Inlet Air Cooling for a Frame 7EA based Combined Cycle Power Plant

by

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ABSTRACT

Refrigerated inlet air cooling is one of the most effective ways to increase the capacity of combustion turbines (CTs) during high ambient temperatures, yet the application has not received wide acceptance for large industrial type turbines. There are more than one hundred installations in which inlet air chilling is successfully applied to aeroderivatives, but only a handful of instances for large industrial type combustion turbines. The benefits are equally applicable to these machines as they are to aeroderivatives. The capacity enhancement will be even more cost effective for the new generation of CTs which are fired to higher temperatures and use less air per kW produced. The objective of this paper is to discuss the effects of chilling the inlet air for a GE Frame 7EA CT. An inlet air cooling system was installed at the Cogen Technologies' Camden Cogen facility during the spring of 1997. The cooling system consists of the latest chiller technology using non-ozone-depleting refrigerant R134a. This paper will address how the turbine capacity could be cost effectively increased during the summer months when demand is usually the highest. Several issues including the feasibility study, system description and its performance at various temperature/humidity conditions will be discussed in this paper. Field data shall be used to illustrate the benefits of inlet air cooling for turbine capacity enhancement.

INTRODUCTION

Turbine output is proportional to the air mass flow rate. At higher ambient temperatures, combustion turbines lose capacity (see Figure 1) since the mass flow rate of air decreases as the air becomes less dense. This degradation in CT output at higher temperatures can be avoided by cooling the inlet air to its compressor. In addition, when the CT is running in combined cycle mode, increased mass flow through the Heat Recovery Steam Generator (HRSG) increases steam production, which enhances steam turbine output in the steam bottoming cycle.

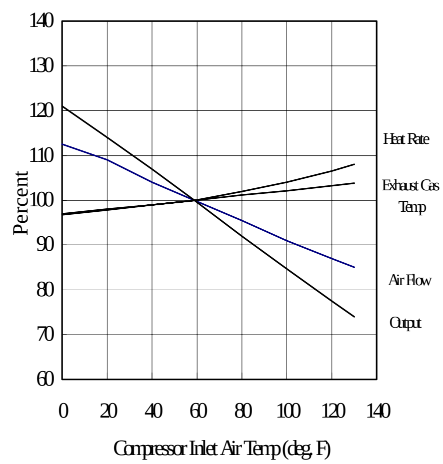

Figure 1: Effect of Compressor Inlet Air Temp on GT Performance

The advantages of increasing the capacity of gas turbines through inlet air cooling are as follows:

- Maximize/optimize the performance of an existing installation.
- Equipment uses relatively small space as compared to additional CTs.
- No new environmental permits may be required to upgrade the performance of an existing turbine. If a permit is required for a particular refrigerant, alternate refrigerants may be used.
- The refrigeration system maintenance requirements are significantly lower than a CT.
- Environmentally friendly (emissions/kWh decrease).
- Ease of operation – proven technology.

INLET AIR COOLING OPTIONS

There are two basic types of inlet cooling - evaporative cooling and refrigerated cooling. With evaporative cooling the water is brought in contact with the incoming air. The water evaporates as it absorbs heat from the incoming air, thereby reducing the dry bulb temperature. These systems work well in relatively drier climates since they can cool the air to near wet bulb temperature. They are of limited use in regions with high humidity.

Refrigerated cooling is utilized when cooling below the wet bulb temperature is desired. This cooling can be provided by continuous means or by using thermal storage. Continuous cooling provides instantaneous cooling and can therefore deliver uninterrupted power enhancement. This cooling is particularly effective when capacity increment for more than 6-8 hours a day is required. There are several ways of providing continuous cooling - direct refrigeration, mechanical chillers, steam absorption chillers, etc.

In a direct refrigeration system, the refrigerant is circulated through the cooling coils. Typically, the system consists of an ammonia liquid overfeed, closed loop system which is driven by screw compressor(s). The coil arrangement can be single or multi-stage. Multi-stage coils – though thermodynamically more efficient – necessitate both increased capital and space. An electric motor or gas engine typically drives the compressor for this system. Refrigeration systems using natural gas driven compressors result in lower power consumption than electrically driven ones, even though some power is still required for auxiliaries. This results in increased net power output over that possible with conventional electro-mechanical compressors. However, gas driven compressors usually have higher initial cost, a larger footprint, and demand more maintenance than electric compressors. Therefore, gas driven compressors are recommended when parasitic power consumption cannot be spared.

The use of mechanical chillers has been successfully applied to a number of installations. Chillers come in standard packages and may therefore require less space than a direct refrigeration system. For sites where permitting of ammonia is a concern and heat source for an absorption system is not available, this option is a cost effective way to increase capacity through inlet air cooling. Chillers are proven technology that has been used for comfort cooling, process cooling, etc. for over fifty years. See Figure 2 for a schematic of a typical chiller-based continuous cooling system.

