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INLET FOGGING OF GAS TURBINE ENGINES PART A: THEORY, PSYCHROMETRICS AND FOG GENERATION.

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ABSTRACT

Gas Turbine output is a strong function of the ambient air temperature with power output dropping by 0.3-0.5 % for every 1°F rise in ambient temperature. This loss in output presents a significant problem to utilities, cogenerators and 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. The paper contrasts the traditional evaporative cooling technique with direct inlet fogging. The state of the art relating to fog generation and psychrometrics of inlet fogging are described.

1. INTRODUCTION

1.1 The Need for Inlet Fogging

Turbine output and efficiency of gas turbine engines are reduced during periods of high ambient temperature. 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 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. Coupled with the need for power augmentation during hot ambients is the need to accomplish this with a minimum capital cost for the incremental power generated. High-pressure inlet fogging fits this niche.

Fog intercooling which has been applied from the early days of gas turbine and jet engine technology (injection of water into the axial flow compressor) is a technique that consists of over-injecting fog into the inlet air stream (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 air stream 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.

1.2 Historical Antecedents

Water injection into gas turbine compressor inlets has been studied and applied since the forties. Early studies were done in the forties by Kleinschmidt (1946), Wilcox (1951). Wet compression was described in detail in several well known text books on gas turbines written in the 40s. Water injection was used on the older jet engines (with zero or low bypass ratios) to boost take –off 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. The first published paper utilizing this technology was reported by Nolan and Twombly (1990) at an ASME Gas Turbine Congress. A review of the technology may be found in Meher-Homji and Mee (1999).

1.3 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:

1. Refrigerated inlet cooling systems- utilizing absorption or mechanical refrigeration
2. Evaporative methods- either conventional evaporative coolers or direct water fogging
3. 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 (1988). Stewart (1999) has provided a design guide for combustion turbine air cooling systems. No single technology can be considered the best and though the focus of this paper is on direct evaporative cooling, the choice of the appropriate technology must be made on a case by case basis and considering detailed climatic conditions and economic parameters. There may be cases where traditional evaporative cooling is appropriate especially where high quality water is not available.

2. CYCLE THERMODYNAMIC CONSIDERATIONS.

2.1 Gas Turbine Cycle Thermodynamics

The compression process consumes as much as 66% 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.

The compressor work per lb of air is given by

$$\left(\frac{W}{J}\right)_{Compr} = \frac{h_2'-h_1}{\eta_c} = \frac{C_p[T_2'-T_1]}{\eta_c} = \frac{C_pT_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}-1\right]}{\eta_c} \qquad (1)$$

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)

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

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 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 1 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

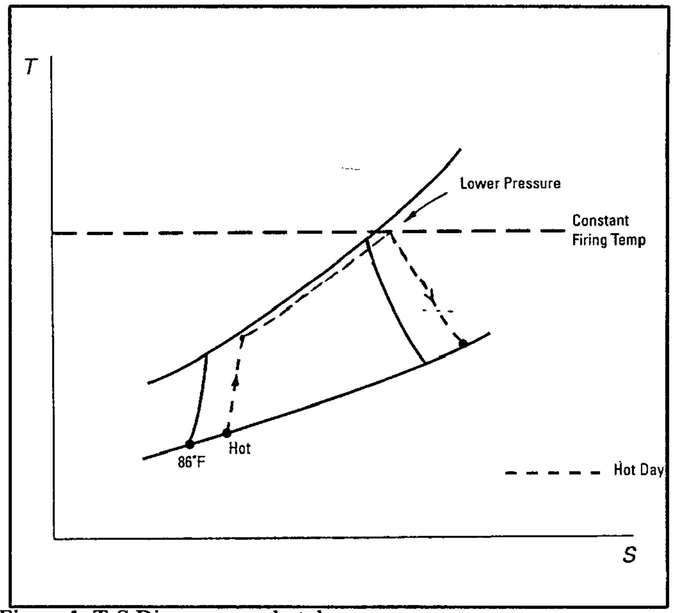

Figure 1 T-S Diagram on a hot day

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 2 (Meher-Homji et al, 1993).

Figure 3 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 where N is the rotational speed and $\sqrt{\theta} = \sqrt{[Tin/Tref]}$. 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.

