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INLET FOGGING OF GAS TURBINE ENGINES - PART A: FOG DROPLET THERMODYNAMICS, HEAT TRANSFER AND PRACTICAL CONSIDERATIONS

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ABSTRACT

The inlet fogging of gas turbine engines for power augmentation has seen increasing application over the past decade yet not a single technical paper treating the physics and engineering of the fogging process, droplet size measurement, droplet kinetics, or the duct behavior of droplets, from a gas turbine perspective, is available. This paper provides the results of extensive experimental and theoretical studies conducted over several years coupled with practical aspects learned in the implementation of nearly 500 inlet fogging systems on gas turbines ranging in power from 5 to 250 MW. Part A of the paper covers the underlying theory of droplet thermodynamics and heat transfer, and provides several practical pointers relating to the implementation and application of inlet fogging to gas turbine engines.

NOMENCLATURE

Cp_a	Specific heat of air ($J.kg^{-1}.K^{-1}$)
Cp_d	Specific heat of water ($J.kg^{-1}.K^{-1}$)
D_d	Droplet diameter (m)
Df_a	Mass coefficient of diffusion for air ($m^2.s^{-1}$)
Df_{mass}	Coefficient of mass diffusion ($m.s^{-1}$)
DBT	Dry Bulb Temperature (°C)
ECDH	Equivalent Cooling Degree Hours (°C-hr)
g	Gravitational acceleration ($m.s^{-2}$)
Gr_m	Mass Grashof number
Gr_t	Heat Grashof number
hcv	Coefficient of thermal convective exchange ($W.m^{-1}.K^{-1}$)
Lv	Latent heat of vaporization of water ($J.Kg^{-1}$)
M	Molecular weight ($kg.mole^{-1}$)
m_d	Mass of the droplet (kg)
Nu	Nusselt number
P_a	Atmospheric pressure (P_a)
Pr	Prandtl number
Pvd	Vapor pressure at the interface droplet-air (Pa)
Pvap	Partial pressure of the vapor (Pa)
Re	Reynolds number
R	Universal gas constant (8.31 $J.mol^{-1}.K^{-1}$)
RA	Active Radius
RH	Relative Humidity (%)
S_d	Droplet surface (m^2)
Sc	Schmidt number
Sh	Sherwood number
T	Absolute Temperature (K)
T_a	Temperature of air (K)
T_d	Droplet Temperature (K)
We	Weber number
WBT	Wet Bulb Temperature
ΔC_{mass}	Different of mass concentrations of water vapor at the interface droplet-air and the air ($kg.m^{-3}$)
Δm_d	Mass variation of the droplet (kg)
V_{rel}	Droplet relative velocity ($m.s^{-1}$)
Δt	Time Step (s)

GREEK ALPHABET

β	Coefficient of thermal dilatation (K^{-1})
Φ_{conv}	Flux exchanged by thermal convection (W)
Φ_{lat}	Flux exchanged by latent heat (W)
γ_w	Water Surface tension ($N.m^{-1}$)
λa	Thermal conductivity, air ($W.m^{-1}.K^{-1}$)
μa	Dynamic viscosity, air ($kg.m^{-1}.s^{-1}$)
ν_a	Kinetic viscosity, air ($m^2.s^{-1}$)
ρ_a	Density, air ($kg.m^{-3}$)

SUBSCRIPT

a	air
d	droplet
t	time

INTRODUCTION AND BACKGROUND

Gas Turbine output is a strong function of ambient air temperature, with power output dropping by 0.54-0.90 % for every 1°C rise in ambient temperature. This loss in output presents a significant problem to utilities, cogenerators and merchant power plants when electric demands are high during the hot months. One way to counter this drop in output is to cool the inlet air.

Over the past decade and especially over the past five years, the application of inlet fogging for the power augmentation of gas turbines has become increasingly popular. It is estimated that approximately 700 gas turbines have been fitted with fogging systems at this time including many modern F class gas turbines.