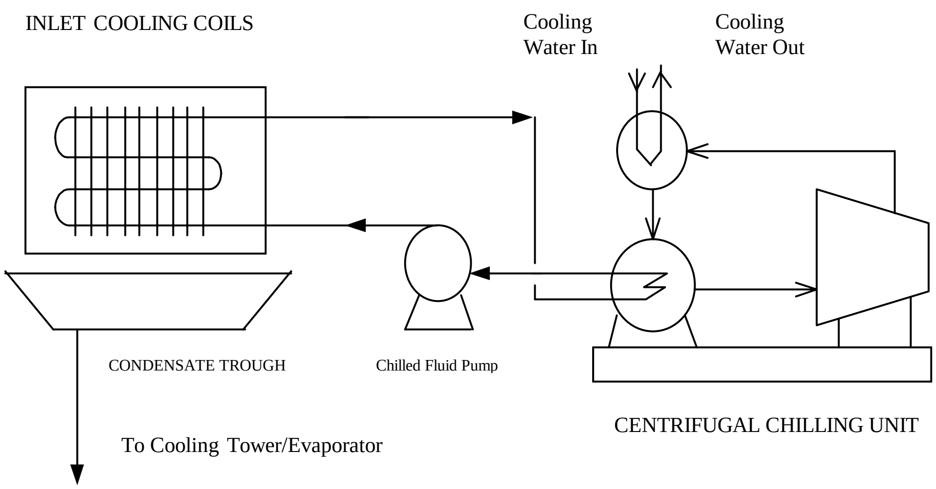

Figure 2: Schematic of a Chiller System for turbine inlet air cooling.

Absorption refrigeration, especially that using lithium bromide, has been successfully applied to provide cooling on many existing installations for a number of years. This system can cool the inlet air to about 50°F with the cooling effect being produced using very few moving parts. The heat source for the generator can be gas, steam, hot water or turbine exhaust gases. The choice of a particular heat source depends upon its availability. It is to be noted that although a heat source spares the use of electric power for a compressor, electric power is still needed to drive the pumps, condenser fans and other auxiliaries.

Energy from a turbine's exhaust is significant and for turbines operating in simple cycle mode, this energy is wasted to the atmosphere. It would be beneficial if this energy could be utilized for some useful purpose. The turbine exhaust gases could be directly used as a heat source for the absorption system's generator or they can be used to produce steam or hot water in a heat exchanger at the turbine exhaust. Some absorption system suppliers recommend exhaust gas temperatures of 1200°F or more for its successful operation. Care should be taken when steam or hot water usage requires the addition of a heat exchanger or heat transfer sections in HRSGs since additional pressure drop occurs through these sections. This pressure drop can be significant in combined cycle units if the sections are installed as retrofits, since the overall plant performance may not be optimized.

Cooling with thermal energy storage (TES) is most economically employed to maximize peak capacity by utilizing energy available during the off-peak hours to build a cold energy reserve. This cold reserve is used during the peak hours to cool the inlet air and thereby increase the CT capacity (see Figure 3). Ice, water, or other heat transfer fluids can be used as the energy storage media. Since the cold reserve is built over a greater number of hours than the cooling provided, the size of the refrigeration system is considerably reduced. The main advantage of a TES system is that almost the entire gain in the power output during the peak hours is available as dispatchable power. Only power to pump the heat transfer fluid through the cooling coils is required.

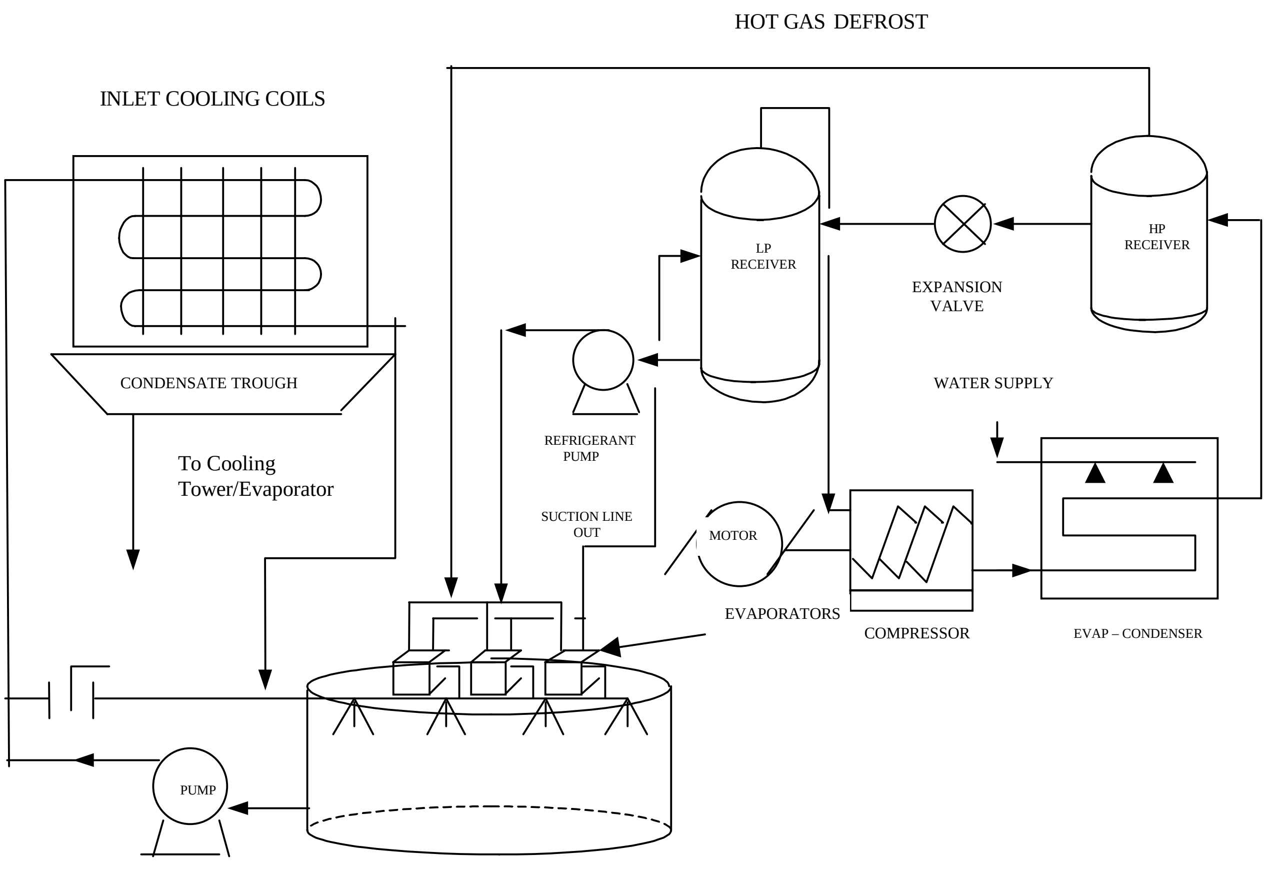

Figure 3: Schematic of a typical turbine inlet air cooling system with Thermal Energy Storage.