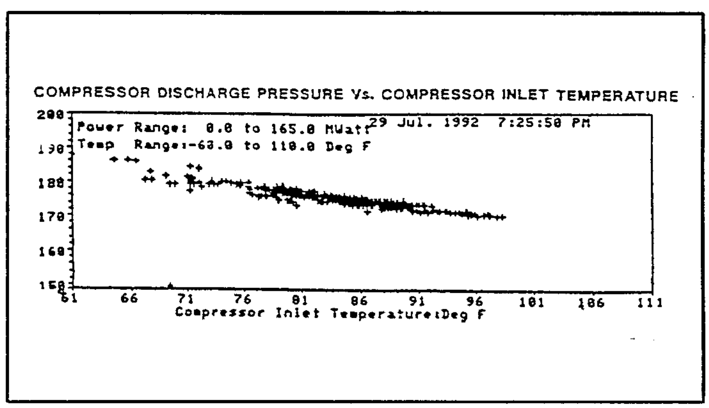

Figure 2. CDP- Ambient temperature dependency of a Frame 7F Gas Gas Turbine (Meher-Homji, et al 1993)

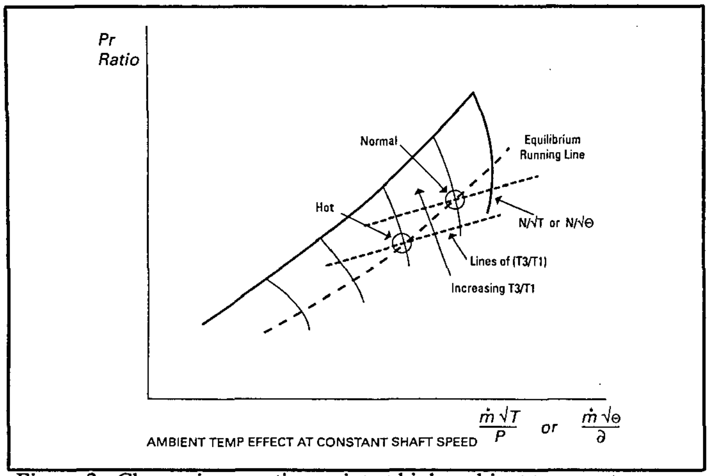

Figure 3. Change in operating point at high ambient temperature.

Power reduction during hot ambients can be as high as 20-40% 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.

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, lbs/ft^3

It is evident from the equation that the ambient temperature will have considerable impact on the air density.

The single shaft gas turbine operating at constant speed is essentially a constant volumetric flow machine assuming that the variable geometry is 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 also 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).

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. 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 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.

3. HIGH PRESSURE FOGGING COMPARED TO TRADITIONAL EVAPORATIVE COOLING

3.1 Traditional Evaporative Cooling

Traditional evaporative coolers that use media for evaporation of the water have been widely used in the gas turbine industry especially in hot arid areas. The basic principle of this type of cooling is that as water evaporates, it consumes 1,160 BTUs 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 4. Water is distributed over the media blocks which are made of fibrous corrugated material. The airflow through this block evaporates the water.

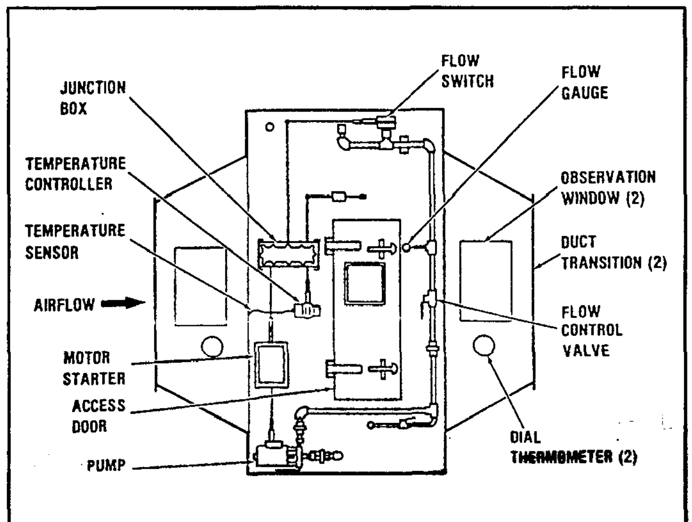

Figure 4. Traditional Evaporative Cooler (Johnson, 1988)

Evaporative cooler effectiveness is given by:

$$E = \frac{T_{1DB} - T_{2DB}}{T_{1DB} - T_{1WB}} \qquad (4)$$

Where,

T_1 = inlet temperature
T_2 = exit temperature of evaporative cooler
DB = dry bulb
WB = wet bulb

A typical value for effectiveness is 85-90% which means that the Wet bulb temperature can never be attained.

The temperature drop is given by:

$$\Delta T_{DB} = 0.8\,(T_{1DB} - T_{1WB}) \qquad (5)$$

A psychrometric chart can be used to obtain the values. The exact power increase depends on the particular machine type, site altitude and ambient conditions. The power increase is the greatest at low relative humidities. Typical numbers are presented in Table 1 below

Ambient Temp, F	Power Boost, % RH= 60%	Power Boost, % RH= 40%	Power Boost, % RH= 20%
70	2.4	4.9	6.5
80	3	5.5	7.8
90	3.5	6	8.8
100	4	7	10

Table 1. Typical power boosts for traditional evaporative cooler.

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).

3.2 Issues related to Traditional Evaporative Cooling

In order to compare traditional evaporative cooling with direct fog evaporative cooling, it is helpful to review some key issues relating to evaporative cooling.

