GAS TURBINE POWER AUGMENTATION BY FOGGING OF INLET AIR

by

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ABSTRACT

Gas turbine output is a strong function of the ambient air temperature with power output dropping by 0.3 percent to 0.5 percent for every 1°F rise in ambient temperature. While this characteristic is inherent in any gas turbine, the effect can be more severe on certain aeroderivative engines. This loss in output presents a significant problem to utilities, cogenerators, and independent power producers (IPPs) when electric demands are high during the hot months. In the petrochemical and process industry, the reduction in output of mechanical drive gas turbines curtails plant output. One way to counter this drop in output is to cool the inlet air. This paper briefly reviews cooling technologies and focuses on *direct water fogging of the gas turbine inlet air.* The paper provides a comprehensive overview of the state-of-the-art of inlet fogging systems and how they have been applied to gas turbines. The paper assists readers to make an assessment of the benefits that are derived from applying this technology to their gas turbines.

INTRODUCTION

Gas turbines have gained widespread acceptance in the power generation, mechanical drive, and gas transmission markets. Their compactness, high power to weight ratio, and ease of installation have made them a popular prime mover. Improvements in hot section materials, cooling technologies, and aerodynamics have allowed increases in firing temperatures. Consequently, thermal efficiencies are currently very attractive, with simple cycle efficiencies ranging between 32 percent and 42 percent and combined cycle efficiencies reaching the 60 percent mark.

The fact that both turbine output and efficiency are reduced during periods of high ambient temperature poses a significant problem for turbine operators. For mechanical drive applications in the petrochemical industry, process output is considerably reduced when output power from the turbine falls on hot days. This often occurs when market demand is high and the power margins originally designed into the driver have been exhausted. In such cases, augmenting power can have a notable impact on profitability.

In the rapidly deregulating power generation segment, the structure of supply agreements and the dynamics of an open market usually mean that power producers are paid significantly more for power generated during high demand periods (typically hot summer afternoons). This creates an additional incentive to attempt to overcome the inherent loss of gas turbine power output during periods of high ambient temperature. Peaking power plants also need to augment power during high demand periods. An increase in peaking capability is typically needed during the late afternoon when temperatures are at the highest levels and turbine output at the lowest. As the power generation industry becomes more competitive, there is a growing interest in increasing power capability without the large capital investment associated with the addition of new capacity.

Power augmentation methods available for existing gas turbines include:

- *Steam or water injection into the combustor*—While commonly applied for NO_x control, it also boosts power due to the increased mass flow and higher specific heat of the products of combustion going through the turbine. The increased specific heat of the products of combustion and better heat transfer results in higher blade metal temperatures, and control systems often compensate for this by backing off on the firing temperature.
- *Inlet cooling of the compressor*—Different types of inlet cooling systems have been applied including the use of evaporative coolers, direct high-pressure inlet fogging, inlet refrigeration (mechanical compression and absorption), and the use of thermal energy storage.
- *Increasing the firing temperature (this calls for the implementation of an upgrade package from the original equipment manufacturer (OEM))*—In this case, hot section durability must be carefully considered.

Additionally, in order to counter the drop in power output during high ambient temperature conditions, different types of inlet cooling approaches have also been applied. These include the use of evaporative coolers, direct high-pressure inlet fogging, inlet refrigeration (mechanical compression and absorption), and the use of thermal energy storage.

The focus of this paper is on direct inlet fogging as a means to:

- Provide 100 percent evaporative cooling capability (i.e., attain the wet bulb temperature at compressor inlet).
- Provide fog intercooling, a technique that consists of over-injecting fog into the inlet airstream (i.e., spraying more fog than will evaporate under the given current ambient temperature and humidity conditions). The desired quantum of unevaporated fog is carried with the airstream into the compressor where it evaporates and gives an intercooling effect. The resulting reduction in the work of compression can result in a significant additional power boost.

Water injection into gas turbine compressor inlets is an old concept. Early studies were done in the forties by Kleinschmidt (1946), and Wilcox and Trout (1951). Water injection was used on the older jet engines (with zero or low bypass ratios) to boost takeoff thrust when aircraft were operating on hot days or from high altitude airports. The power gain came mainly from the cooling of the air (i.e., lower inlet temperature) and from the intercooling effect in the compressor, as opposed to the increase in mass flow rate caused by the injected water itself. Recently, with the advancement in high-pressure water fog technology, this concept has gained popularity in the industrial market and is being applied in the power, cogeneration, and IPP industries.

GAS TURBINE CYCLE THERMODYNAMICS AND DEPENDENCY ON INLET TEMPERATURE

In order to understand the concepts of power augmentation, it is helpful to review the thermodynamics of gas turbine cycles to understand the underlying dependence of the gas turbine cycle output and efficiency on ambient temperature.

Gas Turbine Cycle Thermodynamics

A representation of a single shaft gas turbine utilized for power generation applications is presented in Figure 1, which will be used for the discussion here. The behavior of split shaft aeroderivative machines is different and will be addressed ahead. In the pressure-volume (P-V) diagram shown in Figure 2, process 1-2 represents compression, 2-3 represents heat addition in the combustor, and 3-4 represents the expansion in the turbine. The shaded area represents the work of the cycle. The corresponding temperature entropy (T-S) diagram, shown in Figure 3, shows ideal or insentropic compression from 1-2 with the actual compression process being 1-2'. Line 2'-3 represents heat addition in the combustor. The expansion in the turbine is represented by process 3-4 (isentropic) with the actual expansion being 3-4'. The compression work per lb of air is given by C_p [T_2'-T_1] and the work of expansion is given by C_p [T_3-T_4']. The very reason that a gas turbine can produce output power is that the work of expansion must be greater than the work of compression, i.e., the line described by 3-4 must be greater than 1-2. This occurs because the constant pressure lines in Figure 3 diverge as shown. This is an important feature of the T-S diagram.

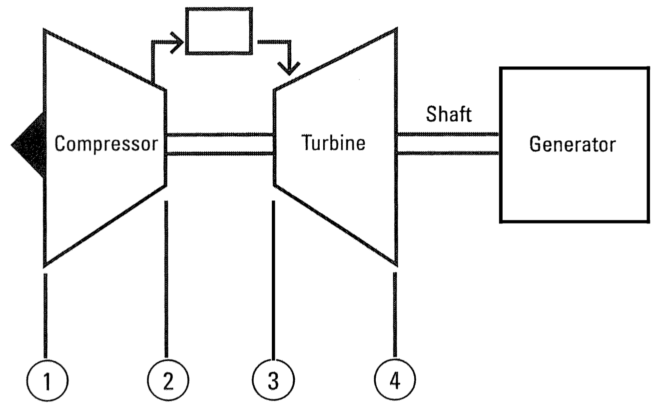

Figure 1. Representation of a Single Shaft Gas Turbine.

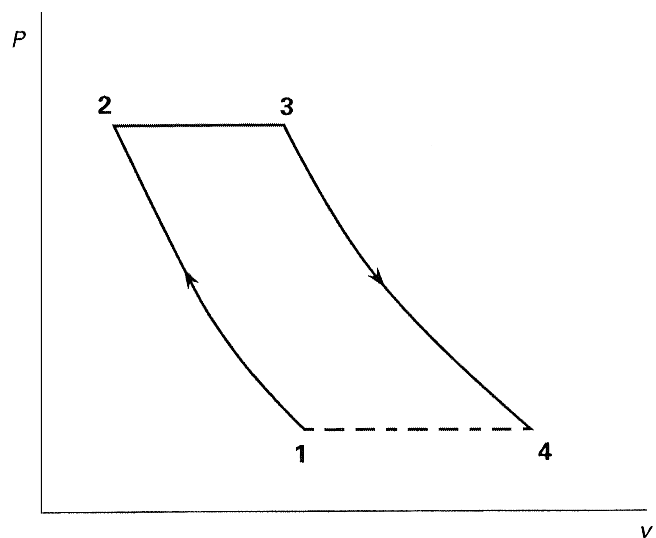

Figure 2. Pressure Volume Diagram for a Brayton Cycle. (Moving the compressor line to the left will reduce the work of compression (area under the curve).)

On a hot day, an examination of the simple T-S diagram (Figure 4) shows that the compression line 1h -2h is longer (i.e., more work) than the cold day case. The starting temperature at point 1h is hotter than a cooler day. Hence the compressor work (on a per lb basis), which is C_p [T_2-T_1], i.e., h_2-h_1, is more because of the shape of the diverging constant pressure lines. We will examine ahead how the drop in mass flow rate caused by the increase in ambient temperature further diminishes the output.

The compression process consumes as much as 66 percent of the total work produced by the gas turbine, and therefore any means of reducing the work of compression will enhance the power output of the gas turbine. Figure 5 shows an actual T-S diagram for a 10:1 pressure ratio industrial gas turbine indicating the temperatures involved.

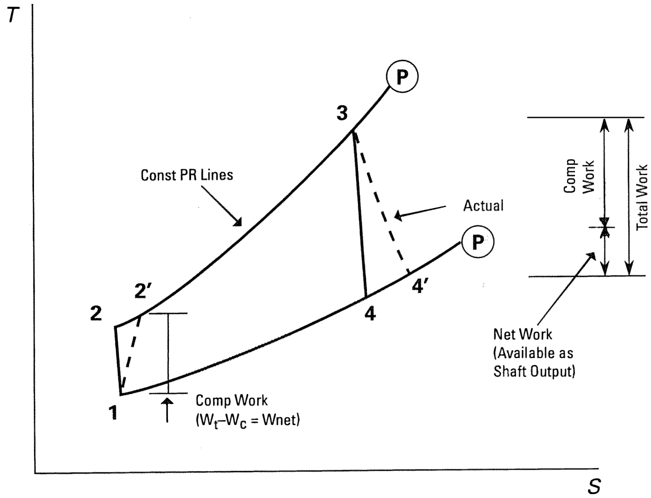

Figure 3. Temperature Entropy Diagram (T-S).

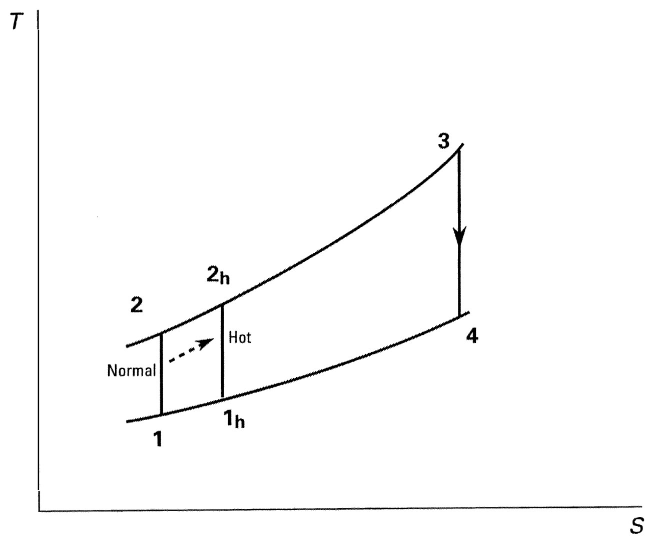

Figure 4. T-S Diagram Showing Cold and Hot Day Compression.

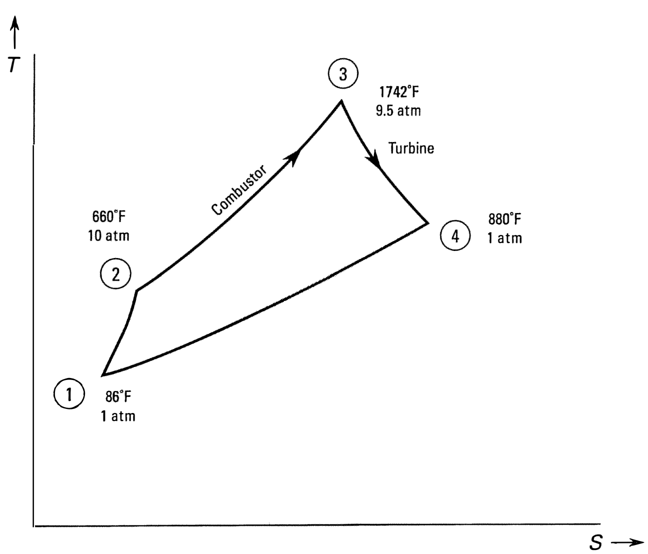

Figure 5. Actual T-S Diagram of a Gas Turbine with Temperatures and Pressures.

While these diagrams are predicated on some simplifying assumptions, actual gas turbine cycle performance is based on the following thermodynamic variables:

- Pressure ratio
- Compressor inlet temperature
- Turbine inlet temperature
- Compressor and turbine efficiencies
- Other factors such as pressure drops and combustor efficiency

The compressor work per lb of air is given by:

$$\left(\frac{W}{J}\right)_{Compr} = \frac{h_2' - h_1}{\eta_c} = \frac{C_p\left[T_2' - T_1\right]}{\eta_c} = \frac{C_p T_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{\eta_c} \quad (1)$$

In examining this equation, it can be seen that increasing T_1 would increase the compression work.

Assuming no pressure losses, and equal specific heats in the cycle, gas turbine thermal efficiency can be reduced to the equation below (Sorenson, 1950):

$$\eta_{th} = \frac{\frac{\eta_t T_3}{B} - \frac{T_1}{\eta_c}}{\frac{T_3 - T_1}{B - 1} - \frac{T_1}{\eta_c}} \quad (2)$$

where:

$B = (P_2/P_1)^{\frac{\gamma-1}{\gamma}}$
η_c = Compressor efficiency
η_t = Turbine efficiency
T_3 = Turbine inlet temperature
P_2, P_1 = Cycle pressure limits

Examination of this equation shows that the cycle efficiency decreases with an increase in compressor inlet temperature. When the inlet temperature goes up, the compressor discharge pressure and temperature drops and more fuel is required to attain the same turbine inlet temperature (TIT). An increase in ambient temperature, T_1, causes the numerator, representing net work output, to decrease at a faster rate than the denominator, representing the heat supplied. Hence the cycle efficiency decreases with increase in compressor inlet temperature.

Figure 6 shows a T-S diagram on a hot day. The drop in pressure ratio that occurs on the hot day can be seen on this diagram. The cycle peak temperature is still limited as before and so the expansion ratio also drops, meaning that less work is extracted from the turbine. As the compressor work increases and the turbine work decreases, the output work drops.

Further, as the ambient temperature increases, the pressure ratio of the compressor decreases and more fuel has to be added to attain the same TIT. This further decreases the output of the machine and causes the cycle efficiency to drop. Actual data obtained from a GE Frame 7 F machine showing a scatter plot of variation in compressor discharge pressure for different ambient temperatures is presented in Figure 7 (Meher-Homji, et al., 1993).

Figure 8 shows a typical compressor map with corrected speed lines. As the ambient temperature goes up, the compressor mechanical speed stays the same (assuming a generator drive), but the corrected speed defined as $N/\sqrt{\theta}$ changes as indicated on the map. Constant T_3/T_1 lines are shown on the map and these drop from a cold to a hot day because of the increase in T_1.

Power reduction during hot ambients can be as high as 20 percent to 40 percent depending on the engine match point design. Excessively high ambient temperatures also impose severe loads on the turbine cooling systems and this could have an impact on hot section life.

