APPLICATION OF WET COMPRESSION FOR AERODERIVITIVE COMBUSTION TURBINES

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by

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ABSTRACT:

While the application of wet compression to boost turbine capacity has been successfully applied to industrial frame machines, only recently has this technology been applied to aeroderivatives like the LM-2500. The principles of applying wet compression are similar to the industrial frame machines, but there are subtle differences that affect the design of the spray system, inlet duct modifications, control system integration, and turbine protective devices. This paper discusses design differences and the performance enhancement of an LM-2500PE that was retrofit with wet compression at PurEnergy's Kingsburg Cogeneration Facility in Kingsburg, California. The LM-2500 is an interesting application because the compressor and power turbine are on different shafts which introduce speed and pressure variations between the gas generator and power turbine that affect the performance augmentation resulting from wet compression. Operating limits for surge and speed were also encountered which were not previously encountered on the single shaft industrial frames. The incremental performance per gallon of water is notably different from the industrial turbines which will be shown for comparison of the augmentation results along with a comparison of wet compression to other power augmentation technologies.

INTRODUCTION

Wet compression is the process in which water in the form of fine droplets is intentionally sprayed into the compressor inlet, where they will evaporate within the blade path to provide a thermodynamic intercooling affect. As the air gets heated by the work of compression, its tendency to absorb moisture increases. The resulting adiabatic process causes the air temperature to drop, relative to dry operating conditions. Since it takes less energy to compress relatively cooler air, there is savings in compressor work. Any reduction in compressor work translates to increase in net turbine out because one-half to two-thirds of turbine output is typically used to drive the compressor.

The "cooling" of the compressor air causes a reduction in the compressor discharge temperature. With single shaft industrial gas turbines a 1% overspray causes a 50°F reduction in compressor discharge temperature, where the percent overspray is the amount of water carried into the compressor is expressed as a percentage of inlet air flow. Measurement of compressor discharge temperature were not available within the LM-2500 and testing of compressor discharge temperature was not monitored in this first application, although testing on a subsequent unit installed with wet compression showed a smaller reduction in compressor discharge temperature.

A reduction in compressor discharge temperature allows more fuel to be fired in the combustor maintain a constant firing temperature. This further boosts the mass flow through the turbine and contributes to the overall gain in turbine power output.

The increase in power output of the LM-2500 was expected to be similar to or greater than a single shaft industrial turbine because the gas generator spool piece is able to accelerate as compressor work is reduced and turbine power is increased, thereby resulting in a greater increase in mass flow rate through the compressor and turbine. Whereas, the power turbine would be operating at a fixed speed which is coupled to the electrical grid. In testing, the LM-2500 showed the expected increases in speed and pressure, and at times limited the amount of water that could be injected into the inlet air based on speed and pressure limits, but the power turbine did not produce the expected incremental gain power output or achieve the heat rate improvement exhibited by industrial turbines with similar percentages of overspray.

In the Kingsburg application, the wet compression system was installed and operated downstream of a media type evaporative cooling system, which means that the water injected through the wet compression system was providing an intercooling affect and did not significantly reduce compressor inlet air temperature. An equivalent amount of water used for evaporative cooling will produce greater output and efficiency gains than if it is used for wet compression, but the gain from wet compression is not as dependant on ambient weather conditions or and can be used in conjunction with fogging, evaporative cooling, or chilling for a compounded increase in turbine power output.

As part of the wet compression upgrade, Caldwell Energy made additional changes to the inlet duct system to clean it up and reduce the inlet pressure drop. Prior to the wet compression upgrade the inlet air system included weather hoods, an inertial type filtration system, media type evaporative cooler and mist eliminators, filter elements, silencing panels, and a trash screen placed immediately upstream of the compressor bellmouth. PurEnergy had requested that the inertial filtration system be removed since it was not considered to be effective and was causing a continuous pressure loss. PurEnergy also asked that the evaporative cooling unit be replaced with a fogging system to further reduce pressure losses and to increase the effectiveness of the system. Given the short inlet duct configuration and installation of wet compression spray headers, and full duct trash screen Caldwell recommended that the evaporative cooling unit be refurbished with the PurEnergy's stock of media elements. The silencers were also removed from the inlet since sound levels at this industrial site were not an issue resulting in a further reduction in inlet pressure losses and gain in performance.