3.2.1 Blow down issues. There are two types of evaporative coolers- circulating and non-circulating. The coolers used for most gas turbine operations are of the recirculating type and consequently there is a requirement for blowdown in order to avoid the accumulation of minerals in the water. Thus make up water will equal the blowdown water plus the water evaporated. The blowdown rate is dependant on the hardness of the water available and a curve is provided by Johnson (1988) which is widely used in the industry This curve indicates that the blowdown rate should equal 4 X the evaporation rate.

3.2.1 Mist eliminator. As the water is not treated, it is imperative that none enter the compressor and so a mist eliminator is provided on the downstream side to ensure that the air entrained water droplets are removed.

3.2.3 Water flow Rates. Water flow rates are typically between 1-2-gal/min for each square foot of surface area of the distribution pad but this number can be higher for larger evaporative coolers. Higher flow rates minimize the potential of mineral build up but increase the risk of entrainment of the water in the air stream. Thus the amount of water should be carefully adjusted during commissioning and per Johnson, should not be "tuned" constantly as this often leads to excessive dry spots or the other problem of water carryover.

3.2.4 Water Carryover - Even with the use of mist eliminators, water carryover must be minimized. Johnson (1988) has listed the causes of water carry over some of which are provided below.

- Incorrect media polarity
- Damaged media (can occur after field reinstallation and can result in improper alignment and cracks between media. Poor handling often crushes Media.
- Improperly aligned media strips - If strips are not properly aligned together the resultant gap may allow water carry over.
- Poor media seal against retainers.
- Excessive water flow- media flooding can cause carryover.
- Uneven water distribution from the header- this is often caused by improper initial design of the holes or clogging of the holes resulting in an imbalance of flow over the media.
- Uneven or distorted airflow throughout the evaporative cooler
- Scale deposits on the media.

Media type evaporative coolers do not normally require demin water, in fact demin water can damage the media. However some operators have reported compressor fouling caused by carryover of water with high levels of dissolved minerals. This can be avoided by installing mist eliminators downstream of the cooler or by ensuring that the air velocity through the media is not so high as to cause carryover. Some form of water treatment may be required in order to deal with potential problems of microbiological fouling, corrosion and of course scaling. It is also advisable to clean and flush the header on a regular basis.

3.3 Inlet Differential Pressure Drop.

The presence of a media type evaporative cooler inherently creates a pressure drop and this will create a drop in turbine output. As a rough rule of thumb, a 1" WG increase in inlet duct losses will result in a 0.48% drop in power and a 0.12% 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.

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 that those required for direct fog cooling systems and this may be an important factor in some site locations when demin water is not easily available or is expensive.

3.4 Direct Inlet Fogging

Direct inlet fogging is a method of cooling where demineralized water is converted into a fog by means of special atomozing nozzles operating at 1000-3000 psi. This fog then provides cooling when it evaporates in the air inlet duct of the gas turbine. This technique allows 100% 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.

A comparison between traditional evaporative cooling and fog intercooling is shown in Table 2.

	TRADITIONAL EVAP COOLING	*HIGH PR INLET FOGGING*
First Cost	x	0.25-.3 x
Need for high quality water	Not requiredPotable water ok	Demin water is a must.
Incremental inlet Delta P	higher	Low- practically nil
Size Foot Print	Large	small
Effectiveness[1]	.85 to 0.9	.97-1.0
Maintenance activities	Higher	Comparatively lower
Aux Power consumption	Requires pump	High pressure pumps needed
Sensitivity to Relative Humidity	High	Lower
Installation down time	Extended outage	Can be done in 2-3 days
Possibility to intercool compressor	Not Possible.	Possible
Risk due to FOD	Minimal potential	Potential exists.

Table 2. Qualitative comparison between traditional Evaporative Cooling and High Pressure Cooling

3.5 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. For instance, the possibility of using fogging systems in conjunction with mechanical inlet refrigeration should be considered, either for fog intercooling (overspray) after the chiller or for supplementing the output of an under-designed chiller. This may not always be intuitive, since evaporative cooling is an adiabatic process[2], which occurs at constant enthalpy. When water is evaporated into an air stream, any reduction in sensible heat is accompanied by an increase in the latent heat of the air stream (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 air stream, the result would yield no thermodynamic advantage.

If, however, the chiller is under-designed 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.

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.

[1] Effectiveness would depend on design features and in the case of fog systems, the location of the fog nozzles and residence time.

[2] This statement is true only if the water supplied is at or below the inlet air temperature.

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. *Statements that one type of evaporative cooling method is "better" should be avoided as the choice is very site specific and depends on economic parameters, degree of capital available etc.*

3.6 Selection Criteria for Inlet Cooling

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 which should be taken into account in doing a study are:

- Climatic Profile (Discussed in detail ahead)
- 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 PPA have to be carefully considered
- Potential impact on existing emission licenses

Economic analysis for inlet cooling systems may be found in Utamura et al (1996), Ondryas (1990), van Der Linden, Searles (1996) and Guinn (1993).