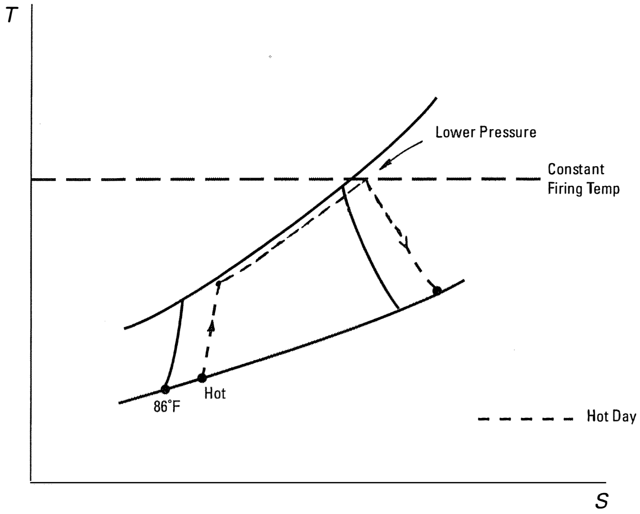

Figure 6. T-S Diagram on a Hot Day. (Note that the starting point is to the right and, because of the diverging pressure lines, work/lb is therefore more.)

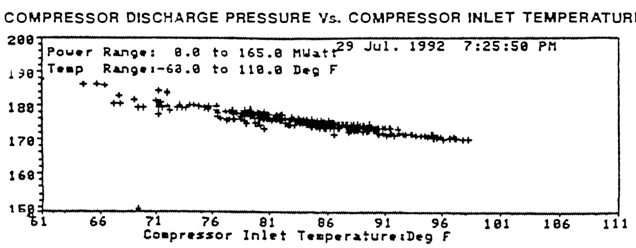

Figure 7. Pressure—Ambient Temperature Dependency of a Frame 7 F (2300°F Firing Temp) Gas Turbine. (Courtesy of ASME)

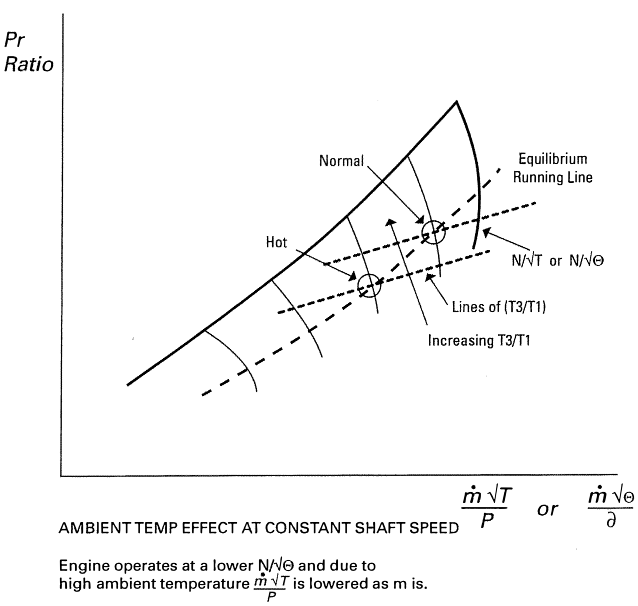

Figure 8. Compressor Map Showing Change in Operating Point at High Ambient Temperature.

Effect of Ambient Temperature on Air Density

The density of the air is given by:

$$\rho = \frac{P \times 144}{RT} \qquad (3)$$

where:

P = Pressure, psia
R = 53.3
T = Temperature, °R
ρ = Density, lb/ft^3

High ambient temperature causes a drop in density of the air as shown in Figure 9, and this causes a drop in the mass flow rate through the gas turbine.

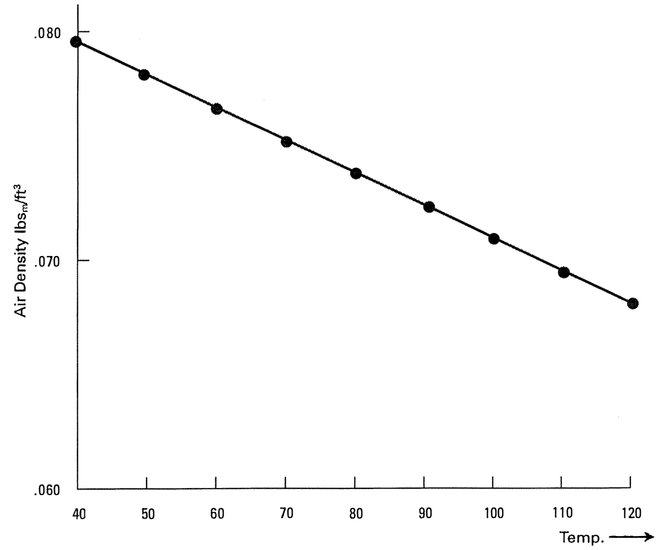

Figure 9. Effect of Ambient Temperature on Density.

The single shaft gas turbine operating at constant speed is essentially a constant volumetric flow machine, assuming that the speed and variable geometry are kept constant. These single shaft turbines are typically used in power generation applications and are held at synchronous speed by isochronous control or by the grid frequency. The amount of power generated is controlled by the fuel flow, which in turn affects the turbine inlet temperature. The mass flow rate is proportional to the absolute compressor inlet pressure and inversely proportional to the absolute inlet temperature. The mass flow is proportional to the absolute pressure (P_3) at the turbine inlet nozzle and is subject to a Mach number limitation here. This makes it inversely proportional to the square root of the turbine inlet temperature (T_3).

$$\dot{m} = K_1 \frac{P_1}{T_1} = K_2 \frac{P_3}{\sqrt{T_3}} \qquad (4)$$

Typical correction factors for a large heavy duty gas turbine are depicted in Figure 10, showing the effect of increased ambient temperature on output power and heat rate.

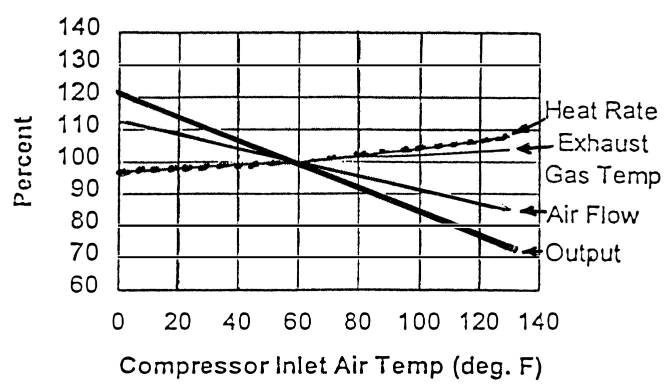

Figure 10. Effect of Ambient Temperature on Salient Gas Turbine Parameters.

One way to recover the lost power would be to increase the turbine inlet temperature, but there are obvious limitations to doing this due to hot section life considerations.

Aeroderivative machines in general exhibit a greater sensitivity to ambient temperature, but this distinction is blurring rapidly with several of the new heavy duty advanced gas turbines operating at high firing temperatures and pressure ratios.

Split Shaft Gas Turbines

Aeroderivative engines are widely used both for mechanical drive applications and also for power generation. Typically, these units will have at least two shafts, with the compressor being driven by the gas generator turbine and the load being driven by the power turbine (also known as the free turbine). Typical of this would be the Avon (pressure ratio of 12:1) or the LM2500 (pressure ratio of 18:1). Some aeroderivative engines, such as the RB211 or the LM5000, are three-shaft machines where the compression is split into two spools, each driven by their own turbine. This arrangement is preferred for higher pressure ratios. The LM6000 engine (pressure ratio of 30:1) is a two-shaft design, but employs the low-pressure compressor shaft as the drive shaft and is therefore unique since no power turbine is used.

In split shaft aeroderivative machines that are applied for power generation, the power is controlled by modulation of the fuel flow, which in turn controls the gas generator (N_1) speed, and this consequently affects the airflow rate. The gas generator speed is free to move to the level required to attain the power required, subject to limits on maximum speed or maximum turbine rotor inlet temperature (TRIT). In such machines, the volumetric flow rate does change due to the change in the mechanical speed of the gas generator. The power turbine speed (N_2) is fixed as it is coupled to the generator. In mechanical drive applications, both the power turbine and N_1 speeds can change in accordance to the matching laws.

Gas Turbine Intercooling

There are several ways to enhance the performance of a gas turbine cycle. Two approaches that have now become popular are the use of intercooling and reheat. Both of these increase the net work from the cycle. Intercooling decreases the compressor work, while reheat increases the turbine work.

In examining the P-V diagram of the compression process shown in Figure 11, it can be seen that the work for an isothermal process is less than that for an isentropic process between the same two pressures. The difference in work is shown as the shaded area in the figure.

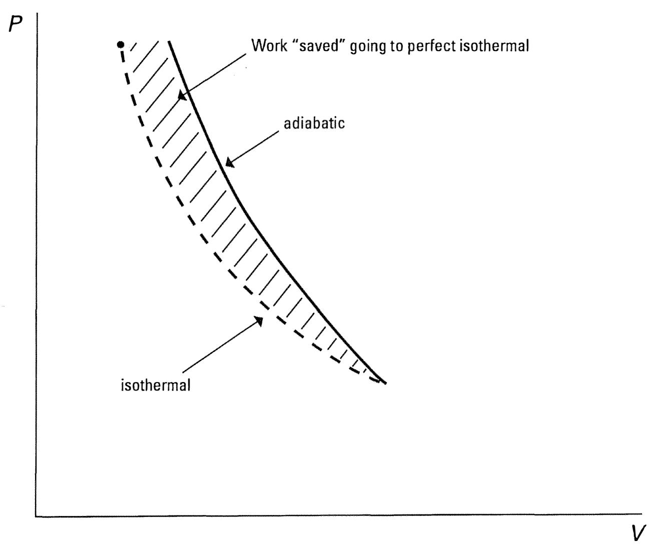

Figure 11. Reduction in Work by Movement Toward Isothermal Compression by Means of Intercooling.

Traditional intercooled cycles, as shown in Figure 12, utilize an intercooler (external heat exchanger) where air extracted from the first compressor is cooled and then returned for additional compression. In an ideal case, there is no pressure drop in the intercooler and the entry temperature to the second stage of compression is the same as the inlet temperature to the first. Common thermodynamic principals show that the optimal intermediate pressure ratios are those that derive equal stage work. A pressure-volume (P-V) diagram of a traditional intercooled cycle is shown in Figure 13, and the corresponding temperature-entropy (T-S) diagram in Figure 14. Intercooling has been covered by Tanaka and Ushiyama (1970).

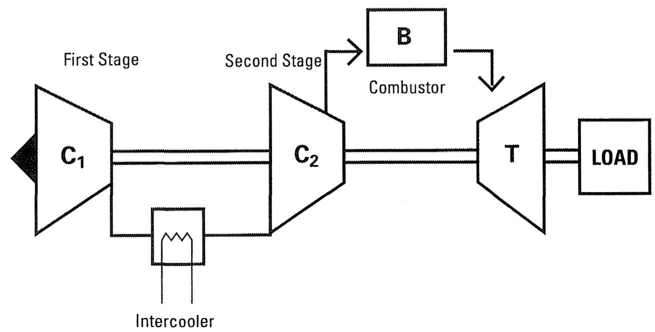

Figure 12. Traditional Intercooled Gas Turbine with External Intercooler.

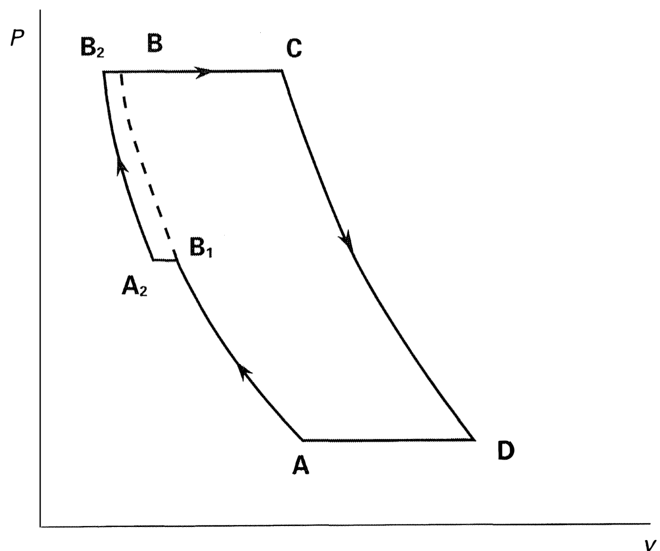

Figure 13. P-V Diagram of Traditional Intercooled Cycle B1-A2 Representing Intercooling.

The cost of the intercooler and the complexity involved are prohibitive for a retrofit situation, and so this solution is only offered on certain new gas turbine designs of a major European manufacturer. An interesting and successful case where intercooling has been applied to an aeroderivative machine is the LM6000 Sprint cycle, where direct intercooling is done between the low-pressure and high-pressure compressor spools. As this is the only commercially available engine with direct intercooling, it is described in detail ahead.

For practical retrofit applications on existing heavy duty or aeroderivative machines, the most effective and increasingly popular way to derive an intercooling effect is by fog intercooling, wherein a predetermined amount of deionized water fog is injected into the compressor section.

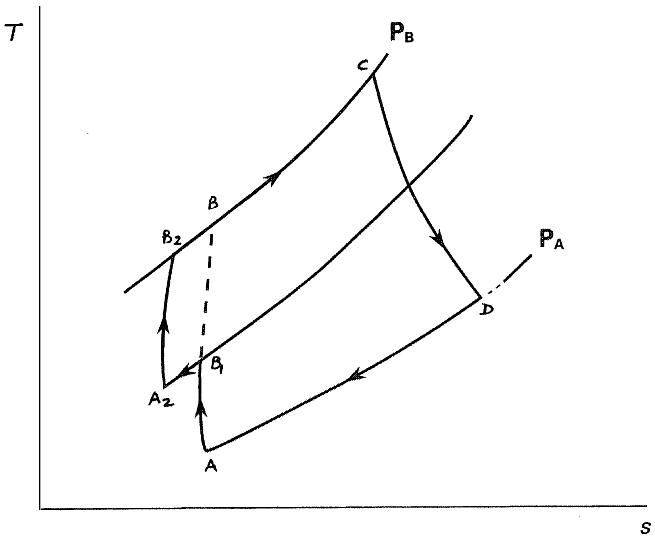

Figure 14. T-S Diagram of Intercooled Cycle.

Rules of Thumb Relating to Performance Effects of Ambient Temperature, Inlet Pressure, and Inlet/Exit Differential Pressure

Very often when working with gas turbines and evaluating power enhancement approaches, it is useful to get a quick feel for the performance sensitivity to site parameters. These parameters must be considered when evaluating any inlet cooling option. The turbine OEM can provide runs from its cycle deck to evaluate the different options. There are several excellent modeling codes commercially available that have a high degree of accuracy. However, for a quick analysis, some basic rules of thumb are provided here to give the user a feel for the numbers:

- *Ambient temperature effect*—As a rough rule of thumb, consider a 0.3 percent to 0.5 percent drop in output per °F increase in ambient temperature. It is the mitigation of this effect that is the main focus of this paper. As the temperature is lowered (i.e., air becomes more dense), the machine will operate at a higher mass flow rate and higher pressure ratio, and this results in increased power output and improved heat rate. The opposite occurs when the ambient temperature increases. Some simulated results for a variety of gas turbines are provided ahead.

- *Altitude effect*—The power loss for every 1000 ft of altitude is between 3 percent and 4 percent. A change in altitude causes a change in density and therefore has the effect of reducing output. However, as the inlet and exhaust are at the same altitude, the pressure ratio and expansion ratio are not affected.