DESIGN PARAMETERS

The main parameters that influence the suitability of an inlet air cooling system are as follows:

- Ambient temperature
- Air flow to turbine output ratio
- Slope of turbine performance curve
- Hours of operation
- Space consideration for new CTs

The wet bulb temperature has a significant impact on the size and cost of a cooling system. Care should be taken in specifying the design wet bulb temperature. Normally, the design ambient conditions are defined in terms of dry bulb temperature and relative humidity. When the combination of these parameters is not chosen properly, the resulting system may be overdesigned. This overdesign may change the feasibility of an inlet cooling system. It is recommended to calculate the wet bulb temperature from the dry bulb temperature and relative humidity, and to compare the value to those specified in the ASHRAE or other handbooks. If the calculated value is higher, lower the design relative humidity such that the wet bulb temperature corresponds to a more realistic number. The design conditions should be specified in terms of dry and wet bulb temperatures to avoid confusion.

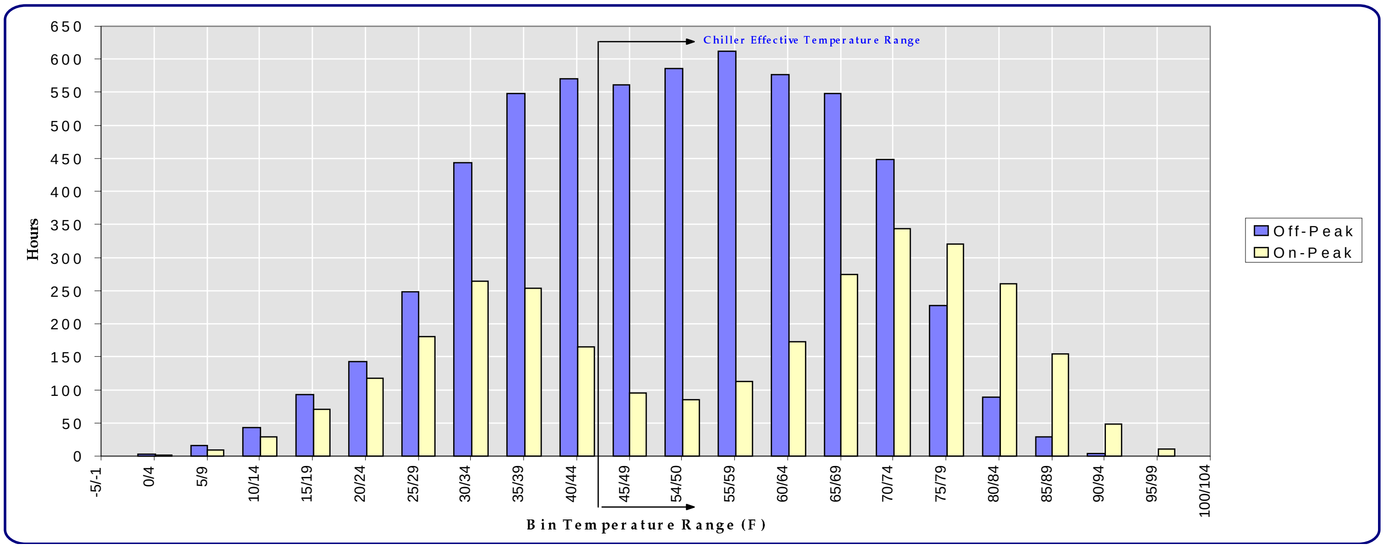

Figure 4: No. of hours in a year vs. 5°F temperature bins.

Another important criterion is the airflow to kW ratio. The cooling load is directly proportional to the airflow. The lower the ratio, the more effective inlet cooling is. This ratio is normally expressed as lb/hr-kW. For a ratio less than 30, the cooling option is very cost effective; for a ratio between 30 and 35, the cost effectiveness is moderate and for a ratio higher than 35, the cost effectiveness diminishes. The newer, more efficient gas turbines have low air flow to kW ratios and therefore provide considerable capacity enhancement when inlet air temperature is reduced. Another important consideration while looking at the airflow to kW ratio is the size of the turbine. The installed cost in $/ton of cooling capacity goes down as the size of the system goes up. Therefore, it is possible that larger units with higher ratios may prove to be more cost effective than smaller units with a lower ratio.

The slope of the turbine performance curve as a function of its inlet air temperature describes the capacity enhancement that can be achieved by cooling inlet air. The steeper the curve, the more the output per degree reduction in inlet air temperature. Care should be taken to evaluate output enhancement because the output may be limited by turbine shaft horsepower or generator kVA rating.

If more than 8 hours of cooling is needed, continuous cooling will most likely be a more cost effective option. The reason is that the size of a TES system is directly proportional to the hours of operation. Even though continuous cooling requires larger refrigeration capacity than a TES system, the cost is approximately the same. This is due to savings in refrigeration system being offset by storage tank cost. The economics of on-peak and off-peak rates have to be determined in order to judge the suitability of one system over the other.