In the rapidly deregulating power generation market, 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 peaks). 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 need to augment power during high demand periods. Coupled with the need for power augmentation is the requirement to accomplish this with a minimum capital cost for the incremental power generated. High-pressure inlet fogging fits this niche. The concept is a simple one in which a direct evaporation effect is derived by the use of fog generated by high-pressure pumps and atomizing nozzles. The fog evaporates in the inlet duct and cools the air down to the wet bulb temperature.

Fog intercooling[1], which has been applied from the early days of gas turbine and jet engine technology is a technique that consists of spraying more fog than will evaporate under the given ambient temperature and humidity conditions so that liquid water droplets enter the compressor. The desired quantum of unevaporated fog is carried with the air stream into the compressor where it evaporates and produces an intercooling effect. The resulting reduction in the work of compression can result in a significant additional power boost and an improved heat rate.

A review of the technology may be found in Meher-Homji and Mee [1,2]. A detailed parametric study of inlet fogging in the context of gas turbine design parameters has been conducted by Bhargava and Meher-Homji [3]. Early papers on fog intercooling and wet compression started to appear in the late 1940s including Kleinschmidt[2] [4], and Wilcox & Trout [5]. Other references include Hill [6] Arsen'ev and Berkovich [7], Nolan and Twombly[3] [8] and Utamura et al [9].

To this point however, <u>not a single technical paper exists comprehensively covering the physics and engineering of the fogging process, droplet measurement methods, droplet kinetics, and the duct behavior of droplets, from a gas turbine perspective.</u>

This paper provides the results of extensive experimental and theoretical studies conducted over several years, coupled with practical aspects learned in the design and implementation of nearly 500 inlet fogging systems on gas turbines ranging from 5 to 250 MW. Part A of the paper covers the underlying theory of droplet thermodynamics, and heat transfer and provides several practical pointers relating to the application of inlet fogging to gas turbine engines. Parts B covers details on nozzle technology and droplet measurement, putting the topic in the context of gas turbine fogging. Finally, Part C covers fog behavior in inlet ducts and provides results of several wind tunnel experiments.

COMPONENTS OF A HIGH PRESSURE FOGGING SYSTEM

A typical high pressure fogging system consists of:

- A high-pressure pump skid with associated pumps and controls as shown in Figure 1.
- A set of fog nozzles located in the intake duct after the filters as shown in Figure 2. Each impaction pin nozzle in the array produces billions of droplets per second creating a fog as shown in Figure 3.
- A PLC based control system.

Figure 1. Typical Gas Turbine Inlet Fogging Skid

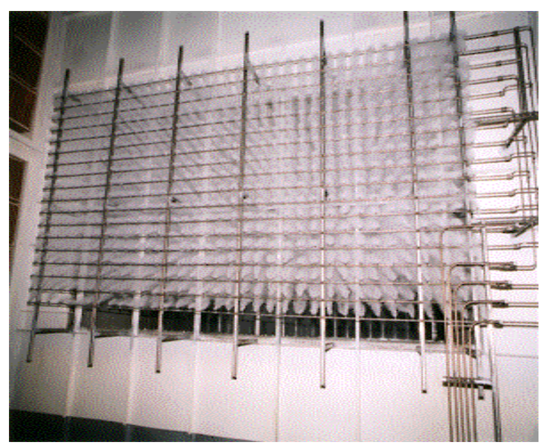

Figure 2. Fog Nozzle Manifold operating in the inlet duct of a GE-7EA gas turbine

[1] Also known as overspray or wet compression.
[2] This paper coined the term "wet compression".
[3] Nolan and Twombly covered an application of fog intercooling on GE Frame 5 engines.