4. CLIMATIC AND PSYCHROMETRIC ASPECTS OF INLET FOGGING

4.1 Modeling of Climatic Data

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 "evap. cooling degree hour" numbers for each hour of the year and allow a turbine operator to make a very detailed and accurate analysis of potential power gain from inlet fogging.

It is most important to avoid the use of a single design point for the design and evaluation of the cooling system and though this is intuitively obvious, it is often done. This point has also been made by Guinn (1993). Very often the site conditions are used for the analysis and this often represents the worst case scenario of temperature and humidity and is really not experienced during normal operation. High relative humidity conditions do not occur with high dry bulb temperatures. Using a single point design temperature can lead to overpriced and overdesigned systems.

A common mistake made by potential users is to take the reported high relative humidity and temperature for a given month and base the design on these. The problem is that the high relative humidity generally occurs time-coincident with the lowest temperature and the lowest relative humidity occurs with the highest temperature. This mistake results in the erroneous conclusion that very little evaporative cooling can be accomplished.

A detailed analysis for a large gas turbine based CCPP is provided in Figure 5 This shows the incremental power attainable by the use of a fogging system for several months.

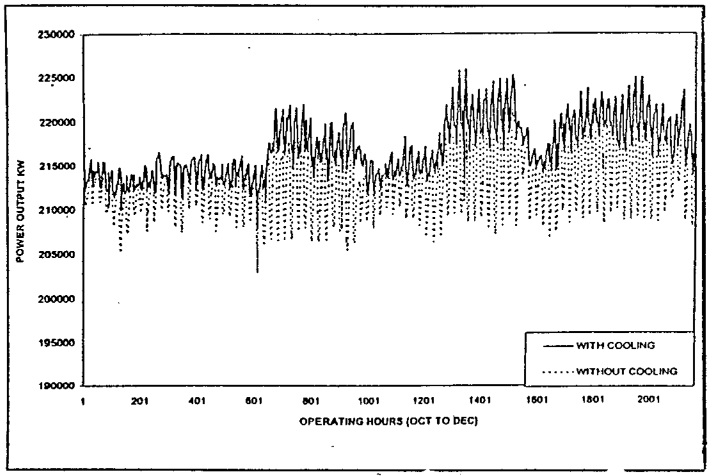

Figure 5. Output changes predicted for a CCPP over time by Fog evaporative cooling.

Table 3 shows a sample calculation that computes degrees F of evaporative cooling potential for a site based on DOE climatic data. This table provides one months worth of data and a summary for the years operation. The total indicated is the total annual degree F – hours of cooling potential by the use of fog. A barchart showing composite data considering all the months of the year from another site is shown in Figure 6.

DB [F]	Hrs	Avg Coincident WB [F]	WB Depression [F]	Evap Cool Potential F-hrs
110-86	0	0	0	0
85	1	71	14	14
84	1	70	14	14
83	0	0	0	0
82	2	72	11	21
81	7	73	8	55
80	9	73	7	61
79	9	73	6	55
78	13	72	6	73
77	10	72	5	53
76	22	69	7	154
75	29	69	6	163
74	37	70	4	140
73	41	68	5	203
72	38	66	6	221
71	62	66	5	281
70	76	66	4	286
			TOTAL:-	**1794**

Table 3. Data taken for a typical month .

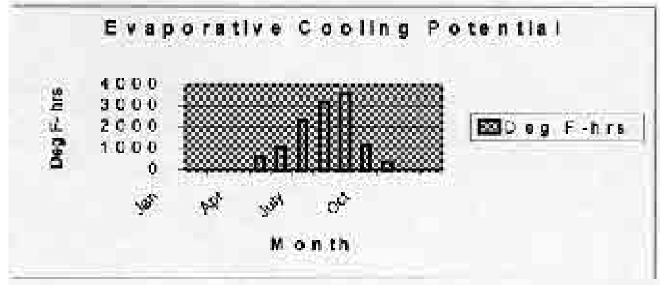

Figure 6. Evaporative Cooling Potential for a year.

4.2 Fog Evaporative Cooling in High Humidity Regions.

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 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 air stream, and are relatively inexpensive to install, they have been successfully applied in areas with very high summer time 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`

4.3 Psychrometrics of Inlet Fogging.

Direct high pressure inlet fogging involves direct evaporative cooling wherein the fog, which consists of billions of particles generated at sizes of 5-20 microns, is injected into the airstream where it evaporates and provides cooling. Tests and application data have shown that this process can be 100% 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 micron while mist particle sizes are 40- 100 micron. 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 , 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.

A psychometric chart is shown in Figure 7 and will be used as an example to determine the cooling requirement for a gas turbine with a mass flow rate of 280 lbs/sec. The following conditions are assumed

Ambient conditions: Houston, TX (ASHRAE climate data, figures not exceeded more than 30 hours per year) – 96°F DB & 77°F WB, RH = 43 %

Step 1: Find the ambient condition on the chart (Start point). Note that the moisture content at this condition is 111 grains (H_2O)/lb (dry air).