Figure 1A – Kingsburg Inlet Duct Configuration

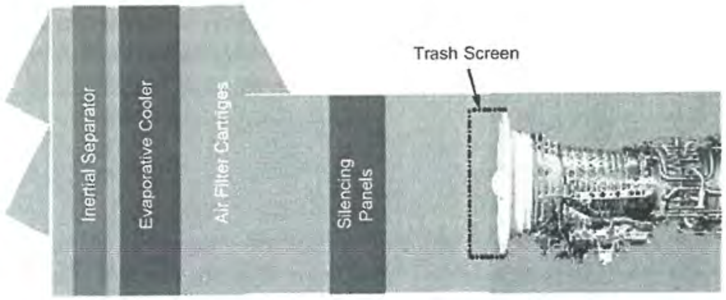

Figure 1B – Pre-Wet Compression Inlet Duct Configuration

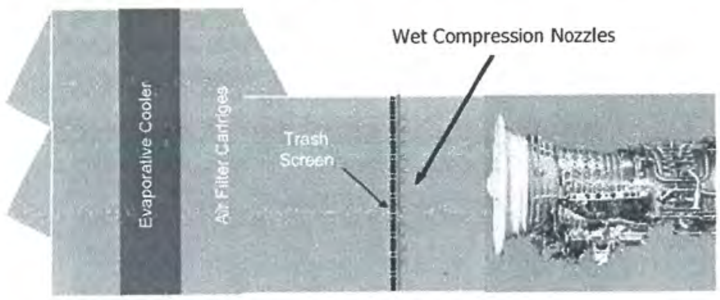

Figure 1C – Post-Wet Compression Inlet Duct Configuration

Wet Compression versus Fogging

Terminology like "high-fogging" for wet compression tends to suggest that fogging and wet compression are synonymous. They are not. Although, the process of spraying water into the air is similar but there are important differences. The goal of the fogging system is to evaporate all the water before the air reaches the compressor inlet. Therefore it is desirable to place the fogging nozzles as far away from the compressor inlet as possible so as to allow sufficient residence time for the droplets to evaporate.

For wet compression operation, the nozzle arrays are placed close to the compressor inlet to allow the droplets to go into the compressor without agglomeration and to minimize the wetting of duct walls, floors, silencing panels and structural members within the inlet duct, which have promoted compressor blade erosion with fogging system. The amount of water sprayed for wet compression is 2 to 5 times that required for fogging. At the same time, the cross-section available for array installation is smaller than for fogging system. There are design challenges with wet compression, with the higher air velocity and proximity of spray system hardware to the compressor inlet, which if liberated could damage the compressor.

Spraying water into a compressor should only be done after an engineering failure modes and effects analysis of doing this to the turbine and its auxiliary and control systems . The main issues with wet compression like blade erosion, uniform water distribution, compressor casing distortion, combustor dynamic pressure monitoring and electrostatic charge build-up on the rotor need to be addressed. It must be remembered that wet compression is not merely spraying water in the compressor inlet air and that there is a sophisticated turbine downstream that must be carefully monitored. All these issues that influences the design makes wet compression system different than a fogging system, which are generally based on the premise that the only effect to the turbine will be a decrease in the compressor inlet air temperature, which in applications such as the LM-2500 with its short inlet duct length and internal components would invalidate such an assumption.

SYSTEM DESIGN CONDITIONS:

The wet compression system was installed on LM-2500PE combustion turbine at the Purenergy's combined cycle plant located in Kingsburg, California. For this turbine, the gas generator (see Figure 2) is decoupled from the power turbine. The axial flow compressor is 16-stages with a compression ratio of 20:1 and is driven by a two stage high pressure turbine. Steam is injected into both combustor and Compressor Discharge Pressure (CDP) manifolds. The combustor is annular diffusion flame type. These fore mentioned components represent the gas generator. The power turbine consists of 6-axial flow stages. The LM-2500 turbine is 21.4' long with a diameter of 6.7' at the power turbine exhaust flange and weighs 10,300 lbs.