Figure 3. Fog from an impaction pin-fogging nozzle operating at 2000 psig (138 barg)

FUNDAMENTAL RESEARCH EFFORTS CONDUCTED

In spite of fogging being a popular power augmentation strategy there is a scarcity of solid technical data covering the theoretical and practical aspects of the fogging process. This paper, along with Parts B and C, will be the first time that a solid set of experimental and empirical data are provided with information that can help users understand the issues and complexities involved in the fogging process. The paper is based on several years of fundamental and applied research programs conducted in the area which include:

- Droplet kinetics and thermodynamics.
- Droplet collision, coalescence and shattering dynamics.
- Droplet/airflow interface dynamics.
- Computational Fluid Dynamics (CFD) studies of gas turbine inlet ducts.
- Theoretical and experimental studies of droplet trajectories both in still air and in a moving airflow.
- Optimization studies relating to nozzle array placement in gas turbine inlet air ducts.
- Comparative experimental analysis of nozzle performance and standardization approaches for droplet measurement for gas turbine applications.
- Studies of drains, silencer and trash screen interactions.
- Development of design tools for ensuring the structural integrity of the fogging nozzle manifolds in intake ducts.

Much of the work presented here has been validated by the use of the wind tunnel that is shown in Figure 4. In additional to experimental verifications, considerable field experience and lessons learned are also embedded in this paper. The authors hope that turbine operators will find the information in this paper helpful when specifying equipment and evaluating specific fog system designs.

Figure 4. Experimental Wind Tunnel (10.5 m long, 25 m/sec) used to study droplet kinetics and thermodynamics. This wind tunnel provides dynamic similarity to gas turbine inlet ducts

ISSUES OF IMPORTANCE IN INLET FOGGING

From a gas turbine operator's perspective, there are several issues that are of importance when using, or considering the use of, inlet fogging systems. These include:

- An understanding of what actually happens with the evaporation of water in the inlet duct and droplet thermodynamics that result in the cooling of the inlet air.
- An understanding of the physics of droplet sizing, and the different terminology used, and the significance to gas turbine operation. This is of particular importance because of the wide range of diameter characterizations used. This is dealt with in this paper briefly; part B will cover the topic in greater detail.
- An understanding of the different nozzle types offered and the proper testing and evaluation of inlet air fogging nozzles. With the growing popularity of gas turbine inlet air fogging it is becoming increasingly obvious that a standard for measuring droplet sizes generated by fog systems is needed so that gas turbine operators can make informed comparisons of different systems. Fog system manufacturers make various claims based on different testing methods and there is little agreement from one manufacturer to another. The result is confusing and contradictory information that makes it impossible for the gas turbine user to understand the relative performance of competing systems.
- The behavior of fog in gas turbine inlet ducts in terms of rates of evaporation, velocity profiles, and uniformity of distribution.
- A close understanding of the climatic conditions and the coincident wet bulb and dry bulb temperature history for a particular location. While this may seem to be a trivial topic, there is very little data available in an easy to use form to enable gas turbine users to estimate the amount of power boost that can be attained through the year, or during typical peaking periods, at a particular site. Chaker et al [10,11] have provided comprehensive data of evaporative cooling potential in numerous locations in the USA and internationally, using the concept of Evaporative Cooling Degree Hours (ECDH). Tabulations provided in these papers allow users to quickly

estimate the cooling potential available at their site and to estimate the power boost available for their turbines.

- Numerous practical aspects relating to gas turbine operation including issues such as droplet collision and agglomeration, duct wetting, duct drainage, icing considerations, avoidance of Foreign Object Damage (FOD), corrosion considerations in the duct, compressor inlet distortion (uneven temperature and pressure profiles at the compressor inlet which can cause blade vibration in the axial-flow compressor), etc. This set of papers covers these important practical issues.

ISSUES AND COMPLEXITIES RELATING TO DROPLET BEHAVIOR

The study of the sizing and the thermo-kinetics behavior of fog droplets is a complex subject. Care has to be taken when studying the continuous changes that occur in both space and time with the billions of droplets that are emitted every second by each nozzle.

Small changes in any test parameter, the measurement methodology, environmental conditions, and the measurement device itself, can result in significant differences in the characterization of nozzles.