Step 2: Assuming 100% 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% RH, "Finish Point"). A wet bulb temperature of 77°F at 100% RH. Note that the moisture content at this condition is 142 grains.

Step 3: Calculate how much moisture has to be added to the air stream to reach the wet bulb temperature. [142-110] = 32 grains/lb.

Step 4: Compute the required water for gas turbine air-mass flow rate of 280 lbs/sec. The water required would be: 280 lb/sec x 32 gr/lb x [1/7005 gr/lb] = 1.279 lb/sec.

Step 5: Convert to gpm, 1.279 X 60/8.345 = 9.2 gpm. This is the water flow required to cool 280 lbs./sec of air by 11°F.

This number would be increased depending on the amount of fog intercooling desired and also on empirical factors based on experience.

5. FOG GENERATION

Fog is generated by the application of high pressure demineralized water between 1000 to 3000 psi to an array of specially designed fog

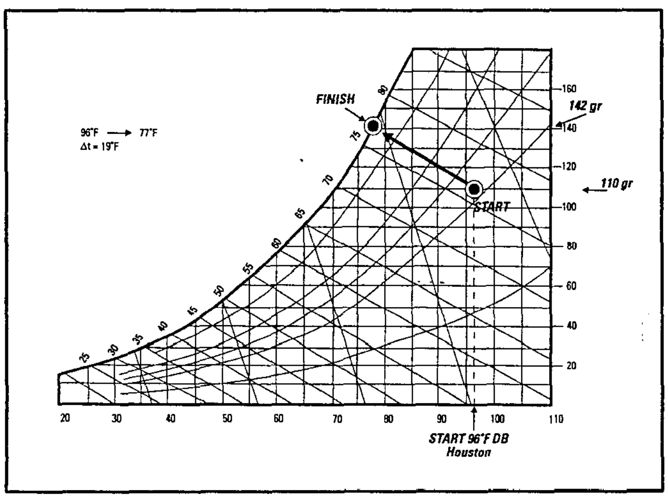

Figure 7. Psychrometric chart showing Fog evaporative effect.

nozzles. A photograph of a typical fog nozzle is shown in Figure 8 and the fog generated is shown in Figure 9. The nozzle is made of 316 SS and consists of a small orifice from 5 – 7 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 micro-fine 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 ten times more surface area than the same amount of water atomized into 100 micron droplets.

Figure 8. Fog Nozzle showing impaction pin.

Figure 9. Generation of High Pressure Fog

Figure 10 shows a typical distribution manifold made up of 1/2 inch SS tubes and a number of fog nozzles. These manifolds are all stainless steel and induce a very low pressure drop ususally less than 0.02 inch WG. Considerable care 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 10. High Pressure Fog Manifold.

5.1 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 11 which is 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% of the droplets generated are below10 microns in size and almost none are greater than 20 microns. As higher atomization pressures are used, droplet size decreases as indicated in Figure 12. 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 %.

5.3 Fogging Equipment- Nozzles, Pumps and Auxiliaries

The key to effective fogging is the design of the fogging nozzle. The use of proper nozzle filters and means to avoid 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.

Typically the pumps used to derive the 2000-3000 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 SS 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

