensers. The screw compressor package is self-contained, with the ability to control the vapor pressure and temperature to the set points.

The oil cooling method for these compressors is liquid ammonia injection into the discharge gas line [9]. The liquid refrigerant boils at the same pressure as the discharge pressure of the compressor. As the liquid refrigerant evaporates, it cools the discharge gas and the oil to the same temperature. This type of cooling is referred to as V-Plus oil cooling. A variable frequency drive runs the liquid injection pump. This method of injection saves more pump horsepower compared to a conventional injection system, which injects refrigerant at some point in the compression cycle, since the vapor does not need to be recompressed.

Recirculation System

The liquid recirculation system consists of a 72" OD x 18' vertical LP receiver, level controls and liquid pumps. The liquid/vapor that is returned from the evaporators is drained into the LP receiver, where the liquid and vapor are separated. The liquid is pumped back to the evaporators with liquid refrigerant pumps, and the vapor is sent to the compressors. There are two liquid refrigerant pumps - one primary and one stand-by. The capacity of these pumps is 450 gpm at 48 psig pressure of liquid per recirculation package. Each of these hermetically sealed pumps has 30 HP motors. Figure 6 shows the building in-between the two storage tanks housing the refrigeration system.

Figure 6: The refrigeration system is placed inside the building between the ice storage tanks.

There is one recirculation package for two compressors and therefore there are two recirculation packages per site. The liquid level in the LP receiver is maintained by liquid make-up from the high-pressure (HP) receiver. There is one 48" OD x 20' high-pressure receiver per refrigeration package. The HP receiver serves as a storage vessel for condensed liquid from the evaporative condensers and supplies hot gas for the harvest cycle. When required, liquid is taken from the HP receiver and sent to the LP receiver via expansion valves. All flow from the HP receiver is due to pressure differential between the high and low side of the receiver.

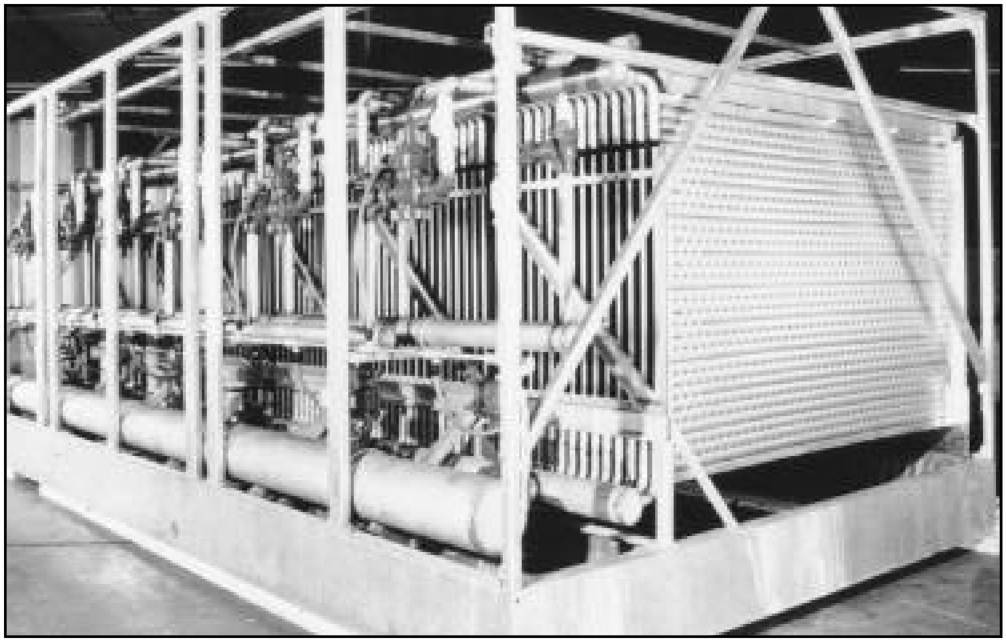

Figure 7: Evaporator section showing the plates for making ice. (courtesy of Vogt Ice)

Evaporators

The evaporator can work either as a chiller or ice maker depending upon the signal sent to the controller. This signal is based on the temperature of water entering the distribution system of the evaporator. In either mode, water is circulated over the evaporator plates via the water distribution system located above each evaporator section. The water is delivered with the help of recirculation pumps. There are two recirculation pumps per site. Each pump is capable of delivering 3,200 gpm at 100 ft. head.