The reason for injecting water droplets is to increase the relative humidity of the airflow by evaporation of the droplets, which in turn leads to a decrease in the temperature resulting in an increase in the air mass flow[4] and more power for the gas turbine. Compressor work is also reduced during inlet fogging as work of compression is proportional to compressor inlet temperature, this also results in more output power. While this may intuitively seem to be a simple matter, there are several interacting factors that define the success and efficiency of the process.

An analysis of droplet sizes produced by inlet air fogging nozzles must take many factors into account and it is therefore important to have a complete understanding of the atomization process. Results can vary dramatically depending on several key parameters, such as the velocity of the air stream in which the nozzle is located, the properties of the water itself (including temperature, dissolved mineral content, etc.), the pressure applied on the liquid, the geometry of the nozzle, and consequently the spray angle and even the location in the plume where the measurement is taken. Some of these factors are highly critical, as will be discussed ahead.

Droplet size itself is a critical factor for the inlet air fogging process. The fog droplets quickly take the velocity of the airflow (within a few centimeters from the nozzle orifice) and they typically spend just one to two seconds in the airflow before they reach the compressor inlet. The droplets must be small enough that they evaporate, or substantially evaporate, before reaching the compressor inlet. There are many potential drawbacks to using large droplets for the inlet fogging process. These include:

- Large droplets may not completely evaporate leading to under-saturated air at the compressor inlet (i.e. less than 100% evaporative efficiency).
- Large droplets are more likely to distress compressor blades and/or blade coatings.
- Large droplets are more likely to be removed from the air stream by gravity or by impaction on obstructions in the inlet ducts, such as silencers, trash-screens, etc. (Small droplets more readily follow the airflow around obstructions.).
- More fallout can lead to excessive pooling of water on the duct floor, thus requiring more extensive drainage systems and leading to more potential for corrosion of the duct surfaces.
- Excessive pooling at the compressor inlet can result in large slugs or sheets of water being suctioned off the duct floor and ingested by the compressor. This can lead to blade erosion and/or coating wear and, in the most severe cases, may cause compressor blade tip deformation.

Smaller droplets, in the range of 5 to 15 microns, have many advantages for gas turbine inlet air fogging applications. Experience has shown that droplets in this size range do not cause erosion of compressor blading or wear of blade coatings (provided the nozzles are installed so that excessive pooling is avoided). Small droplets are also more likely to follow the airflow around obstructions in the duct and less likely to fall out (i.e. they cause less pooling in the duct) and they provide more evaporative surface area. Evaporation rate is a strong function of the exposed surface area because evaporation occurs only at the water/air interface.

To put it in perspective, a ten-micron droplet has only about 13% the mass of a 20-micron droplet and a given mass of water divided into 10-micron droplets results in four times more surface area than the same mass of water divided into 20-micron droplets. Obviously a ten-micron fog will evaporate much faster, and is much less likely to impact on obstructions or fall out on the duct floor, than a 20-micron fog.

There are several factors that influence droplet size distribution:

- Type of nozzle—impaction-pin and swirl-jet nozzles are the types currently offered by GT inlet air fogging suppliers. These are described in more detail ahead.
- Operating pressure—typically ranges from 1000 to 3000 psig (70 to 210 barg).
- Nozzle capacity—typically ranges from about 2 gph to about 5 gph (7 to 18 l/h).

Nozzle flow rate is an important issue. Nozzles with relatively high flow rates have some advantages (fewer nozzles to install and service, etc.) but also have disadvantages (fewer points of emission can mean poor distribution of fog in the inlet air stream, resulting in poor air/fog mixing, uneven cooling or incomplete cooling of the air stream). Also, other factors being the same, a larger flow rate nozzle will, as a general rule, produce larger droplets than a small flow nozzle. An exception to this general rule is discussed in Part B of this paper—swirl-jet nozzles have higher internal frictional losses than impaction-pin nozzles so that the latter can produce smaller droplets at slightly higher flow rates.