Space for additional CTs may not be readily available and peak capacity may be the primary need for adding capacity. In that instance inlet air cooling should be considered as a viable alternative to new CTs.

ECONOMIC ANALYSIS

A cost benefit analysis is required to determine the economic feasibility of an inlet cooling application. The economic evaluation criteria will most likely vary from one power producer to the other. From some it may be revenues from enhanced capacity. For others, it may be the bonus for meeting or exceeding capacity, avoiding any penalties for not meeting capacity, or improving heat rate. Payback and net present value over the life of the equipment are two important factors to judge the economic feasibility of a project.

Major factors that influence the economics of a project are as follows:

- installation costs
- maintenance
- rate structure (on-peak, off-peak)
- fuel costs
- revenues
  - additional capacity
  - bonus for meeting or exceeding capacity

There is an improvement in heat rate with a decrease in CT inlet air temperature. Whether there is a net improvement in heat rate depends upon the type of turbine, mode of operation (simple or combined cycle), and the power consumption by the cooling system. For a majority of CTs operating in simple cycle mode, there is a slight improvement in heat rate. For combined cycles, the improvement in heat rate is usually offset by the auxiliary power consumption of the cooling system.

FEASIBILITY STUDY

An initial screening study was performed to review the suitability of the various inlet cooling methods for the plant. Available space for system installation was a concern. Systems reviewed included evaporative cooling, electric motor driven chillers with environmentally safe refrigerants (mechanical chillers), absorption chillers using excess or waste steam or natural gas for heat, direct ammonia refrigeration chilling and thermal energy storage using ice. Results of this phase of study determined that evaporative cooling and mechanical chillers were the most cost effective options and justified further investigation.

The following parameters were included in the analysis:

- On-peak and off-peak revenues per kWh.
- Plant availability of 97% on-peak and 90% off-peak.
- A design life of 20 years.
- Weather data compiled by the U.S. Air Force, tabulating monthly hours in temperature bins of 5 degrees, dry bulb with coincident wet bulb.

Many chilled water systems are designed to produce 42^{o}F to 45^{o}F chilled water to supply process and comfort cooling loads. Cooling gas turbine inlet air to within 10^{o}F of freezing could potentially form ice at the inlet bell. Therefore, the design point for the cooled inlet air was set at 48^{o}F to accommodate a reasonably efficient chiller system and maintain inlet air temperature setpoint a few degrees above the design minimum. This value was used as the lower limit for economic evaluation purposes for both evaporative cooling and chilled water systems.

For the evaporative cooling option, the use of evaporative media as well as fogging systems were evaluated. These systems are typically capable of cooling inlet air to within 85-90% of saturation. For this location, evaporative cooling would therefore only provide effective cooling to the inlet air stream when the ambient dry bulb temperature is above 75^{o}F, thereby limiting the benefit of the capacity increase.

Weather data for the nearby McGuire Air Force Base in New Jersey was compiled in temperature bins of 5°F increments. The number of hours per month that the dry bulb temperature occurs in each temperature bin was also determined. The coincident wet bulb temperature was then averaged for each temperature bin. Grouping the data in this manner provided a convenient and effective method of evaluating the benefit of reducing the inlet air temperature. A plot of the number of hours in each 5°F temperature bin resembles a normal distribution curve and is shown in Figure 4.

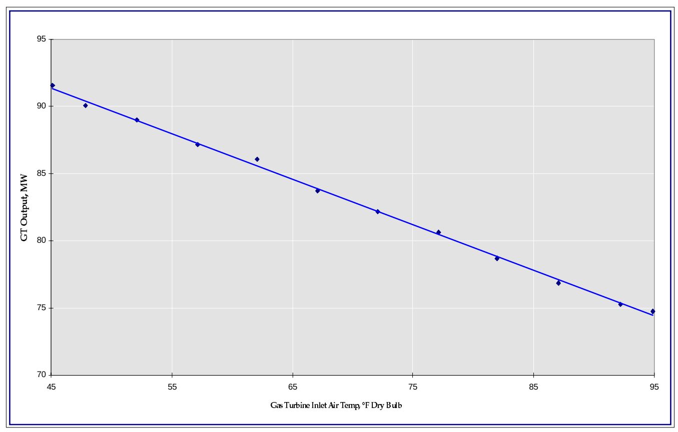

Figure 5: Gross GT Output vs. Inlet Air Temperature

In order to determine the expected increase in gross electric power, kW, from the actual combustion turbine, plant data was obtained for a full year, tabulated and reduced to obtain a curve of combustion turbine output, in kW, versus turbine inlet air temperature (see Figure 5). The benefits were calculated for each of the four periods of the year during on-peak as well as off-peak hours. The cooling load, in refrigeration tons, for each temperature bin was determined by using the difference in moist air enthalpy, Btu/lb, of the ambient air and chilled air (48°F) multiplied by the estimated turbine inlet air flow. When combined with the number of hours of on-peak and off-peak operation in each temperature bin, the cooling load in ton-hours was produced for a given period. Calculation of the combustion turbine power increment was based on actual gas turbine performance data. Although chiller parasitic load will vary with the cooling capacity of the machine and its design, it was estimated to be 0.65 kW per ton of refrigeration for the purpose of evaluation of various system capacities. The power requirements for the chilled water and cooling water pumps were then included separately in the model for each chilled water case.

