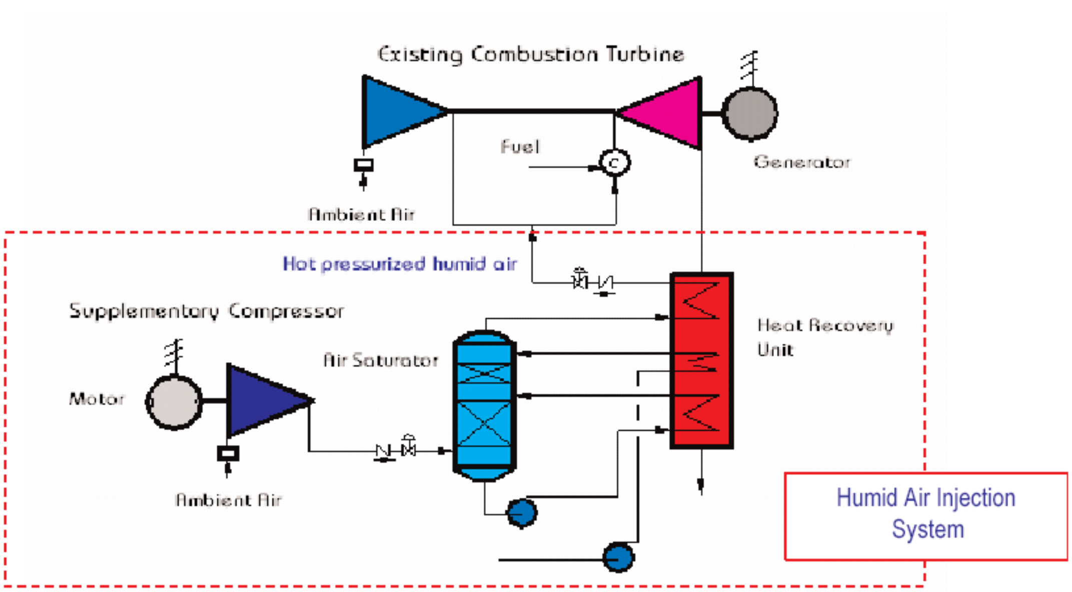
Assessment of Emerging Combustion Turbine Capacity Enhancement Technologies

Technical Report



Assessment of Emerging Combustion Turbine Capacity Enhancement Technologies

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Final Report, December 2003

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REPORT SUMMARY

This study evaluated the potential for exploiting emerging capacity enhancement technologies to deal with the new challenges to combustion turbine (CT) operators in the upcoming decade. It also looked at ways of optimizing maintenance intervals to minimize the increased cost of repair and replacement of parts that goes along with high-efficiency CTs.

Background

Currently, combustion turbine operators must deal with over-capacity, fuel cost volatility, high maintenance costs, and containing greenhouse gases (GHG). There are a number of emerging technologies that may be brought to bear on these issues. The technologies cover the gamut of inlet cooling and mass injection options, synergistic hybrid combinations of the options, and emerging maintenance practices.

Objective

To assess the potential for near term emerging capacity enhancement technologies to deal with the challenges faced by combustion turbine operators in the next decade.

Approach

The project team examined the potential for either increasing the efficiency of simple or combined cycles and/or increasing the capacity of the higher efficiency units. By raising the level of efficiency through projects that adapt these technologies, fuel use and GHG are reduced. The team reviewed aspects of maintenance practices that can materially reduce costs, as well as increase efficiency.

Results

This report contains an assessment of emerging technologies that can enhance the operating flexibility of simple or combined cycle combustion turbines to achieve higher efficiency and/or higher capacity at higher efficiency. The technologies include: direct and indirect evaporative cooling, fogging, chilling, inter-stage cooling, supercharging, humid air injection, and steam injection. The report identifies needs for further technological advances both for technologies that are well known and for others that are in various stages of maturing. Hybrid applications of different technologies are shown to have the potential to further increase efficiency and capacity at high efficiency. The report identifies maintenance practices that may help to reduce operating costs and achieve high efficiency.

EPRI Perspective

There is an interrelationship between capacity, performance, and emissions inherent to CT design. The general goal of the Operational Flexibility area is to assess, develop and test improved technology and related experience-proven guidelines that address a wide variety of interconnected mechanical, control, and emissions issues arising from cyclic operation.

This report examines the potential for increasing the output of CCs while maintaining or increasing their efficiency over a wide range of site and weather conditions. The efficiency of combined cycle (CC) units is at the upper limits of all installed generation. Therefore, the preferred power generation option using natural gas fuel is large efficient combined cycles. One of the disadvantages of CCs is de-rating at high ambient temperatures and low site ambient pressures. With added CC flexibility, the use of the lower efficiency CTs can be deferred until the demand catches up to the capacity.

Other EPRI engineering and economic guidelines and evaluation procedures are available as reports and software. EPRI's Strategic Capacity Analysis and Database (SCAAD) software for CT capacity enhancement analysis and design and its supporting application guidelines provide 12 capacity enhancement techniques that can be screened for any site. Recent key products include:

- Combustion Turbine Spray Cooler Guide, EPRI report TR-113983
- Chiller and Storage Technology for Additional Capacity, EPRI report 1004352
- SCAAD V3.0, EPRI software 1004583

Keywords

Combustion turbines
Combined cycles
Overcapacity
Energy efficiency
Maintenance

ABSTRACT

This report contains an assessment of emerging technologies that can enhance the operating flexibility of simple or combined cycle combustion turbines to achieve higher efficiency and/or higher capacity at higher efficiency. The technologies include: direct and indirect evaporative cooling, fogging, chilling, inter-stage cooling, supercharging, humid air injection, and steam injection. While some of these technologies have been used for many years, there is need for improvements to achieve high efficiency and reduce operating costs. The report also reviews the importance of maintenance practices in light of the improved and lower cost parts that may be available from the changing parts supply industry.

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1 INTRODUCTION

Objective

The objective of this study is to evaluate the potential for exploiting emerging technologies to deal with the new challenges to combustion turbine (CT) operators in the upcoming decade [1]. The key near-term issues are:

- Over-Capacity

The upwards economic momentum of the 1990s followed by the downturn in the economy has resulted in over-capacity, exceeding 30% in some regions.

- Volatility of Fuel Costs

The fuel of choice is natural gas, which is increasing the demand for natural gas and concurrently the price volatility.

- Greenhouse Gases (GHG)

Although the U.S. has not signed the Kyoto agreement, global political pressures may well result in CO_2 caps or sequestration targets in the U.S.

- Operating Costs

The introduction of high-efficiency CTs using advanced technology has resulted in a corresponding increase in the cost for repair and replacement of parts. The volatility of fuel costs and the need to contain the cost of parts justifies economic optimization of maintenance intervals.

Emerging Technologies Considered

As the size of combined cycle (CC) units has grown, their efficiencies have increased as shown in Figure 1-1. The efficiency of these units is at the upper limits of all installed generation. Therefore, the power generation approach using the preferred natural gas fuel is to use large efficient combined cycles. One of the disadvantages of CCs is the de-rating at high ambient temperatures and low site ambient pressures. This report examines the potential for increasing the output of CCs while maintaining or increasing their efficiency over a wide range of site and weather conditions. With added CC flexibility, the use of the lower efficiency CTs can be deferred until the demand catches up to the capacity. Note that in Figure 1-1, a 250 MW CC may have a thermal efficiency of 55%, but if there is a 20% loss in power due to high ambient temperatures, then a 50 MW simple cycle turbine with a typical efficiency of 33% might be needed to make up the lost power. The fuel use for 50 MW SC units is roughly 67% more than if the power could be generated by the CC.

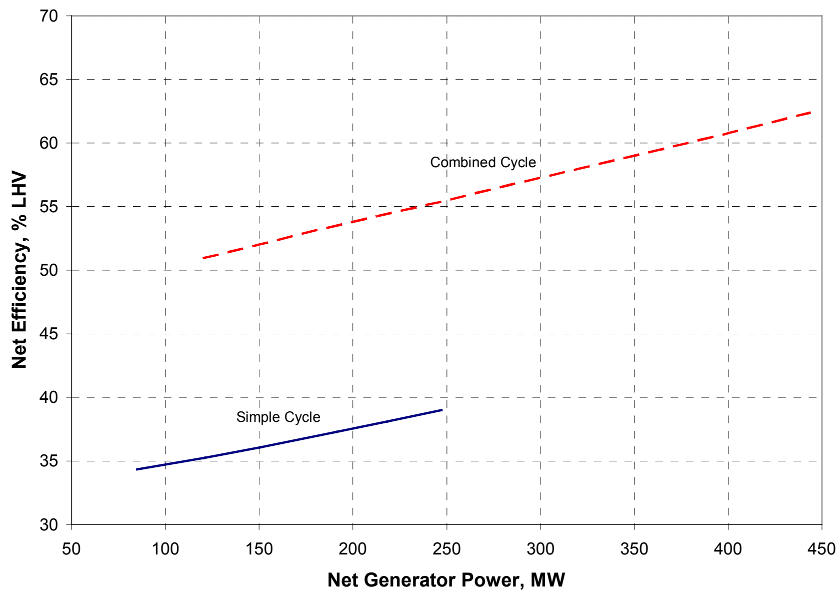


Figure 1-1
Increase in Efficiency and Size of Large Combustion Turbine and Combined Cycle Plants [2]

Retrofitting a simple cycle unit could defer the need to purchase new capacity and also reduce fuel consumption. The potential fuel reduction for simple cycle CTs is not as large as for combined cycles. If an older, less efficient CT unit was needed for more peaking capacity, then the increase in fuel for the shortfall in power could be as little as 12%, but the added capacity to high efficiency units may defer adding new CTs.

Examples of the benefits of each technology are viewed with respect to their enhancements on the GE PG7241FA, but the results are, of course, generally applicable to other comparable heavy duty CTs. The baseline combined cycle model has two CTs and a steam cycle with two reheat three pressure level HRSGs and one two-casing steam turbine. Two sites were selected, Las Vegas and Miami, that reflect differences in climate conditions and altitude. The comparative combined cycle performance for two sites [3] is given in Table 1-1. As shown in Table 1-1, the effect of the Las Vegas site elevation of 2179 feet (664 meters) is a loss in output of 38 MW.

The effect of the CT efficiency and the steam turbine power (STP) on the CC efficiency is shown in the equations below.

$$ECC = ECT * [1 + STP/CTP]$$

$$STP/CTP = 0.53,\ ECT = 35.0\%,\ ECC = 53.5\%$$

Note that the steam turbine power is more than one-half of the combustion turbine power (CTP). The effects on the CT efficiency and the ratio of STP/CTP differ for each enhancement technology. For example, when steam is extracted for injection into the combustion turbine, the ratio STP/CTP may decrease markedly, but the CT thermal efficiency may increase. Other enhancement techniques may show an increase in steam turbine power, but the ratio of STP/CTP tends to decrease, unless the steam cycle is also enhanced. Therefore, to retain a high CC thermal efficiency, an enhancement that increases combustion turbine efficiency is important. The combustion turbine efficiency implied in the equations above is the net value, so that high auxiliary power losses are undesirable.

Note that the steam turbine power is more than half of the combustion turbine power, so combined cycle capacity enhancements that approximately maintain this ratio are desirable in order to have high thermal efficiency. Also shown in Table 1-1 is the capacity enhancement required at the maximum ambient temperature to achieve a flat-rated operational capability over the range of site temperatures. This enhancement is 19% at a site such as Miami and 30% at a site such as Las Vegas.

Table 1-1
GE PG7241FA Combined Cycle Performance

Site	Temp., °F (°C)	CC Thermal Effcy., %	CC Net Power, MW	CT Net Power, MW	ST Net Power, MW	Site Temp. Range Tc to Th, °F (°C)	Power Ratio, MWc/MWh
Las Vegas	82.5 (28.1)	53.4	437.0	285.3	151.7	20 to 110 (-6.7 to 43.3)	1.30
Miami	82.5 (28.1)	53.2	475.5	312.9	162.6	35 to 95 (1.7 to 35)	1.19

The operating limits of each retrofit technology are set by the site weather conditions, the OEM-specified safe operating limits and critical CT parts limits. The operating limits are briefly summarized in Table 1-2. The generator and transformer temperature rise limits are common to all the technologies and require assessment of their capacity to handle increased power output.

Table 1-2
Operating Limits of Various Technology Options

Technology	Operating Limits
Evaporative Cooling - Media Type	Site Wet Bulb Temperatures
Evaporative Cooling - Fogging	Site Wet Bulb Temperatures
Chilling/TES - Peaking	Compressor Icing Temperature
Chilling - Base Load	Compressor Icing Temperature
Compressor Inter-Stage Cooling	Compressor Saturation
Supercharging	CT OEM Mechanical
Steam Injection	CT OEM Injection
Humid Air Injection	CT OEM Injection
O&M Practices	CT Parts Upgrades

The CT impact issues are summarized in Table 1-3. To overcome the limits and impacts of an individual technology, the use of synergistic hybrid approaches are considered. The curves shown in Figure 1-2 illustrate how a large total enhancement may be achieved by the hybrid approach.

Table 1-3
Impact Issues of Various Technology Options

Impact Issues	Potential Changes
Stall/Surge Margin Envelope	Compressor Pressure Ratio, Stage Matching
Compressor Maintenance	Erosion and Fouling Rates
Hot Section Parts Maintenance Interval	Pressure, Cooling Air Temperature, Heat Transfer
Mechanical Design Envelope	Pressure, Force, Torque
Emissions	Combustion Air Composition, Temperature, Pressure, Fuel Use

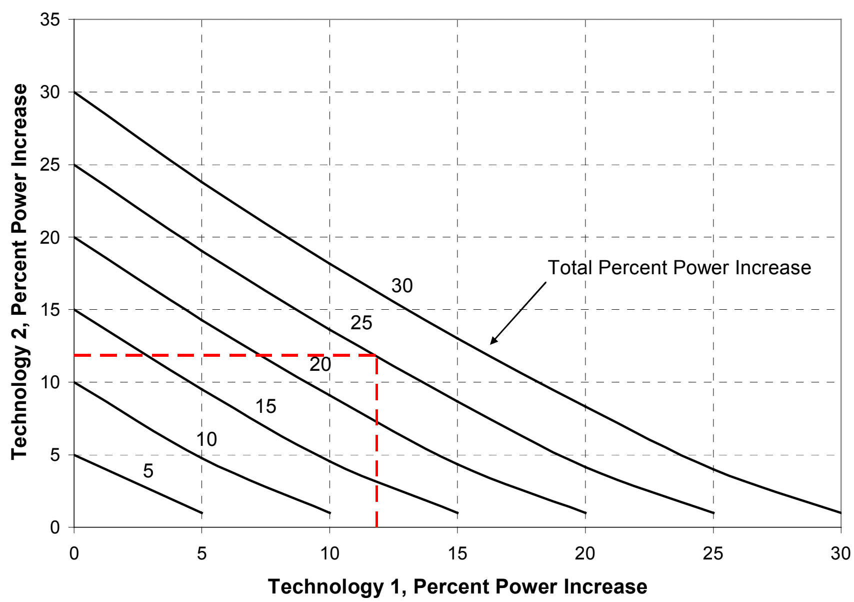


Figure 1-2
Hybrid Technologies Concept

The curves shown in Figure 1-2 assume that the ratio of the total enhanced power to the basic CT power is equal to the product of the separate power ratios for each approach, i.e.:

$$MW_T/MW_B = MW_1/MW_B \times MW_2/MW_1$$

For example, if fogging reduces the inlet temperature from 100°F to 80°F (37.8°C to 26.7°C), then there is a corresponding performance increase of MW_1/MW_B. If steam injection is also used, then there is a second performance increase of MW_2/MW_1 based on the 80°F (26.7°C) inlet condition.

2 INLET AIR COOLING

Evaporative Cooling with Rigid Cooling Media

Background of Rigid Cooling Media

The newest and most efficient media type evaporative coolers use media that is typically 12 inches (305 mm) thick and perform with 90% efficiency. On a day with 100°F (37.8°C) dry bulb temperature and 80°F (26.7°C) wet bulb temperature; the media may cool the air to within 2°F (1.1°C) of the wet bulb temperature with a pressure drop of ~0.2 inches of water (0.5 mbar). The media type coolers are very compact, typically requiring only approximately 30 inches (762 mm) in length to install a complete module. Rigid media evaporative coolers may have an installed cost/benefit ratio as low as $10/kW. The life of media may be as much as 5 years with high quality maintenance.

The potential CT impacts of rigid media evaporative coolers are relatively benign except for the potential of water carry-over that can foul and erode the compressor airfoils. Air leakage around the sealing perimeter of the cooling media can produce air jet velocities up to 1800 fpm (548.6 mpm), increasing the potential for carry-over. The water chemistry must be maintained to limit plugging of air passages by organic and inorganic deposits.

Performance of CTs Using Rigid Cooling Media

The additional operational flexibility for simple and combined cycles was studied for dry and humid sites [3]. The performance results are summarized in Table 2-1 for maximum site ambient temperatures.