- *Inlet differential pressure drop*—A 1 inch water gauge (wg) increase in inlet duct losses will result in a 0.48 percent drop in power and a 0.12 percent increase in heat rate. These numbers would be somewhat higher for an aeroderivative machine. Increases in inlet duct differential pressure will cause a reduction of compressor mass flow and engine operating pressure. Increase in inlet differential pressure results in a reduction of the turbine expansion ratio. This factor is important when considering the application of any inlet cooling technology such as evaporative systems, refrigeration, etc., and the effect of the increase in inlet differential pressure must be factored into the evaluation.

- *Exhaust differential pressure drop*—A 1 inch wg increase in exhaust losses will result in a 0.15 percent drop in power and a 0.12 percent increase in heat rate. It can clearly be seen that engines are more sensitive to the inlet pressure drop than the exhaust drop. An increase in the exhaust differential pressure would cause a change in the turbine expansion ratio for a single shaft constant speed machine, which would reduce the amount of work extracted from the turbine and would consequently result in a higher exhaust gas temperature.

- *Effect on heat recovery steam generator (HRSG) behavior for cogeneration and combined cycle applications*—As inlet cooling will often effect the exhaust conditions of a gas turbine, some rules of thumb are provided here. It is best to model the effect of changes in operating parameters from OEM performance curves. A 1 percent drop in exhaust flow will result in a reduction of steam flow by approximately 1 percent. A drop of 5°F in exhaust gas temperature would result in a steam flow reduction of 1 percent or a 2°F drop in steam temperature.

Gas Turbine Control Scheme and Typical Power-Temperature Relationships

In understanding the impact of inlet temperature and fog intercooling on a gas turbine, it is important to understand the different control methodologies used by manufacturers. The fundamental parameter that is controlled is the turbine inlet temperature (TIT or TRIT) defined as the total temperature at the inlet to the first stage rotating turbine. For base load operation, the control system will try to maintain this temperature at a constant setting, while ensuring that other constraints and limits are not exceeded. These constraints are typically:

- Compressor surge limits
- Maximum exhaust gas temperature (EGT) or interturbine temperature, also know as power turbine inlet temperature (PTIT)
- Torque or output limit
- Compressor discharge pressure or temperature limit
- Mechanical overspeed of compressor spools on a multishaft gas turbine

All gas turbines available today infer the turbine inlet temperature based on measurements of exhaust gas temperature or, in the case of aeroderivative engines, interturbine temperature. The control system is based on a control curve such as the one shown in Figure 15. This curve would be appropriately adjusted for items such as steam or water injection into the compressor. Thus the EGT is controlled (by fuel flow modulation) to stay on the control curve for different pressure ratios.

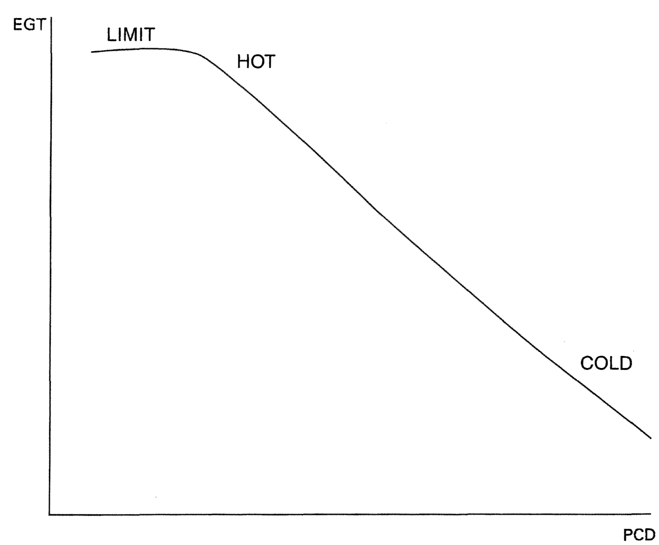

Figure 15. Control Curve of Gas Turbine Controlling Temperature.

Some temperature-power relationships for a variety of gas turbines are shown plotted in Figure 16. These were derived by the use of the Thermoflow GT PRO simulation program. Programs such as these allow a high level of modeling of the gas turbine and provide a good insight into performance parameters.

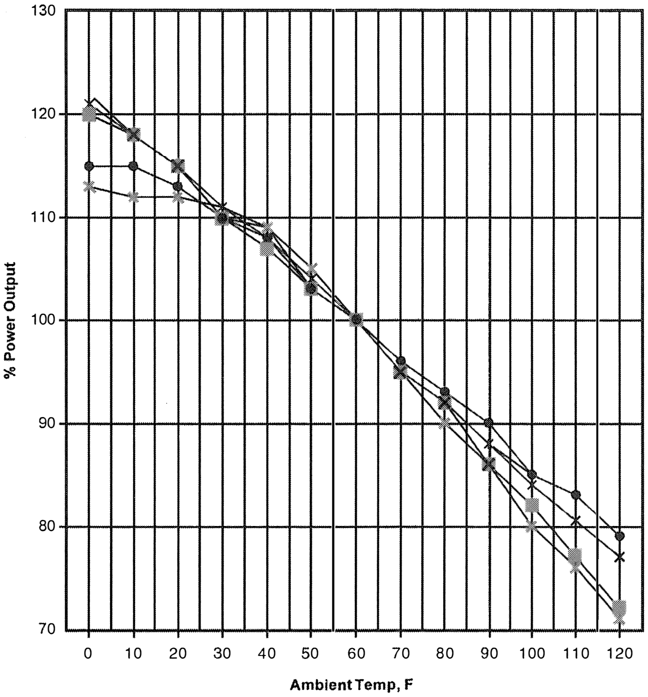

Figure 16. Typical Temperature-Power Relation for a Variety of Gas Turbines—GT PRO Simulation.

The Concept of Specific Power as a Measure of Evaluating Turbines Suited for Inlet Cooling

The power density concept is a useful construct to get a feel for engine sensitivity to ambient temperature and of the advantages that can be derived by inlet cooling. The power density, also known as the specific power (SP) of a gas turbine, is defined as:

$$SP = kW/m_{air} = kW/lb/sec \tag{5}$$

For example, a 165 MW gas turbine with a mass flow rate of 924 lb/sec would have a specific power of 179 kW/lb/sec. Since the specific power of an engine increases with higher firing temperature and pressure ratio, it is obvious that *inlet cooling is more effective with the high temperature and pressure ratio machines*. A scatter plot of a wide range of engines based on published data is shown in Figure 17. This plot shows how the new generation of machines (high output and high temperature) on the right of the plot has high specific powers and are therefore good candidates for inlet cooling. Whereas inlet cooling on machines with lower specific power also results in augmented power output, the gain is higher on the new generation of gas turbines. Kitchen and Ebeling (1995) have conducted a study qualifying different gas turbines for inlet cooling, while Van Der Linden and Searles (1996) have conducted an interesting analysis considering different sites in the U.S. and in evaluating different cooling approaches.

AN OVERVIEW OF GAS TURBINE INLET COOLING TECHNOLOGIES

There are several methods available for power augmentation by inlet air cooling (Meher-Homji and Mani, 1983). In general, they can be classified into three broad classes:

- *Refrigerated inlet cooling systems*—Utilizing absorption or mechanical refrigeration

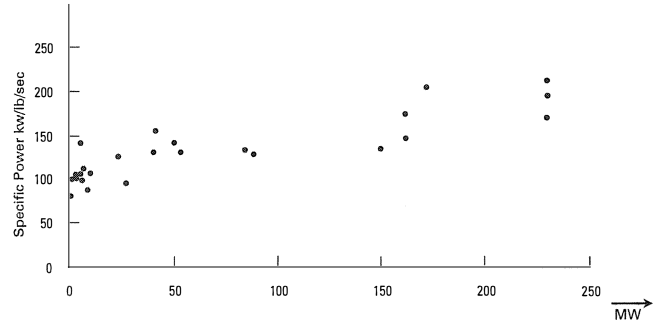

Figure 17. Scatter Plot Showing Specific Work Versus MW Output.

- *Evaporative methods*—Either conventional evaporative coolers or direct water fogging
- *Thermal energy storage systems*—These are intermittent use systems where the cold is produced off-peak and then used to chill the inlet air during the hot hours of the day.

A detailed discussion of cooling techniques is covered in Bacigalupo, et al. (1993), De Lucia, et al. (1993, 1995), De Piolene (1993), Giourof (1995), Kohlenberger (1995), and Weismantel (1998). Stewart (1999) has provided a design guide for combustion turbine air cooling systems.

Refrigerated Inlet Cooling Systems

Two techniques for refrigerating the inlet of a gas turbine are vapor compression (mechanical refrigeration) and absorption refrigeration.

Mechanical Refrigeration

In a mechanical refrigeration system, the refrigerant vapor is compressed by means of a centrifugal, screw, or reciprocating compressor. Centrifugal compressors are typically used for large systems in excess of 1000 tons and would be driven by an electric motor. Mechanical refrigeration has a significantly high auxiliary power consumption for the compressor driver and pumps required for the cooling water circuit. After compression, the vapor passes through a condenser where it gets condensed. The condensed vapor is then expanded in an expansion valve and provides a cooling effect. The evaporator chills cooling water that is circulated to the gas turbine inlet chilling coils in the airstream.

Chlorofluorocarbon (CFC) based chillers are now available and can provide a large tonnage for a relatively smaller plot space and can provide cooler temperature than the lithium-bromide (Li-Br) absorption based cooling systems. The drawbacks of mechanical chillers are high capital and operation and maintenance (O&M) cost, high power consumption, and poor part load performance.

Direct expansion is also possible wherein the refrigerant is used to chill the incoming air directly without the chilled water circuit. Ammonia, which is an excellent refrigerant, is used in this sort of application. Special alarm systems would have to be utilized to detect the loss of the refrigerant into the combustion air and to shut down and evacuate the refrigeration system. Mechanical refrigeraton for gas turbines is covered by Sadek (1981).

Absorption Cooling Systems

Absorption systems typically employ lithium-bromide (Li-Br) and water, with the Li-Br being the absorber and the water acting as the refrigerant. Such systems can cool the inlet air to 50°F. The heat for the regenerator can be provided by gas, steam, or gas turbine exhaust. Absorption systems can be designed to be either single or double effect. A single effect system will have a coefficient of performance (COP) of 0.7 to 0.9, and a double effect unit, a COP of 1.15 (Jolly, et al., 1998). Part load performance of absorption systems is relatively good, and efficiency does not drop off at part load like it does with mechanical refrigeration systems. Absorption systems have lower O&M costs than mechanical refrigeration systems. Details of absorption systems as applied to gas turbines may be found in Malewski and Holldorff (1986).

Traditional Evaporative Cooling

Traditional evaporative coolers that use media for evaporation of the water have been widely used in the gas turbine industry over the years, especially in hot arid areas. The basic principle of this type of cooling is that as water evaporates, it consumes 1160 Btu's of heat (latent heat of vaporization), and in doing so reduces the ambient air temperature.

Physically, media type evaporative cooling consists of an arrangement as shown in Figure 18 (Johnson, 1988). Water is distributed over the media blocks, which are made of fibrous corrugated material. The airflow through this block evaporates the water.

Evaporative cooler effectiveness is given by:

$$E = \frac{T_{1DB} - T_{2DB}}{T_{1DB} - T_{2WB}} \quad (6)$$

where:

T_1 = Inlet temperature
T2 = Exit temperature of evaporative cooler
DB = Dry bulb
WB = Wet bulb

A typical value for effectiveness is 80 percent to 85 percent, which means that the wet bulb temperature can never be attained.

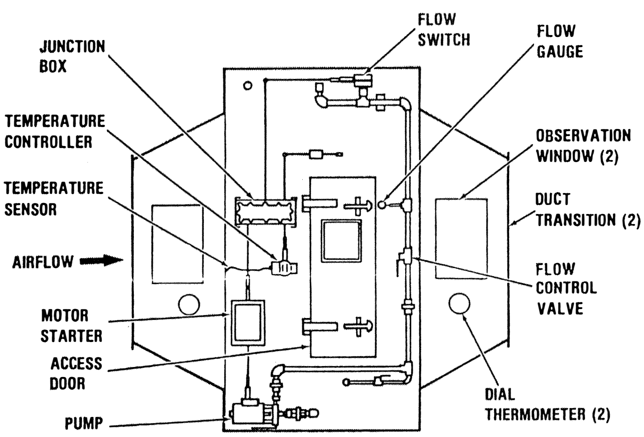

Figure 18. Traditional Media Based Evaporative Cooling Arrangement. (Courtesy of ASME)

The temperature drop is given by:

$$\Delta T_{DB} = 0.8\ (T_{1DB} - T_{1WB}) \quad (7)$$

A psychrometric chart can be used to obtain the values. Figure 19 (Tatge, 1980) shows typical power increases attainable at different relative humidities taking into account typical media evaporative cooler effectiveness. The exact power increase depends on the particular machine type, site altitude, and ambient conditions. It can be seen from this figure that the power increase is the greatest at low relative humidities. For a typical industrial gas turbine with an air mass flow rate of 250 lb/sec, the water evaporation rate can range from 200 to 800 gallons/hr, depending on the relative humidity. A detailed treatment of evaporative cooling as applied to gas turbines may be found in Johnson (1988).

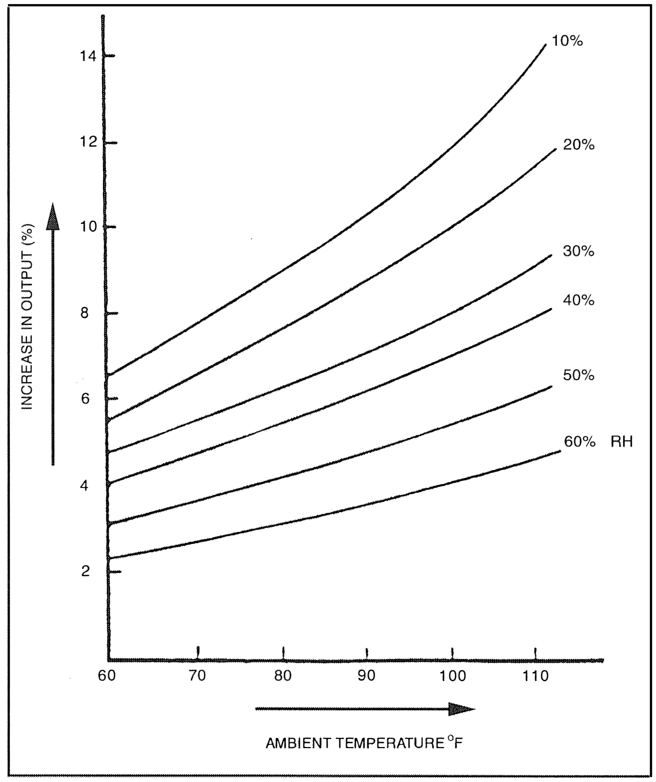

Figure 19. Evaporative Cooler Graph. (Courtesy of GE)

The inherent loss of efficiency and increased inlet pressure loss in a traditional evaporative cooling system never allows for the maximum cooling effect to be attained. Water quality requirements, however, may be less stringent than those required for direct fog cooling systems.