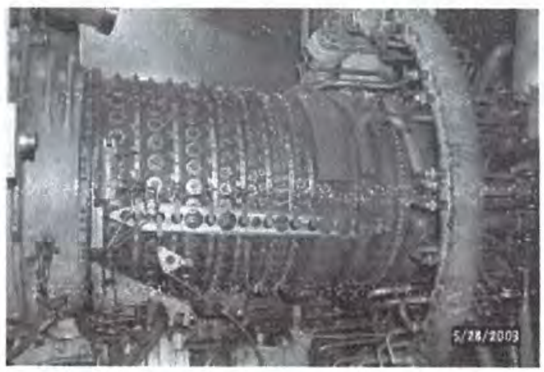

Figure 2 – LM-2500 Compressor Section with Steam Manifold

The design conditions for the Kingsburg plant were for an ambient dry bulb of 59ºF with air mass flow rate at 529,200 lb/hr. The site elevation was 30' above sea level and therefore insignificant. The ambient air temperature at the site will exceed 100 ºF, but due to the presence of an evaporative cooling unit will generally have compressor inlet temperatures in the range of 50ºF to 70ºF Water at a rate of approximately 22 gpm was sprayed into the compressor inlet for the purpose of wet compression. This flow corresponds to 2% overspray.

SYSTEM DESCRIPTION:

The wet compression system consists of high pressure pump skid, high pressure water transfer piping, a water spray array module using pressure atomization, inlet duct treatment and electronic control system. Contract award, system design, installation and commissioning for the project took place within a period of 30 days and helped the facility to offset power losses from the loss of its steam turbines last row blades on several occasions.

Because the flow rate for the system is 22 gallons per minute, the most cost effective pumping system is a positive displacement system similar to that used for fogging. Two pumps were required for full system augmentation; however a three pump skid (See Figure 3) was supplied allowing one to be used as a redundant spare or to allow for a future conversion to fogging if desired by the customer, along with other inlet duct modifications to promote evaporation of the water droplets.

Figure 3 – Wet Compression Pump Skid

The pumps were all connected to a common discharge manifold to enable pump operation to be scheduled to balance starts and hours of operation. The water spray distribution system was divided into 5 equal zones so that water could be gradually staged into service to minimize thermal transients associated with normal operation.

Since demineralized water is used for spray, all piping is stainless steel. There is a fine filter on the water supply to the pump skid. Water is delivered at a pressure of 3,000 psig to the spray zones. There are a total of 5 spray stages. Each stage is designed to provide uniform water distribution into the air stream. These five stages are operated independently and are brought on line one by one to full operation. The nozzles arrays consists of stainless steel piping, welded fittings and proprietary nozzles. The nozzles utilize machine threads with "O" ring seals for ease of maintenance and to reduce the chance for nozzle clogging from Teflon tape or pipe dope associated with sealing nozzles with NPT fittings. The nozzles are safety wired to prevent them from becoming loose and cause foreign object damage to the compressor.

The inlet duct downstream of the nozzle arrays was lined with stainless steel panels. A drain was added to the inlet duct plenum to carry away any accumulated water. A trap was provided in the drain to prevent objects from outside being pulled into the ductwork due to negative pressure inside. This drain is also used for disposing off water during initial system flushing commissioning activities.

As a result of reduction in compressor work, the speed of the compressor increased above the set point and it restricted the output. In order to realize higher gains, a change was made to increase the set point to allow higher compressor shaft speeds.