Table 2-1
Estimated Incremental Performance for Evaporative Cooling

Site	SC Power, %	SC Heat Rate, %	CC Power, %	CC Heat Rate, %
Las Vegas	21.0	-5.0	16.0	~0
Miami	6.0	-1.3	4.5	~0

Evaporative Cooling with Fogging

Background of Fogging

Fogging is an alternate technology for evaporative cooling based on the use of a fine spray of water droplets. The fine spray is generated by an array of nozzles that atomize the water into an opaque plume of fine droplets and vapor. The droplets require a certain amount of residence time in the air stream to evaporate. For a typical fogging system [4] as shown in Figure 2-1, the inlet ducting air flow path may be complex and a CFD analysis is valuable for assessing the residence time of the droplets coming from different locations in the array. When the nozzles are located downstream of a high efficiency filter, perimeter seal leakage results in flow non-uniformity and turbulence. Sharp corners in the flow path short-circuit the flow, speeding up the flow around the corner and creating velocity changes across the inlet ducting.

Hydraulically atomized nozzles are commonly used with an impact pin or swirl to breakup the droplets into a fine spray. A standard method for testing comparing alternative hydraulically atomized nozzle designs has been developed [5]. A Swirl-Flash nozzle has been developed [6] that is capable of producing droplets ~3 microns in diameter. A SwirlFlash tube assembly, shown in Figure 2-2, has nozzles that are aimed upwards and downwards with respect to the air flow, increasing the spacing between plumes, with the potential of getting finer droplets due to the added shear forces from the air flow. Flashing requires a water temperature of ~400°F (204°C) and results in hot water droplets and vapor which tend to raise the wet bulb temperature slightly. Flashing is a well known technique for recovering steam from the blow down of boiler drums. For water at 400°F (204°C), flashing to atmospheric pressure results in converting 20% of the water to steam and the water outlet is at 212°F (100°C). Therefore, for a SwirlFlash fogger, additional de-mineralized water is needed to compensate for the flashing losses. The flashing losses add vapor to the air and a psychrometric study indicates that the cooling effectiveness may be reduced by as much as ~4%. Nevertheless, the small droplet size may be able to achieve a higher cooling effectiveness by approaching the wet temperature more closely and achieving a dry condition at the compressor inlet. The other benefit of the smaller droplet size is the reduced risk of erosion. Comparative droplet size distributions are shown in Figure 2-3 and the comparative evaporation rates based on a model [7] for diffusion-controlled evaporation are shown in Figure 2-4. The results shown in Figure 2-4 are based on a simulation model that accounts for a droplet size distribution, droplet heating, and diffusion of the vapor from the droplet surface to the air [4].

The installed cost/benefit ratio of fogging systems may be as low as $10/kW, depending on the installation complexity, water supply, and site weather conditions. The expected life estimate of a system may be as high as 20 years with good quality maintenance. The impact of fogging on CTs is similar to that for media type evaporative coolers, except de-mineralized water is used, so compressor fouling is negated.

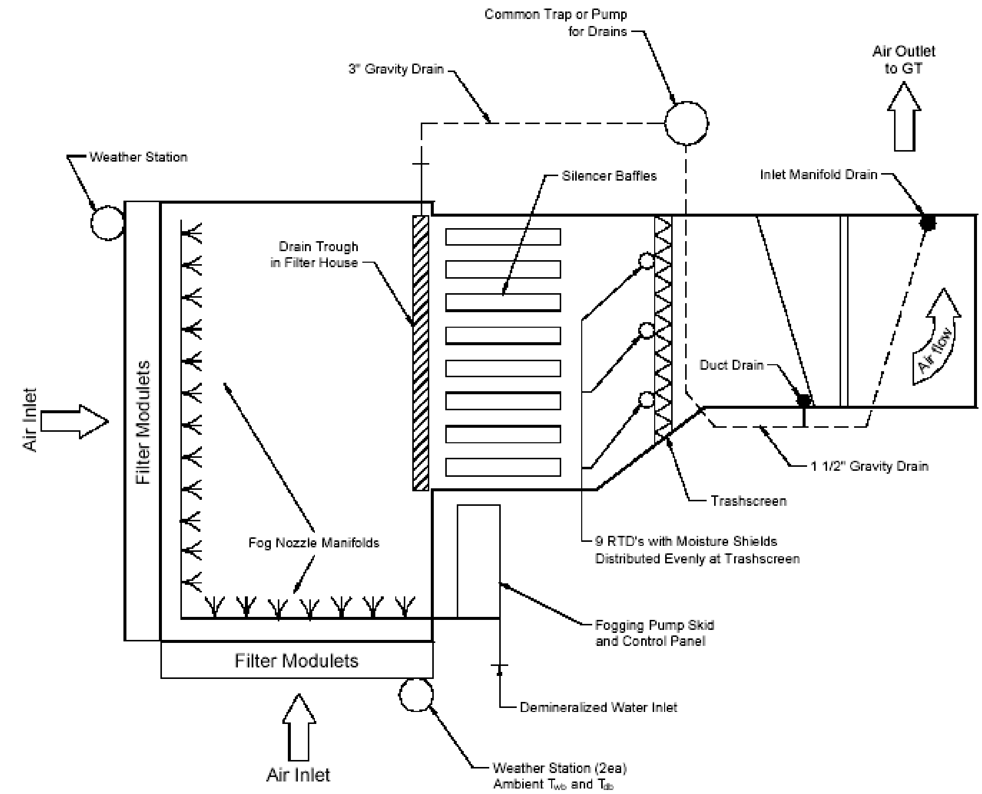


Figure 2-1
Plan View of Two-Sided Filter and Duct with Fogging System [4]

Figure 2-2
SwirlFlash Tube Assembly

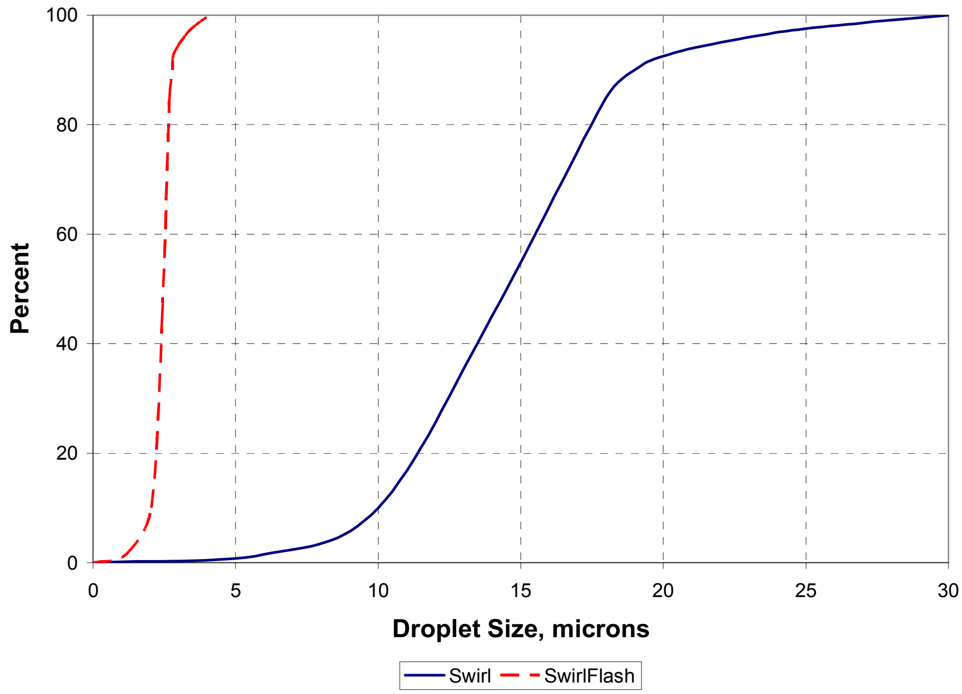


Figure 2-3
Distribution of Water Droplets from SwirlFlash [5] and Swirl Nozzles

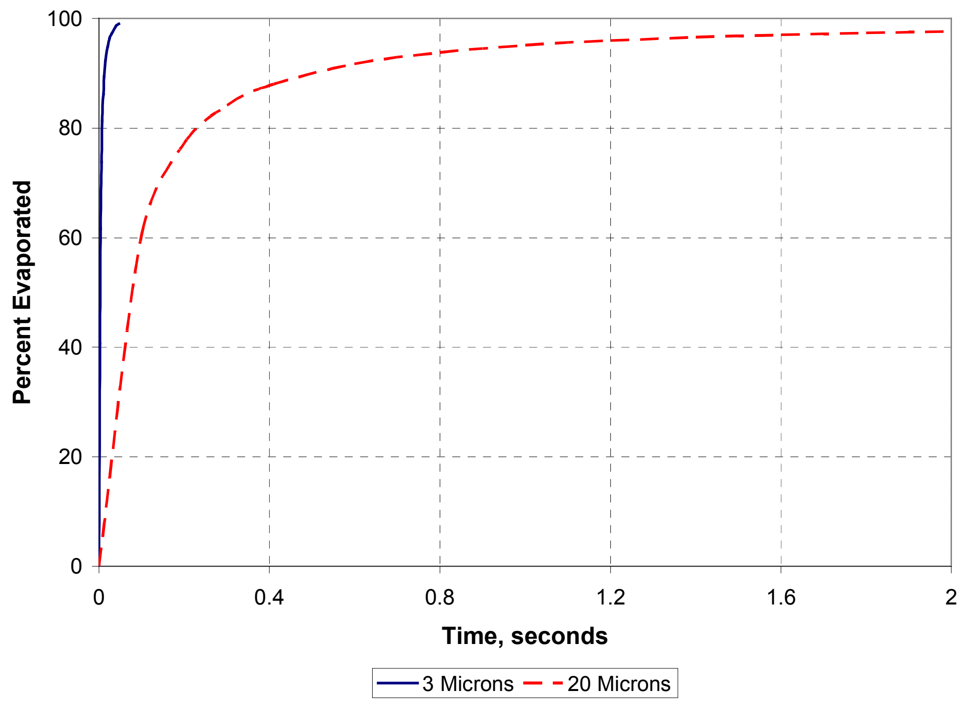


Figure 2-4
Effect of Droplet Size Distribution on Evaporation Time (Tdb = 95°F (35°C), Twb = 65°F (18.3°C), Saturation Water/Air Ratio)

Impacts of Fogging on CTs

The impact of fogging on CTs is similar to that for media type evaporative coolers. There may be as many as 500 nozzles or more, so the quality of the installation must be high to ensure no leakage and few plugged nozzles. Maintenance involves visual inspections for leakage and signs of water carry-over. The array must be well-designed to promote acceptable cooling uniformity across the entire duct for all operating conditions.

The nozzles can be located in high speed air streams as the pressure drop introduced by the nozzle array is very small. On the other hand, there is a need for sufficient droplet evaporation residence time to ensure a dry condition (dryness) at the compressor inlet even though the water is de-mineralized. The concern with dryness involves the potential effects of erosion and roughness caused by droplet impacts around the leading edge of the compressor blades. Erosion limits are specified on highly stressed compressor components and roughness resulting from droplet impacts may reduce the aerodynamic efficiency of the compressor [8]. The degree of dryness may also influence the operation of a DLN combustion control system, which may depend on the inlet air temperature. If there is two-phase flow, then measuring the “dry” inlet air temperature requires special attention such as a droplet shield for the temperature sensor [3].

Methods to achieve dryness include the use of a demister. A pad-type demister may eliminate all remaining droplets, whereas an inertial-type demister may be tailored to remove the larger droplets such that the remaining smaller droplets evaporate to dryness. The location of a demister is near the outlet of the inlet house to keep the pressure drop low.

One or more floor drains are generally used to collect water due to leakage or wetting of the inlet duct structure. Other CT impacts are the same as noted above for the media type coolers. The nozzle array structure results in very little blockage, so the pressure drop is negligible. The fogger systems require de-mineralized water to prevent deposits and fouling of the compressor. The application is most favorable where there is an existing water de-mineralizing plant.

Performance of CTs Using Fogging

The performance of simple cycle CTs using media-type evaporative coolers shown in Table 2-1 applies to fogging applications as well, keeping in mind differences may arise due to the detailed design of each system. The performance for combined cycles is discussed later in the report.

Indirect Evaporative Cooling

There are a number of different inlet air cooling processes that may be considered. Direct evaporative cooling with media or fogging places the air in direct contact with the water and moisture is added to the air. Indirect evaporative cooling separates the air and water such that no moisture is added to the air. Chilling does not add moisture and may remove moisture if the air becomes saturated during cooling and condensation occurs. The inlet cooling processes are depicted in Figure 2-5. The effect of site altitude on the psychrometric processes is also shown. The differences between a dry and humid site, as shown in Figure 2-5, illustrate the potential effect on the cooling equipment. For a dry site such as Las Vegas, the dew point temperature may actually be lower than the specified icing limit, so the saturation condition (on average) is not encountered when using chilling.

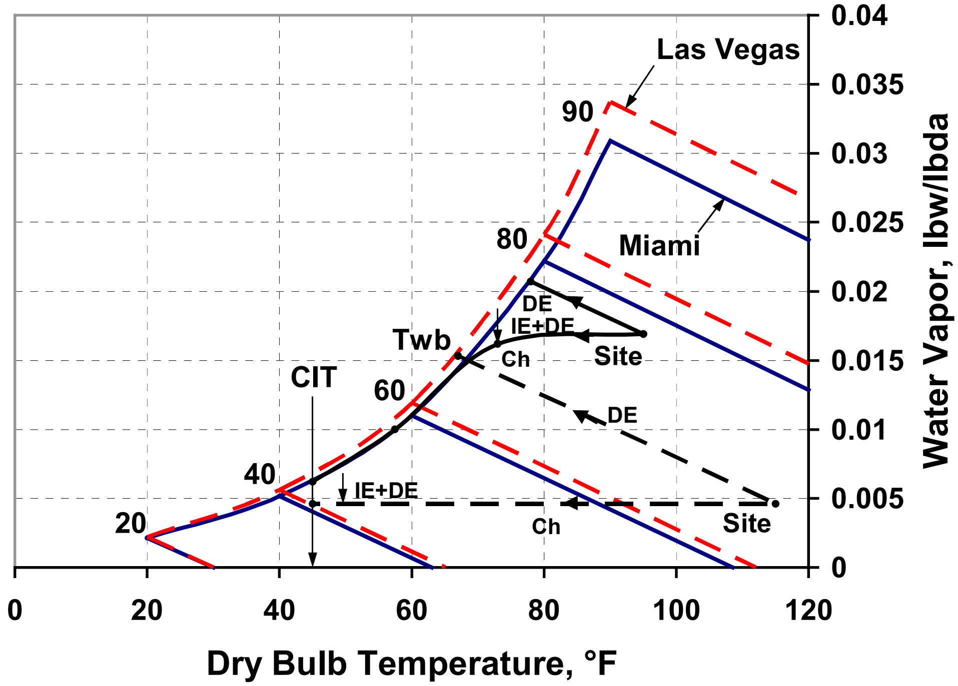


Figure 2-5
Inlet Cooling Psychrometric Processes

Generally, indirect evaporative cooling is limited by the wet bulb temperature and the incremental temperatures required for heat transfer. In the case of a cooling tower and a cooling water coil, one may achieve an inlet temperature as low as the wet bulb temperature plus 15°F (8.3°C). For a dry site such as Las Vegas with a possible site condition of 110°F (43.3°C) dry bulb and 66°F (18.9°C) wet bulb, cooling would be limited to about 81°F (27.2°C). Using a wet surface heat exchanger with the air inside the tubes may result in an inlet air temperature as low as 76°F (24.4°C). The use of combined indirect cooling and chilling may reduce the chiller size by as much as 30% [3].

Indirect evaporative cooling has the effect of lowering the wet bulb temperature of the inlet air as it is cooled. By adding a stage of direct evaporative cooling following a stage of indirect evaporative cooling, the inlet air temperature may be reduced to as low as 54°F (12.2°C) without refrigeration. The application of staging indirect evaporative cooling can further reduce the inlet temperature. The Everest Cycle [9] adds a second stage of indirect evaporative cooling followed by a stage of direct evaporative cooling. By using an additional air stream, a second stage of cooling is possible that can be used to cool the combustion inlet air further. The second cooled air stream is available for other cooling needs also.

Details of the specific Everest Cycle approach are not available as the system is under development. A potential arrangement is shown in Figure 2-6 which may be suitable for large CTs using cooling water coils, a cooling tower, cooling tower packing and a final stage of direct evaporative cooling for added cooling and air-washing. For an ideal case at a dry site, consider TDB1 = 110°F (43.3°C) and TWB1 = 66°F (18.9°C). The first stage of indirect cooling can reduce the inlet air temperature to as low as 66°F (18.9°C) in the ideal case, but the moisture content is unchanged, so the exiting wet bulb temperature TWB2 is 51°F (10.6°C). A second stream of air has the same arrangement, but also has cooling tower packing that can cool water to 51°F (10.6°C) (ideally). This cooling water is then used in a second stage cooling coil to cool the inlet air to TDB3 =51°F (10.6°C) and TWB3 = 43°F (6.1°C). The addition of direct evaporative cooling can reduce the inlet air temperature to 45°F (7.2°C). The estimated performance for the system is shown in Figure 2-7 for 2-stages of indirect evaporation plus a final air washer stage. The potential performance depends on the incremental temperature differences required for the heat transfer in a practical system and the pressure drops needed to accomplish the heat transfer. For the case described above, the inlet temperature may be as low as 49°F (9.4°C). De-mineralized water is not required. A prototype of the system is presently under development for small gas turbines such as the RR 501 series.