DROPLET DIAMETER NOMENCLATURE

The spray from any water-atomizing nozzle contains a range of droplet sizes and many statistical techniques are employed to characterize sprays. The more obvious, but least useful, is the average droplet size. Taking the arithmetic average of all the droplets found in a spray leads to a deceptively small diameter number because, as discussed above, the larger droplets offer much less surface area and carry far more mass.

[4] By increasing the air density

Therefore, several statistical methods have been developed for defining the range of droplet diameters found in a spray. These statistical numbers give a fictional droplet diameter, which "characterizes" the spray in a meaningful way. Most often these diameters are given in microns.

Two of these methods are most applicable to gas turbine inlet fogging applications. They are:

- **Sauter Mean Diameter (SMD or D32)** is the diameter of a hypothetical droplet whose ratio of volume to surface area is equal to that of the entire spray. Since it deals with surface area, Sauter mean diameter is a good way to describe a spray that is used for processes involving evaporation. To enhance droplet evaporation one has to maximize the active surface area and minimize the internal volume of the droplet, thus the lower the Sauter Mean Diameter, the more rapid the evaporation process.

- **Dv90** is a diameter for which 90% of the water volume in the spray is less than or equal to. A small value of this number indicates that a very small number of larger droplets are present. A small Dv90 minimizes the potential for impaction on obstructions and droplet fallout due to gravity (both of which reduce water pooling on the duct floor) and reduces the potential for compressor blade distress.

DROPLET TESTING STANDARDS

As mentioned above, there are no existing testing standards for fog sprays in the gas turbine inlet air fogging industry. Different fog system suppliers measure droplets under different conditions, at different locations in the spray plume and with various kinds of droplet measuring instruments.

This has resulted in a situation in which fogging system users find it impossible to meaningfully compare competing nozzle designs. Statements relating to droplet sizes without a solid analytical description of what specifically was measured and how readings were taken make no engineering sense and must be qualified carefully before a gas turbine user can make an intelligent decision.

For example, if a fog system supplier reports droplet sizes that were taken at a position 2 to 3 inches (50 to 75 mm) in front of the nozzle orifice in a high-velocity air stream, this results in extremely misleading information because it ignores the significant part of the spray plume. Furthermore, all atomizing nozzles currently in use for gas turbine inlet fogging applications have a conical spray plume. The jet-action of the spray creates a low-pressure area in the middle of the conical plume and the smallest droplets are readily pulled into this area, because they more quickly follow the airflow, while all of the larger droplets remain in the edge of the plume, outside of the area where measurements were taken. Even if the nozzle is tested in a high-velocity airflow, most of the large droplets never enter the measurement area.

For these reasons, it is important that an industry standard be developed to provide a ***uniform and repeatable approach that makes meaningful comparisons possible.*** Part B of the paper covers one such proposed standard.

FACTORS AFFECTING DROPLET SIZE MEASUREMENTS

Several key factors have a significant impact on droplet size measurements:

- Airflow velocity during the measurement.
- Location in the nozzle spray plume where the measurements are taken (i.e. center or edge).
- Ambient temperature and humidity conditions around the nozzle.
- The difference in liquid characteristics may also account for differences in size distribution but, since inlet fogging always deals with ambient temperature demineralized water, differences in liquid surface tension and viscosity can be largely ignored.

To illustrate the effect of measurement location and flow velocity on droplet size, diameter curves of an impaction-pin nozzle made by our company, for gas turbine inlet fogging applications is shown in Figure 5. This data was developed from measurements taken in the aforementioned wind tunnel and provides data for a variety of measurement locations over a range of velocities. This figure clearly indicates the problem faced by a gas turbine user when he is told, for example, that a nozzle produces a 20 microns droplet without also being told how and where and under what conditions the measurements were taken.