Effects of the steam turbine were also modeled in the optimization portion of the study to account for the increased exhaust flow from the gas turbine to the HRSG. This value is small but positive, depending upon the temperature bin. It should be noted that the study also assumed that high pressure steam production of the HRSG was maximized via duct burning. This assumption was not related to the inlet air cooling system benefits. The additional pressure drop due to the cooling coils, prefilters or evaporative media was also taken into consideration. For the analysis, a conservative additional pressure drop of 1.5 in w.c. was used for the cooling coils when used with a chiller. For the evaporative media, an inlet loss of 0.5 in w.c. was used. Other factors considered in the analysis included additional fuel consumption to produce additional power, increased maintenance costs and water and chemical costs required to support higher CT NOx injection rates. The CT manufacturer was consulted to determine if the generator cooling system was adequate during summer months to handle the additional power. The present generator system was found to be capable of handling the load.

The final optimization phase of the study focused on the chiller system capacity. The evaporative cooling system did not require optimization. Four cooling system capacities were evaluated: 1000, 2000, 3000, and 4000 tons. The additional generation capacity (kWh) and expected revenues were calculated for each system. Based on average ambient temperature, the maximum cooling load of 4,000 tons was determined. This load, however, was required for less than 3% of the total hours in a year. The study determined that the number of hours of operation for a 4,000 ton system was not cost effective due to the higher first cost and reduced efficiency at part load operation.

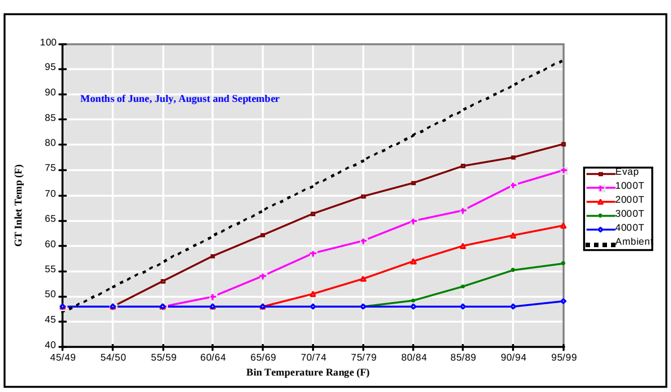

Figure 6: GT Inlet Temperature vs. Bin Temperature Range

Combustion turbine inlet air temperature with inlet cooling versus ambient temperature is shown in Figure 6 for each chiller system investigated as well as the evaporative cooling system. The corresponding kW increase is shown in Figure 7. A net present value (NPV) analysis was made for on-peak and off-peak kWh production of the various chiller system sizes in each temperature bin. The calculation included capital costs for equipment, annual net power generation revenues, and annual operating and maintenance costs. The 2000 ton inlet air cooling system was found to be most cost effective system for the Camden plant. The NPV for the 2000 ton chiller was more attractive than that for the evaporative coolers. Therefore, the 2000 ton inlet cooling system was selected for detailed design.

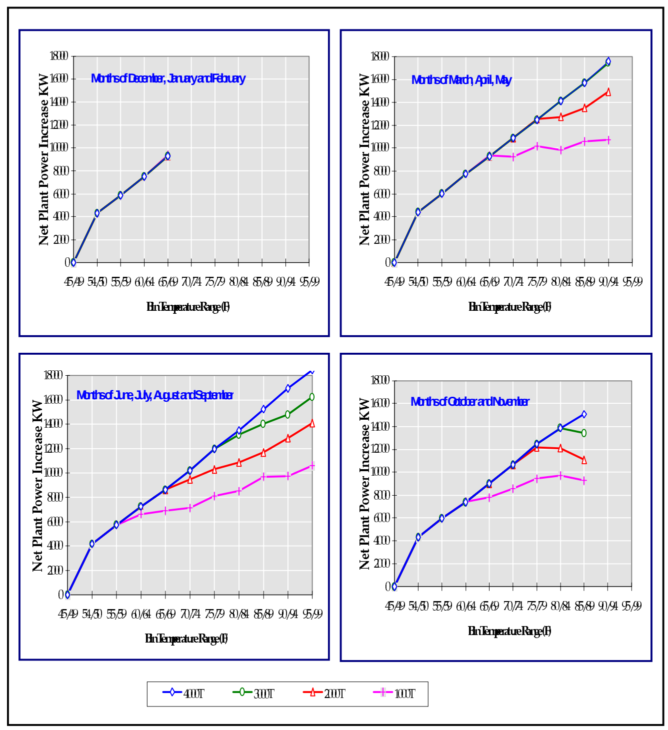

Figure 7: Net Plant Power Increase

PROJECT DESCRIPTION

The 165 MW nameplate rated combined cycle power plant located in Camden, New Jersey, consists of one General Electric Frame 7EA gas turbine, one General Electric auto extracting condensing steam turbine, one dual pressure heat recovery steam generator, one multi-cell mechanical draft cooling tower, and balance of plant equipment. An annual capacity factor of 91% was used for modeling purposes. Steam is supplied to an adjacent paper-making facility and electric power is supplied to Public Service Electric and Gas Corporation.

The project consisted of a complete inlet air cooling system and included a mechanical chiller, cooling coils and their installation, chilled glycol/water pumps and piping, condenser cooling water pumps, piping, electrical and mechanical tie-ins to existing systems. The cooling system was required to meet the following performance criteria:

Combustion turbine inlet air flow	2,353,551 lbs/hr
Ambient dry bulb/wet bulb temperature	72°/66°F
Desired inlet air temperature	50°F
Allowable airside pressure drop	1.0 inwc

No appreciable increase in noise at plant boundary.

Note: An additional requirement to cool from dry/wet bulb temperature of 97°/76°F to 64°F was required to address those hottest days during summer beyond design.