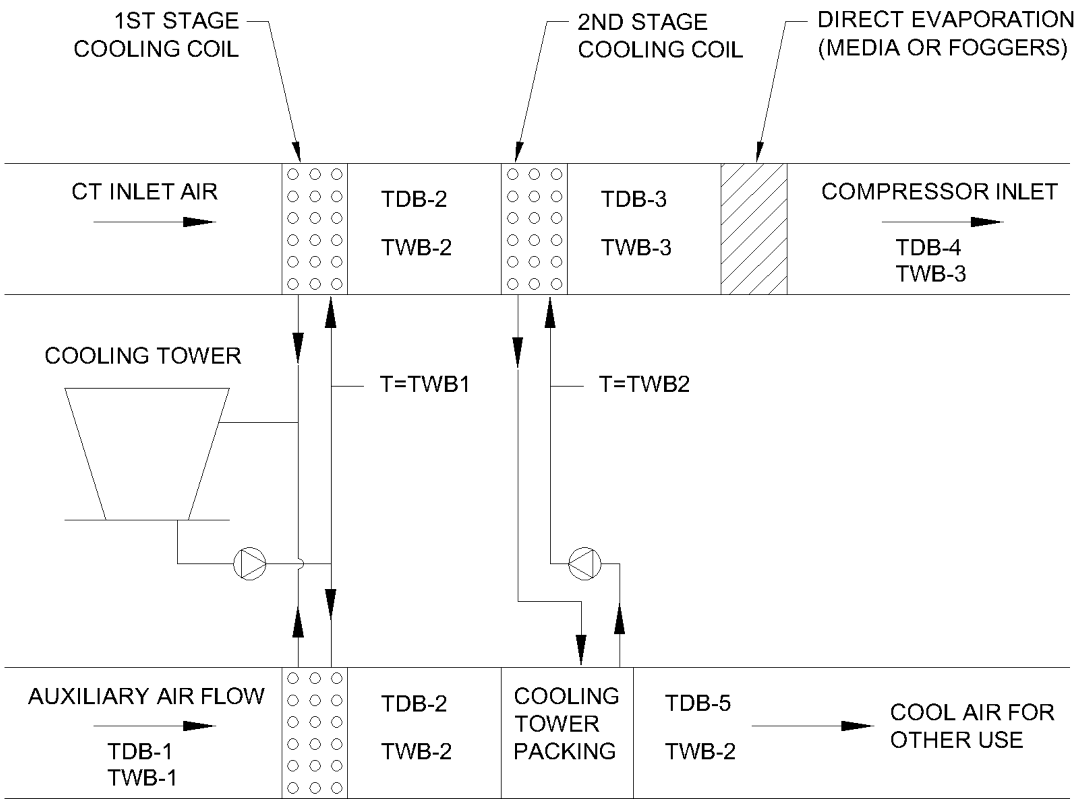


Figure 2-6
Ideal 2-Stage Indirect Evaporative Cooling Concept

Inlet Air Chilling for Peaking CTs

Background of Chilling for Peaking CTs

Inlet air cooling by chilling using refrigeration equipment and thermal energy storage (TES) is being used to obtain a larger increase in power than is obtainable by using evaporative cooling. An EPRI study [3] reviewed the state-of-the-art of inlet air cooling by fogging or chilling. The inlet chilling concept for peaking operations utilizes TES, as depicted in Figure 2-8. The use of TES results in a much smaller chiller and eliminates the chiller parasitic power (and the associated demand costs) during peak operation. Whereas evaporative cooling can reduce the inlet air temperature close to the wet bulb temperature, which ranges from 65°F (18.3°C) to 80°F (26.7°C) on hot days, chilling systems can cool the inlet air temperature down to the icing limit which may be as low as 40°F (4.4°C). Estimated cost/benefit ratios for chilling systems for peaking CTs range from $100/kW to $200/kW depending on site weather conditions, CT performance, size of the installation and peaking duty.

Ambient Dry Bulb

Wet Bulb	**50**	**55**	**60**	**65**	**70**	**75**	**80**	**85**	**90**	**95**	**100**	**105**	**110**	**115**
85									84	83	82	80	79	78
80								79	77	76	75	73	72	71
75							74	72	71	69	68	66	65	63
70						68	67	65	64	62	**60***	58	56	55
65					63	62	60	58	56	54	52	50	47	45
60				58	56	54	52	50	47	45	43	40		
55			53	51	49	46	44	41	38	36				
50		48	45	43	41	40	40							
45	43	40	40	40	40									
40	40	40	40											

Figure 2-7
Everest Cycle Performance at Sea Level, Temperatures in °F (°C = (°F – 32)/1.8) [9]

As a result of chilling, fuel use increases along with the increased air flow and power output. The NO_x mass flow may either increase or decrease depending on the ambient conditions. The lower combustion air temperature tends to reduce the flame temperature and NO_x formation. If the ambient humidity is high, chilling condenses out some of the moisture, tending to increase the NO_x concentration. On the other hand, for dry sites, there is little, if any, condensation and the cooling lowers the combustion air temperature and the NO_x concentration. Therefore, depending on site conditions, the chilling may either increase or decrease the mass flow of NO_x for CTs with diffusion flame combustors. DLN combustors may need to be tuned to retain their low emissions performance.

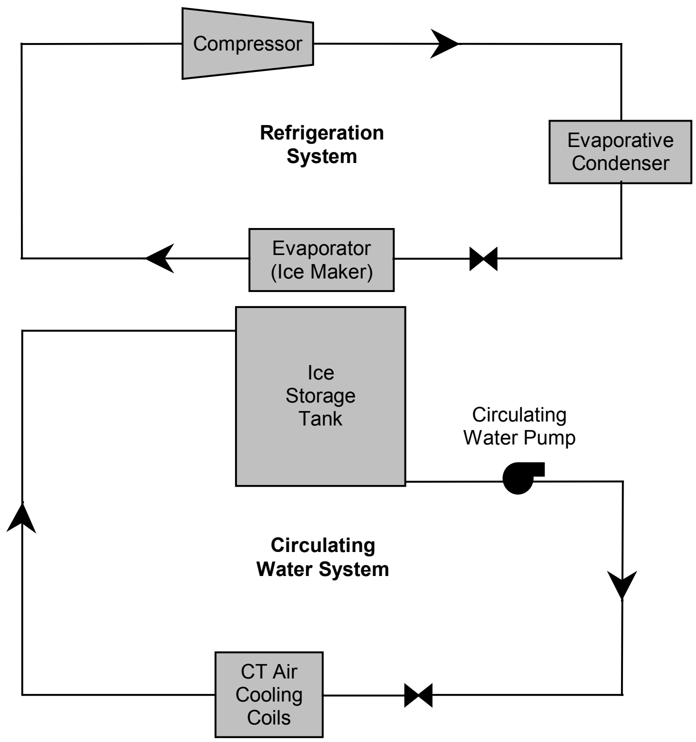


Figure 2-8
CT Peaking Chiller Schematic

Performance of Peaking CTs Using Chilling or Fogging

Comparisons of the performance of fogging and chilling are depicted in Figure 2-9 for a dry site such as Las Vegas and in Figure 2-10 for a humid site such as Miami. The fogger performance is based on CIT = Twb + 2°F (1.1°C) as a dryness and fogging system control consideration. The performance gains at a dry site using a fogging are competitive with using chilling. At humid sites, the performance gains using chilling are far greater than using fogging.

Water injection may be used to further augment power in a fogging or chilling system. The potential power gain in the hybrid system may be as high as 25% as shown in Figure 2-11. Water injection reduces the hot section life [10], but for peaking systems, this may be acceptable, as the total number of operating hours is likely to be low.

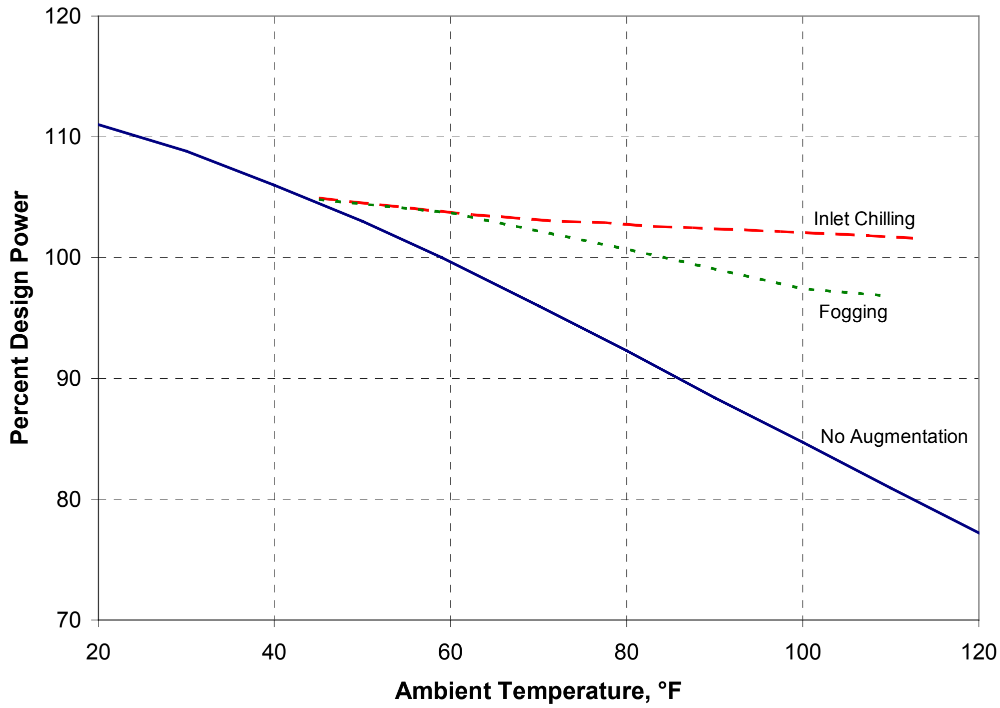


Figure 2-9
Simple Cycle Peaking Operational Range, GE PG7221FA, Las Vegas (°C = (°F – 32)/1.8)

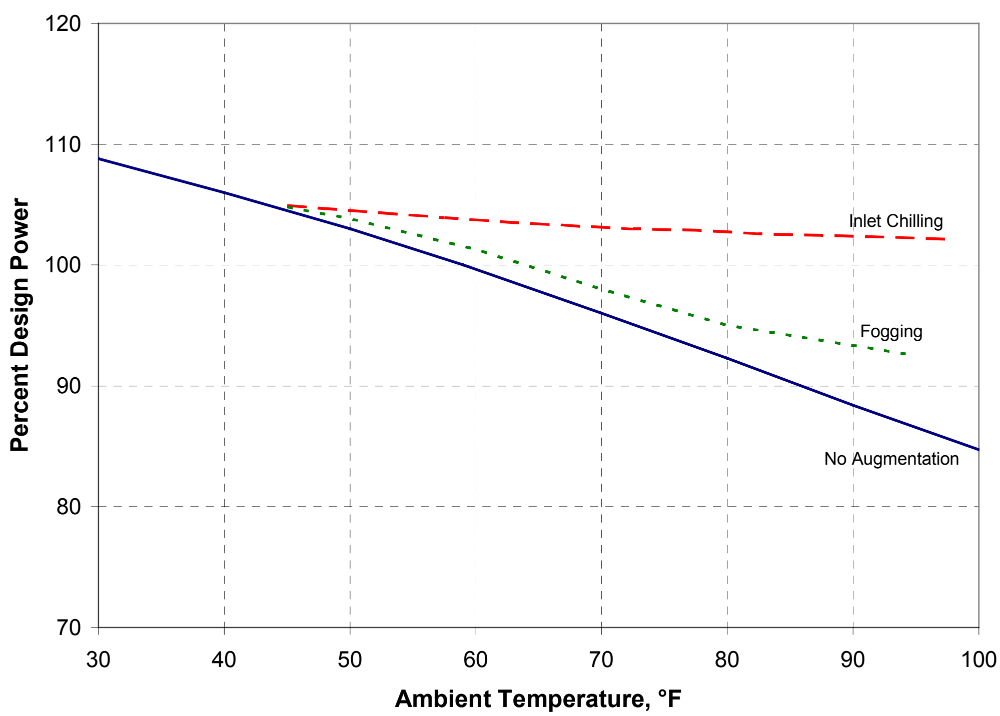


Figure 2-10
Simple Cycle Peaking Operational Range, GE PG7221FA, Miami (°C = (°F – 32)/1.8)

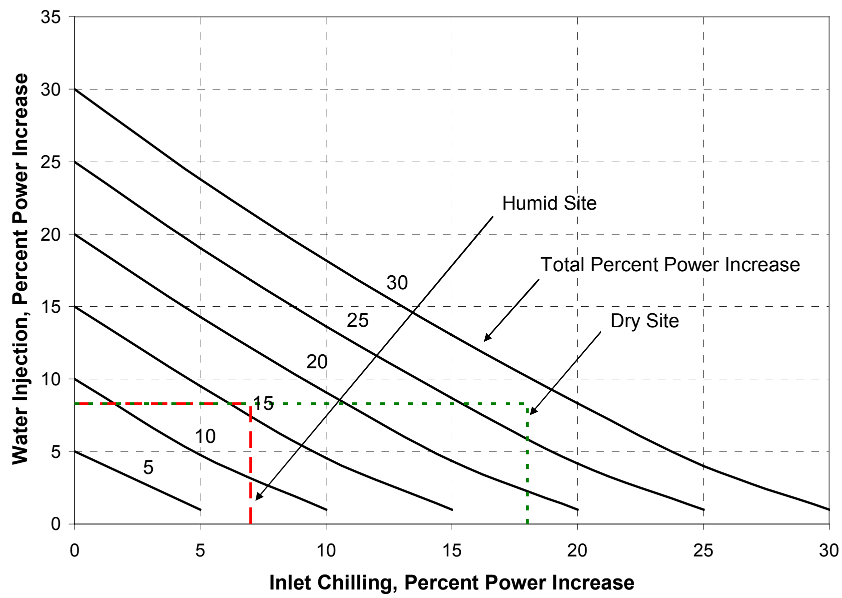


Figure 2-11
Hybrid Inlet Fogging/Water Injection Concept, Peaking

Inlet Air Chilling for Base Load Combined Cycles

Background of Chilling for Base Load Combined Cycles

Mechanical chiller systems are applicable to base load combined cycles. Chiller systems are supplied with packaged skids as depicted in Figure 2-12 with three modules: 1) refrigeration equipment skid, 2) water coils, and 3) cooling tower (or evaporative cooler).

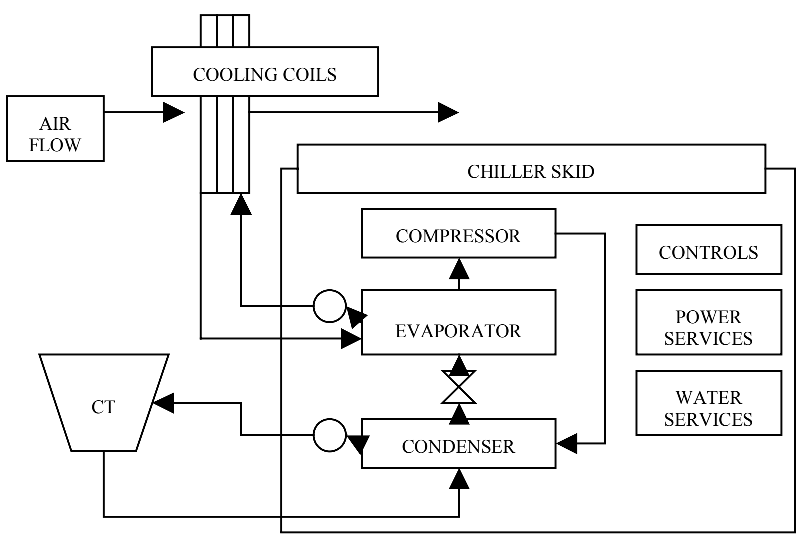


Figure 2-12
Chilling System Equipment

Impacts of Chilling on CTs in Combined Cycle

The CT impacts of chilling for combined cycle operation are similar to those for peaking, except the effects on hot section maintenance interval are more important. The lower compressor discharge air temperature results in a lower temperature for the hot section cooling air. The lower cooling air temperature has a favorable effect on metal temperatures such that the equivalent creep life operating time may be reduced. The increase in pressure and the stresses on the static parts tends to partially offset the benefits of the lower cooling air temperature.

Performance of Combined Cycles Using Chilling or Fogging

The impact of chilling systems on the operating range is shown in Figure 2-13 for a dry site such as Las Vegas and in Figure 2-14 for a humid site such as Miami. The fogging system performance is based on a 2°F (1.1°C) approach to the wet bulb temperature.