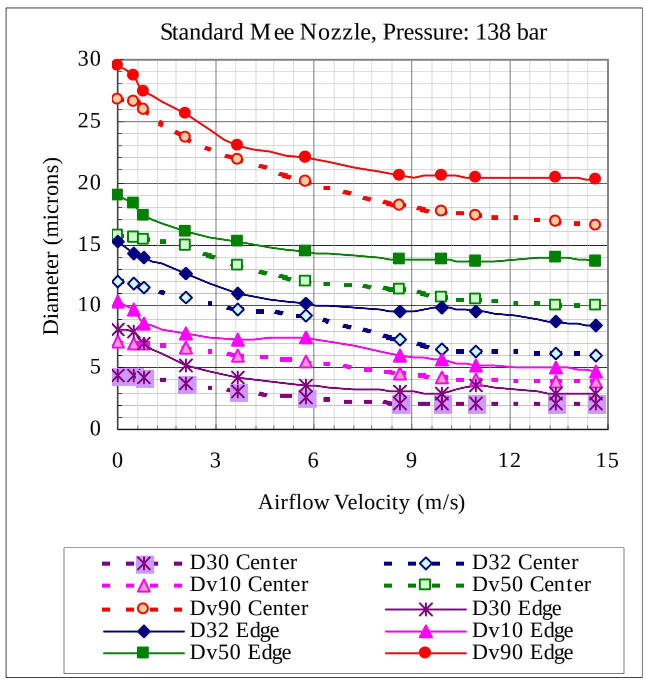

Figure 5. Effect of air velocity, measurement locations and diameter definition on droplet size

Note for instance the red curve at the top of the chart in Figure 5. The Dv90 diameter, with zero airflow velocity, measured at the edge of the plume is 30 microns, while the same Dv90 diameter for the same nozzle operating at the same pressure is about 16 microns when measured with high airflow velocity in the middle of the plume (second curve from the top, right side of the chart) velocity of 9 m/s (~1800 ft/min). It's easy to see how someone trying to evaluate one nozzle against another could reach a totally false conclusion unless they have an understanding of the subject and the information on how, were and under what conditions the measurements were taken.

Measurement location and air velocity alone can account for a very large variation in the reported diameter of a droplet. The reason for the decease in droplet size at higher airflow rates has to do with the fact that airflow moves the smaller droplets out of the path of the larger droplets and, thus, reduces collision and agglomeration. This is discussed in more detail in Part B of the paper.

DROPLET HEAT AND MASS TRANSFER MODEL

Prior to the study of the complex properties and behavior of the whole mass of fog droplets emitted from the nozzle, it is helpful to study the properties and behavior of individual droplets, and then extend this study to the interaction between a single droplet and the other droplets and the interaction with the carrying phase (air). The study of an isolated droplet is first treated using the classical assumption of spherical symmetry for both the liquid droplet and the surrounding air. The behavior of a single droplet injected into the airflow will be analyzed then a study of the effect of the velocity on droplet thermodynamic and trajectory will be done.

The study of big number of droplets is currently being analyzed using the Lagrange/Euler approach (Computational Fluid Dynamics (CFD) Analysis) and will be reported on in a later paper.

Model for an Isolated Droplet

In order to cover the underlying basis of the model developed ahead, the basic theory and equations used are provided here. Heat and mass transfer occur at the interface of the droplet and the surrounding air, mainly by convection and diffusion. This transfer depends on the thermal characteristics of the air (hygrometry, total pressure and temperature) and the droplet (volume, liquid characteristics and temperature).

A model has been developed below to study the evaporation process of the droplets. The main object of this model is to show the lifetime of a droplet in a limited volume as it reaches saturation within this volume. Details of the model can be found in Chaker [12].

In this model we take into account the fact that the thermo-kinetic behavior of the water droplets, which represents the dispersed phase will be influenced by that of the air, which represents the continuous phase.