Refrigeration System

The chiller chosen was a hermetically sealed industrial grade machine, which uses R-134a, non-ozone-depleting refrigerant. The chilled fluid circulated to the cooling coils is an ethylene glycol water mixture. The solution is 35% glycol and 65% water by volume. This concentration was chosen to allow the system to have burst protection down to –100°F while not being severely penalized with poor heat transfer characteristics.

The chiller package consisted of a two-pass shell and tube condenser and evaporator, economizer and pump out vessel with a self-contained pump-out unit. The copper tubes in the condenser and evaporator are enhanced with rifling and fins. The condenser and evaporator have marine water boxes with hinged covers to allow for easy maintenance. The overall dimensions of the chiller package are 10'-6" wide by 12'-5" tall and 21'-11 3/4" long. Chiller sound level was 90 dBA at 3 feet. Chiller motor is rated at 2000 horsepower and uses approximately 1950 horsepower. The main motor starter is an auto-transformer type with a 66% tap.

The chiller utilizes an economizer section, which allows liquid to be sub-cooled before being fed to the evaporator and vapor to be fed to the compressor at an intermediate pressure. This is done to improve cycle efficiency. The lubrication system is force fed, which uses a two horsepower pump to force oil into the bearings of the centrifugal compressor.

Fluid Systems

Pumps for the circulation of chilled fluid to the cooling coils are end suction centrifugal with 350 HP motors. The brake horsepower at design condition is 302. One pump is primary and one is for stand-by. The chilled fluid system is a pressurized closed loop, with a total dynamic head of 150 feet. The pumps have an efficiency of 83% and the motors have an efficiency of 94.5%. The pumps are 480 volt three phase and are started across the line.

Pumps used for condenser cooling water are vertical turbine pumps installed over the existing plant cooling tower sump. A three-section screen grid was added at the basin to sump entrance, just upstream of these pumps to provide an additional level of protection. The discharge piping runs from the cooling tower, under the plant parking lot and through the turbine hall to the condenser approximately 530 feet away. Return from the chiller condenser was tied into the plant's return line to the cooling tower. Isolation valves were installed at the condenser inlet and outlet and the point of connection back into the plant's cooling water. The total dynamic head for the cooling water system is 116 feet. The pumps have 250 HP motors installed, while using 232.5 BHP at design conditions. The pumps have an efficiency of 86% and the motors have an efficiency of 94.5%. As with the chilled fluid pumps, one primary and one stand-by pump is used. Both these and the glycol pumps are oversized on capacity.

There are two condensate pumps that are located under the inlet structure, a primary and back-up. The pumps are stainless steel 3-1/2 HP end suction centrifugal. They are operated by a float switch in the condensate basin under the cooling coils. The pumps are capable of pumping 80 gallons per minute against an 80 foot head. They take suction from the condensate collection basin. The condensate can be pumped to one of three places: the neutralization tank, the glycol storage tank, or the cooling tower basin. Primary discharge is into the cooling tower basin. Should a glycol leak develop inside the inlet structure, the condensate pump discharge can be lined up to divert glycol flow to the storage tank, thus preventing a glycol spill.

If it becomes necessary to empty the glycol solution from the system, the glycol charge can be pumped to the storage tank with a portable transfer pump. The glycol storage tank is a double containment tank made of reinforced fiberglass. The tank is 72 inch outside diameter by 18 feet long and holds the entire system charge of 4000 gallons.

The controls system of the chiller unit is utilized to control the complete chiller plant. The microprocessor starts/stops pumps and controls the air temperature to a set point of ±2°F. This system allows the chiller to operate at infinite steps between 10 and 100 percent load. It requires about 15 minutes to cool from ambient conditions to the maximum cooling point. In order to allow CT NO_x injection to track CT fuel flow, a 15-minute ramp rate was programmed. The control system also can be monitored remotely for maintenance purposes. The system has several redundant safeties to protect the combustion turbine and plant personnel. One safety is to prevent the chiller system from running if the inlet temperature drops below the freeze protection set point. As an additional safety, a highly sensitive fluid level measurement device is used to alarm and shut the system down should a glycol leak occur.

A new electrical equipment room was required to keep the same number of spare starters in the plant. The new room is inside the chiller enclosure at approximately 100 feet from the existing electrical equipment room. A 600 amp fused load breaking disconnect was placed into the plant's existing electrical equipment room to feed the new electrical equipment room. A 1333 kVA air-cooled 4160-3-60 to 480-3-60 transformer and 600 amp load breaking fused disconnect were installed in the chiller enclosure. The transformer is used to supply power to the chilled fluid pumps, condenser cooling water pumps, condensation pumps, lights, exhaust fans, and controls. A new motor control center (MCC) was installed in the chiller enclosure for all 480-volt loads. The new MCC has a 2000 amp buss with breaker protection set at 1600 amps. Also located in the new electrical room was the chiller auto-transformer starter and disconnect. All electrical equipment, pumps, chiller and building were tied into the existing ground grid.

Chiller Enclosure

In order to meet environmental permit criteria regarding noise contribution at the plant boundary, a 1060 square foot chiller enclosure was used which includes a separate electrical room. The chiller enclosure also protects the chiller, chilled fluid pumps, and electrical equipment. The electrical equipment is separated by a firewall from the rest of the equipment. The chiller enclosure was designed in accordance with the following standards: NFPA, NEC, ASHRAE, and state and local building codes. The exhaust fans and louvers are of the low sound emitting type. The enclosure walls were designed to cut the sound levels by 39 dBA. The chiller enclosure roof was designed to handle a seven foot snow load and 75 mph wind.

Figure 8: Inlet filter housing looking Southeast.