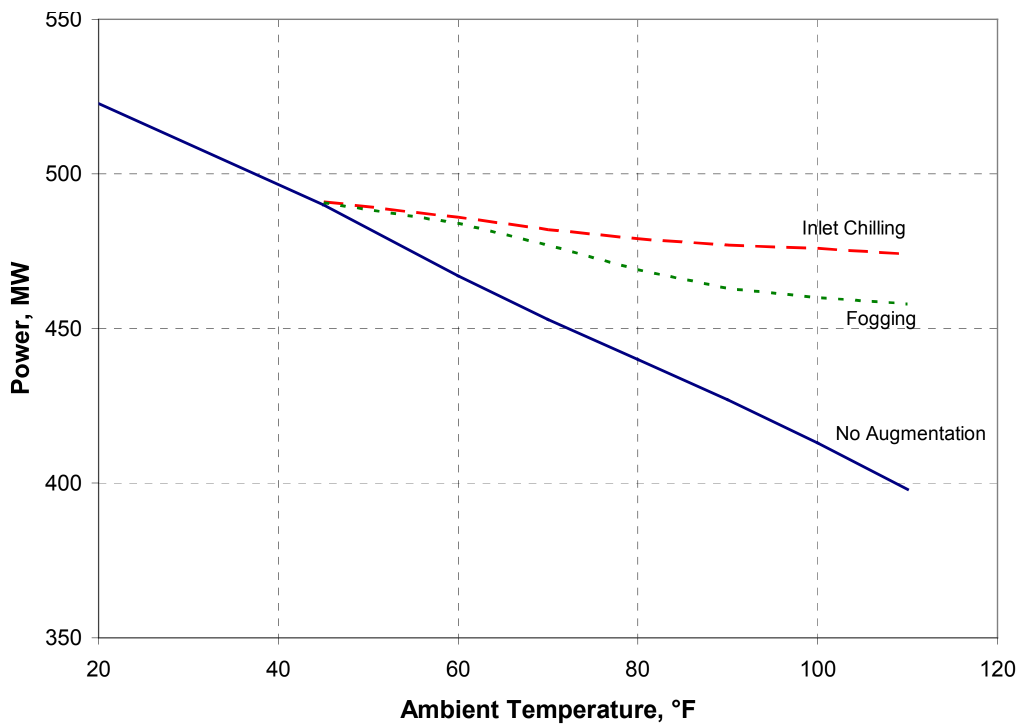


Figure 2-13
Combined Cycle Operational Range, GE PG7221FA, Las Vegas, 2 Units (°C = (°F – 32)/1.8)

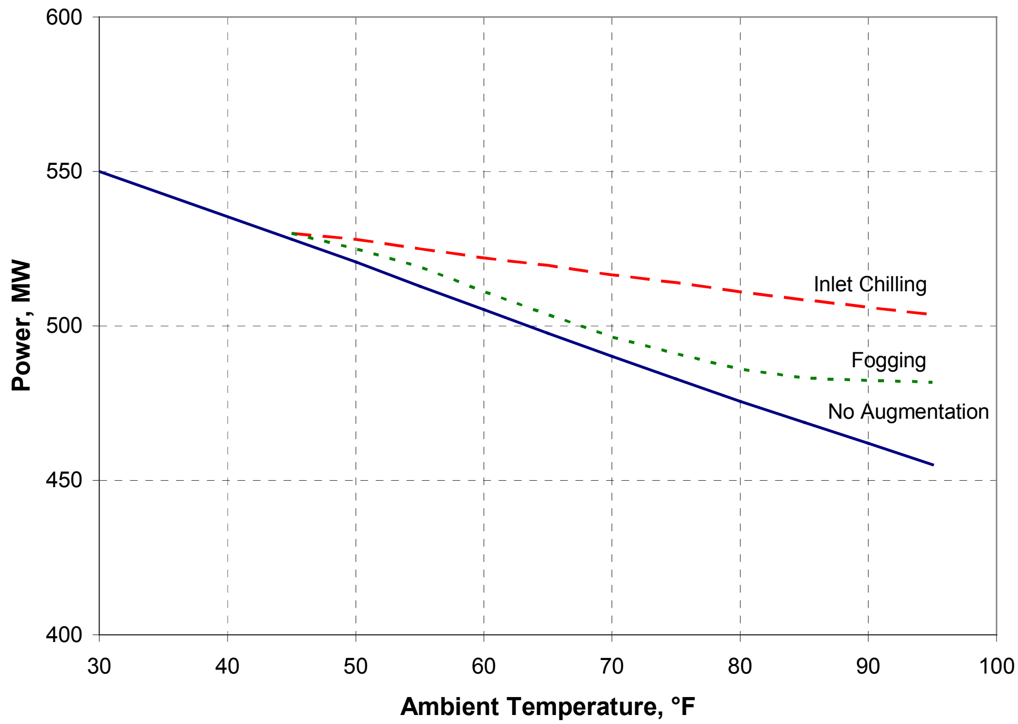


Figure 2-14
Combined Cycle Operational Range, GE PG7221FA, Miami, 2 Units (°C = (°F – 32)/1.8)

The results of the EPRI study [3] comparing fogging and chilling for combined cycle operation are shown in Figure 2-15 and Figure 2-16. The results show that the economic benefits of chillers for base load are sensitive to the capacity factor (annual percent of use of the rated capacity) so they are best suited for climates with long warm seasons. Combining foggers and chilling is thermodynamically incompatible, but for sites with very low humidity, the use of indirect evaporation with chilling may be advantageous. The following sections of the report examine the options for flat-rating the combined cycle over the complete range of ambient temperatures by employing other technologies in combination with inlet air cooling.

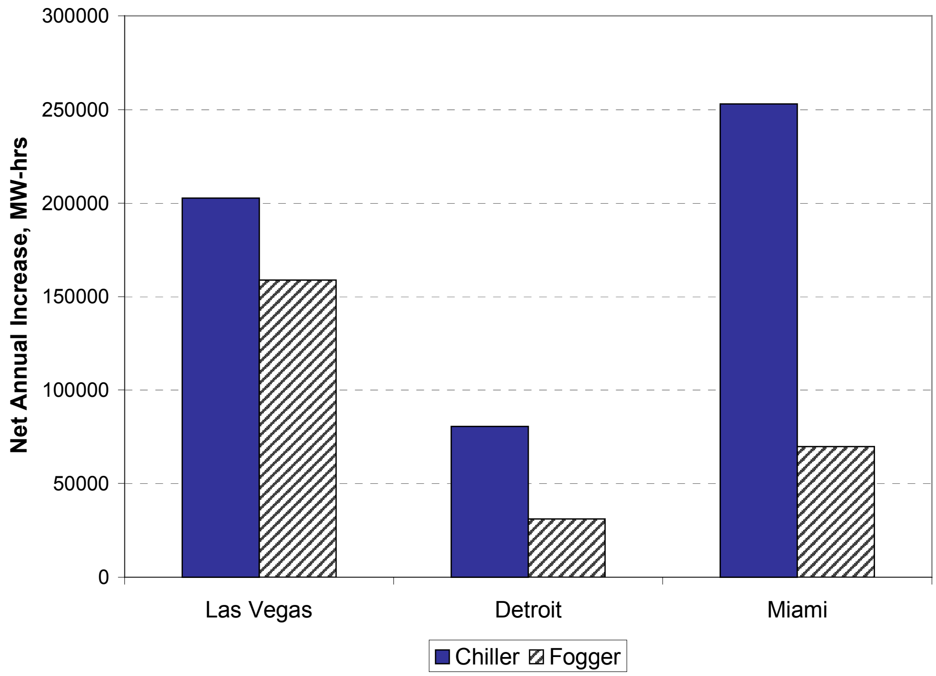


Figure 2-15
Combined Cycle Electric Energy Augmentation for Chillers and Foggers at Three Sites (GE 7FA, 2 Units, Designed for Maximum Ambient Temperature) [3]

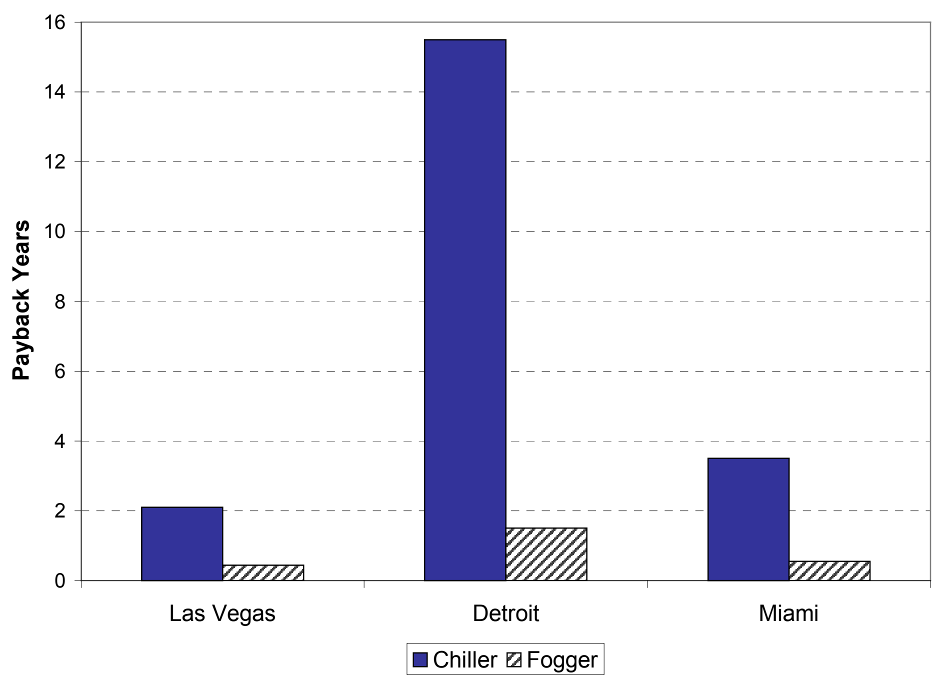


Figure 2-16
Combined Cycle Payback for Chillers and Foggers at Three Sites (Designed for Maximum Ambient Temperature) [3]

3 INTER-STAGE COOLING

Background of Inter-Stage Cooling

Two W501A CTs were operated with inter-stage cooling from 1994 to 1999 with water injection into the compressor inlet [11]. The water injection levels varied up to 1.7% of the inlet air flow and confirmed the estimated power augmentation benefits. Testing has shown that injecting water into an axial compressor can reduce the work of compression, reduce the compressor outlet air temperature, increase output power and reduce NO_x emissions. Issues of concern are the potential for erosion of the compressor blade elements and the reduced stall margin. Estimates for the installed cost/benefit ratio are as low as $35/kW, excluding the water supply [6].

Impacts of Inter-Stage Cooling on CTs

Stall/Surge Margin Envelope

Studies on the effects of water injection on the axial compressor aerodynamics show that the water injection may be limited to less than 3% to avoid the potential for stalling as shown by sample results [12] shown in Figure 3-1 and Figure 3-2. The results shown are based on a pitch-line stage stacking simulation that sets the stage loadings and losses to match a design point condition for a specified pressure ratio, flow, efficiency, and stage geometry. The evaporation is based on a polydisperse droplet distribution and the effects of surface impacts are also modeled [12]. When the front stages of the axial compressor become saturated, the front stages unload and the rear stages become more loaded. The flow coefficient for the first stages increases with a corresponding increase in air flow, while the flow coefficient for the rear stages decreases with a trend towards stall. If each stage is saturated as the water droplets evaporate, then because of the limit on the amount of water injection, the evaporation will be completed by the fifth stage. The benefits of water injection are greatest when complete evaporation is achieved as soon as possible in the compressor.

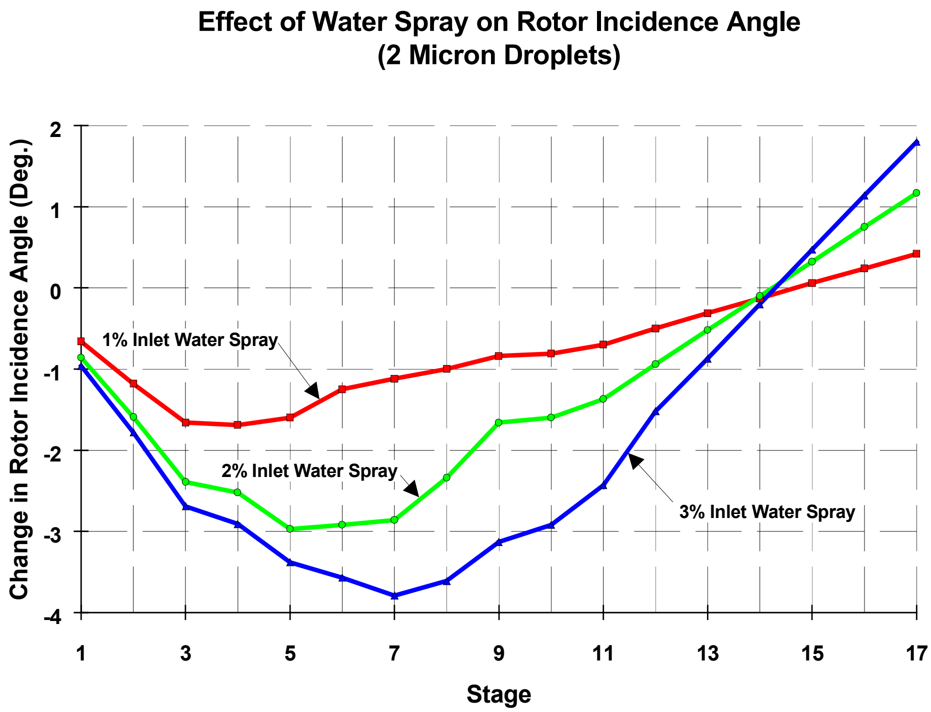


Figure 3-1
Effect of Inter-Stage Cooling on Compressor Rotor Air Flow Incidence Angles [12]

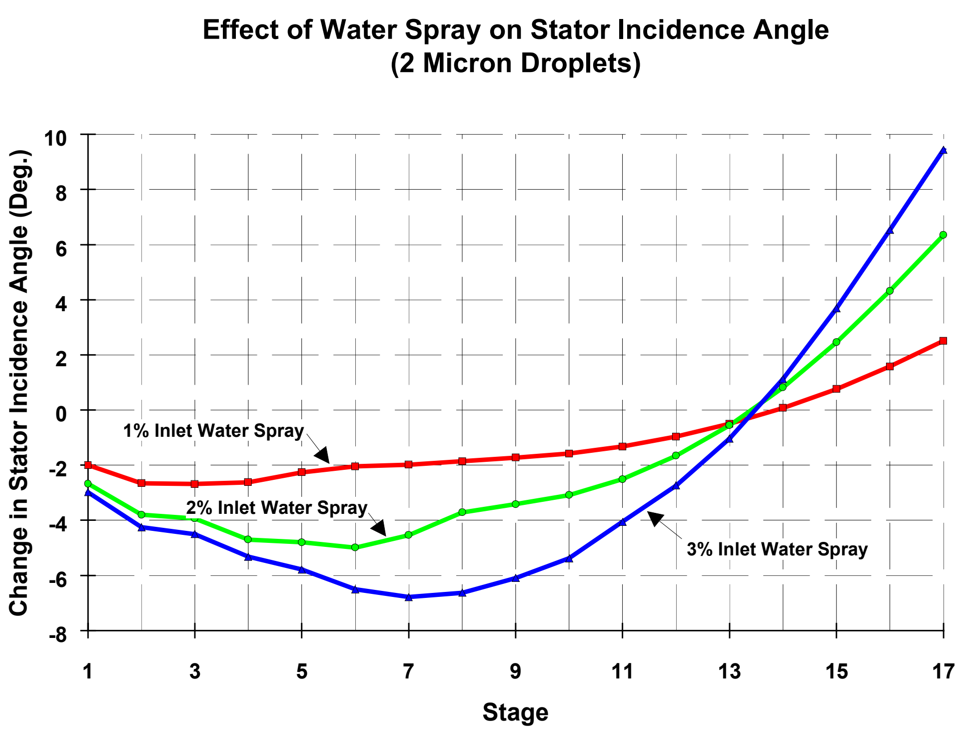


Figure 3-2
Effect of Inter-Stage Cooling on Compressor Stator Air Flow Incidence Angles [12]

Compressor Maintenance

The impact of inter-stage cooling on the CT is primarily the potential erosion and roughness that may occur. There is emerging technology to limit erosion and maintain compressor surge margin and structural integrity. The use of very fine droplets that are less than 3 microns in diameter has been shown to reduce erosion to a negligible rate [6]. Alternatively, an erosion resistant chrome carbide coating on the leading edge of the first five blade rows of the compressor has been applied and is in operation [13].

Hot Section Parts Maintenance Interval

A key benefit of inter-stage cooling is the significant reduction in compressor extraction and discharge air temperatures. As shown in Figure 3-3, a temperature reduction of up to 140°F (77.7°C) may occur for an injection rate of 2.5% and up to 80°F (44.4°C) for an injection rate of 1.5%. The cooling air temperatures often limit the firing temperature to achieve the desired hot section parts life. Incorporating inter-stage cooling may enable the firing temperature, the power output, and the efficiency to be raised significantly. When used in a hybrid power augmentation system, the lower air temperatures will offset effects such as an increase in pressure ratio to retain the hot section parts life. However, recent tests [14] indicate a potential for increased wear of the combustor components.