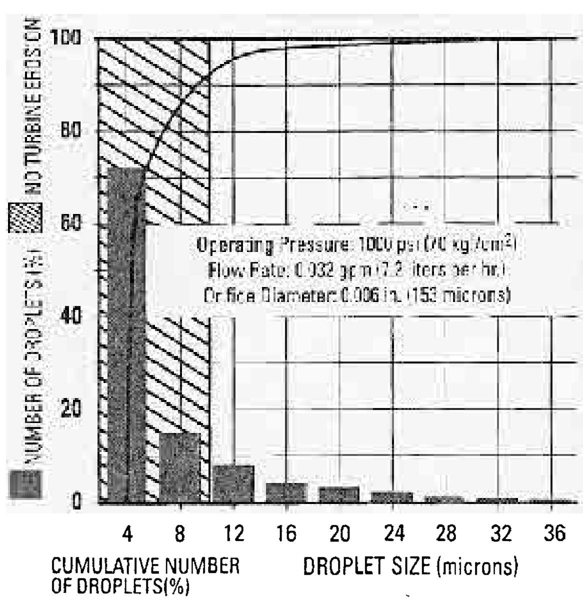

Figure 11. Droplet size distribution

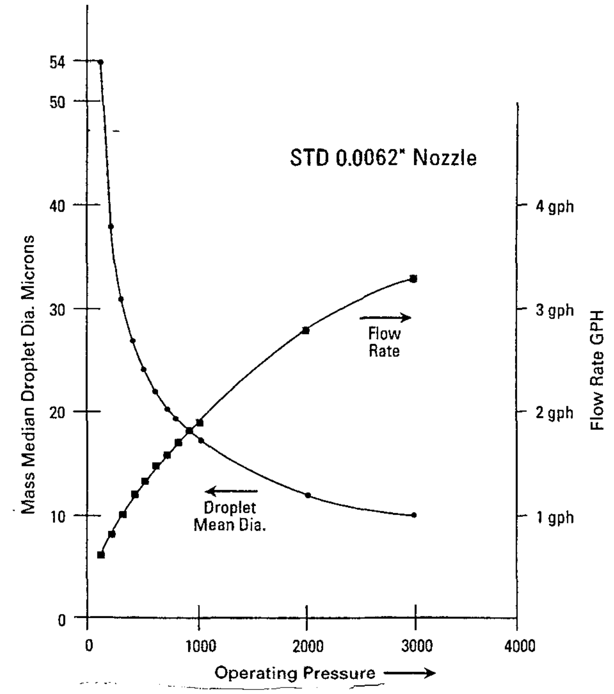

Figure 12. Droplet size and flow rates as a function of pressure

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 fogging skid is shown in Figure 13.

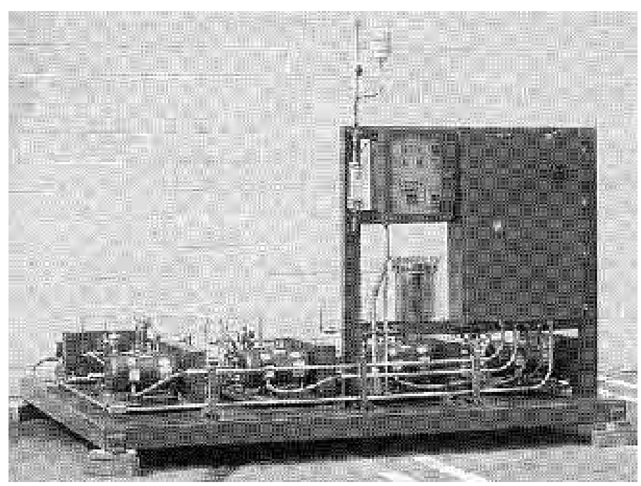

Figure 13. High pressure fogging skid.

5.4 Parameters for Fog Evaporative Cooling

5.4.1 Test curves

A discussion of the evaporative efficiency factors based on rig tests[3] is presented here. The test rig consisted of a fog pump and nozzles that placed fog in a duct. A fog filter was provided that allowed the spray distance to be varied. After the fog filter, there was no fog and the leaving air relative humidity was measured. These tests were conducted using a 1000 psi operating pressure and airflow velocity of 500 ft/min. The graph shown in Figure 14 shows the fog nozzle spray distance on the x-axis and the percent fog water evaporated on the y axis. The two curves show the 50% RH leaving air situation and the 100 % RH leaving air situation . The graph shown in Figure 15 shows operating pressure variation on the x axis and the percent of fog waer evaporated on the y axis. Curves for spray distances of 1 ft, 3 ft and 5 ft distances are provided. As one would expect, at a given pressure, the percent of fog water evaporated (i.e., the fog evaporative efficiency) is a function of the distance and also the pressure (which in turn determines droplet size).

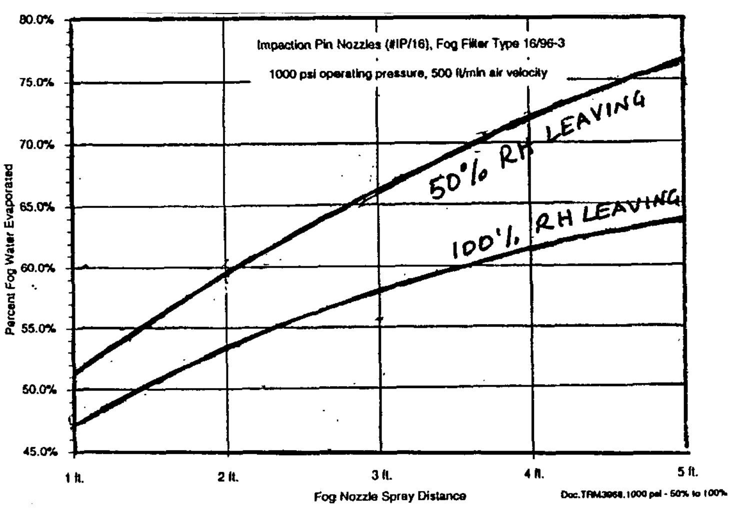

Figure 14. Evaporative Efficiency Curve for Varying Spray Distances, 1000 psi Nozzle Pressure.

5.4.2 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. It the critical Weber number for breakup is sufficiently exceeded, the breakup occurs within a very short time. The time to rupture is given by:

$$T_{rupture} = (0.3 - 1)\ \frac{\pi}{4} \sqrt{\frac{\rho_f D^3}{\sigma}} \qquad (6)$$

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 rupture time drops.