Inlet Structure Modification

The original filter housing was a three level Donaldson TTD Huff-N-Puff Self-Cleaning Air Filtration system (see Figure 8). Placement of cooling coils downstream and upstream of the filter house was evaluated. With coils placed downstream of the filters, the face velocity would exceed accepted limits required to prevent moisture carryover. This problem could have been addressed with the installation of moisture eliminators, but that would have added to the airside pressure drop. With the higher face velocity, a larger pressure drop through the coil was also calculated. Economics on the impact of pressure drop on CT performance indicated unnecessary MW penalty at times of year when the chiller was off. Downstream coil installation required a longer turbine downtime since the filter house would have to be relocated to accommodate a coil section. There was no provision in the existing filter house for installation of coils.

Placement of cooling coils upstream of the filters was found to be a more viable option. The cross-sectional area was selected such that the face velocity of the air stream was below 450-500 fpm which eliminated the need for moisture eliminators. This concept also eliminated the need to relocate the existing filter housing and reduced the dependence on a scheduled outage by allowing fabrication of the housing at a nearby laydown area. Any extension on the turbine outage schedule would mean loss of revenue for Camden Cogen since it is a base load plant. Ten specially designed cooling coils were placed in two banks of five with nozzles towards the center and end turn access on the blind end of the coils. The coil bank was ducted into the existing filter house. All erection and coil placement work was done

Figure 9: Inlet filter housing looking Southeast after the inlet structure modification.

while the turbine was still running. Final welding and primer painting was done during the scheduled outage. Hatch type doors were installed in both the plenum walls and floors, allowing ambient air to by-pass the cooling coils. This was done to eliminate airside pressure drop across the coils in winter when the cooling system was not in operation. Moreover, this upstream concept allows any future repair work on the coils to be accomplished while the turbine is running. Pre-filters were installed in front of the coils. Individual coil condensate drain trays were installed on the downstream side. The completed filter house modification is shown in Figure 9.

Table 1: List of major equipment installed for the project

Item	Quantity	Capacity	Type
Chiller	1	2,000 tons	Centrifugal
Chiller controls	1		Micro-processor based remote monitoring capability
Cooling Coils	10 coils	2,035 tons	Finned tube
Valves and instrumentation	As required	As required	
Major Electrical:			
4160 V disconnects	2	600 amps	Load breaking
4160 V motor starters	1	2,000 HP	Auto-transformer
Transformer (4160V/480 V)	1	1,333 kVA	Air cooled
480 V MCC	1	1,600 amps	CH 2100 series
Glycol pumps	2 (one stand-by)	350 HP	End suction centrifugal
Cooling water pumps	2 (one stand-by)	250 HP	Vertical turbine
Condensate pumps	2 (one stand-by)	3 ½ HP	End suction centrifugal
Glycol storage tank	1	4,000 gallons	Reinforced fiberglass

PERFORMANCE GUARANTEES

The guarantees for the inlet cooling system performance were as follows:

Chiller Capacity:	2,000 tons
Δp across coils and pre-filters	1.05 inwc

Parasitic losses:

Chiller	1,450 kW
Cooling water pumps	220 kW
Chilled water pumps	262 kW

Inlet air temperature control set point ±2°F

A four-hour test was conducted to verify whether the performance of the installed system met the guarantees per the commercial contract. All the instruments used for the test were calibrated within 30 days of the test to meet the test requirements for valid recorded data. The airflow was based on standard manufacturer curves. However, this airflow was verified by two methods: from heat balance around the cooling coils and from turbine exhaust flow.

1. Heat balance around the coils:

$$W_{air} = W_w \times (h_{w2} - h_{w1}) / (h_{a1} - h_{a2}),$$

where

W_{air} = air flow, lb/hr

W_w = glycol/water flow, lb/hr

h_{w2} = enthalpy of glycol/water mixture at coil outlet, Btu/lb

h_{w1} = enthalpy of glycol/water mixture at coil inlet, Btu/lb

h_{a2} = enthalpy of air mixture at coil outlet, Btu/lb
h_{a1} = enthalpy of air mixture at coil inlet, Btu/lb

2. From turbine exhaust flow: The inlet airflow was calculated by subtracting the steam injection and fuel flow from the turbine exhaust flow. All the above quantities were being measured by the existing plant instrumentation.

$$W_{air} = W_{exh} - W_{fuel} - W_{inj\ stm}$$

Where
W_{exh} = turbine exhaust flow (lb/hr)
W_{fuel} = fuel flow (lb/hr) = fuel flow (ft^3/hr) x density (lb/ft^3)
$W_{inj\ stm}$ = injection steam flow (lb/hr)

The ambient conditions during the test were expected to be different from the design conditions. Therefore, corrections need to be applied to the measured values before comparing them to the guarantee values. For example, if the wet bulb temperature during the test was 72°F instead of 66°F, the inlet cooling system will cool the air to a temperature higher than the guarantee value of 50°F. The corrected inlet air temperature as a function of ambient wet bulb temperature is shown in Figure 10. During the test, if the measured inlet air temperature is within 2°F from the set point temperature as determined from Figure 10, then the performance for the turbine inlet air temperature is met. An uncertainty of 2% was allowed for the pressure drop and power consumption, and of ±2°F for the temperature set point to account for measurement and instrument uncertainties.

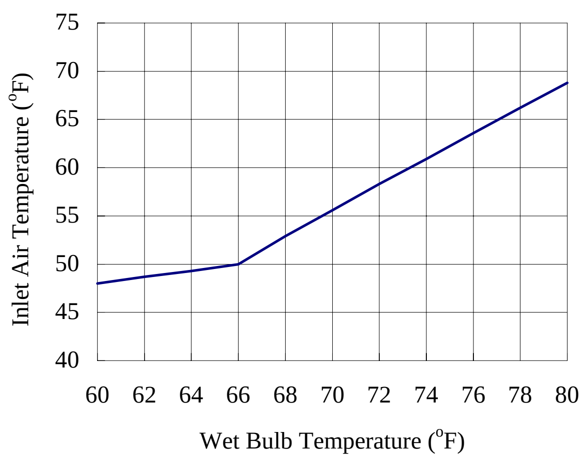

Figure 10: Expected turbine inlet air temperature vs wet bulb temperature.