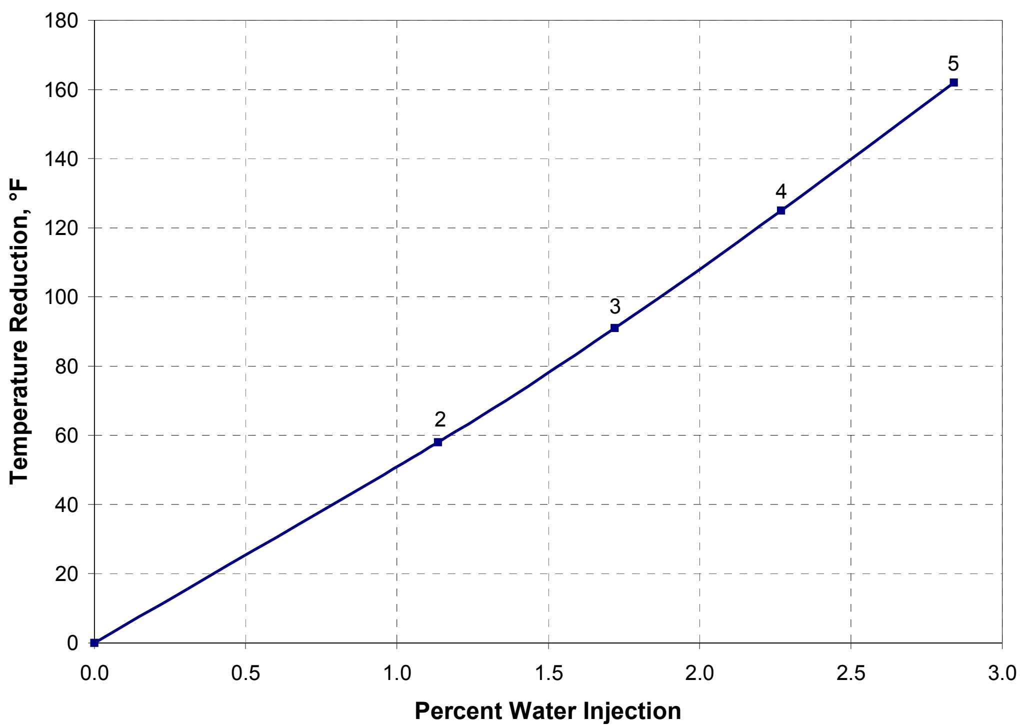


Figure 3-3
Reduction in Compressor Discharge Temperature with Inter-Stage Cooling (Numbers Indicate Stage Numbers when Saturation is Achieved, 14:1 Pressure Ratio, 18 Stages,°C = (°F – 32)/1.8)

Emissions

The large reduction in compressor discharge air temperature plus the addition of moisture results in lower flame temperature and a reduction in NO_x for diffusion flame combustors.

Performance of CTs Using Inter-Stage Cooling

A comparison of the power augmentation of simple cycle CTs with inter-stage cooling, inlet fogging and water injection into the combustor is shown in Figure 3-4. As shown in Figure 3-4, the most effective use of water is for inlet evaporative cooling, but this is limited by the wet bulb temperature. On the other hand, the use of water injection into the combustor is seen to be the least efficient use of water for power augmentation. The results shown are for a GE 7FA. Note that the maximum amount of water injected to saturate the air throughout the compressor is nearly 10% of the air flow. The saturation model is based on a numerical integration of the compression process with continuous saturation and includes the effects of the added mass of water vapor and the properties of the moist air mixture. The mechanism for saturation may be by droplet or surface evaporation. The model assumes that the compressor discharge pressure rises as mass is added based on a choking condition of the 1st stage turbine nozzle. The cycle analysis includes the effects of the added mass flow, pressure ratio, and gas properties.

The potential benefits of inter-stage cooling come from several sources: 1) the cooling of the air reduces the work of compression, 2) an increase in the compressor inlet air flow due to re-adjustment of the stage loadings, and 3) added mass flow through the turbine section increasing the power output. The injection of water adds mass flow and increases the specific heat, so the actual compression power reduction is somewhat offset by these effects. Nevertheless, when it is considered that the compressor power for a GE7FA is approximately equal to the output power, there is a potential for a significant increase in power output. The added mass flow and moisture boosts the power of the CT as the turbine section output is increased, yielding the total benefits estimated in Figure 3-4.

Estimated performance results for simple and combined cycles with inter-stage cooling with 1.5% water injection show an incremental power of ~10% for simple cycle CTs and ~7% for combined cycles. The estimated heat rate changes for simple cycles are small and increase slightly for combined cycles. The cycle and heat balance simulation included a model for the effects of saturation, the reduced compressor power and the addition of moisture to the gas path. The results are based on a control algorithm that maintains a constant firing temperature.

Test results for inter-stage cooling using a SwirlFlash nozzle are shown in Figure 3-5 and Figure 3-6. The power augmentation compares favorably with the calculated results in Figure 3-4 for saturated conditions.

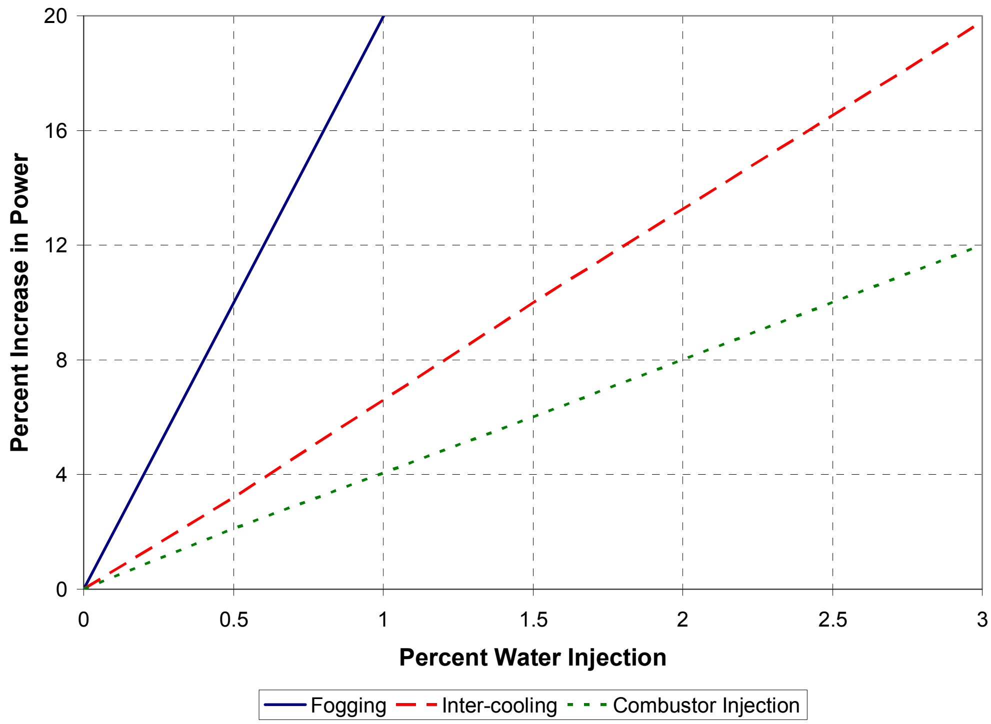


Figure 3-4
Estimated Power Augmentation Due to Fogging or Water Injection

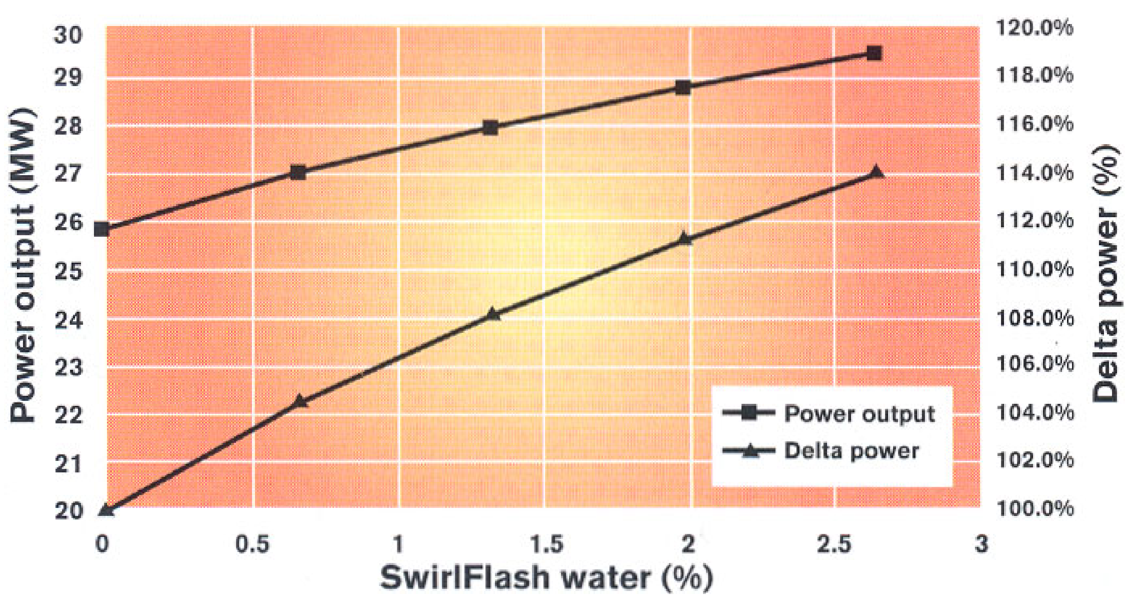


Figure 3-5
Power Augmentation of an Alstom/ABB 9D Gas Turbine Due to SwirlFlash, as a Function of the Amount of Injection Water [6]

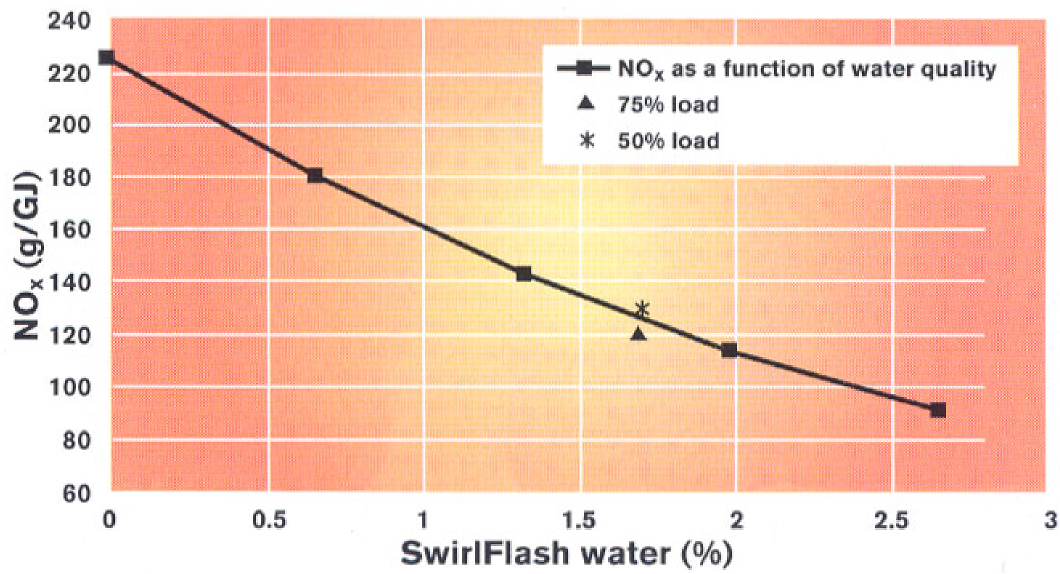


Figure 3-6
Reduction of NO_x Emissions from an Alstom/ABB 9D Gas Turbine Due to SwirlFlash, as a Function of the Amount of Injection Water [6]

The use of inter-stage cooling will most likely always be done as a hybrid with fogging as the introduction of nozzles upstream of the compressor will tend to evaporatively cool the inlet air to the wet bulb temperature prior to droplets entering the compressor. As shown in Figure 3-7, a 2.5% water rate plus evaporative cooling can result in up to a 25% boost in power with a lower heat rate and lower emissions.

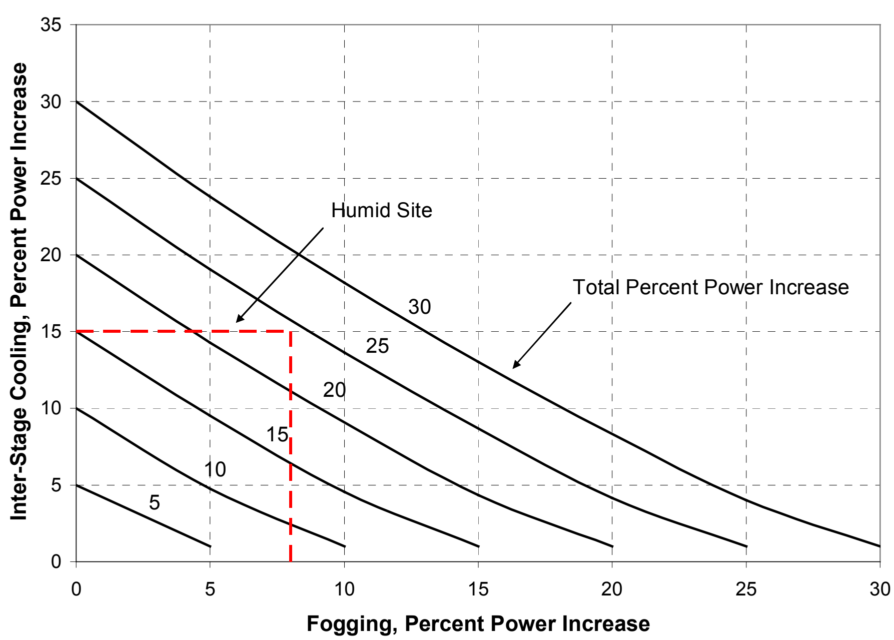


Figure 3-7
Hybrid Fogging/Inter-Stage Cooling Concept

4
SUPERCHARGING

Background of Supercharging

Supercharging rounds out the technologies based on inlet air conditioning such that the inlet temperature, humidity and pressure may be controlled. An electric motor-driven fan (or blower) strategically located in the CT inlet air flow path increases the inlet pressure at the axial compressor scroll. A temperature rise accompanies the pressure rise, so after-cooling is desirable to capture the maximum benefit of the pressure rise. Evaporative cooling is also beneficial in reducing the fan power, but this is considered a separate technology as discussed previously. The motor, fan and after-cooler ensemble constitute the supercharger.

During the late 1960s, Westinghouse offered their W301 gas turbine with supercharging. The W301 was offered in a fully fired combined cycle configuration, so the supercharger blower was also used for operating the furnace and steam turbine when the CT was not operating. The Westinghouse performance data was modeled with SCAAD (Strategic Capacity Analysis and Design) software developed for EPRI [15]. The results are given in Table 4-1 for the following conditions:

- Site @ 80°F (26.7°C) db, 67°F (19.4°C) wb, 14.7 psia (1.0135 bara)
- Tfire = 1460°F (793°C)
- 40 inches of water (100 mbar) supercharging
- 85% effective aftercooler

The net gain in power is 3.7-4.4 MW or 13-15%. The SCAAD calculation is based on a high-efficiency fan and aftercooling to the wet bulb temperature. The fan temperature rise is 17°F (9.4°C).

Table 4-1
W301 Supercharger Performance Enhancement

Item	Westinghouse (1968)	SCAAD
Power w/o Supercharging, MW	27.7	27.7
Heat Rate w/o Supercharging, Btu/kW-hr (kJ/kW-hr)	13,900 (14,662)	13,900 (14,662)
Gross Power with Supercharging, MW	34.2	34.1
Net Power, MW	31.4	32.1
Fan Power, MW	2.8	2.0
Net Heat Rate, Btu/kW-hr (kJ/kW-hr)	13,440 (14,177)	13,100 (13,818)

The use of a variable pitch axial fan has been proposed [16] to control the pressure rise and ensure that the fan operates efficiently when the fan and compressor air flows are matched. Utility-grade variable pitch fans as shown in Figure 4-1 have been used for well over 25 years in steam boilers in power stations. There are multiple supplier sources and qualified field services available for maintenance. A pressure rise of 40 inches of water (100 mbar) may be accomplished in a single stage.