The heat quantity stored in a droplet between instant t and instant t+Δt is given by

$$\Phi = m_d Cp_d \frac{\left(T_{d_{(t+\Delta t)}} - T_{d_t}\right)}{\Delta t} \tag{1}$$

Therefore, the droplet temperature at the instant t+Δt is

$$T_{d_{(t+\Delta t)}} = T_{d_t} + \frac{\Delta t \times \Phi}{m_d Cp_d} \tag{2}$$

The thermal equilibrium between the droplet and the surrounding air occurs when the quantity of heat received by the droplet Φ, in a given time, is equal to the flux exchanged by convection and latent heat

$$\Phi = \Phi_{conv} + \Phi_{lat} \tag{3}$$

Convective exchange: The rate exchanged by thermal convection between the droplet and the air is given by

$$\Phi_{conv} = h_{cv} \times S_d \times \left(T_a - T_d\right) \tag{4}$$

h_{cv} is the coefficient of thermal convective exchange ($w.m^{-2}.K^{-1}$), and is derived from the Nusselt number

$$Nu = \frac{h_{cv} D_d}{\lambda_a} \tag{5}$$

λ_a is the thermal conductivity of the air ($W.m^{-1}.K^{-1}$). Using a linear regression of the data shown in the Handbook of Chemistry [13], for T_a varying between (253°K < T_a < 373°K), we find

$$\lambda_a = (46.766 + 0.7143\,T_a) \times 10^{-4} \tag{6}$$

For natural convection, the Nusselt number is function of the thermal Grashof and Prandtl numbers and is given by the relation [14]

$$Nu = 2 + 0.6.Gr_t^{0.25} \times Pr^{0.33} \tag{7}$$

where, Pr is the Prandtl number, given by

$$Pr = \frac{\mu_a \times Cp_d}{\lambda_a} \tag{8}$$

μ_a is the dynamic viscosity of the air ($kg.m^{-1}.s^{-1}$). For 273K<T_a<373K it is given by [13]

$$\mu_a = \left(0.004823 \times T_a + 0.3976\right) \times 10^{-5} \tag{9}$$

The thermal Grashof number Gr_t is given by:

$$Gr_t = \frac{\rho_a^2 .g.\beta.(T_a - T_d).(D_d)^3}{\mu_a^2} \tag{10}$$

Where, g is the acceleration of gravity (9.81 $m.s^{-2}$).

β is the thermal dilatation coefficient, and (for a perfect gas) is equal to

$$\beta = \frac{1}{\rho_a}\left(\frac{\partial \rho_a}{\partial T}\right)_{C,p} = \frac{1}{T_a} \tag{11}$$

By combining the equations (5, 7, 10, 11), the coefficient of convective exchange becomes

$$h_{cv} = \frac{\lambda_a}{D_d} . \left[2 + 0.6 \times \left(\frac{\rho_a^2 .g.(T_a - T_d).(D_d)^3}{\mu_a^2 .T_a} \right)^{0.25} \left(\frac{\mu_a}{\rho_a Df_a} \right)^{0.33} \right] \tag{12}$$

Latent heat exchange: The latent heat exchange is equal to

$$\Phi_{lat} = \frac{\Delta m_d \, L_v}{\Delta t} \tag{13}$$

L_v is the latent heat of water vapor ($J.kg^{-1}$) given by

$$L_v = 1000\,(2498 - 2.413 \times T_d) \tag{14}$$

The mass variation of the droplet (Δm_d) which occurs due to the flux exchanged by evaporation between the droplet surface and the surrounding air during the interval time Δt is given by

$$\Delta m_d = -\Delta t \times S_d \times \phi_{evap} \tag{15}$$

where, S_d is the exchange surface of the droplet with the surrounding air (m^2).

We assume that the droplet and its environment make an isolated system with their temperatures being uniform and designated as T_d and T_a respectively [15]. When the ambient air is not saturated with water vapor, the droplet evaporates naturally.

The mass flux may then be written as

$$\phi_{evap} = Df_{mass} .\Delta C_{mass} \tag{16}$$

where, Df_{mass} is the coefficient of mass diffusion ($m.s^{-1}$).