Similarly, the pressure drop needs to be corrected before comparing it to the guarantee value. The correction factor was read from Figure 11. Electric/power demand analyzers measured the power consumption for the glycol and cooling water pumps. The plant's power analyzer located at the new chiller enclosure main feed measured the entire system's power consumption. In order to meet the guarantee for the power consumption, it was also important to verify that the chiller capacity during the test was 2,000 tons. The chiller capacity was calculated from the glycol/water flow and supply and discharge temperature as follows:

$$CC_{measured} = W_w \times (h_{w2} - h_{w1})/12{,}000.$$

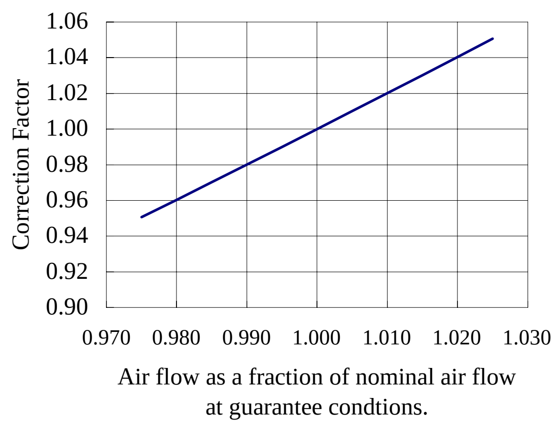

Figure 11: Pressure drop correction factor for air flow.

PERFORMANCE TEST RESULTS

The performance test was conducted on July 15, 1997, between the hours of 2:40 AM and 6:40 AM. During this time, the average ambient conditions were as follows:

Dry bulb temperature	78.7°F
Wet bulb temperature	73.2°F
Barometric pressure	1023 mbar

The other parameters recorded during the test are as follows:

Turbine inlet air temperature	60.1°F
Duct temperature	59.1°F
Condenser cooling water:	
Chiller, in	86.9°F
Chiller, out	97.7°F
Flow rate	6,016 gpm
35% glycol/water mixture:	
Chiller, in	63.8°F
Chiller, out	52.4°F

Flow rate	5,768 gpm

Airside pressure drop:

Prefilters and cooling coils	0.45 inwc
Prefilters only	0.05 inwc

Power consumption:

Entire system	1934.8 kW
Cooling water pump	198.1 kW
Glycol/water pump	262.3 kW

The standard airflow of 2,276,546 lb/hr at the inlet air temperature of 60.1°F was used in all the calculations. It was verified by heat balance around the cooling coils and from measured turbine exhaust flow.

Chiller Capacity

The chiller capacity was measured by various methods: a) based on the measured glycol flow rate, b) based on measured condenser flow rate and c) based on glycol flow rate calculated by heat balance around the cooling coils. The chiller capacity by the three different methods was found to be 2,398 tons, 2,290 tons and 2,102 tons, respectively.

Airside Pressure Drop

The correction of the pressure drop based upon difference in test conditions versus design conditions was read from Figure 11. The airflow during the test was 1.9% less than the guaranteed value and a correction of 1.04 was applied to the pressure drop reading. The corrected pressure drop across the coils and prefilters was found to be 0.42 inwc.

Parasitic Loads

The average parasitic load of 1934.78 kW for the entire chiller system included all motors, fans, controls and lights. The average chiller power consumption was calculated by subtracting the two pumps from the total system power consumption.

Chiller kW = 1934.78 - 262.32 - 198.06 = 1474.4 kW.

The parasitic power of the lights, fans, controls, and condensation pumps is included in this calculation. The guaranteed values were to be met when generating 2000 tons of cooling. During the test, the actual cooling capacity was calculated to be 2102 tons, 5.1% greater than that guaranteed. When corrected to 2000 tons, the chiller parasitic load is 1402 kW (0.70 kW/ton).

Results of corrected values as compared to guaranteed values:

	Guarantee	Actual
Chiller	1450 kW	1402.0 kW
Cooling water pumps	220 kW	198.1 kW
Chilled water pumps	262 kW	262.3 kW

Chiller Set Point: The set point of the chiller was checked prior to the performance test. The chiller performed according to the ±2°F requirement.

CONCLUSION

The concept of cooling the inlet air to increase the capacity of the combustion turbine was successfully applied for the Camden Cogeneration plant. The installed inlet cooling system consisted of a 2,000 ton electric driven chiller using R134a as the refrigerant. The guaranteed total power consumption of the chiller, glycol pump and condenser cooling water pump was 1932 kW for 2000 tons of cooling. The total corrected measured power consumption of 1862.4 kW is 3.6 % less than the guaranteed value. The measured chiller capacity of 2,102 tons exceeded the guarantee requirement of 2,000 tons by 5.1%. The corrected measured pressure drop increase of 0.42 inwc was 60% better than the guarantee value. Therefore, the actual system performance was better than the predicted performance. Combustion turbine performance with inlet air cooling met the expected increment of 7.0 MW at the design ambient conditions.

ACKNOWLEDGEMENTS

The authors wish to thank Richard Brody, John Snell and Robert Licato of Cogen Technologies, John Kraft of CE&E, and Christopher Balance and Allan White of Lockwood Greene for their continued support and assistance in writing this paper.

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