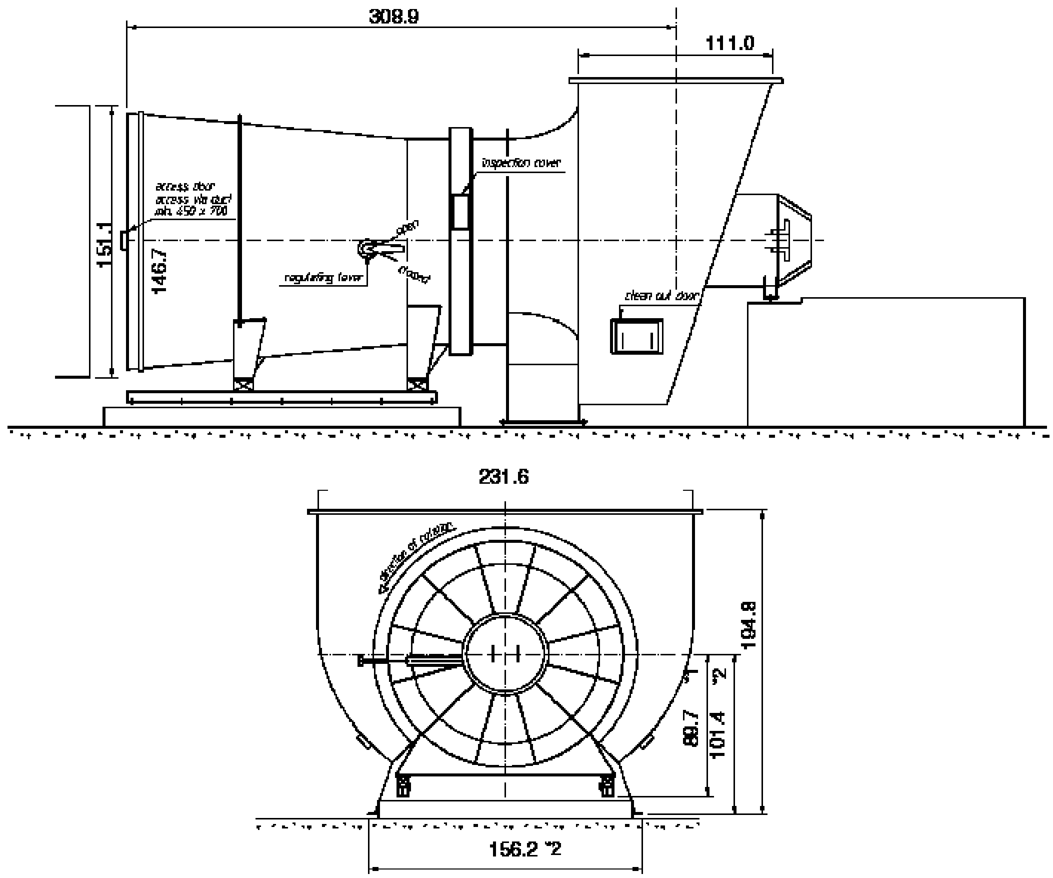
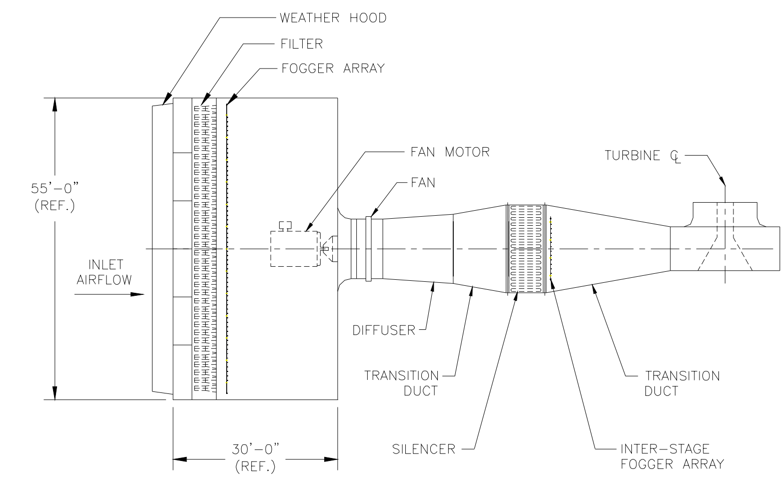
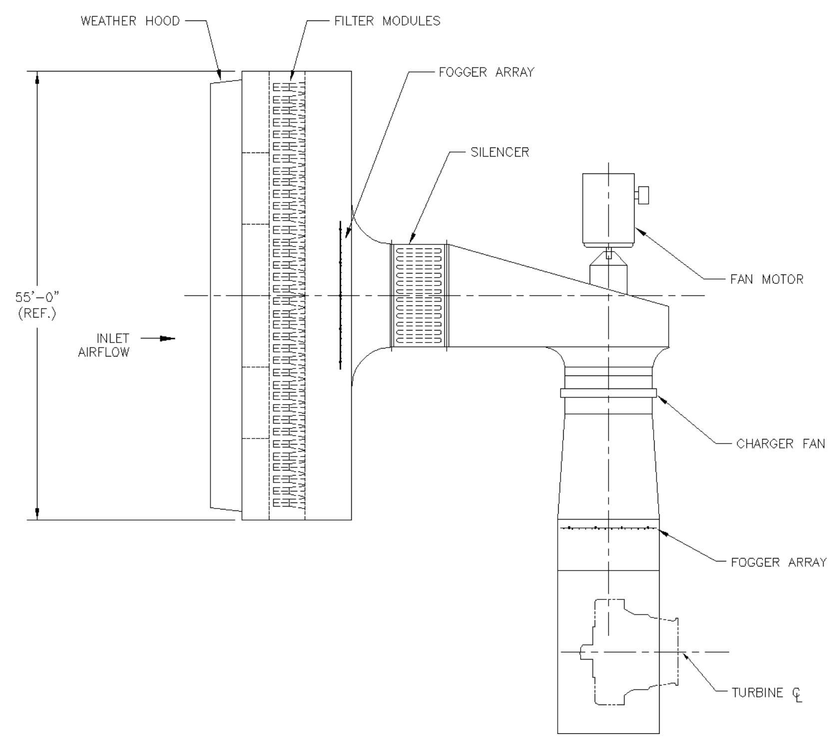


Figure 4-1
Typical Variable Pitch Axial Fan (Courtesy Howden)

Optional inlet duct arrangements with a supercharger are shown in Figure 4-2. The installed equipment consists of: 1) a fan-motor set, 2) a step-down transformer, a motor starter, protection and controls, and 3) a fogger aftercooler. The size noted is for an F-type CT with an air flow of 800,000 cfm (22,656 cmm) and a pressure rise of 40 inches of water (100 mbar). A 4.4 MW, 6-pole, 1200 rpm motor is applicable. The estimated cost of the fan-motor set is ~$1 million.



(a) Ducted Fan-Motor Set (Plan View)



(b) External Motor Drive (Plan View)

Figure 4-2
Schematic Supercharger Arrangements (m = ft/3.280833)

The fan performance is shown in Figure 4-3. The peak efficiency is 89% at the design point. The startup schedule would minimize the fan power until the CT is at base load and the full fan power load is then taken off the CT generator output. The variable vane angle would be scheduled to match the CT air flow at minimum power until the CT is at base load. As noted in Figure 4-3, the change in flow can be accommodated by the fan.

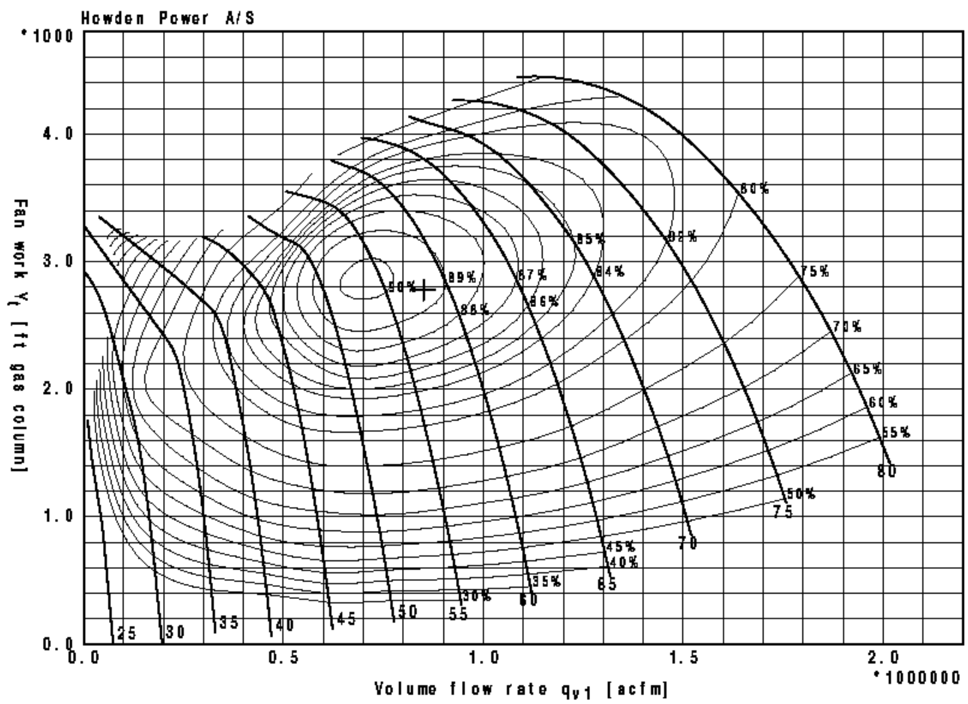


Figure 4-3
Typical Variable Pitch Axial Fan Performance (Courtesy Howden)

The design difficulties of a supercharger mount as the pressure rise increases such as:

- A fan pressure rise over 40 inches of water (100 mbar) requires two stages when using aluminum blades and a low speed motor.
- Hot section stresses increase with increased pressure rise.
- The inlet filter house and ducting must accommodate a higher airflow as the pressure rise increases.
- A higher pressure rise increases the pressure ratio across the turbine and lowers the exhaust temperature.
- Duct stiffening requirements increase with pressure rise.

Impacts of Supercharging on CTs

Stall/Surge Margin Envelope

The compressor pressure ratio is not expected to change significantly as the increase in flow is proportional to the increase in the compressor inlet pressure and the corrected flow tends to stay the same. Therefore, for the same inlet temperature, increasing the inlet pressure increases the compressor discharge pressure in proportion to the inlet pressure increase. This observation is important as it implies little change in the stall margin and the compressor discharge temperature unless the compressor inlet temperature changes.

Hot Section Parts Maintenance Interval

Compressor Discharge Temperature and Pressure

As the pressure ratio is not anticipated to change, the compressor discharge temperature is proportional to the compressor inlet temperature. As the discharge pressure rises approximately 10% for supercharging with 40 inches of water (100 mbar), there may be a reduction in hot section parts life due to higher stresses on the static parts. Modeling the effect on hot section parts life based on fatigue, a lower discharge temperature may offset the effects of increased pressure (stress). For example, for a 1st vane material such as FSX414, an increase in stress of 10% may be offset by a decrease in metal temperature of 25°F (13.8°C), which is estimated to correspond to a decrease in the cooling air temperature of 50°F (27.7°C).

Exhaust Flow and Temperature

The air flow increases in proportion to the increase in the compressor inlet pressure, but if the compressor inlet temperature rises due to fan heating, the net gain in flow is reduced. The exhaust flow may not increase in proportion to the pressure rise unless the compressor inlet temperature is cooled to the fan inlet temperature or lower. The exhaust temperature decreases due to the increase in the pressure ratio across the turbine section. A reduction in exhaust temperature impacts combined cycle performance in two ways: 1) the amount of energy in the exhaust is reduced and 2) the steam rate (lbs steam/kW-hr) increases. Sensitivity coefficients relating the loss in steam cycle power to a decrease in exhaust temperature were derived from parametric combined cycle studies. The exhaust temperature decrease may be as much as 25°F (13.8°C) with a corresponding steam cycle power loss of as much as 3.5%, offsetting some of the potential benefit of the increased flow.

Mechanical Design Envelope

The maximum pressure design limits (for low ambient temperature operation) limit the amount of supercharging. Increasing the compressor inlet pressure and air flow changes the thrust bearing loads. The design limits must be checked to verify that the thrust bearing is sufficiently robust to absorb the change.

Emissions

For diffusion flame type combustors, NO_x emissions are approximately proportional to the square root of the pressure, but very sensitive to moisture content and compressor discharge temperature (which affects the flame temperature). For the inter-stage hybrid case, the compressor discharge temperature is reduced markedly, so the NO_x emissions are likely to be lower. For the fogger case, as moisture is added and the compressor discharge temperature is reduced, the NO_x concentrations are reduced.

Icing Protection

In cold weather operation, the temperature rise resulting from the fan may be sufficient to avoid the need for additional inlet heating. Typical heating requirements are shown in Figure 4-4. If the full pressure rise of the fan is used, the temperature rise is approximately 16°F (8.9°C) for 40 inches of water (100 mbar), which appears adequate to prevent icing.

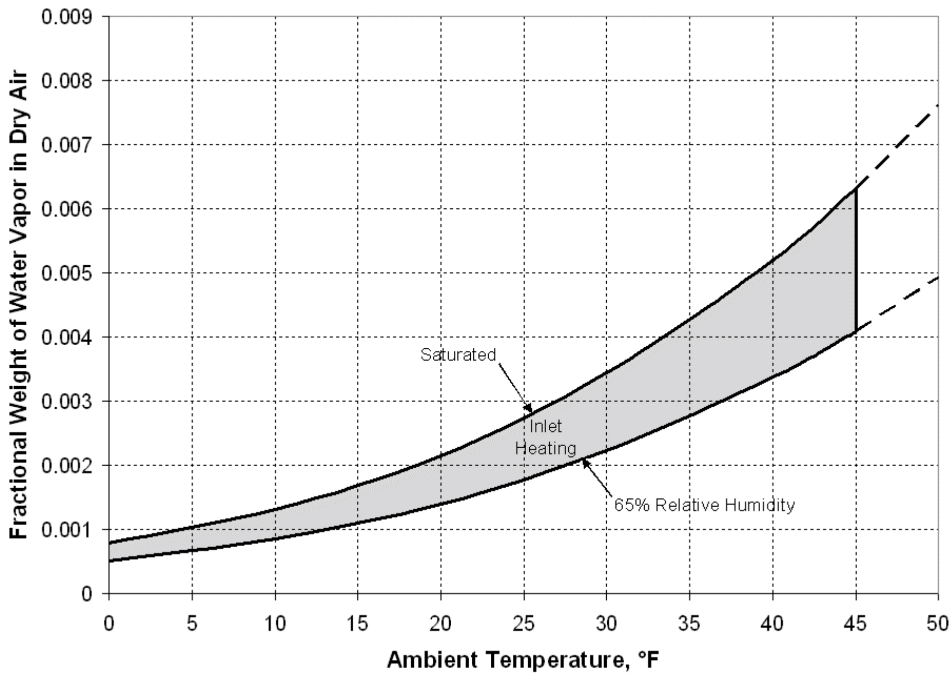


Figure 4-4
Heating Requirements to Avoid Icing (°C = (°F – 32)/1.8)

Performance of CTs Using Supercharging

Performance results with supercharging a simple cycle CT show an increase of ~10% and a decrease in heat rate of ~1.5%. For a combined cycle, the incremental power is ~8% and the heat rate increases ~1%. The results are based on a control algorithm that maintains a constant firing temperature. As noted earlier, the cycle model is based on evaporative cooling upstream of the fan, so the air entering the fan is nearly saturated and the incremental performance is for the supercharger only. The results include the auxiliary power requirements which are modest, as the ratio of net incremental combined cycle power output to the auxiliary power is ~3.8:1. When the benefits of evaporative cooling are included, the total increase in combined cycle power at the maximum site temperature condition is 22% for a dry site such as Las Vegas and 12% for humid site such as Miami. Hybrid system performance is given for dry and humid sites and simple and combined cycles in Figure 4-5, Figure 4-6, Figure 4-7, Figure 4-8 and Figure 4-9.