The ice making starts when the water temperature falls below 39ºF. The process consists of two cycles – ice making and ice harvesting. The process starts by circulating water from the bottom of the tank with the help of two evaporative recirculation pumps.

Figure 8: Four enclosed evaporator sections are placed on top of each ice storage.

The ice making cycle is about 14-15 minutes long and the ice harvesting cycle is about 50 seconds. At an interval of approximately 45 seconds, a section of any one of the four evaporators goes into the harvest cycle. Each evaporator has two harvest sections. Each evaporator has 60 stainless steel serpentine pattern plates. Each plate is 4'x 8' approximately. There are four evaporators per ice tank (see Figures 7 & 8). The harvest cycle is discontinued when the evaporator is running in the chiller mode.

Ice is produced on the outside of the plates while liquid refrigerant is circulated on the inside. The liquid refrigerant is pumped into the plates at a rate 3.5 times faster than the evaporation rate. As the water circulates over the plates, it transfers heat to the refrigerant circulating inside the plates and the temperature of water begins to drop. When the temperature of water approaches 32ºF, ice begins to form on the outside of the plates. Ice thickness is approximately 3/8" thick prior to harvesting. Once the ice reaches desired thickness hot gas is introduced into the plates for approximately 50 seconds to raise the plate temperature and harvest the ice.

The source of hot gas is the HP receiver. The temperature of the hot gas is 96ºF. The hot gas circulating inside the plates helps to melt ice on the outside of the plates. As the ice loosens from the plate, it drops into the tank. The hot gas condenses into liquid as it melts the ice. The resulting discharge refrigerant liquid is drained into the LP receiver.

Heat Rejection

The evaporative condenser rejects heat of compression and condenses the vapor ammonia into a liquid. There are two evaporative condensers per site and each section has two cells. Each evaporative condenser cell has two fans driven by a single motor and a single water pump. The water pump supplies water over the exterior of the condenser tubes. As the water is evaporated on the tubes, the heat is rejected to the atmosphere. Each evaporative condenser cell rejects 1050 tons of heat. See Figure 9 for location of the evaporative condenser.

Chilled Water System

The chilled water system consists of ice storage tanks, chilled water pumps, cooling coils, chilled water control valves, return water distribution system for the tanks, and associated electrical and control gear.

Figure 9: Evaporator condensers for refrigeration system heat rejection.