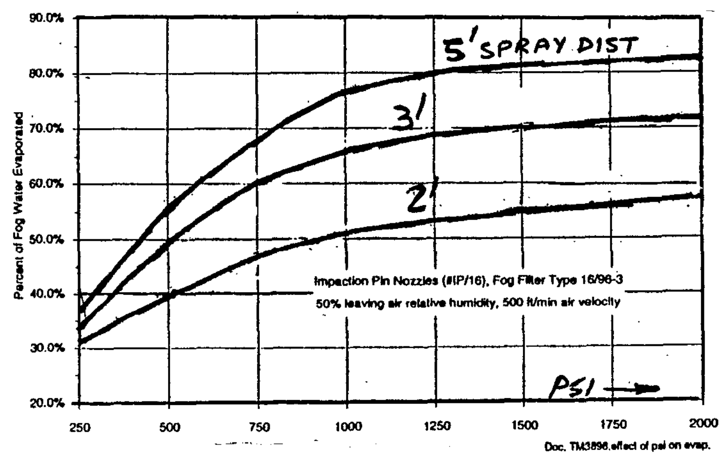

Figure 15. Test Data – effect of Nozzle operating pressure on evaporative efficiency. Pressure varied upto 2000 psi.

5.4.3 Droplet Evaporation Times

Figure 16 (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 in the event of fog ingestion into the compressor,.would exceed the residence time in the compressor[4]. Further with the high pressure ratio units in operation today the gas temperatures are considerably high implying more rapid heat transfer to the droplet.

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 17. (Tsuchuia and Murthy, 1980).

[3] These rig tests were conducted at Mee Industries. Graphs courtesy of Mee Industries

[4] typical residence time of an air particle in an industrial gas turbine compressor may be 10 ms.

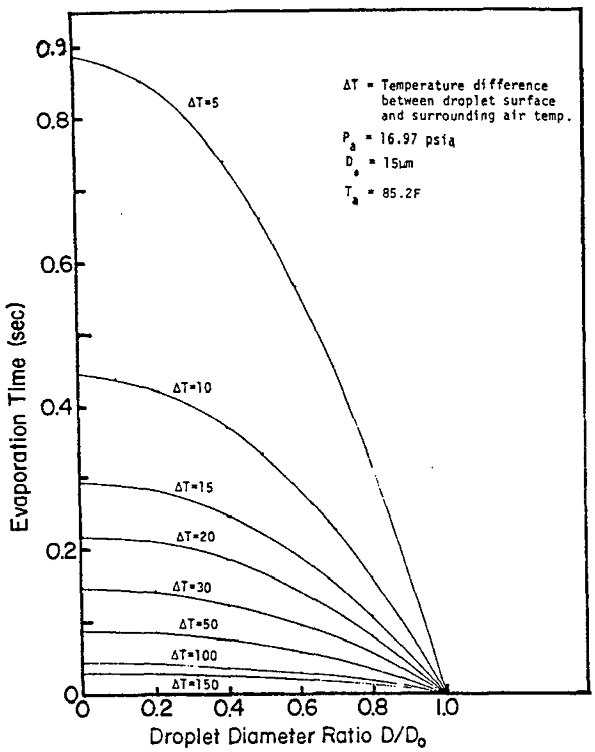

Figure 16. Droplet evaporation time with negligible relative velocity effect- 15 micron diameter. (Tsuchyia and Murthy, 1980)

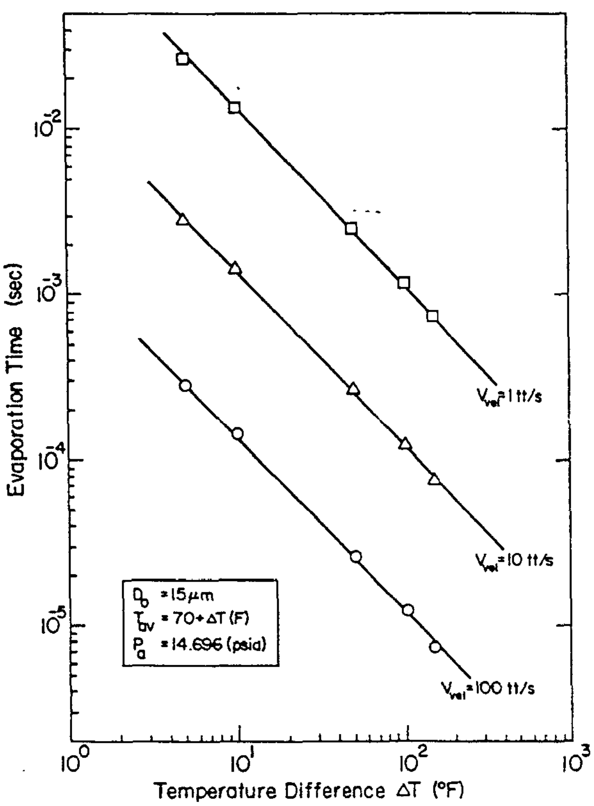

Figure 17. Evaporation time with relative velocity effect. (Tsuchyia and Murthy, 1980)

6. FOG INTERCOOLING OF GAS TURBINES

6.1 Traditional Intercooling

Traditional intercooled cycles 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. Intercooling has been covered by Tanaka and Ushiyama (1970).