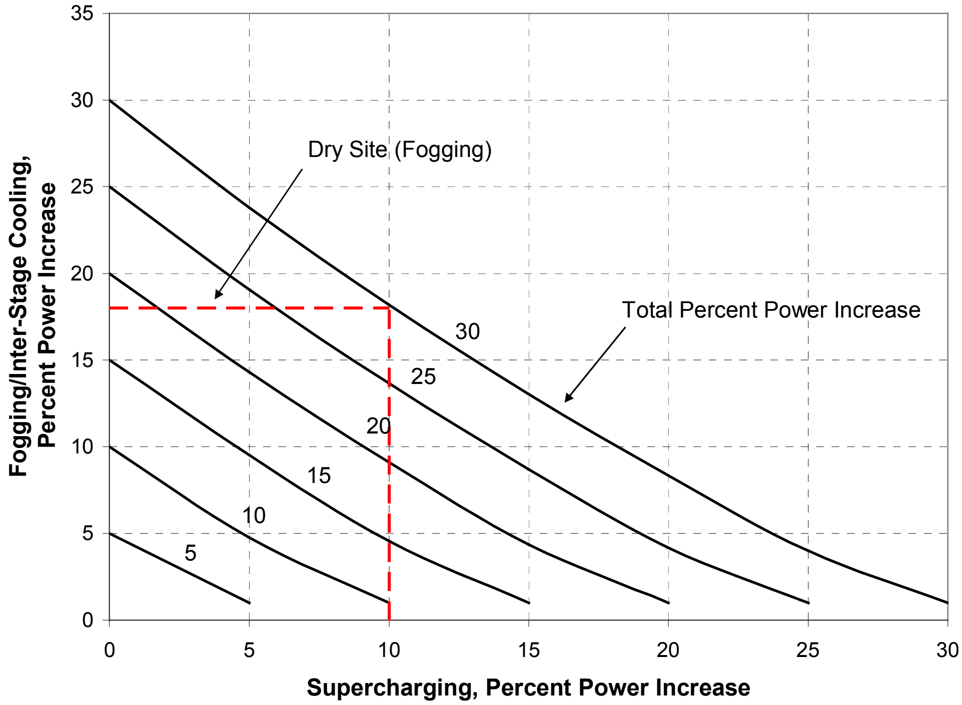


Figure 4-5
Hybrid Simple Cycle Fogging/Inter-Stage Cooling/Supercharging Concept (at High Ambient Temperature)

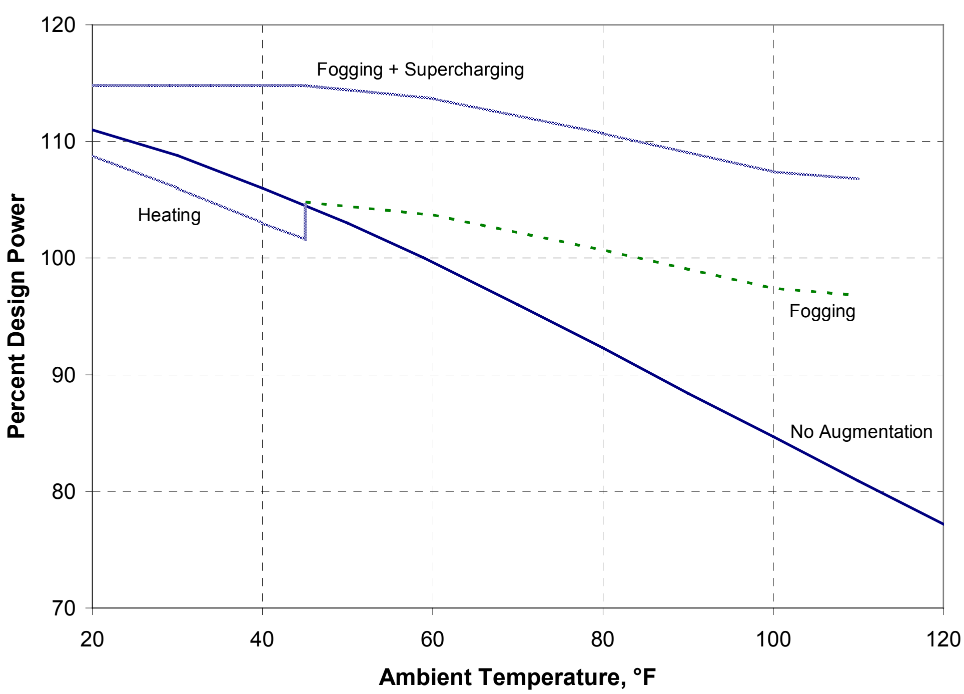


Figure 4-6
Simple Cycle Maximum Capability of Fogging + Supercharging Example, Las Vegas (°C = (°F – 32)/1.8)

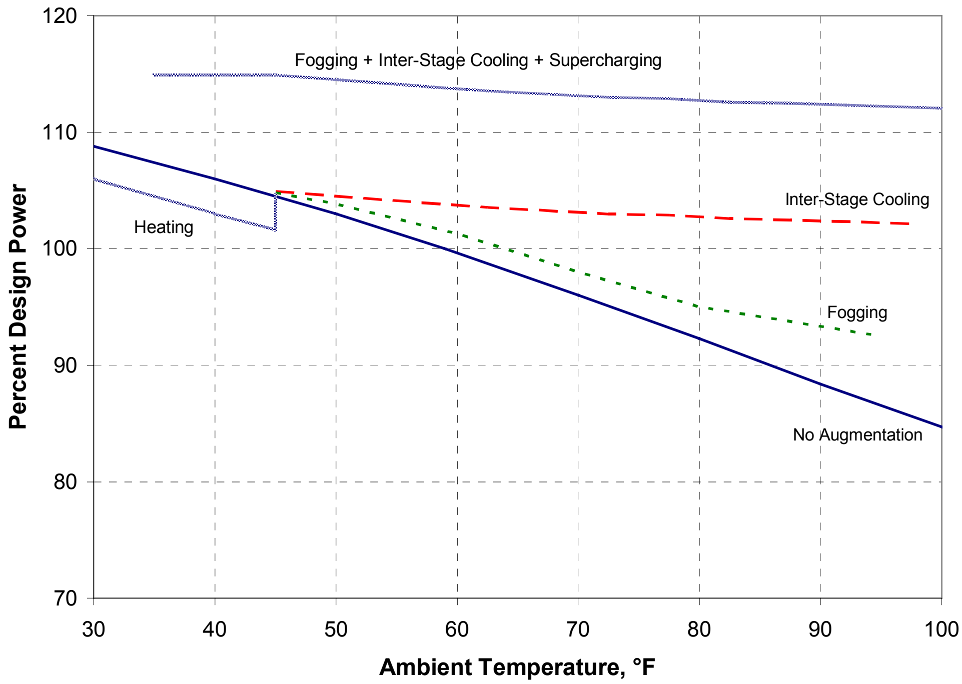


Figure 4-7
Simple Cycle Maximum Capability of Fogging + Inter-Stage Cooling + Supercharging Example, Miami (°C = (°F – 32)/1.8)

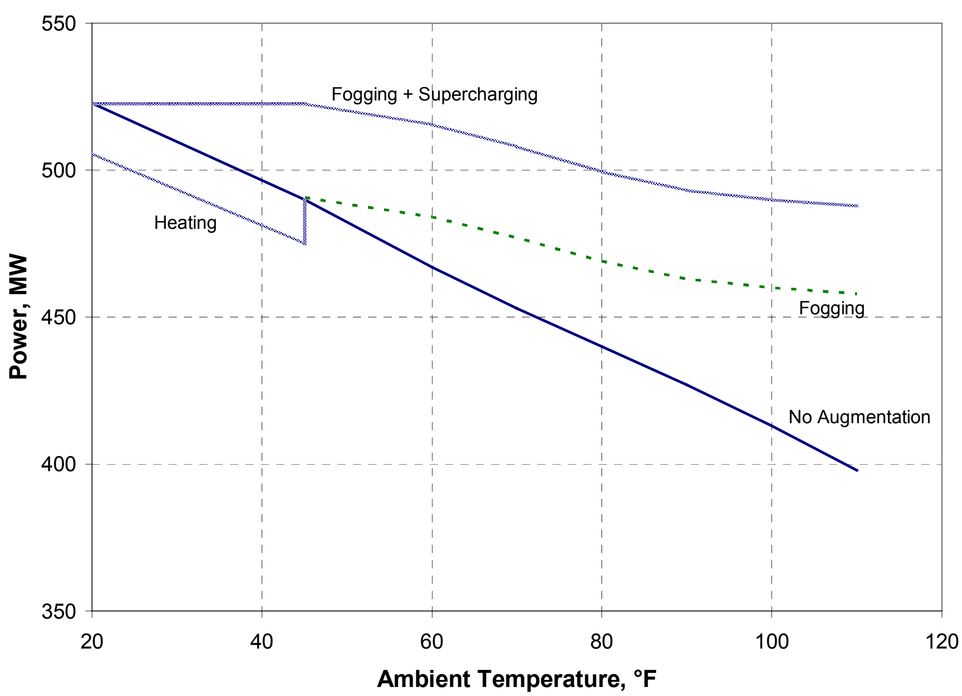


Figure 4-8
Combined Cycle Maximum Capability of Fogging + Supercharging Example, Las Vegas (°C = (°F – 32)/1.8)

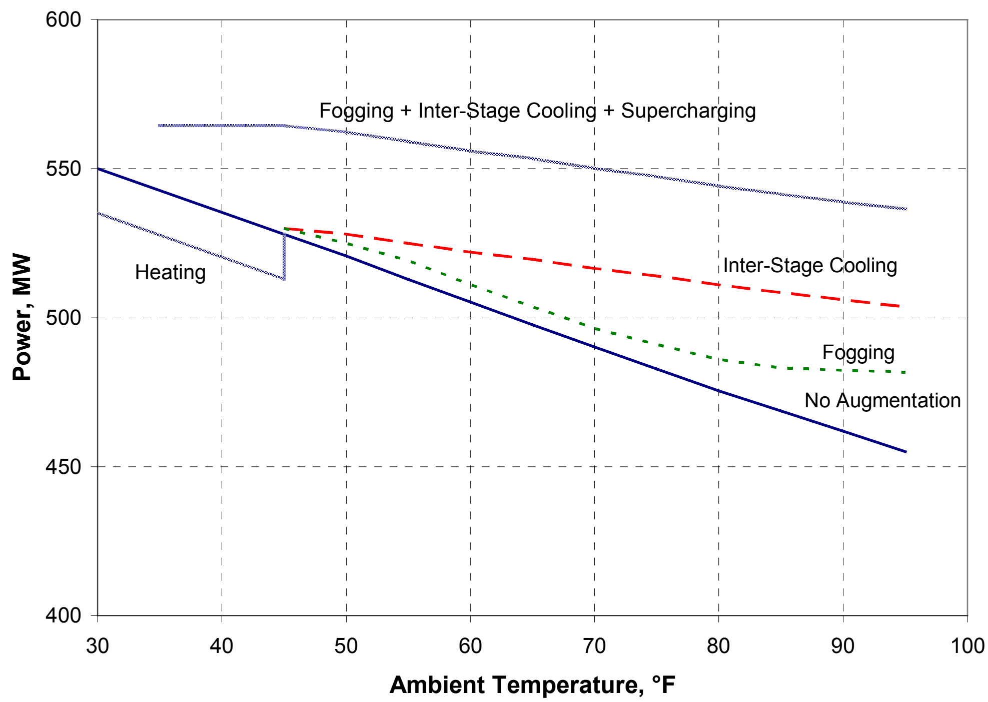


Figure 4-9
Combined Cycle Maximum Capability of Fogging + Inter-Stage Cooling + Supercharging Example, Miami (°C = (°F – 32)/1.8)

Because the use of inter-stage cooling results in a low compressor discharge temperature, an opportunity may exist to increase the firing temperature without affecting the hot section life or increasing emissions beyond the base case. Preliminary analyses show that an increase of the firing temperature of 35°F (19.4°C) results in an exhaust temperature that is the same as for the base case. For a combined cycle with: supercharging 40 inches of water (100 mbar), 1.5% water injection for inter-stage-cooling and an increase in firing temperature of 35°F (19.4°C), the estimated net combined cycle power increases 20%, while the net heat rate increases 1.5%. The inlet condition is saturated air at 80°F (26.7°C) on the basis that evaporative cooling can reduce a higher ambient temperature to 80°F (26.7°C). The auxiliary power is primarily that required for the supercharger and is less than 10% of the net gain in power. The estimated net cost/benefit ratio is $100/kW.

5 HUMID AIR INJECTION (HAI)

Background of Humid Air Injection

The evaporative gas turbine cycle, also known as the humid air turbine (HAT) cycle, has been studied extensively [17]. There are many variants to the cycle using the added moisture to increase the CT power output, increase thermal efficiency and reduce NO_x emissions. Typically, part of the compressor discharge air is passed through a humidification tower or saturator. The saturator recovers heat from the exhaust to heat the humid air mixture to achieve the desired saturation level. The humid air mixture may then be further heated with a heat recovery super-heater to achieve a high thermal efficiency. A retrofit package design suitable for adapting a simple cycle CT to humid air injection that would provide for air extraction of the compressor discharge air and return to the combustor after humidification is patented [18].

A proof-of-concept demonstration test engine has validated the theoretical models [19]. The test engine is a VT 600 Volvo gas turbine with a rated output of 600 kW. The HAT modifications resulted in a output power increase of 50%, an increase in thermal efficiency of 15% and a reduction in NO_x emissions of 95%. A flue gas water condenser is included in the test configuration such that no additional water makeup is required. The HAT approach is a potential replacement to a combined cycle such that there is no bottoming steam cycle. Economic studies show that the thermal efficiency performance is comparable to that of a combined cycle. There may be an economic issue of scale, as economic studies are oriented to small or medium size combustion turbines where combined cycle costs are high. A major cost factor is the saturator, which can be a relatively large custom design packed bed humidifier. An estimate of the costs of the humidifier indicated a cost of $400,000 for an air flow of approximately 60 lbs/sec (27.2 kg/sec) [20]. Part flow humidification of ~30% of the inlet air is thermodynamically advantageous to reduce the flue gas exhaust temperature to low levels suitable for water recovery. For a CT the size of a GE 7FA, the air flow requirement through the humidifier would be approximately 300 lbs/sec (136 kg/sec), so the humidifier cost could be as much as $2 million. The optimum CT would be of a new design tailored for the optimum pressure ratio, recuperator, humidifier, super-heater and water condenser.

To bring the technology of humid air injection to the market, a retrofit approach is being developed and a proof of concept test was successfully completed on a GE 7FA CT [21]. Initial testing has shown that the technology is promising, but costs are high and there is a need for de-mineralized water. A dry air injection version of this approach is also being explored.

A new approach for simple cycle CTs [22] that is expected to lead to lower costs and avoid the need for de-mineralized water is shown in Figure 5-1. This approach eliminates the need for a large custom-engineered saturator and utilizes developed components. A once-through partial boiler or wet steam HRSG is proposed. Wet steam HRSGs are widely used in CT cogeneration systems for enhanced oil recovery operations. The steam output may have a quality as low as 80%, such that the water serves to continuously blow down deposits in the piping. The wet steam is sent to a separator and mist eliminator and is then mixed with compressed air to obtain the desired humid air mixture ratio. The humid air mixture is superheated prior to injection into the CT. The new approach is aimed at using lower cost materials and developed components and reducing complexity. The performance of the HAI system may be unchanged from the previous approach. The estimated installed costs for a simple cycle CT are targeted for ~$150/kW.

An application to combined cycles is shown in Figure 5-2, where intermediate pressure steam is extracted for humidifying the air and a recuperator is installed at the front of the HRSG for superheating the humid air mixture. Humid air injection increases the compressor pressure ratio and reduces the stall margin. The addition of moisture is important for reducing emissions, but tends to increase the hot section heat transfer, which may reduce the hot section parts life [10].

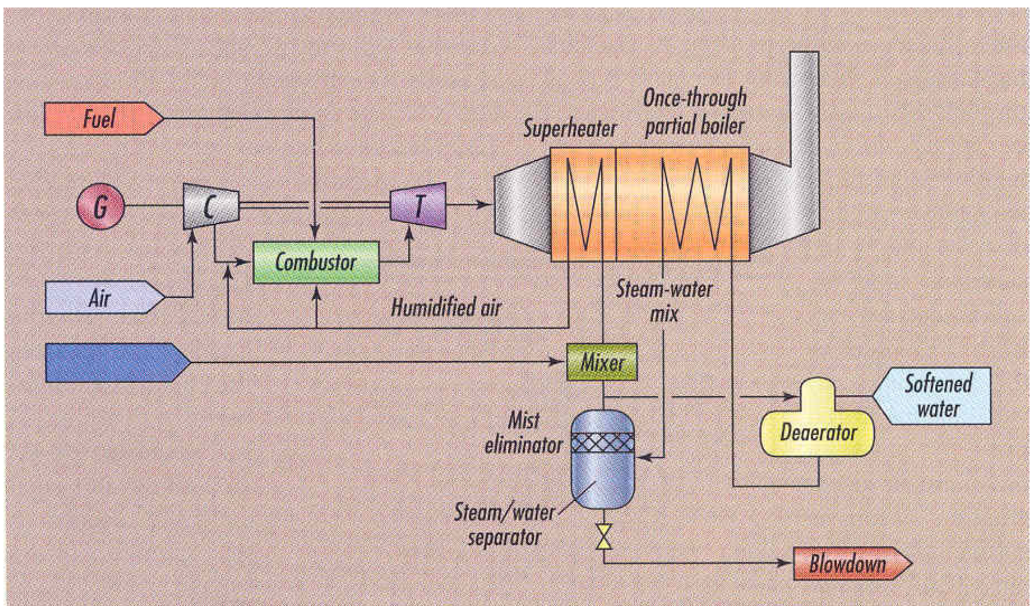


Figure 5-1
CT-HAI with Once-Through Partial Steam Generator [22]

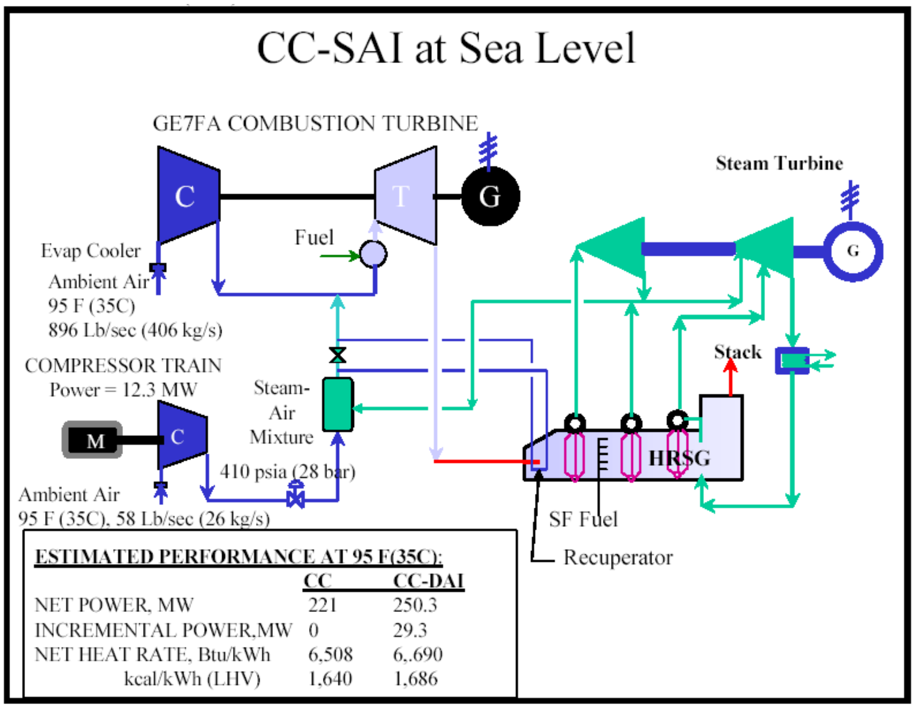


Figure 5-2
Steam/Air Injected Combined Cycle Concept [22]

Impacts of Humid Air Injection on CTs

The impacts of humid air injection are similar to those for straight steam injection. The injection mass flow is limited by OEM specifications. The injection increases the pressure ratio of the compressor, the cooling air temperatures, and the heat transfer to the hot section parts. OEMs have estimated the reduced maintenance intervals of the hot section [10] due to steam injection. For the same mass flow, humid air injection introduces less moisture, so the impact on heat transfer is reduced, but on the other hand, the benefits for NO_x reduction are also reduced.