Direct Inlet Fogging

Direct inlet fogging is a method of cooling where demineralized water is converted into a fog by means of high-pressure nozzles operating at 1000 psi to 3000 psi. This fog then provides cooling when it evaporates in the air inlet duct of the gas turbine. This technique allows a 100 percent effectiveness in terms of attaining 100 percent relative humidity at the gas turbine inlet, and thereby gives the lowest temperature possible without refrigeration (the wet bulb temperature). Direct high-pressure inlet fogging can also be used to create a compressor intercooling effect by allowing excess fog into the compressor, thus boosting the power output considerably. This technique is the focus of this paper and is covered in detail ahead. A discussion of this can be found in Stambler (1997) and Mee (1988). Cooling by means of injection of liquid air is covered by Kishimoto, et al. (1977). Useful charts and calculations have been provided by Wilcox and Trout (1951). While this paper has been written for turbojets, it is a useful reference for land-based gas turbines and enables rapid estimations for wet compression.

Thermal Energy Storage Systems

This method is used when power augmentation is required only for a few hours in a day. In this approach, a cold reserve is built up during the nonpeak hours and this cold energy is utilized during the peak hours to chill the inlet air, thus increasing turbine output. As it operates in this intermittent mode, it is possible to reduce the size of the refrigeration system compared with a system that has to provide continuous cooling. The energy storage media can be ice, water, or other heat transfer liquids.

Hybrid Systems

Depending on the specifics of the project, location, climatic conditions, engine type, and economic factors, a hybrid system utilizing a combination of the above technologies may be the best. The possibility of using fogging systems in conjunction with mechanical inlet refrigeration should be considered. This may not always be intuitive, since evaporative cooling is an adiabatic process that occurs at constant enthalpy. When water is evaporated into an airstream, any reduction in sensible heat is accompanied by an increase in the latent heat of the airstream (the heat in the airstream being used to effect a phase change in the water from liquid to the vapor phase). If fog is applied in front of a chilling coil, the temperature will be decreased when the fog evaporates, but since the chiller coil will have to work harder to remove the evaporated water from the airstream, the result would yield no thermodynamic advantage.

However, if the chiller is underdesigned and is not capable of bringing the air temperature down to the ambient dew point temperature, the addition of fogging in front of the chiller will result in a colder finished temperature, with no additional work being required by the chiller as indicated in the psychrometric chart shown in Figure 20.

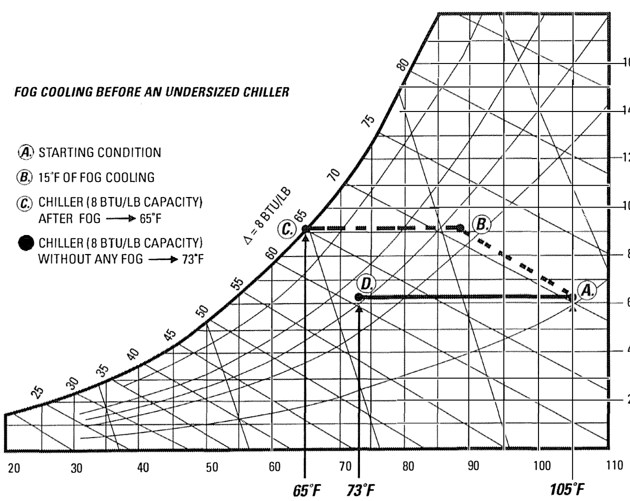

Figure 20. Psychometric Chart Showing Hybrid Application with Chiller.

Direct fogging systems have also been applied in conjunction with traditional media type evaporative cooling systems. Fogging systems have been used to augment evaporative coolers by adding a few degrees of additional evaporative cooling to make up for the fact that media type coolers cannot normally reach the wet bulb temperature. They have also been used to provide fog intercooling to turbines that already have evaporative coolers.

Each technology has associated economic costs and technical pros and cons, and a careful evaluation must be done to select the best technology or mix of technologies.

Selection Criteria for Inlet Cooling Systems

The choice between alternative cooling technologies is essentially an economic one and the total project cost must be evaluated over the life cycle. Dominating factors that should be taken into account in doing a study are:

- Climatic profile.
- Installed cost of the cooling system in terms of $/incremental power increase.
- Amount of power gained by means of inlet air cooling. This should take into account parasitic power used, the effect of increased inlet pressure drop from the cooling coils, or evaporative media.
- Fuel costs and costs of incremental power, i.e., what benefit is attained by the power boost.
- Projected O&M costs for the system.
- Environmental impact—this is especially important with ammonia-based refrigeration systems and CFC-based systems.
- For cogeneration applications, the time of use electric rates and the public power alliance (PPA) have to be carefully considered.
- Potential impact on existing emission licenses.

It is advisable that the site's temperature profile for a full year of hourly data with the 30 year average wet and dry bulb temperature be considered in the analysis. These data can be used to generate evaporative cooling degree hour" numbers for each day of the year, and allow a turbine operator to make a very detailed analysis of potential power gain from inlet fogging.

Economic analysis may be found in Utamura, et al. (1996), Ondryas (1990), Van Der Linden and Searles (1996), and Guinn (1993).

INLET FOGGING FOR GAS TURBINE APPLICATIONS

High-pressure fogging of gas turbine inlets has been applied for 10 years. In essence, it involves direct evaporative cooling wherein the fog, which consists of billions of particles generated at sizes of 5 to 20 microns, is injected into the airstream where it evaporates and provides cooling. Tests and application data have shown that this process can be 100 percent effective (i.e., wet bulb temperatures can be reached) even in high humidity regions. This is a major difference between fogging systems and traditional evaporative cooling methods. By meteorological definition, fog particle sizes are less than 40 microns, while mist particle sizes are 40 to 100 microns. True fog tends to remain airborne due to Brownian movement—the random collision of air molecules that slows the descent of droplets. In still air, a fog particle with a diameter of 10 microns would fall at a rate of about one meter in five minutes, while a 100 micron diameter particle would fall at the rate of about one meter in three seconds.

A typical fog system consists of a series of high-pressure pumps that are mounted on a skid, programmable logic controller (PLC) based control system with temperature and humidity sensors, and an array of fog nozzles installed in the inlet air duct. Fog can also be used to cool ancillary equipment such as generators, lube oil transformers, and coolers.

Generation of Fog

Fog is generated by the application of high-pressure demineralized water between 1000 psi and 3000 psi to an array of specially designed fog nozzles. A photograph of a typical fog nozzle is shown in Figure 21, and the fog generated is shown in Figure 22. The nozzle is made of 316 stainless steel (SS) and consists of a small orifice from five to seven thousandths of an inch for gas turbine applications. The water emanating from this orifice impacts a specially designed impaction pin that breaks up the jet into billions of microfine fog droplets. Other factors being equal, the rate of evaporation of the droplet essentially depends on the surface area of the water exposed to the air. With high-pressure fog, the surface area of the billions of droplets is very large, allowing rapid evaporation. Because of the geometry of a sphere, a given amount of water atomized into 10 micron diameter droplets yields 10 times more surface area than the same amount of water atomized into 100 micron droplets.

Figure 23 shows a typical distribution manifold made up of 1/2 inch SS tubes and a number of fog nozzles. These manifolds induce a very low-pressure drop estimated at less than 0.02 inch wg and are all stainless steel. Considerable care and design features must be incorporated to avoid flow-induced vibration. The manifold is configured to provide multiple stages of fog cooling with each stage typically being supplied by a dedicated high-pressure pump.

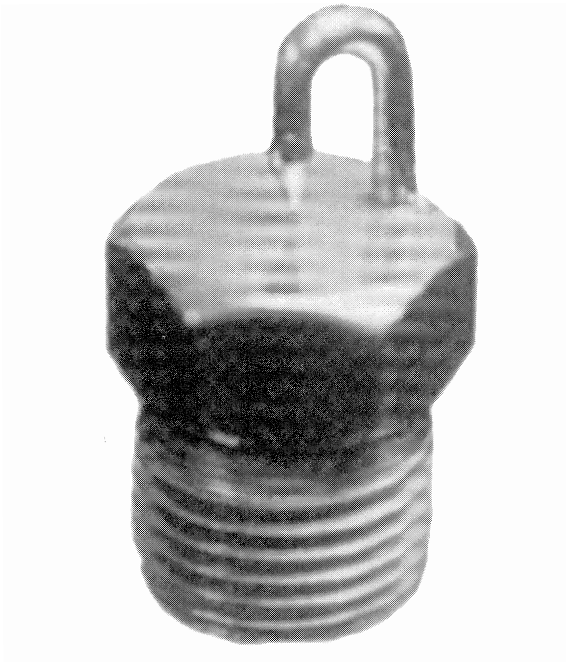

Figure 21. Photograph of a High-Pressure Fog Nozzle.

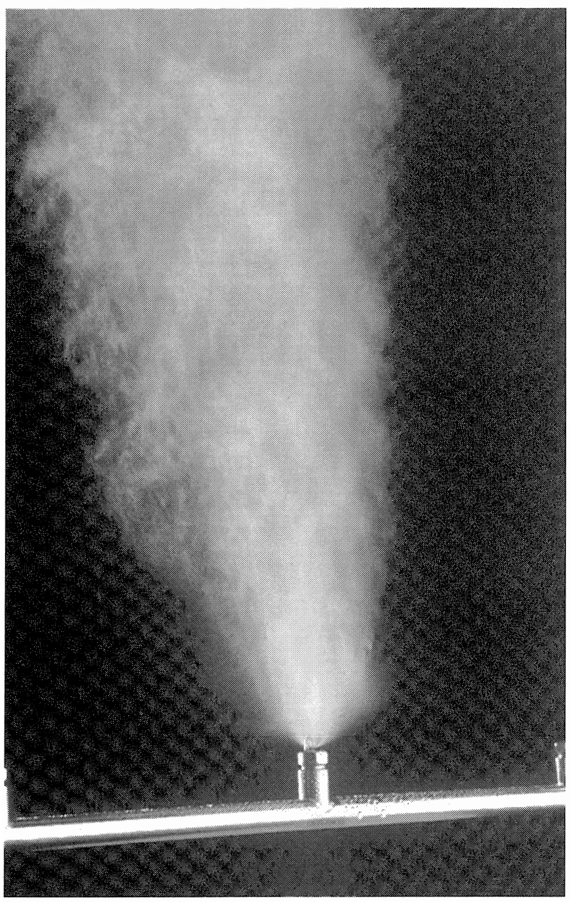

Figure 22. Fog Generated by High-Pressure Nozzle.

Droplet Size Distribution

Because of the importance of small droplet size to maximize evaporative efficiency and to avoid erosion problems within the compressor, considerable focus must be placed on designing and testing nozzles to evaluate fog droplet size. Tests have been conducted on fog nozzles and typical results are shown in Figure 24, which are data taken with an atomization pressure of 1000 psi and a flow rate of 0.032 gpm. The nozzle that was tested had an orifice diameter of 0.006 inch. In examining this figure, it can be seen that 85 percent of the droplets generated are below 10 microns in size and almost none are greater than 20 microns. As higher atomization pressures are used, droplet size decreases, as indicated in Figure 25. Within limits, droplet size is inversely proportional to the square root of the pressure ratio of the nozzle. For example, doubling the pressure would reduce the droplet size by about 30 percent.

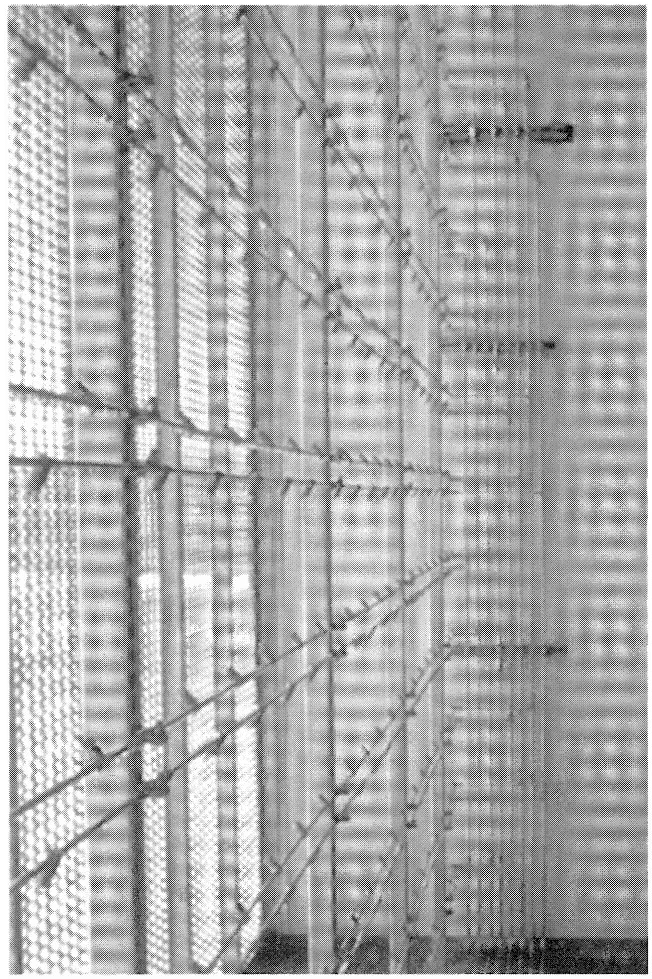

Figure 23. Typical Distribution Manifold.

Gas Turbine Axial Compressor Erosion Considerations

An area of primary concern in the earlier years for the application of fogging systems was the possibility of blading erosion. With the small droplet diameters being generated and use of demineralized water (DM), erosion related to inlet air fogging and fog intercooling has been shown not to be a problem. This has been documented in several references, including Sexton, et al. (1998). There is no detrimental effect on the blading from an erosion standpoint when high-pressure fogging is employed, and indeed some earlier studies have indicated that even higher droplet sizes do not have any detrimental erosion effect.

It is interesting to note that compressor water injection has actually been suggested to improve performance of an eroded axial flow compressor (Tabakoff, et al., 1990). This can be done because the water injection increases the total pressure ratio, thus recovering some erosion-related deterioration.