The control system is based on a PLC based control system that is mounted on the pump skid (see Figure 4) and interfaces with the distributed process control system. Interface with the plant control system allows start/shut down signal for wet compression system and obtain relevant operating parameters from the wet compression skid. Interface with the Woodward turbine control system was not required since its reaction time to upset transients was much more rapid than industrial gas turbines which require special logic helps to protect the turbine if the wet compression system trips. In the event the wet compression system trips, the gas supply to the turbine needs to be stepped down immediately to avoid an over-temperature event.

Figure 4 – Electrical Cabinet with PLC Controller and HMI Interface

Combustion Turbine Monitoring:

Uneven water distribution at the compressor inlet can cause uneven metal temperatures along the circumference of the casing that can lead to casing distortion. This can lead to uneven spacing between the casing and the blades and may cause "blade rub". It is very important to make sure that this undesirable phenomenon is avoided by all means. The application of a wet compression system to the LM-2500 was simplified by the axial inlet system design since the air did not have to make a 90 degree turn to enter the compressor as must be done with most industrial frame gas turbines.

In a subsequent LM-2500 application, the nozzle arrangement was modified to reduce the wetting of the nose cone at the inlet to the compressor. Further refinements of nozzle locations are being investigated and will likely be included in new applications. In addition, modifications will likely be made to the floor drainage system to be more effective in removing water from the duct while the system is operating.

Kingsburg operators have reported that the some water has been observed bubbling from the variable stator vane shafts on the exterior of the compressor cylinder (See Figure 2). This occurs because not all the water has evaporated in the front end of the compressor and if there are air leaks at these points water will be carried along with it. Studies performed by Dr. Michael Sexton, a professor at the Virginia Military Institute, suggest that the water will pass through more stages of the LM-2500 compressor since there is not as much axial length or residence time for the droplets to evaporate in an equivalent number of stages. Although this indicates that the water has not fully evaporated by the 6^{th} stage of the compressor, deposits and erosion patterns on industrial turbines suggest that the water does not fully evaporate by this stage either when 2% overspray systems are used.

Another important factor to consider in the design is the electrostatic charge build-up on the rotor due to the shearing of water droplets when they are impacted by the rotating compressor blades.. Discharge of electro static charges between the rotor and bearings of industrial gas turbines has resulted in a loss of journal bearing material and frosting appearance of the rotor surface supported by the bearings. In these cases, the loss of bearing material resulted in compressor blade rubs as the rotors settled within the compressor cylinder. For the LM-2500 application, the generator rotor already had a grounding brush and the bearings were roller type which provide a direct path to ground to take place before the static charges could build to a point where arcing at higher voltages would harm the bearings. Therefore, there was no need for an additional grounding brush.

Dynamic Pressure Monitoring:

Monitoring of dynamic pressure is important because excessive pulsations can lead to wear and fatigue of combustion system components. It could also affect flame stability and may require combustion tuning to richen the flame that may increase NOx emissions. The factors that effect dynamic pressure are reduction in compressor discharge temperature, increase in moisture content of compressor discharge air. While these influences are more pronounced when wet compression is used in conjunction with dry low NOx combustion systems, it is still advisable to monitor the combustion system response for at least first time applications.

The method used for inserting a dynamic pressure transducer into the combustion system was to machine a combustion system anti-rotation and fit it with a pressure transducer. Our industry research had indicated that this technique had been done on other LM series turbines, but this was a first for the LM-2500.

The results of the testing are shown in Figures 5A-C. Figure 5A represents operation without wet compression and only steam injection for NOx emission control. This point had a peak amplitude of 0.35 psi at a frequency of 352 Hz. Figure 5B shows combustion system dynamics with steam power augmentation showed an increase in both amplitude and the frequency of the pulsations. Figure 5C shows combustion system dynamic pressure with steam for NO control and power augmentation in conjunction with full wet compression system operation. Here the amplitude has risen to 1.12 psi at a corresponding frequency of 404 Hz. Test results actually showed that wet pressure levels were reduced when operating with 3 or 4 zones of wet compression and increased with the use of stage 5. These dynamic pressure levels at the frequencies observed were judged to be within the experience range of other turbines, with which we were familiar and were considered safe for continuous operation. Subsequent combustion system inspections have confirmed this evaluation.