The cost of the intercooler and the complexity involved is 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 LM 6000 Sprint cycle where direct intercooling is done between the low pressure and high pressure compressor spools (Johnson, 1998).

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.

6.2 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 into the compressor has been covered in detail by Utamura et al, (1998), 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 et al (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 of 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.

[1] 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.

[2] 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. Note that the boiling point of the water is a function of the pressure and therefore increases as the gas heats up during compression.

There has been several excellent papers published by Utamura on the subject. Simulation results for a 150 MW, 14.5:1 pressure ratio machine utilizing direct spray intercooling shows that the turbine output can be boosted by 23% by a spray flow rate of 2.3% with an associated increase in the thermal efficiency.

Sexton et al (1998) has modeled a naval LM-2500 engine and concluded that a 34% power boost could be obtained by injecting 24 gpm of water into the compressor inlet along with a drop in specific fuel consumption.

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 Fortin and Bardon (1983).

6.3 Compressor Performance with Fog Intercooling.

Hill (1963) has conducted an excellent study of the aerothermodynamic effects of coolant injection and has derived several useful equations. The compression process can be described as:

$$\frac{P}{P_2} = \left(\frac{T}{T_2}\right)^{\frac{\gamma}{\gamma-1} + \frac{L}{R}\cdot\frac{dW}{dT}} \qquad (7)$$

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 18 (Hill 1963) indicates the relative ideal compression work for pressure ratios of 6 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

$\theta = T_{in}/520$

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

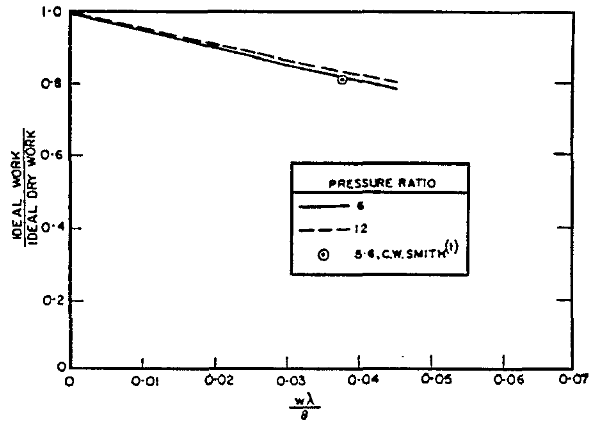

Figure 18. Comparison of Ideal Wet Compression and Dry Compressor Work (Hill, 1963).

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

$$\alpha_t = \frac{30\,\eta_p \log_e \frac{P_3^*}{P_2}}{\left[\left(\frac{T_3^*}{T_2}\right)-1\right](1-\psi^*)}\cdot\frac{w\lambda}{\theta} \qquad (8)$$

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
$\theta = T_{in}/520$
T_3^* = Dry Compressor Discharge Temperature
T_2 = Compresser Inlet temperature
ψ = Average Work Coeff = $1 - \varphi^* (\tan\alpha_1 - \tan\alpha_2)$, α_1 and α_2 are blade angles.
$\psi^* = \Delta h/U_2^{\,2}$
$\varphi = Cx/U$

6.4 Droplet Acceleration and droplet behavior in the Compressor

Studies by several researchers have indicated that at lower droplet sizes erosion will not be a problem in the compressor. Recent CFD studies conducted by Utamura et al (1999) have indicated that droplets 10 to 20 microns in diameter would follow the airpath. Larger droplets would as was shown by Hill (1963) strike the blades.

7. OTHER POSSIBILITIES OF FOG COOLING IN POWER PLANT APPLICATIONS

In any project involving power augmentation, the generator curves must be examined to ensure that the generator has the capability to match the power augmentation. This is especially important on air cooled generators which are now being increasingly used. Similarly, lube oil coolers should be evaluated. In most cases however, the existing capacity will be more than adequate as there is some overdesign in the first place. Fog evaporative cooling can be used to improve the performance of these heat exchangers.

8. CLOSURE

This paper has described the use of high pressure inlet fogging to augment gas turbine power. Direct high pressure evaporative cooling has been compared with traditional evaporative coolers. Fog evaporative cooling which allows attainment of close to the wet bulb temperature and 100% RH at the inlet. Further if desired, an additional boost can be attained by fog intercooling by allowing micron size particles to enter the compressor, thus reducing gas turbine compressor work. As with any technology its use must be evaluated based on the turbine characteristics, climatic conditions, power augmentation expectations and project economics. Part B of this paper (Meher-Homji and Mee, 2000) covers some specific practical aspects relating to inlet fogging systems and details some O&M issues.

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