Performance of CTs Using Humid Air Injection

Performance estimates [22] for a simple cycle CT show that the net output power is increased 18% for a power benefit ratio of 1.8% increase in power for each 1% of injected mass flow. This compares with steam injection, which may be as much as 3.6% per 1% of steam injection (including the added exhaust pressure drop). The ratio of incremental power to auxiliary power is 2.22.

The estimated performance for a combined cycle [22] shows an incremental power increase of 13.1% and a ratio of incremental power to auxiliary power of 2.38. The combined cycle application uses a steam turbine extraction and requires demineralized water. The recuperator increases the HRSG pressure drop and lowers the exhaust gas temperature for the steam cycle.

Independent performance estimates agree closely with those noted above. The auxiliary power is relatively high as the ratio of incremental output to auxiliary power is 2.4:1 for a combined cycle. The performance of a fogger/humid air injection hybrid system is shown in Figure 5-3 for a humid site.

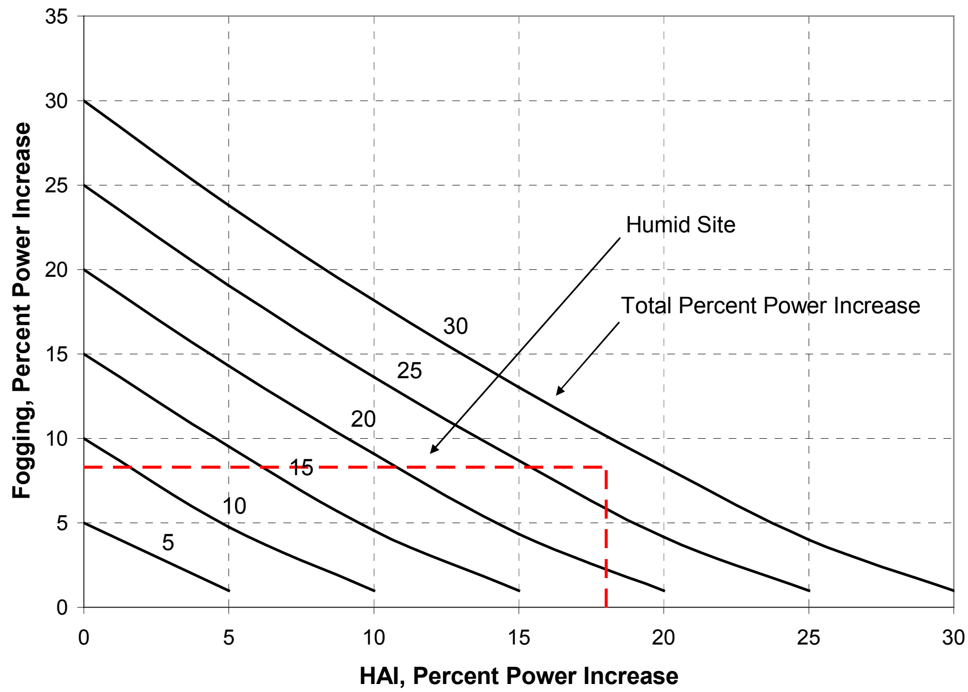


Figure 5-3
Hybrid Fogging/HAI Concept

6
STEAM INJECTION

Background of Steam Injection

The use of steam injection with simple cycle CTs is a mature technology, but there are important developments that may see an expansion of this application. The use of the once-through steam generator (OTSG) is compatible with the cycling requirements of peaking applications. The installed OTSG cost/benefit ratio may be as low as $170/kW for a simple cycle retrofit.

A new concept [23] known as the Cheng Low NO_x (CLN) system is retrofitted to existing combustors, but with steam injected in a more optimum manner [24]. The CLN concept is shown in Figure 6-1. Pre-mixed steam and fuel are injected into the front end of the combustor and steam is also diluted with the air flow by injection through the liner. The CLN system achieves NO_x emissions in the single-digit range [24] without the need of an SCR.

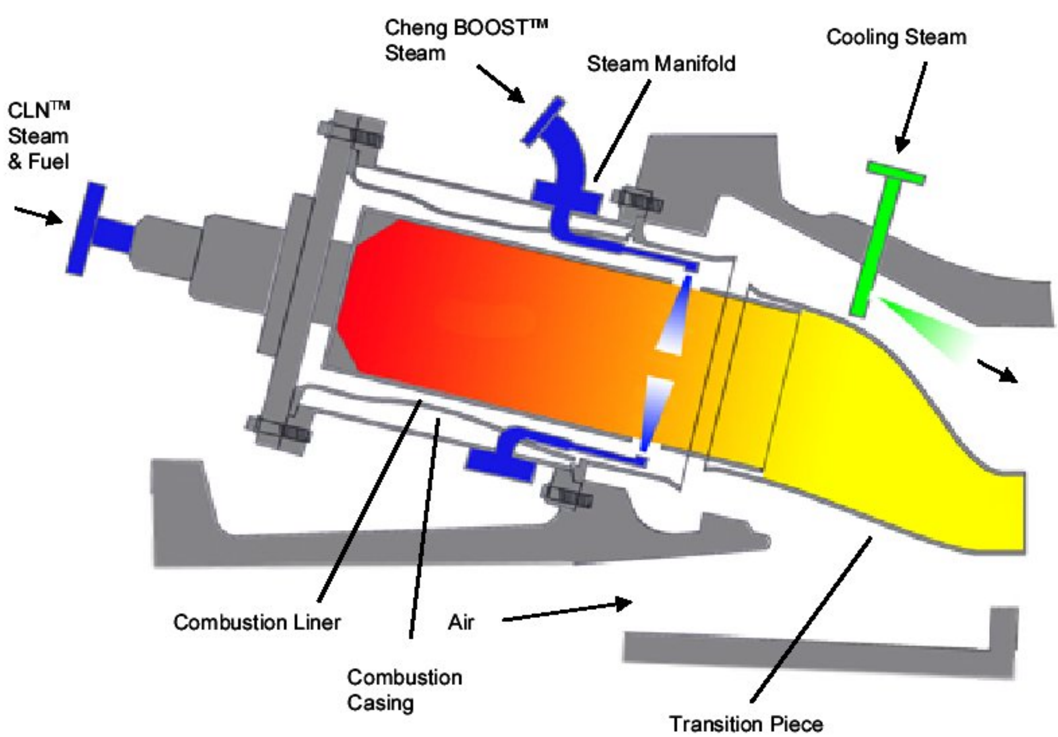


Figure 6-1
Cheng Low NO_x (CLN) System Concept

Impacts of Steam Injection on CTs

The impact of steam injection has been linked to a reduced life for the hot section parts due to the change in transport properties (added moisture increases heat transfer) and the increased compressor discharge pressure and temperature due to the added mass flow [10]. The expanded use of steam injection is also being pursued by seeking axial compressor blade designs with increased stall margin [25].

Performance of CTs Using Steam Injection

The estimated incremental steam injection performance estimates for simple and combined cycles show an incremental power increase of ~13.5% for simple cycles and 8.5% for combined cycles. The heat rate for simple cycles decreases ~4% but the change for combined cycles is small. A constant firing temperature control algorithm is assumed. The trend of the results shows little effect of the ambient temperature.

The potential use of steam injection in concert with chilled water inlet air cooling may be attractive option at humid sites such as Miami. The combined cycle power benefit for chilling with steam injection is estimated at 19%. As seen in Table 1-1, this would enable the plant to be flat-rated, so the same amount of power output is available at all ambient conditions (based on the weather data used).

In cases where there is a stringent limit on NO_x emissions, the use of steam injection and an SCR may be applicable. This approach to NO_x abatement is expected to achieve NO_x concentration levels in the single-digit range. As the exhaust temperature of a simple cycle CT such as the GE 7FA may be above 1100°F (593°C), a HRSG can reduce the exhaust to a temperature of ~800°F (427°C) for efficient conventional SCR catalysts. To reduce the temperature to 800°F (427°C) would require an HRSG capacity of ~200,000 lbs/hr (90,718 kg/hr) and an injection rate of 6% of the air flow. Using a high temperature catalyst would enable the steam injection rate to drop to 3% and meet the OEM injection limits.

Testing of a CLN low NO_x emission modification to existing diffusion type combustors has demonstrated single digit emission levels [17, 18]. Development testing was done on a cogeneration plant with a GE Frame 6 CT. Very low NO_x and CO emission levels were obtained as a result of tuning the fuel nozzle configuration. The installed cost for the 1st unit was $244/kW using steam from the existing HRSG.

7
O&M PRACTICES

Parts Repair & Improvements

As a result of the rapid growth in the installed fleet of CTs in the 1990-2000 time period, new opportunities have opened up for parts suppliers so that investments are being made in special tooling, testing, and analytical capability in order to supply aftermarket parts. Improved hot section parts are being offered that [26]:

- Improve cooling by changing the internal flow paths
- Improve life by changing materials
- Improve efficiency by changing design
- Improve output power by changing design
- Reduce NO_x to single digits

Fully-compatible OEM replacement parts and low NO_x combustion systems are available from aftermarket parts suppliers for: 5P, 6B, 7FA, 9E, 501F, and 11N CTs. It is estimated that aftermarket parts may be lower in price than OEM parts by as much as 40% [26].

Repair and design improvements for the following CTs have been evolved [27] to provide life extension and uprating:

- 7FA 1st Stage Bucket (Blade)
- V64.3 1st Stage Blade
- V84.2 2nd Stage Vane
- V84.2 Hot Gas Casing
- V64.2 and V84.3 Burner Bezel Rings
- W251B12 Transition Piece
- W501F 1st Stage Blade
- 7EA 1st Stage Bucket (Blade)

Improved parts may result in raising the firing temperature limit. If the firing temperature can be increased 50°F (27.8°C), then the projected increased performance for a SC GE 7FA is a 4% improvement in output, a 0.5% reduction in heat rate, and an increase in exhaust temperature of 27°F (15°C). As the exhaust temperature increases, a combined cycle would also have an output increase of approximately 4% and a small reduction in heat rate.

On-Condition Maintenance

Improved modeling and monitoring techniques are being evolved to better estimate the condition of critical parts and their remaining life [28]. Real-time data collected by the CT control system is processed by a computerized engine model for estimating the average metal temperatures and stresses of hot section components. A running assessment of total of the total equivalent operating hours is compared with the rated allowable equivalent hours. The life fraction rule for variable load conditions is used to calculate the equivalent hours.

Conditions monitored for maintenance may include:

- Heat Rate Increase
- Air Flow Decrease
- Power Decrease
- Fatigue Life of Hot Section Parts
- Compressor Efficiency Decrease
- Exhaust Temperature Spread
- Lubrication Oil Sample Analysis
- Fuel Contamination
- Emissions Increase
- Combustor Noise Amplitudes
- Vibration Protection
- Parts-Specific Health Criteria
- Borescope Inspections that Show Coating Condition, Cracks, FOD
- Surface Inspections of Erosion and Roughness
- Visual Inspections of Oil Leaks
- Calibration Interval Checks

Axial Compressor Performance Maintenance

Improved on-line water wash systems are being developed by optimizing the nozzle design and locations such that the cleaning is more complete and erosion is avoided [29]. An assessment of the relative sensitivity of the degradation of each blade row in an axial compressor revealed that the 1st row is most important and the sensitivity decreases for higher rows. The increase in blade roughness was found to be a critical factor in causing a loss in compressor efficiency. The potential benefits of periodic cleaning and/or refurbishment to achieve hydraulic smoothness suggest that monitoring the roughness of the airfoils [30] is useful to establish an optimum maintenance interval.

Greenhouse Gas Reduction

Greenhouse gas reductions arise when projects are instituted that improve efficiency and reduce fuel use [31]. A wide array of options to reduce fuel consumption are presented in this study whereby the efficiency of existing units may be increased or whereby the output of high-efficiency units may be increased, displacing older, less efficient units.

8
SUMMARY

Technical Concerns Summary

A summary of the technical concerns for the capacity enhancement technologies studied is given in Table 8-1. The impact on NO_x emissions is small for inlet cooling and supercharging technologies. Strong reductions in NO_x emissions are likely for technologies that inject moisture into the compressor and/or the combustor. Single digit NO_x emissions with diffusion flame combustors may be obtained with steam injection and a SCR or with the Cheng CLN technology.

Table 8-1
Summary of Technology Concerns

Technology	Concerns
Evaporative Cooling - Media Type	Compressor Airfoil Fouling & Erosion
Evaporative Cooling - Fogging	Compressor Airfoil Erosion
Indirect Evaporative Cooling	Scale for Large CTs, Need for Validation Testing
Chilling/TES - Peaking	Off-Peak Auxiliary Power, Complexity
Chilling - Base Load	Auxiliary Power, Complexity
Compressor Inter-Stage Cooling	Compressor Airfoil Erosion, Hot Section Maintenance Interval
Supercharging	Mechanical Design Envelope, Need for Validation Testing
Steam Injection	Hot Section Maintenance Interval, De-mineralized Water Requirements
Humid Air Injection	Auxiliary Power, Stall Limits, De-mineralized Water Requirements
O&M Practices	Warranties and Guarantees

Performance Summary

The incremental performance estimates for dry and humid sites are given in Table 8-2 and Table 8-3. In each case, the results are for the maximum site temperature, unless noted. Several hybrid systems have the potential for achieving a flat rated capability under variable site weather conditions. However, some of the promising technologies are in need of validation testing before commercialization may start.

Table 8-2
Simple Cycle CT Incremental Performance Summary

Technologies	Las Vegas Site		Miami Site	
	Net Pwr., % (Aux. Pwr., %)	HR, %	Net Pwr., % (Aux. Pwr., %)	HR, %
Evaporative Cooling	21	-5	6	-1
Chilling - Peaking	28	-6	19	-4
Supercharging (40 in. water) & Evaporative Cooling	31 (3)	-6	16 (3)	-2
Inter-Stage Cooling (1.5% water injection) & Evaporative Cooling	31	-5	16	-1
Humid Air Injection (3.5% moisture, 10% total) & Evaporative Cooling @ 95°F, 60% RH	18 (8)	-4		
Steam Injection (3.5%) & Evaporative Cooling	34	-9	15	-5

Table 8-3
Combined Cycle Incremental Performance Summary

Technologies	Las Vegas Site		Miami Site	
	Net Pwr., % (Aux. Pwr., %)	HR, %	Net Pwr., % (Aux. Pwr., %)	HR, %
Evaporative Cooling	15	~0	5	~0
Chilling	20 (1)	~0	10 (2)	~0
Supercharging (40 in. water) & Evaporative Cooling	24 (2)	~0	14 (2)	~0
Inter-Stage Cooling (1.5% water injection) & Evaporative Cooling	22	+2	12	+2
Humid Air Injection (3.5% moisture, 10% total) & Evaporative Cooling @ 95°F, 60% RH			13 (5.5)	+2.9
Steam Injection (3.5%) & Evaporative Cooling	23	~0	13	~0

Cost/Benefit Summary

The estimated cost/benefit ratios are given in Table 8-4 and Table 8-5.

Table 8-4
Summary of Simple Cycle Lowest Cost/Benefit Estimates

Technology	Cost/Benefit Ratio, $/kW
Evaporative Cooling - Media Type	10
Evaporative Cooling - Fogging	10
Chilling/TES - Peaking (4 hrs/day)	100
Compressor Inter-Stage Cooling (2.5% water injection)	35
Supercharging (40 in. water)	150
Steam Injection (3.5% steam)	170
Humid Air Injection (3.5% moisture)	150

Table 8-5
Summary of Combined Cycle Cost/Benefit Estimates

Technology	Las Vegas Site Cost/Benefit Ratio, $/kW	Miami Site Cost Benefit Ratio, $/kW
Evaporative Cooling - Media Type	8	8
Evaporative Cooling - Fogging	8	8
Chilling - Base Load	67 (with Indirect Evaporative Cooling)	200
Compressor Inter-Stage Cooling (2.5% water injection) & Evaporative Cooling	12	30
Supercharging (40 in. water) & Evaporative Cooling	60	100
Steam Injection (3.5% steam) & Evaporative Cooling	30	60
Humid Air Injection (3.5% moisture) & Evaporative Cooling (95°F, 60% RH)		100

9
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