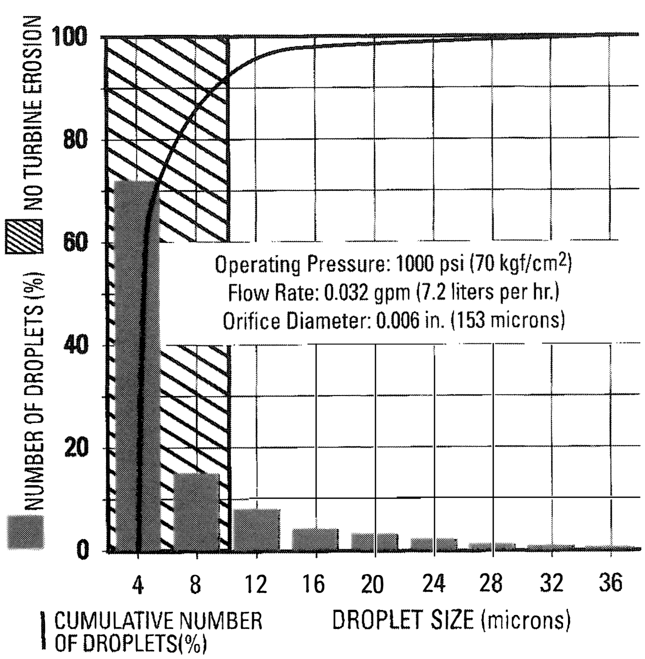

Figure 24. Droplet Size Distribution. (This is an important aspect for good evaporation efficiency and mitigation of erosion problems.) (Courtesy of Mee Industries)

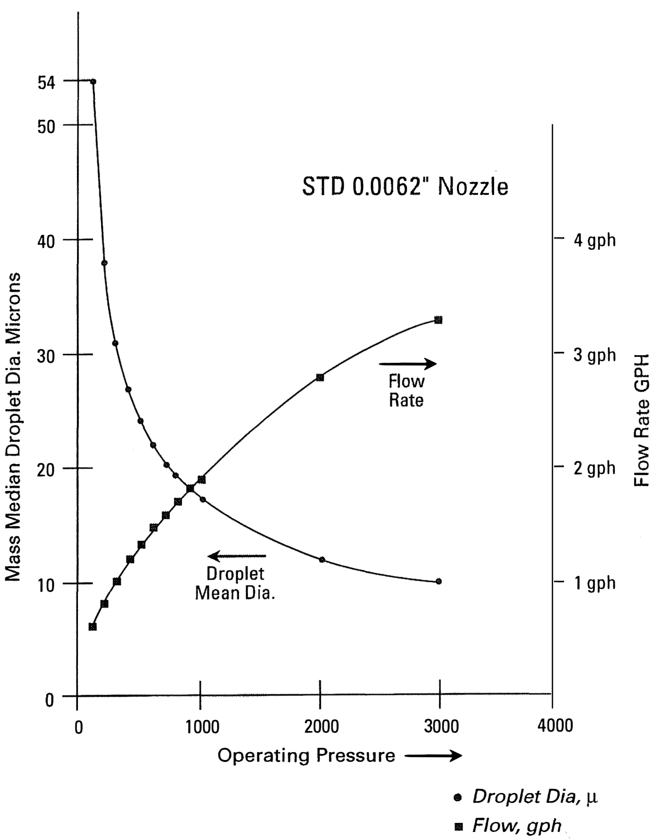

Figure 25. Graph Showing Droplet Size and Flow Rates as a Function of Pressure. (Data Courtesy of Mee Industries)

Compressor Fouling/Washing Considerations

The area of compressor fouling deterioration is indeed one of importance. When good quality DM water is used, no problems of deposits have been noted. In fact, operator experience seems to indicate that there is a washing effect from the fog itself. It is possible that using fog on a nearly continuous basis for power augmentation results in a continuous washing effect. When fog is in use, the likelihood of dirt in the airstream becoming attached to the compressor blades is greatly reduced. This effect may result in savings of online wash costs and washing chemicals. The use of fogging may reduce power losses due to compressor fouling between washes.

High-pressure fog systems have also been successfully applied as air scrubbing systems. In this application, the fogging nozzles are installed upstream of the air filters and a droplet eliminator is employed to remove the fog from the airstream before it reaches the main barrier filters. Dust in the ambient air is captured by the fog droplets, which, in turn, are captured by the droplet eliminator. On installations where this method has been employed, a considerable reduction in the frequency of replacement of barrier filters has been attained.

The area of compressor fouling and control is complex and site dependent. However, a general comment can be made that inlet fogging will not have a detrimental effect, but could indeed help. Details relating to axial compressor fouling may be found in Meher-Homji (1990).

Fog Plenum Efficiency—Factors Affecting Evaporation

Data available today on plenum efficiency would apply only to application of nozzles upstream of air filters (for scrubbing purposes). For application downstream of the filters, nozzle location is important (to avoid wetting the duct floor, etc.). In general when fog intercooling is desired, the nozzles should be installed before the silencers to allow as much time as possible for evaporation. In certain installations involving large heavy duty gas turbines, two nozzle manifolds may be installed, one before the silencers for evaporative cooling and one after the silencers for intercooling. Nozzle orientation is also an important factor. Pointing the fog nozzles into the airstream gives both more residence time and helps with better evaporation in a high velocity airstream.

Nozzles, Pumps, and Auxiliaries

The key to effective fogging is the design of the fogging nozzle. These nozzles are fabricated of 316 SS and have a specially designed impaction pin, as shown in Figure 26. This impaction pin is micromachined and carefully adjusted. The use of proper nozzle filters and means to avoid foreign object damage (FOD) to the gas turbine are very important factors and should be evaluated when considering any fogging system. Also it is imperative that each nozzle undergoes a flow test to ensure proper operation.

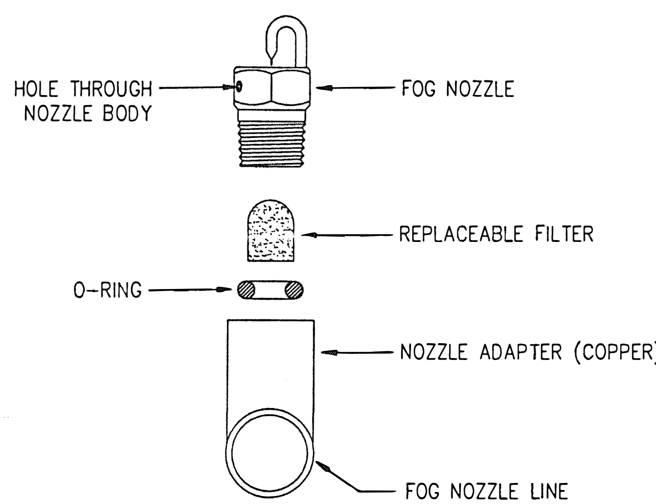

Figure 26. Drawing of Nozzle.

Typically the pumps used to derive the 2000 psi to 3000 psi pressures used for gas turbine inlet air fogging systems are positive displacement ceramic-plunger stainless steel pumps with stainless steel heads. All wetted parts are stainless steel or ceramic. Each high-pressure pump is connected to a fixed number of fog nozzles representing one discrete stage of fog cooling. The pumps can be turned on sequentially to control the amount of cooling. For example, a 20°F drop in temperature may be managed in four 5°F increments. If finer increments are required, more stages are incorporated.

The fog skid should be located as close to the final distribution manifold as possible. A typical skid plan view with four high-pressure pumps is shown in Figure 27, with a photo of a typical skid being shown in Figure 28.

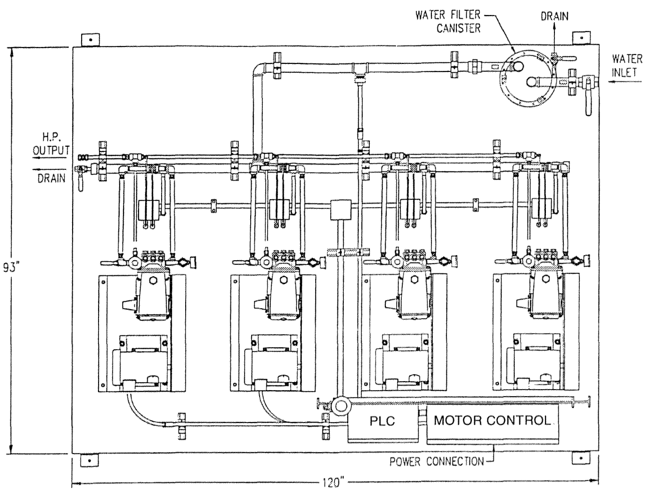

Figure 27. Plan View of a Fogging Skid Showing Components.

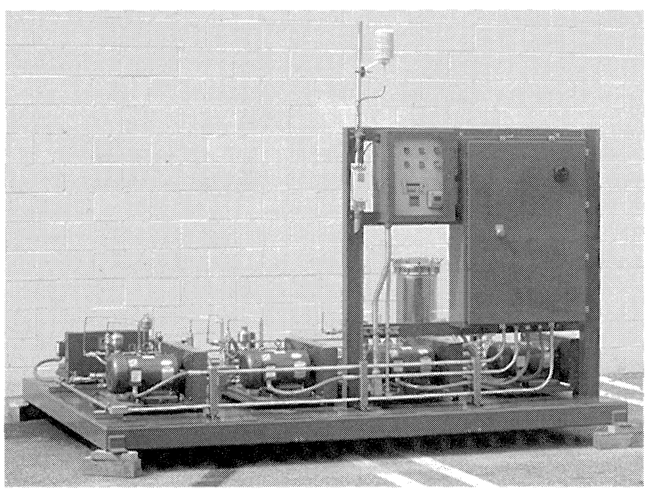

Figure 28. Photograph of High-Pressure Gas Turbine Inlet Fogging Skid.

Control Aspects

The control system incorporates a programmable logic controller (PLC), which is typically mounted on the high-pressure pump skid. Sensors are provided to measure relative humidity and dry bulb temperature. Special proprietary programming codes use these measured parameters to compute the ambient wet bulb temperature and the wet bulb depression (i.e., the difference between the dry bulb and wet bulb temperature) to quantify and control the amount of evaporative cooling that is possible with the ambient conditions. The system turns on or off fog cooling stages to match the ability of the ambient air conditions to absorb water vapor. Typically the system should have a user editable feature parameter that gives the operator the ability to inject a controlled amount of excess fog for compressor intercooling. The control system also monitors pump skid operating parameters such as water flow rates and operating pressures, and provides alarms when these parameters are outside acceptable ranges.

The staging method for the fogging operation is shown in Table 1. In this example, the second and third columns represent the dry bulb and wet bulb temperatures as the day progresses. Additional stages are activated as the day warms up and cooling requirements are increased. In this control scheme example, each stage is activated for 10°F cooling. Table 2 indicates the situation when an intercooling effect is desired. In this control scheme, a user defined parameter "max overcool" is utilized that limits the amount of excess water injection. In intercooling mode, the parameter "max overcool" is set to a positive number and the controller tries to always inject more fog than will evaporate.

Table 1. Typical Fog Cooling Stage Control (Four Stages of 10°F Each).

Time	Dry Bulb	Wet Bulb	Delta	Stage "ON"	Cooling
9:00	70°F	68°F	2°F	none	
10:00	75°F	69°F	6°F	1	5°F
11:00	80°F	70°F	10°F	2	10°F
12:00	86°F	70°F	16°F	3	15°F

Table 2. Typical Fog Intercooling Stage Control (Four Stages of 10°F Each). "Max Overcool" Is Set to +5°F.

Time	Dry Bulb	Wet Bulb	Delta	Stage "ON"	Overcooling
9:00	70°F	68°F	2°F	1	3°F
10:00	75°F	69°F	6°F	2	4°F
11:00	80°F	70°F	10°F	3	5°F
12:00	86°F	70°F	16°F	4	4°F

Location of Fog Nozzles in the Gas Turbine Inlet

To incorporate fog cooling, very little modification of the inlet system is normally needed. However, it may be desirable to install access doors to allow for service of fog nozzles.

There are two main options for installing the inlet fogging system, locating them either upstream or downstream of the filters, as depicted in Figure 29.

- *Upstream of the inlet filters*—One advantage to positioning the fog nozzle manifold upstream of the air filters is that the installation can be accomplished without outage time. In this case, a fog droplet filter must be added downstream of the fog nozzle manifold to remove any unevaporated fog. By definition, the droplet filter would not allow any fog intercooling. Typically about half the water output by the fog nozzles is captured by the droplet filter and drained away. This type of system, while used on some early installations, is rarely applied to gas turbine installations today. It requires more fog nozzles and more water, and is generally more expensive to operate and install. However, turbine operators who have experienced excessive loading of the inlet air filters might find this a cost-effective option, as "fog scrubbing" has been shown to dramatically increase air filter life. A system installed prior to the air filters is shown in Figure 30.

- *Downstream of the inlet filters*—The most common location for the high-pressure fog nozzle manifold is downstream of the air filters and upstream of silencers and trash screens. Installation in this location requires an outage of one to two days and calls for only minor modifications to the turbine inlet structure. This type of installation allows fog intercooling. While the fog nozzle manifolds can be also installed downstream of the silencers, it is generally considered best to locate them upstream of the silencers, as this would allow more residence time for the fog droplets to evaporate. Fog nozzle manifolds are almost always installed upstream of the trash screens to avoid any possibility of FOD.

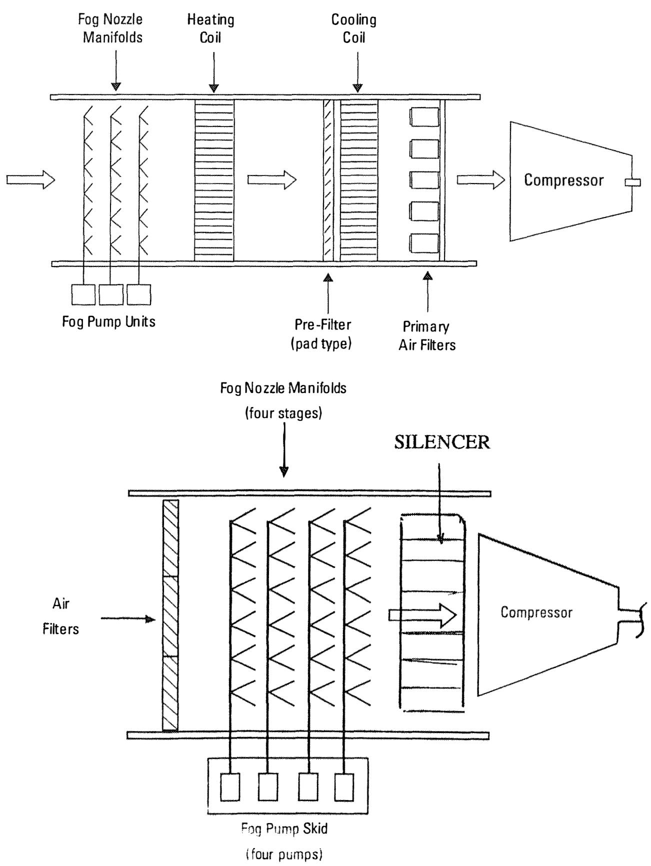

Figure 29. Options for Installing Gas Turbine Inlet Fogging Systems. (For fog intercooling, nozzles must be after the filters.)

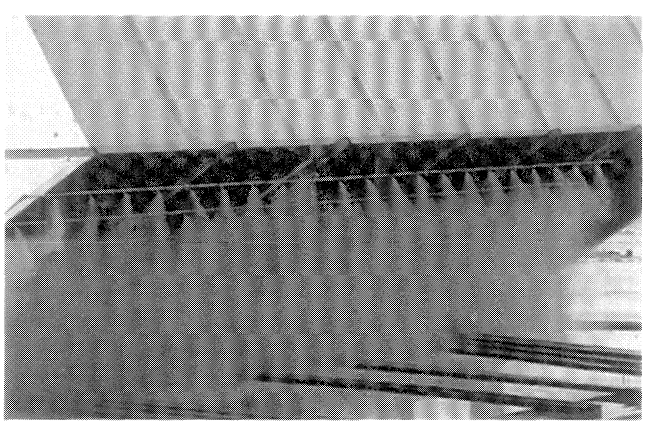

Figure 30. Photograph Showing Fog System in Operation, Upstream of Air Filters.

Consumables and Water Quality

The importance of water quality cannot be overstated, especially when fog intercooling is desired. Experience has shown that demineralized water often requires additional treatment if it is to be used for gas turbine inlet air cooling. Demineralization generally removes ionic material (minerals that are dissolved in the water in ion form) but does not remove colloidal material. Surface waters, such as river or lake-water, and even some well waters often contain silica in colloidal form (i.e., in particles that are small enough that they do not readily settle out of the water). As silica is a very hard element, its presence can damage fog nozzles and possibly even compressor blading. The fog pump skids or water treatment plant should include submicron water filters to remove the silica from the fog supply water.