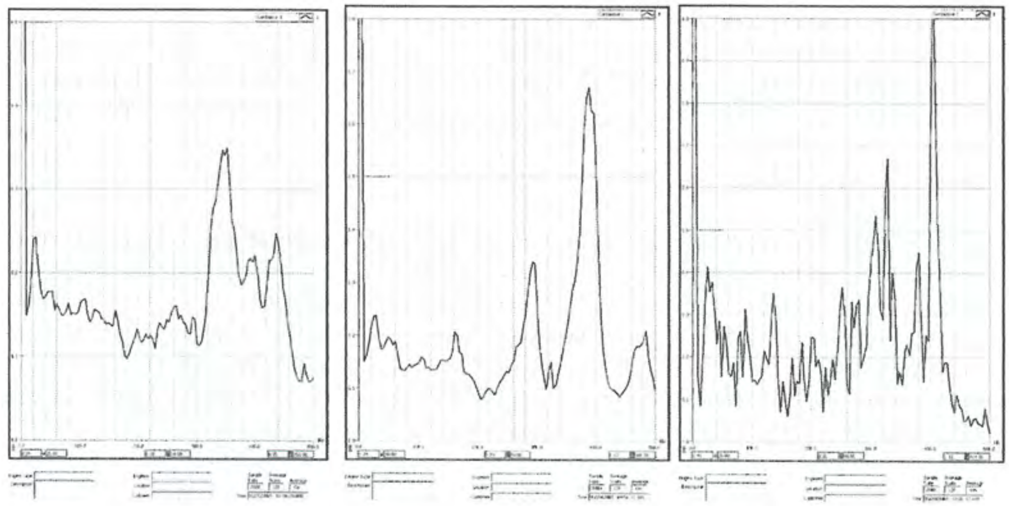

Figure 5A – Nox Steam
Max Pres. 0.35 psi
@ 352 Hz

Figure 5B – Nox & Pwr Aug
Max Pres. 0.67 psi
@ 388 Hz

Figure 5C – Nox PA, WC
Max Pres. 1.12
@ 404 Hz

SYSTEM PERFORMANCE:

In order to minimize the uncertainties associated with the performance test of the wet compression system, the test is run first with the wet compression system off, and then repeating the test with the system turned on, after allowing the gas turbine time to thermally stabilize for about 30 minutes. In doing this, ambient temperature and pressure changes can be minimized and station instrumentation can be used since instrument biases between test points should be negligible. However, the test results do not reflect performance improvements associated with the inlet duct modifications are not reflected in the performance numbers provided in this report because testing was performed with the system off and then on within an hour to minimize ambient variations and other outage factors that would have increased the uncertainties of the test. Had these been included there would have been an additional increase in power output of several hundred kilowatts and a small reduction in heat rate.

The performance test was conducted on June 26, 2003. The discharge water pressure was 3,000 psig when all five zones were on. A set of readings was taken. The spray system was turned off. Another set of readings was taken after few minutes. This was done to let the inlet duct "dry off". The difference between the two readings would give us the power gain due to the WC system. During this time, the firing temperatures were kept the same and the steam flow valves were left completely open to keep the steam injection rates the same. Note that steam is injected at two different locations - GT CDP steam flow and NO_x steam flow. The GT NO_x steam valve was left fully open and steam pressure was kept the same to allow constant steam flow during the test. This installation had media-type evaporative coolers, which were operational during the entire test. The results of the test are summarized in the Table 1.

Table 1: Wet compression test data summary.