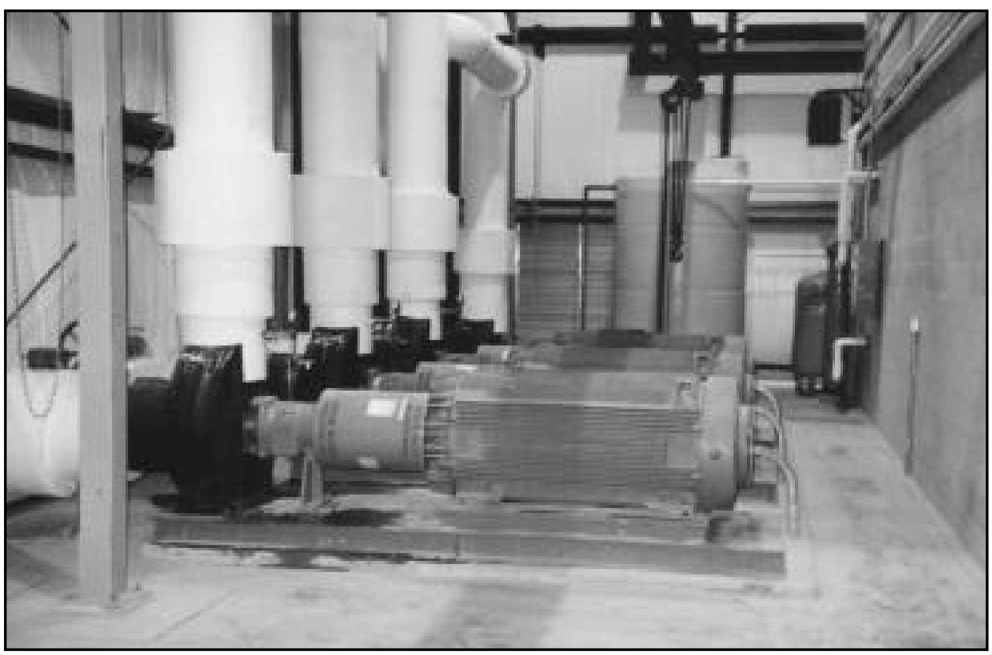

Figure 10: Chilled water pumps to circulate chilled water through the cooling coils.

The inlet filter house is three sided. Six cooling coils are arranged around the inlet structure of each turbine. The coils are designed to maintain the air velocity at 500 fpm or below to prevent water carryover. These coils are piped into common supply and return headers with an isolation valve at each nozzle. The coils are copper tubes with aluminum spiral fins. Drains are provided in the coils to prevent freeze damage during winter. By-pass dampers were installed between the coils and the filters in order to reduce the air-side pressure drop when the cooling system is not in operation. These dampers can be manually opened during system winterization. The anti-icing arrangement was retained on the downstream side of the coils. The exterior of the cooling coils can be cleaned with a hose and nozzle. A mild soap solution can be used and high pressure is not required. The interior of the cooling coils can be cleaned with compressed air.

Drain pans are provided on the leaving side of the air to collect condensate from the air. The condensate is pumped to a 1,100-gallon transfer tank where it is either pumped to the RO system or drained.

Each site includes two 2 1/4 million gallon above ground, all steel ice storage tanks. Four ice harvesters are placed on top of each tank. The tank is well insulated to minimize heat gain from the surroundings. The refrigeration system, with the exception of HP receivers and evaporative condensers, is placed in a building between the two tanks. Five chilled water pumps are provided to circulate chilled water from the bottom of the tanks to the cooling coils and back to the tank. Four pumps are primary and one pump is stand-by (see Figure 10). The chilled water pumps are end-suction types capable of delivering 5,200 gpm at 240 ft. head.

The warm water from the coils is returned to the top of tank and evenly distributed over the ice pile. This is done in order to prevent "short-circuiting" of warm water to the bottom of the tank as mentioned earlier. The return water line is designed to allow even distribution of water between the two tanks. All the chilled water interconnecting piping is insulated to prevent heat gain and formation of condensate on the pipes. A feedback control signal from two flowmeters is used to regulate the control valves for water distribution.

The inlet air temperature is controlled with a signal from the average of the temperature readings taken by RTDs installed at the turbine inlet duct. If inlet air temperature is higher than the set temperature, a control valve on the discharge side of the coils is opened to allow more chilled water flow through the cooling coil bank. The system flexibility allows the operator to change the set point temperature as needed.

EXAMPLES

The performance guarantees based on the design conditions were as follows:

Turbine inlet air temperature = 42ºF
Air-side pressure drop = 1.0 inwc

Auxiliary power consumption
On-peak load = 1,250 kW
Off-peak load = 4,100 kW

In addition, we had to prove that the system is capable of providing cooling for 20 hours a week.

Although the performance testing has not been done so far, the cooling system was utilized throughout this summer. Preliminary indications are that the performance of the system is as expected.

CONCLUSIONS

A CTIAC system was designed to cool the inlet air from a dry bulb temperature of 95ºF and a wet bulb temperature of 74ºF to 42ºF for 20 hours a week with the recharge of the two ice storage tanks during the nights and weekends. With this system, Wisconsin Electric was able to realize a total gain of 112 MW in rated power. However, since ratings are based on 81ºF ambient and the TES continues to cool the inlet air to 42ºF even as ambient temperature goes higher, actual benefits are even greater. The system went on line in June, 1999 just before the start of the summer peak season. During July 1999, WE set electrical generation records six times over 11 days to meet customer demand during a hot spell. The turbine capacity enhancement was instrumental in helping Wisconsin Electric to better handle increased demand during this year's hot and prolonged summer.

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