Typical Maintenance Schedule and Requirements of Gas Turbine Fogging Systems

A typical fog system used for cooling inlet air for a gas turbine requires 15 to 20 hours per year of maintenance. Items requiring maintenance include:

- Inspection of the nozzle manifolds and cleaning or replacement of any damaged fog nozzles.
- Changing crankcase oil for the high-pressure pumps.
- Replacing inlet water filter cartridges.
- Performance of extended shutdown procedures for the pump skids and nozzle manifolds when they are not to be used.
- Seasonal startup procedures (refilling lines, installing new filters, etc.).
- Inspection of water treatment facilities and periodic testing of water quality.
- Calibration of instrumentation.

PHYSCHROMETRICS OF INLET FOGGING

A psychometric chart is shown in Figure 31 and will be used as an example to determine the cooling requirement for a gas turbine with a mass flow rate of 280 lb/sec. The following conditions are assumed—ambient conditions: Houston, Texas (American Society of Heating, Refrigeration, and Air-Conditioning Engineers (ASHRAE) climate data, figures not exceeded more than 30 hours per year), 96°F DB and 77°F WB, RH = 43 percent.

- Step 1—Find the ambient condition on the chart (point "A"). Note that the moisture content at this condition is 111 grains (H_2O)/lb (dry air).
- Step 2—Assuming 100 percent relative humidity (RH) ending conditions (i.e., cooling to the ambient wet bulb condition), proceed left up the constant wet bulb temperature line, until saturation is reached (100 percent RH, point "B"). A wet bulb temperature of *77°F at 100 percent RH.* Note that the moisture content at this condition is 142 grains.
- Step 3—Calculate how much moisture has to be added to the airstream to reach the wet bulb temperature. [142 − 110] = 32 grains/lb.
- Step 4—Compute the required water for gas turbine airmass flow rate of 280 lb/sec. The water required would be: 280 lb/sec × 32 gr/lb × [1/7005 gr/lb] = 1.279 lb/sec.
- Step 5—Convert to gpm, 1.279 × 60/8.345 = 9.2 gpm. This is the water flow required to cool 280 lb/sec of air by 11°F.

This number would be increased depending on the amount of fog intercooling desired.

Climatic Conditions and Applicability of Inlet Fogging

The dictum that fog evaporative cooling is not effective for "high humidity areas" is not true. Even the most humid environments allow for up to 15°F of evaporative cooling during the hotter part of the day. The term "relative humidity" refers to the moisture

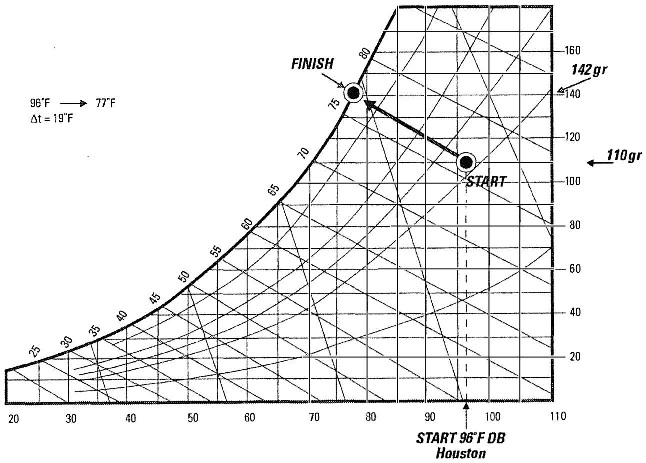

Figure 31. Inlet Cooling Example for "Humid" Gulf Coast Environment.

content in the air "relative" to what the air could hold at that temperature. In contrast, "absolute humidity" is the absolute amount of water vapor in the air (normally expressed in unit mass of water vapor per unit mass of air).

The moisture-holding capacity of air depends on its temperature. Warmer air can hold more moisture than cooler air. Consequently, relative humidity is highest during the cool morning and evening hours and lowest in the hot afternoon hours. As inlet air fogging systems cause a very small pressure drop in the inlet airstream, and are relatively inexpensive to install, they have been successfully applied in areas with very high summertime humidity such as the Texas Gulf Coast region in the USA. Traditional wetted-media type evaporative coolers are considered too inefficient to be applied in these areas.

POWER AUGMENTATION BY FOG INTERCOOLING

As was mentioned earlier, evaporation of fog within the compressor implies a continuous cooling of the air, which leads to a reduction in the compressor work for a given pressure ratio and to a change in the stage work distribution. Intercooling by water injection has been covered in detail by Utamura, et al., (1997), by Arsen'ev and Berkovich (1996), and by Molis, et al., (1997). A detailed investigation of the aerodynamic and thermodynamic effects may be found in Hill (1963). Wilcox and Trout (1951) have presented a detailed analysis of a turbojet thrust augmentation. Fortrin and Bardon (1983) have covered an interstage cooling method using methanol, and Kishimoto, et al. (1977), have covered the use of liquid air injection. A treatment of intercooling in combined cycles is made by Macchi, et al. (1994). Walsh and Fletcher (1998) have detailed water injection effects in addition to treating the whole area of gas turbine performance. This reference is particularly useful as it provides a comprehensive treatment of the gas turbine cycle and contains several useful performance equations and computations.

The process of direct fog ingestion into the compressor does cause a change in the compressor map. It is important to note that the operating point of an engine is determined by the matching between all the components. Thus the swallowing capacity of the turbine/nozzle at a given engine speed and turbine entry temperature will determine the operating point on the compressor map. Any changes in the compressor outlet conditions caused by fog intercooling will change the compressor map and cause a reduction in the surge margin. However, for the amounts of water injection envisioned, this does not typically pose a problem.

In order for the fog droplet to be vaporized, two conditions should be met.

- The vapor pressure in the surrounding air should be less than the vapor pressure at the droplet surface for water to evaporate from the droplet.
- There must be adequate heat transfer from the gas to the liquid phase to provide the required latent heat required to evaporate the droplets (i.e., both droplet size and residence time in the compressor are important). With today's high-pressure-ratio compressors, evaporation is expected. Figure 32 (Tsuchiya and Murthy, 1981) shows the temperature progression through a six-stage compressor. Note that the boiling point of the water is a function of the pressure and therefore increases as the gas heats up during compression.

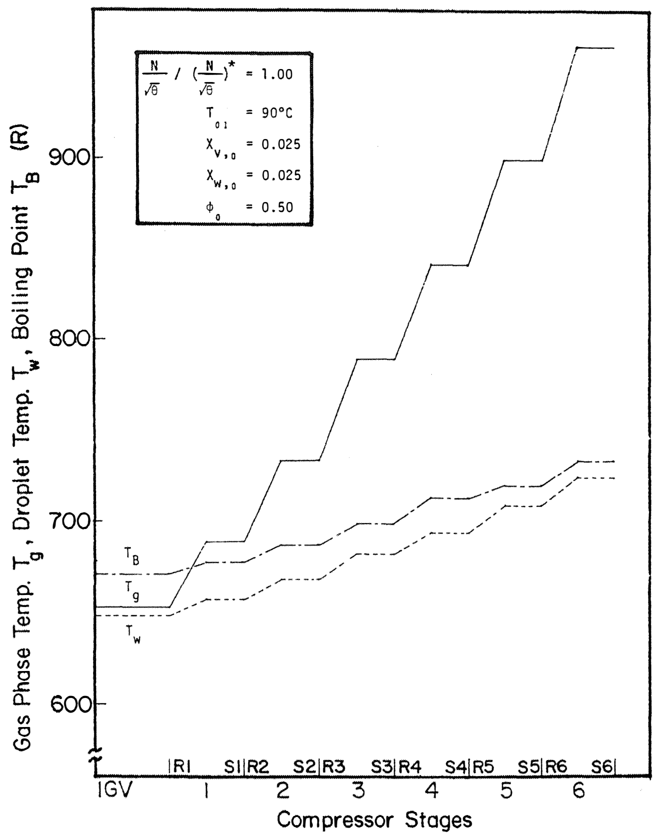

Figure 32. Temperature Progression Through a Six-Stage Axial Compressor. (Courtesy of Aero Propulsion Laboratory)

The drop in compressor work versus spray flow rate derived from a simulation by Utamura, et al. (1997), is shown in Figure 33. In this graph, the abscissa shows the compressor work ratio defined as:

R = Work with evaporation in the compressor/Work without evaporation

Results are shown for two pressure ratios and for different spray flow rate percentages. Simulation results for a 150 MW, 14.5:1 pressure ratio machine utilizing direct spray intercooling are shown in Figure 34 (Utamura, et al., 1997). The graph shows that the turbine output can be boosted by 23 percent by a spray flow rate of 2.3 percent with an associated increase in the thermal efficiency.

An important finding that was also verified experimentally by Utamura was that the adiabatic compressor efficiency actually increased with the addition of water. This finding has been corroborated by Arsen'ev and Berkovich (1996). Both these researchers have concluded that there is an increase in the adiabatic efficiency of the compressor with inlet fogging. Earlier studies done on power augmentation had not considered this and, to be conservative, had assumed a drop in compressor efficiency (Wilcox and Trout, 1951).

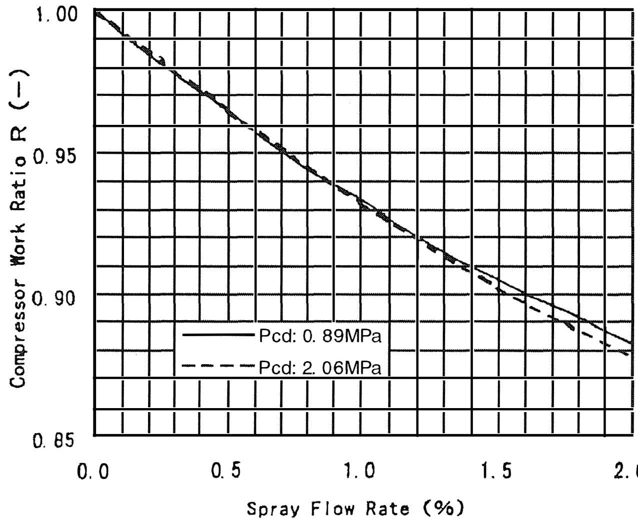

Figure 33. Drop in Compressor Work with Direct Intercooling. (Courtesy of ASME)

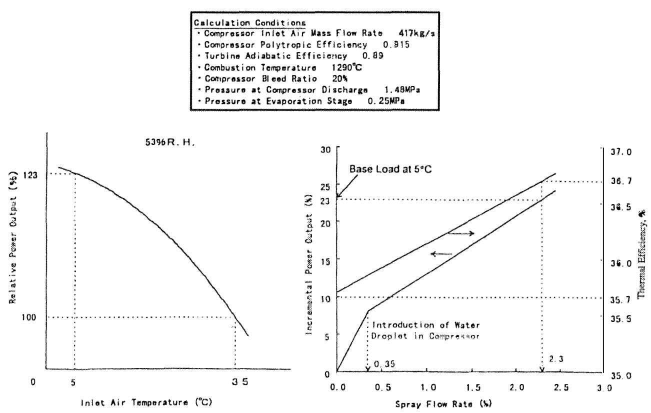

Figure 34. Performance Prediction for a 150 MW Class Gas Turbine with Compressor Water Injection. (Courtesy of ASME)

Sexton, et al. (1998), has modeled a naval LM2500 engine and concluded that a 34 percent power boost could be obtained by injecting 24 gpm of water into the compressor inlet along with a drop in specific fuel consumption (Figure 35, Sexton, et al., 1998). The drop in compressor discharge temperature due to the water fog injection is shown in Figure 36 (Sexton, et al., 1998). The increase in pressure ratio between dry and wet operation (with fog injection of 10 gpm) is shown in Figure 37 (Sexton, et al., 1998). As this figure depicts, operating at the same speed and airflow rates (design airflow rate of 137 lb/sec), the compressor tends to stall at higher airflows than the dry compressor operating at the same speed.

Intercooling studies by injection of liquid air into the compressor inlet is described by Kishimoto, et al. (1977), and by means of methanol spray injection by Fortrin and Bardon (1983).

IMPACT OF FOGGING ON EMISSIONS

There are several studies that have indicated that NO_x emissions can be reduced by fogging of the inlet air. Experimental work on an LM2500 unit indicated NO_x reductions of approximately 33 ppm over the range of engine powers with the addition of 4.5 gpm of water to the inlet of the engine's compressor, as shown in Figure 38 (Sexton, et al., 1998).

Thermal NO_x is an exponential function of flame temperature. Hotspots in the flame are responsible for the bulk of NO_x production. The water vapor introduced by inlet fogging, having a specific heat about four times that of air, has a quenching effect on hotspots.

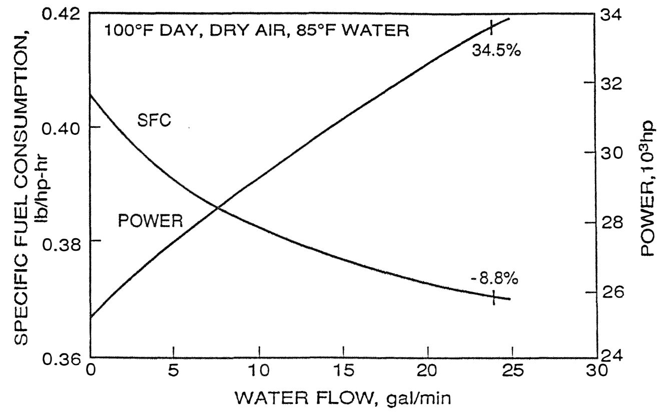

Figure 35. Effect of Compressor Inlet Water Fog Injection on Power and Single Phase Feeding Power Conditioner (SFC). (Courtesy of ASME)

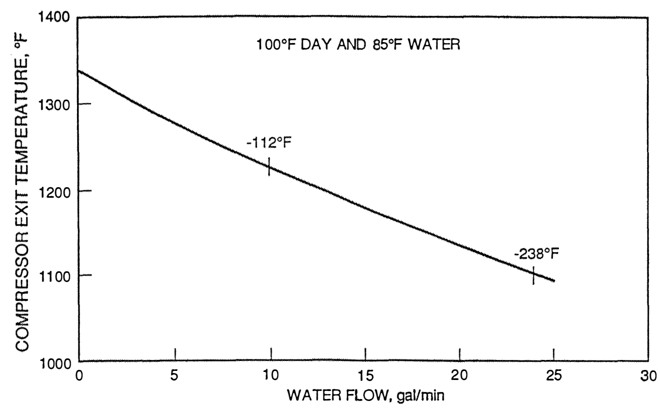

Figure 36. Compressor Discharge Temperature with Water Fog Injection. (Courtesy of ASME)

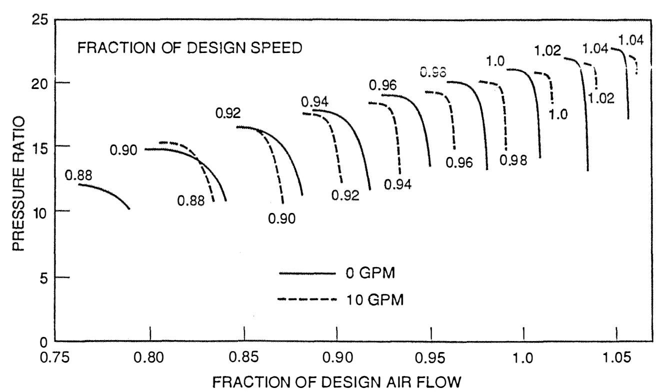

Figure 37. Compressor Map Effect of Water Fog Injection. (Courtesy of ASME)

DESCRIPTION OF THE GE LM6000 SPRINT INTERCOOLED GAS TURBINE

The first commercially available gas turbine that can be purchased utilizing direct mist intercooling is the GE LM6000 Sprint intercooled gas turbine. The machine has a five-stage low-pressure (LP) compressor (pressure ratio of 2.4:1) and a 14-stage high-pressure (HP) compressor (pressure ratio of 12:1) and operates at an overall pressure ratio of 30:1. It has a two-stage HP turbine and a five-stage LP turbine, with drive being available at either end of the LP shaft. The first intercooled gas turbine has

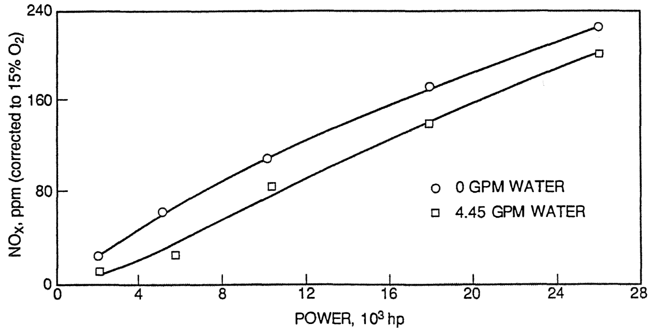

Figure 38. No_x Reduction with Water Fog Injection. (Courtesy of ASME)

been installed at Fort Lupton, Colorado, where it operates in cogeneraton service and provides a boost in power of approximately 9 percent at ISO conditions and more than 20 percent on a hot 32°C day. The cross section of the LM6000 engine in Figure 39 shows the location between the LP and HP compressor sections where the water is injected for intercooling purposes.