	No Wet Compression	Wet Compression
Number of Stages On	0	5
Ambient Dry Bulb (ºF)	100.6	101.4
Ambient Wet Bulb (ºF)	66.1	65.9
GT Output (MW)	23.46	25.02
Plant Output (MW)	33.20	35.04
Change in Plant Net Heat Rate (Btu/kWhr)	---	+3.5%
GT Net Heat Rate (Btu/kWhr)	---	+2.9%
GT Speed	9,511	10,027
GT Governor Control	Temperature	Temperature
Compressor Discharge Pressure (psia)	270	282
GT Exhaust Temperature (ºF)	1,032	1,019
GT NO_x Emissions (ppm)	4.95	4.40
HP Steam Flow (kpph)	158.4	162.9
HP Steam Temp (ºF)	855	850
HP Steam Pressure (Psia)	834	834
IP Steam Flow (kpph)	9.5	10.4
IP Steam Temp (ºF)	377	372
IP Steam Pressure (psig)	123	127

Note – Accuracy of the plant heat rate measurements at the time plant testing was performed was considered by PurEnergy to be questionable. However, the trend was an increase for the gas turbine versus a decrease as had been exhibited with wet compression on industrial gas turbines.

Ammonia Consumption for NOx Control was also recorded to be decreased by 14 to 20 percent during wet compression system operation, indicated a similar reduction NOx on a parts per million basis was achieved.

Highlights of the data provided in table 1 are:

1. The speed of the gas generator increased 5.4%. In a number of cases the shaft speed reached its control limit of 10,050 rpm and automatically reduced firing temperature.
2. Compressor discharge pressure increased 4.4 %, which is approximately twice the gain from operating wet compression in a single shaft industrial gas turbine. This is also believed to be part of the reason why compressor temperatures did not decrease as much as with the industrial frames.
3. The gain in GT output was 1.6 MW and the gain in net plant output was 1.8 MW. This is about
4. There is degradation of 3.6% and 2.9% in the heat rate of net plant and gas turbine respectively.. This too is different from results obtained from applications of the system

to industrial frame machines in which there was an improvement in heat rate for the GT and slight degradation for the net plant output.

A curve illustrating the relative performance of the LM-2500 to industrial frame turbines with wet compression is provided in Figure 6. Independent studies performed by Dr. Sexton et al. and by Fern Engineering have predicted performance gains that are more in line with the industrial turbines. The cause for noticeable difference in performance continues to be evaluated to see if there are system or turbine modifications that can result in higher levels of power output and efficiency.

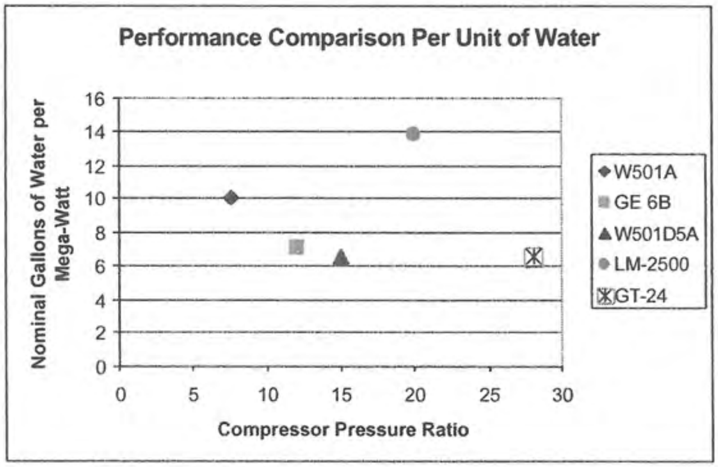

Figure 6 – Relative Power Gain Per Gallon of Water

COST/BENEFIT ANALYSIS:

Although wet compression technology is complimentary with other inlet air cooling technologies, a comparison in often made on the basis of $/kW. Proper care should be taken while making this comparison because the gains realized with wet compression are over a wider temperature range than with other cooling technologies like evaporative cooling, chilling, etc. One should not just look at the $/kW at the design point because most inlet air cooling technologies have wider swings in instantaneous $/kW.

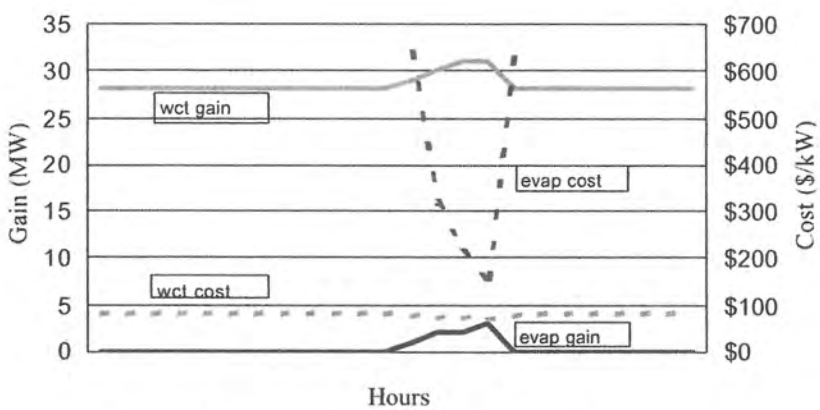

Figure 9: Performance and Cost on a hot humid day.

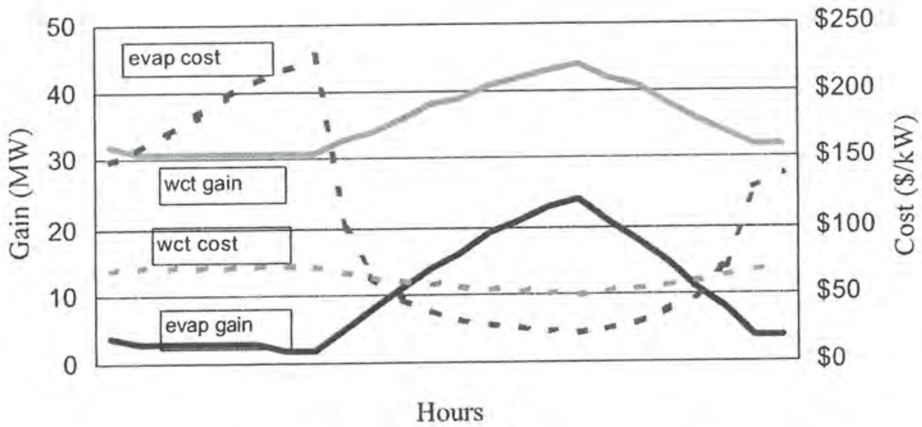

Figure 8: Performance and Cost comparison for a hot day.

For example, on a hot day design point, the cost of wet compression may be $50/kW versus $30/kW for evaporative cooling. If averaged over the whole day, the evaporative cooling cost would be $98/kW versus $63 for wet compression (see Figure 8). It all depends upon the application whether the gains are desirable over few hours or over a wider temperature range throughout the day. For plants that are operational more than 6 hours a day, the MWhr gains that can be realized with wet compression are much higher than with evaporative cooling or even chilling (see Figure 9). On a humid day, evaporative cooling is not of much use because of low ambient wet bulb depression. Even for chilling, the gains are limited by a combination of low ambient dry bulb temperature and higher auxiliary load due to high humidity. In other words, the gains due to wet compression are not as much ambient temperature dependent as other inlet air cooling technologies.

LM-2500 users are advised to consider the heat rate impact of wet compression when evaluating the potential revenues from power the incremental power sales.

CONCLUSIONS:

The application of wet compression was successfully demonstrated on an LM-2500PE. Although the level of performance improvement was not as great as initially expected based on experience with industrial frame gas turbines, it was still demonstrated to be an effective power augmentation technology, and to have added value of reducing NOx emissions. The benefits of wet compression over a much wider range of ambient air temperatures makes it one of the most cost effective means of enhancing the output of a combustion turbine and can be used in conjunction with other cooling technologies. The performance test showed that when wet compression is applied to LM 2500, a gain of 1.6 MW in the net combustion turbine output and 1.8 MW for the net plant output was realized. This gain can be realized whenever the ambient wet bulb temperature is higher than 50°F. No other cooling technology is as effective when operating at such low ambient temperatures, which enables wet compression to be used with evaporative cooling or chilling systems.

Acknowledgements:

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