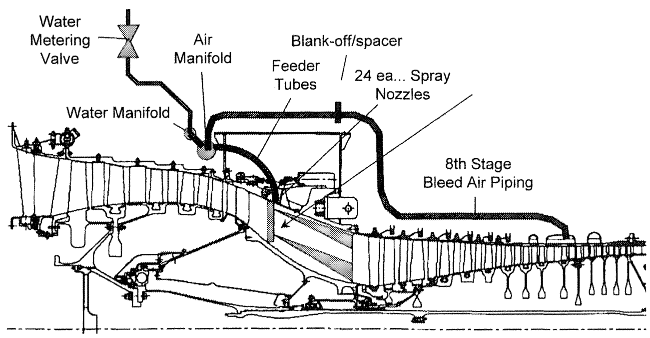

Figure 39. Cross Section of LM6000 Sprint Intercooled Engine. (Courtesy of GE SSEP)

Approximately 6 to 7 gpm are injected into the space between the LP and HP compressor sections by the use of 24 nozzles. Atomization is assisted by the use of eighth stage bleed air that is used to atomize the water into small droplets. A photograph of the piping associated with the injection system located in the HP compressor inlet duct section is shown in Figure 40.

Being a 30:1 pressure ratio machine, this engine is ideally suited to intercooling, causing a reduction in the work of the HP compressor and a lower discharge temperature. Table 3 provides data on the power boost that can be achieved by the use of intercooling and provides an indication of the intercooling water injection flow rates.

This unit has run successfully at the Fort Lupton cogeneration plant. There is also investigative work going on in the incorporation of water injection to the LP compressor of this engine. Details may be found in Johnson and Thompson (1998) and McNeely (1998).

CASE STUDIES AND OPERATING EXPERIENCES

Application of Inlet Fogging to a Frame 7 E Gas Turbine

An application of direct fogging made on a Frame 7001E model turbine has been reported on by Molis, et al. (1997), and is summarized here. The goal of the project was to increase power from the gas turbine during the hot summer months, and a system was demonstrated during August 1996. The subject gas turbine is operated at base load power in peaking service and is base load rated at 60.9 MW.

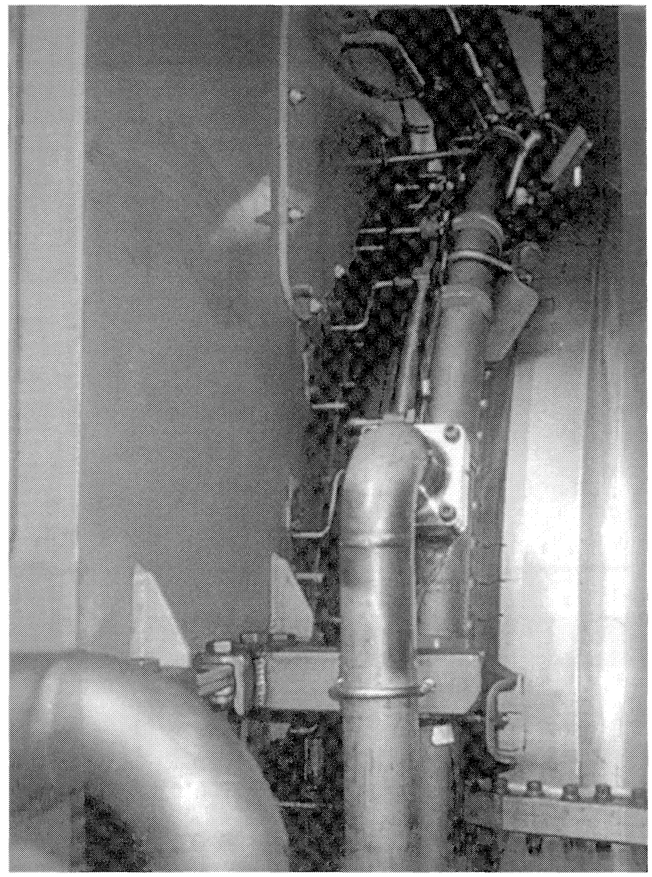

Figure 40. Photo Showing Inlet Piping at Plenum Between LP and HP Compressor. (Courtesy of GE SSEP)

Table 3. Performance Enhancement Comparing the LM6000 PC and Sprint Engines at Different Ambient Temperatures. (Data Courtesy GE SSEP)

Ambient Temperature		-1°C/30°F	7°C/45°F	15°C/59°F	21°C/70°F	27°C/80°F	32°C/90°F
Output (MW)	LM6000PC	51.5	48.2	43.3	38.9	35.2	31.9
	Sprint	51.5	50.0	47.2	44.4	41.1	38.4
Heat Rate (kJ/kWh)	LM6000PC	8773	8835	8982	9195	9440	9686
	Sprint	8756	8836	8936	9042	9163	9310
Exh. Temp. (°C)	LM6000PC	432	435	433	434	438	442
	Sprint	427	438	440	441	444	453
Flow (kg/s)	LM6000PC	140	134	127	121	114	108
	Sprint	141	137	132	127	123	118
Sprint Water (L/s)		0	.25	.47	.59	.63	.69

System Description

In this project, the inlet air enters through an elevated horizontal inlet house. The fog nozzles were located after the bird screen and prior to a mist eliminator located in the inlet duct. Thereafter the flow went via two pantleg ducts, through a silencer, and into the gas turbine inlet. Water drains were installed to the floor of the inlet duct. The fog spray delivery system consisted of a delivery skid and an array of stainless steel tubes containing the spray nozzles. High-pressure water at 2900 psi is delivered by the high-pressure pumps.

The nozzle array is partitioned into eight stages, with each stage configured with multiple horizontal legs of tubing. The system consists of two identical skid mounted water transfer systems that are located under the air inlet house.

Results

As this project was in part a demonstration project, some special test instrumentation was added. For example, a high temperature dynamic pressure sensor was installed in a borescope inspection port near the seventeenth stage of the compressor. This sensor was installed to aid in monitoring compressor stall during high levels of fog overspray. The results of the tests as reported by Molis, et al. (1997), are presented in Figure 41. This figure shows the compressor inlet temperature dropped from 84°F to the wet bulb temperature of around 70°F, as the appropriate number of staged fog nozzles were utilized. The increase in power can also be seen. Once the wet bulb temperature is attained, all additional fog injection is then ingested into the compressor causing an intercooling effect, thus further boosting power. The test results reported by Molis, et al. (1997), include:

- 100 percent saturation achieved on all days of the test allowing attainment of the wet bulb temperature.
- Power increase from the direct fogging evaporative cooling was approximately 3.5 percent for every 10°F of inlet cooling.
- There was a 5 percent power gain attained for each 1 percent (of air flow rate) of overspray.

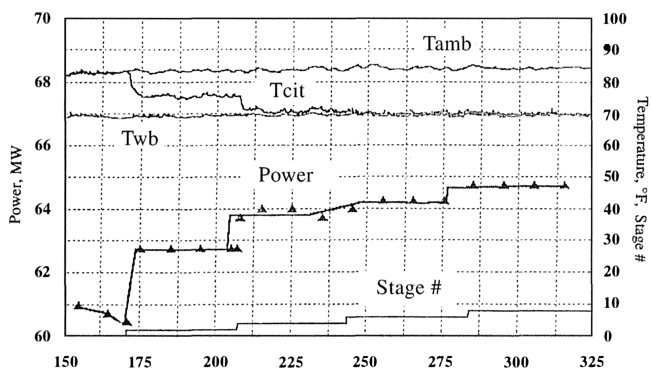

Figure 41. Power Boost by Fog Application. (Courtesy of PowerGen International)

Application of Inlet Fogging on a Frame 7 FA Gas Turbine

Another implementation on a Frame 7 FA advanced gas turbine has been implemented at a combined cycle power plant in 1997, in Oregon. The utility investigated several cooling options and determined that the most cost-effective was an inlet fogging system. The analysis included operating and maintenance costs and also the need for structural modifications required for the inlet system to incorporate traditional evaporative cooling.

System Description

In this installation, the location of the fogging manifold is downstream of the air filters and upstream of the silencers and trash screens. A total of 1120 nozzles is arranged in eight stages with each stage providing 3.75°F of cooling. Operation of all eight stages would result in cooling of up to 30°F. The nozzle atomization pressure is 2000 psi and the total fogging water flow is 50 gpm. In this application, the pump skid has four 20 hp high-pressure pumps. Shutdown time required for installation was less than one week.

Operating Results and History

The facility has reported an output increase of approximately 2 MW per cooling stage (3.75°F per stage). A problem that was noted at this facility was the accumulation of water in the inlet duct. A water drain was installed downstream of the fog nozzles and eliminated this problem.

Inlet Fogging Application to Frame 7 EA Cogen Plant-Hybrid Application

System Description

A cogeneration facility in California operates a large Frame turbine that generates a total of 120 MW of power and also provides steam to a nearby plant. This plant utilizes chillers to cool the gas turbine inlet air regardless of ambient temperature. However, high running costs made it feasible to add some form of supplemental cooling. In 1994, the company decided to drop evaporative coolers and retain chillers and supplement them with a high-pressure inlet fogging system.

Operating Problems and Solutions

Implemented in 1994, this represented one of the earlier fogging systems installed and there were several teething problems.

- Due to the use of high speed positive displacement pumps, excessive vibration was noted on the high-pressure lines. The installation of pulsation dampers resolved this problem. While pulsation dampers are now always used, slow speed pumps inherently reduce vibration and wear on pump valves and seals.
- The water supply stagnated and allowed the growth of bacteria, which caused plugging of the high-pressure fogging nozzles and clogging of the filters. The problem was resolved by maintenance schedules for draining the system. This initial installation also changed from polyvinyl chloride (PVC) water supply piping to stainless steel (SS) piping, which is now a standard feature in fogging systems offered.
- Some fog nozzle manifold failures were experienced possibly by vibration caused by air flow. The problem was resolved by strengthening the attachment of the nozzle manifolds to the structure.

Currently, this installation utilizes its chillers only when economical (at certain times of the day) and uses its fogging system, which is less expensive to operate during off-peak hours. This is a good example of a hybrid system wherein both cooling technologies are integrated and allow an optimal solution.

Inlet Fogging Application to LM6000 Gas Turbine-Hybrid Application

Another application that uses a hybrid approach is a LM6000 installation located in a peaking plant in Las Vegas that starts up 560 times a year. This installation has a 1040 ton capacity absorption chiller but as these units take time to come online, a high-pressure inlet fogging system is used to provide maximum power instantly. The fog system is used exclusively when the ambient temperature is less than 70°F. When temperatures exceed 70°F, the absorption chilling system is also brought online.

In this application, the fogging system is installed upstream of the inlet filters, and results have shown that the final barrier filters remain dry and filters stay cleaner due to the scrubbing effect of the fog. This system consisted of 240 nozzles installed in three stages of cooling and operates at 2000 psi water pressure with a water flow rate of 10.8 gpm. Three high-pressure pump units rated at 5 hp each are installed, one for each stage.

Inlet Fog Application (Upstream of Filters) for Frame 7 EA Turbine

System Description

A cogeneration plant operates a Frame 7 EA turbine with a fog system installed upstream of the air filters. The inlet fogging system was designed to include six stages of cooling with 30 gpm of water pumped through a total of 912 nozzles by two high-pressure (1000 psi discharge) pumps. This was also one of the earliest applications of inlet fogging.

Operating Experiences

The main problem experienced by this cogen facility resulted from its adjacency to a coke plant and the effectiveness of high-pressure fogging as a scrubbing or prefiltering system. Large quantities of airborne coke fines were scrubbed out of the inlet air and captured in the fog water. The water was collected on the fog droplet filters and, as was originally designed, returned to the fog system. As the recycled water contained a large number of submicron coke fine particles, the shear quantity of coke fines caused plugging of the filters located in each nozzle. Several fixes were attempted including using larger orifice nozzles that would allow the coke fines to pass through. While this solved the plugging problems, it did not address the abrasion problem. Further, the larger orifices created larger droplet sizes, which did not provide the evaporative cooling that would have been attained with the smaller droplet sizes. The final solution was to reinstall smaller orifice sizes and abandon water recycling. Fresh demineralized water was used and the wastewater treated and burned off by the plant's stream host. The plant continues to use the fog system for gas turbine power augmentation.

Application of Inlet Fogging to a Frame 5 Cogen Installation

An application on three Frame 5 gas turbines (air mass flow rate of 161 lb/sec) operating in cogeneration service has been reported on by Nolan and Twombly (1990). This facility is located in Rifle, Colorado, which has long hot and dry summers. The facility examined traditional evaporative cooling approaches but chose to use a direct fogging system owing to a low initial cost.

System Description

The system utilized 540 nozzles that allowed a total flow of 15 gpm at 600 psi pressure supplied by three reciprocating pumps.

Operating Experiences

Wet bulb temperatures could be effectively attained (i.e., 100 percent evaporative effectiveness) throughout the temperature range of 70°F to 100°F. Figure 42 (Nolan and Twombly, 1990) shows the average gas turbine output over the July to September time frame. In this figure, the lower curve (A) shows average output for all three units if no fogging would have been employed. The curve in the middle (B) shows the predicted output from the OEMs curves with average wet bulb temperatures attained, i.e., with 100 percent evaporative cooler effectiveness. The uppermost curve (C) was the actual curve of these engines. The difference between curve B and C was due to the fog intercooling effect. As reported by Nolan and Twombly (1990), an economic analysis of this system indicated a payback of 3.2 months.

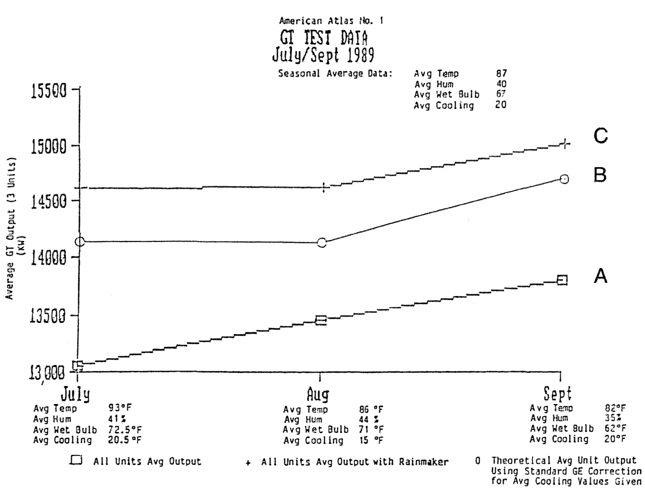

Figure 42. Boost Derived on Engine by Fog and Fog Intercooling. (Courtesy of ASME)

Inlet Fog Installation on 48 Heavy Duty Peaking Gas Turbines

A major utility in Tennessee has installed inlet fogging systems on 48 of its gas turbines including 28 Frame 7 units, five Frame 5N units, and four 501 B units. All these units are peaking gas turbines with no inlet filters. Fog nozzles are located downstream of the silencers and upstream of the trash screens. Several of these gas turbines also utilized fog nozzle lines to cool lube oil heat exchangers to accommodate the augmented turbine output during hot ambients. The high-pressure fogging systems all operate at 2000 psi (144 bar) and were designed to permit 26°F of fog cooling. The utility has estimated that the inlet fog system has resulted in an augmentation of 150 MW to 200 MW of peaking power.

DESIGN AND IMPLEMENTATION CHECKLIST

Table 4 provides a checklist to ensure troublefree operation of an inlet fogging system. While not totally exhaustive, it contains several elements of importance. It covers the following areas:

- Application engineering
- Design
- Choice of vendor
- Manufacture
- Implementation
- Operation
- Maintenance

Table 4. Design and Implementation Checklist.

DESIGN AND IMPLEMENTATION CHECKLIST FOR INLET FOGGING SYSTEMS

	CHECKLIST
APPLICATION ENGINEERING	• Has a detailed climatic study been conducted? • Has economic analysis been conducted to check NPV of system over its life cycle? • What are the economic parameters that are subject to change that might make the project marginal?
DESIGN REVIEW	• Check all the vendors design calculations with regard to flow requirements, off-design conditions, and evaluate the design under different climatic conditions. • Evaluate amount of overspray required (i.e., intercooling effect) and ensure that the compressor can accommodate this. • If overspray is being considered, review generator capability, LO cooler capability. • Insist on all SS piping. • If a high-pressure system is being used, have pulsation dampers been provided considered? • In event of a problem, does the design permit rapid isolation so operation can proceed without the fogging system? • Optimize location of the fogging manifold with respect to air filters and inlet system. • Check that sequential fogging capability (cooling stages) is adequate to meet your demand profile and turndown profile. • What design features has the vendor provided to avoid potential FOD? • Check rigidity of the manifolds to avoid flow induced vibration due to gas turbine airflow. • Review vendors proposed manifold design for structural rigidity and strength. • Review possible tie in of skid PLC with plant DCS. • Ensure that design does not impose a large pressure drop. • Evaluate proposed installation in duct for maintainability and accessibility. • Ensure that appropriate drain lines will be installed in the inlet system.
VENDOR SELECTION	• Has vendor proven experience with application of fogging to gas turbines? Check track record. • What is vendor experience with the size/type of gas turbine that you are considering? • Talk to users of the proposed system. • Evaluate long-term service capability of vendor. • Evaluate after sale support by asking for user experience. • Has vendor supplied systems that can be used as hybrid systems should this need arise in the future? • Check track record of installation and debugging time with different users. • What tests has the vendor done to quantify droplet size for his system?
PRODUCTION AND QUALITY ASSURANCE	• Review Vendors Quality Plan and set up inspection points for the skid and other equip. • Does vendor exhaustively check each fog nozzle? • What inhouse tests are conducted?
INSTALLATION	• Ensure that the appropriate outage has been planned and that the details of the installation and Division of Responsibility has been clearly defined with the vendor. • Has thought been given to maintainability of the system – access doors, and drains?
TESTING	• Has a test plan been clearly defined and guarantee qualifications made. This should be carefully looked at to avoid problems, especially if there is a turbine overhaul instituted.
OPERATION	• What testing program has been instituted to ensure quality demin water is always being supplied? • What sensors and interlocks have been provided to ensure quality water?
MAINTENANCE	• Has vendor provided a detailed manual including PLC logic and code? • Has some training been scheduled for your operators? Are operators aware of the importance of only demin water being used? • Review maintenance of the demin water system.

CLOSURE

The temperature dependency of gas turbine engines, wherein output and efficiency drops with increasing ambient temperatures, makes inlet fogging an effective power enhancement technology. Power augmentation is derived first by direct evaporation, which allows attainment of the wet bulb temperature and 100 percent RH at the inlet. Then, if desired, further boost can be attained by fog intercooling by allowing micron size particles to enter the compressor, thus reducing gas turbine compressor work. This aspect of fog intercooling is unique from the other technologies available. As with any technology, its use must be evaluated based on the turbine characteristics, climatic conditions, power augmentation expectations, and project economics. While fogging may be the optimal solution for some installations, it should also be considered in hybrid systems to augment power by inlet cooling. This paper has covered the theoretical basis for gas turbine power augmentation by inlet fogging and fog intercooling, and has delineated several practical design implementation and operational aspects.

APPENDIX A EQUATIONS, RULES OF THUMB, AND DROPLET BEHAVIOR

Rules of Thumb for Installed Cost per Ton of Cooling and Power Requirements

These numbers in Table A-1 (Giourof, 1995) are budgetary and are only provided to give a feel for the situation.

Table A-1. Rules of Thumb

Technology	Installed Cost $/ton	Power Requirements
Evaporative cooling	100	<0.08 kW/ton
Absorption, Li-Br single effect	800	18 lbm/hr/ton of 15 psig steam
Absorption, Li-Br double effect	1000	12 lbm/hr/ton of 110 psig steam
Ammonia absorption	3000	18 lbm/hr/ton of 15-50 psig steam
Ammonia mechanical chiller	950	0.58 kW/ton
CFC-based centrifugal chiller	800	0.7 kW/ton

Droplet Stability and Rupture

The stability of droplets moving through a gaseous atmosphere depends on the ratio of aerodynamic pressure forces tending to deform it and the surface tension force causing its shape to remain spherical (Tsuchyia and Murthy, 1980). This ratio is expressed by the Weber number. At slow relative speed or small droplet diameter, droplets remain nearly spherical. If the critical Weber number for breakup is sufficiently exceeded, the breakup occurs within a very short time. The time to rupture is given by:

$$T_{rupture} = (0.3 - 1)\ \frac{\pi}{4}\sqrt{\frac{\rho_f D^3}{\sigma}} \qquad \text{(A-1)}$$

where:

σ = Surface tension of droplet

Rupture time for a 100 micron droplet is in the range of 0.03 to 0.01 msec. Obviously as the equation indicates, as the diameter gets smaller, the time drops.

Droplet Evaporation Times

Figure A-1 (Tsuchyia and Murthy, 1980) shows evaporation time of a 15 micron droplet with the effect of relative velocity of the droplet to the airstream. It can be seen from this graph that even with a small relative velocity difference between the droplet and the airstream, the evaporation times are exceedingly small and certainly would exceed the residence time in the compressor. Further, with the high-pressure ratio units in operation today, the gas temperatures are considerably high, implying more rapid heat transfer to the droplet.

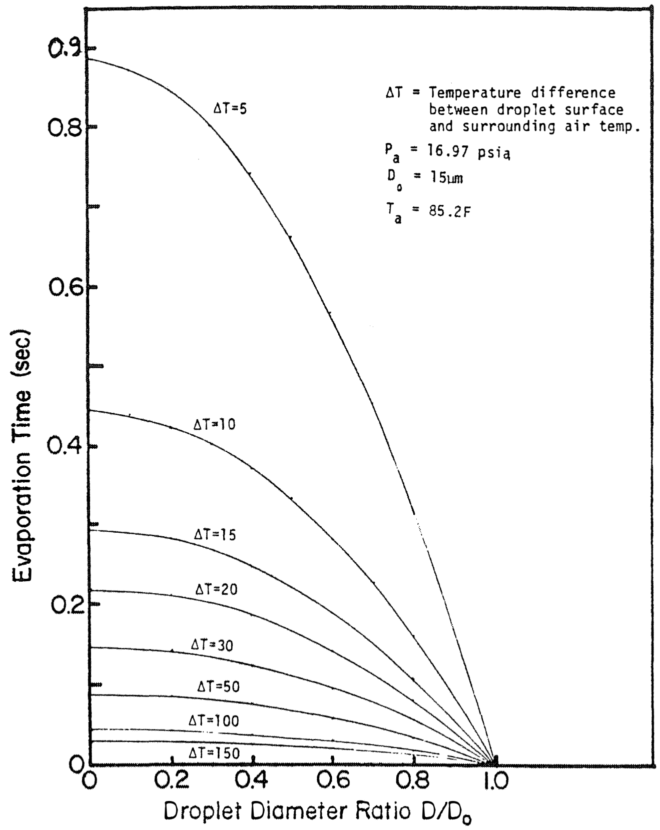

Figure A-1. Droplet Evaporation Time with Negligible Relative Velocity Effect (15 Micron Diameter Droplet)—Compressor. (Courtesy of Aero Propulsion Laboratory)

If the temperature of the water droplet reaches local saturation temperature, the water droplet undergoes boiling, provided there is adequate heat transfer available in the gas phase. The evaporation time with the effect of relative velocity is shown in Figure A-2 (Tsuchuia and Murthy, 1980).

Compressor Performance with Wet Compression

Hill (1963) has conducted an excellent study of the aerothermodynamic effects of coolant injection and has derived several useful equations.

$$\frac{P}{P_2} = \left(\frac{T}{T_2}\right)^{\frac{\gamma}{\gamma-1} + \frac{L}{R}\cdot\frac{dW}{dT}} \qquad \text{(A-2)}$$

where:

L = Latent heat
R = Gas constant
dW/dT = Evaporation rate (with time)

Note that when dW/dT = 0, the equation reduces to the traditional isentropic compression equation.

Figure A-3 indicates the relative ideal compression work for pressure ratios of six and 12 for different coolant flow variables defined as $w\lambda/\theta$.

where:

λ = Latent heat of coolant/latent heat of water = 1 for water as coolant
θ = $T_{in}/520$

The ordinate on this graph is [Ideal Wet Work/Ideal Dry Work].

Hill has provided an equation for the flow increase due to coolant injection:

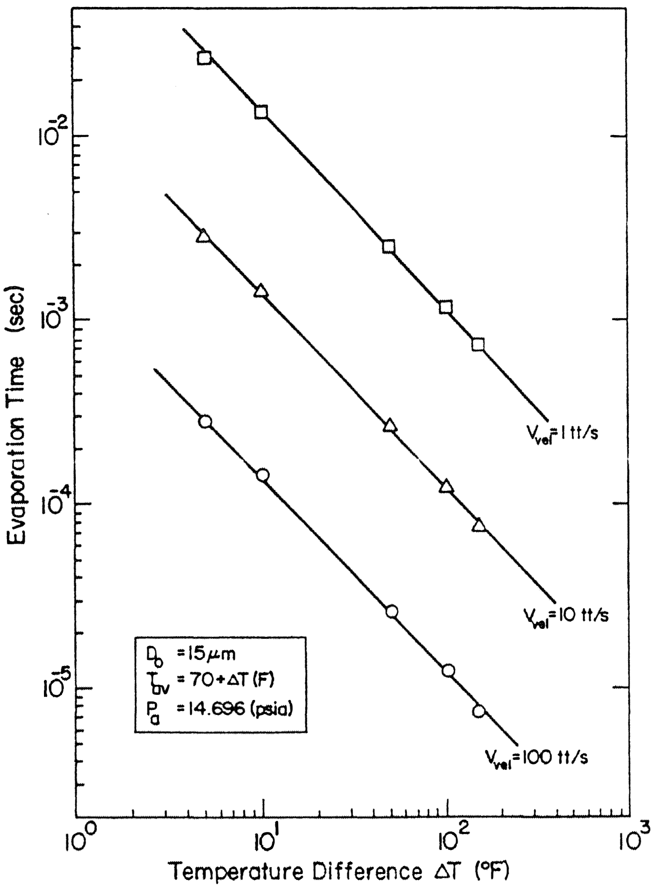

Figure A-2. Evaporation Time with Relative Velocity Effect. (Courtesy of Aero Propulsion Laboratory)

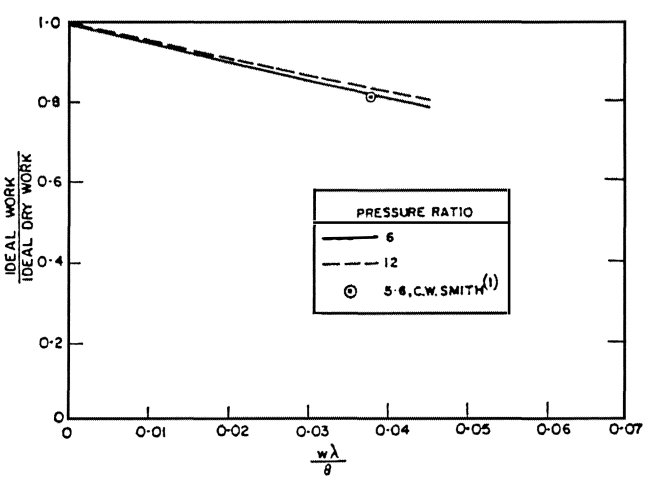

Figure A-3. Comparison of Ideal Wet and Dry Compressor Work. (Courtesy of Aeronautical Quarterly)

$$\alpha_t = \frac{30\,\eta_p \log_e \dfrac{P_3^*}{P_2}}{\left[\left(\dfrac{T_3^*}{T_2}\right) - 1\right](1 - \psi^*)} \cdot \frac{w\lambda}{\theta} \qquad \text{(A-3)}$$

where:

η_p = Polytropic efficiency
P_3^* = Compressor discharge pressure
P_2 = Compressor inlet pressure
λ = (Latent heat of coolant)/(latent heat of water) = 1 (for water coolant)
w = Vapor/air mass ratio
θ = $T_{in}/520$
T_3^* = Dry compressor discharge temperature
T_2 = Compressor inlet temperature
ψ = Average work coeff = $1-\sigma^*(\tan\alpha_1 - \tan\alpha_2)$, α_1 and α_2 are blade angles
ψ = $\Delta h/U_2^2$
φ = Cx/U

Droplet Acceleration and Droplet Behavior in the Compressor

- A 30 micron drop will be accelerated to the stream speed within a length of approximately 1 inch. Hill's studies have indicated that droplets would strike the blades or the outer annular wall.
- Hill summarizes that drops of minimum average size can only be expected to travel with the airstream in the blade passages of an axial compressor if their velocities are much smaller than the gas speed. If this is not so, the drag forces on the drops will be too low to effect substantial turning. Given the rapid acceleration within 1 inch, it is therefore unlikely that it will follow the airstream, and most will impinge on the blade surfaces within the first or second stages.
- Similar studies conducted by Arsen'ev and Berkovich (1996) and Tsuchiya and Murthy (1981) have indicated that the evaporation will occur on the blade surfaces or on the end walls on the outer part of the compressor.
- Recent CFD studies have indicated that droplets with diameters of 10 to 20 microns will follow the flowpath.

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