ΔC_{mass} is the difference of mass concentrations of water vapor between the droplet-air interface and the air (free stream) ($kg.m^{-3}$) and is given by

$$\Delta C_{mass} = C_{mass_d} - C_{mass_a} \tag{17}$$

By assuming the air around the droplet is a perfect gas, the concentration C_{mass_i} may be written as

$$C_{mass_i} = \frac{M.p_i}{R.T_i} \quad (18)$$

Therefore, ΔC_{mass} may be written as

$$\Delta C_{mass} = \frac{M}{R}.\left(\frac{p_{v_d}}{T_d} - \frac{p_{vap}}{T_a}\right) \quad (19)$$

where M is the molar mass (kg.mole^{-1}), and R is the universal gas constant (8.32 j.mole^{-1}.K^{-1}) and p_{vap} and p_{v_d} are the vapor pressure of the air and the interface air-droplet (Pa) respectively.

The exchange coefficient by mass diffusion is given by the relation

$$Df_{mass} = \frac{Sh.Df_a}{D_d} \quad (20)$$

Df_a is the mass coefficient of diffusion, given by

$$Df_a = 2{,}26.10^{-5}.\left(\frac{101325}{P_a}\right)\left(\frac{T_a}{273.15}\right) \quad (21)$$

Where, P_a is the total pressure (Pa) and T_a is the temperature of the air (K).

ρ_a is the density of the humid air (kg.m^{-3}) and for $T_a \le 373$°K. It is given by the relation

$$\rho_a = \left(\frac{353.15}{T_a}\right).\left[1 - 0.377 \times \frac{P_{vap}}{P_a}\right] \quad (22)$$

P_{vap} is the partial pressure of the water vapor. P_a is the total pressure of the air, which depends on the air temperature. P_{vap} reaches its maximum (saturation) because the quantity of the vapor in the air cannot be higher than the maximum value corresponding to the equilibrium between the water at the vapor phase and the liquid phase. By considering air to be a perfect gas, it is given by the following formula [16]

$$P_a = e^{77.3417 - 8.2\times \mathrm{Ln}(T_a - 0.15) + 5.7114\times 10^{-3}\times T_a - \frac{7235.46}{T_a - 0.15}} \quad (23)$$

The Sherwood number Sh is a function of the Grashof Gr_m mass number and the Schmidt number Sc for natural mass transfer, as show by the following empirical relation [14]

$$Sh = 2 + 0.6 Gr_m^{0.25} Sc^{0.33} \quad (24)$$

The Sc is Schmidt number is given by the equation:

$$Sc = \frac{\mu_a}{\rho_a Df_a} \quad (25)$$

The Mass Grashof number is equal to

$$Gr_{mass} = \frac{\rho_a^2.g.\beta^*.\Delta C_{mol}.(D_d)^3}{\mu_a^2} \quad (26)$$

ΔC_{mol}, is the difference in molar concentration of vapor in the ambient air and at the interface droplet-air (mole.m^{-3}). It is equal to

$$\Delta C_{mol} = \frac{1}{R}.\left(\frac{p_{vap}}{T_a} - \frac{p_d}{T_d}\right) \quad (27)$$

β^*, is the coefficient of mass dilatation (m^3.mole^{-1}), and equal to

$$\beta^* = -\frac{1}{\rho_a}.\left(\frac{\partial \rho_a}{\partial C}\right)_{T_a, p_a} = -\frac{M}{\rho_a} \quad (28)$$

By substituting β^* in the equation (16), the Grashof mass number becomes

$$Gr_m = \frac{\rho_a^2.g.(D_d)^3}{\mu_a^2}\left(\frac{p_d.T_a}{p_{vap}.T_d} - 1\right) \quad (29)$$

Finally, the evaporative flux may be written in the form

$$\phi_{evap} = \frac{M.Sh.Dfa}{R.(D_d)}\left(\frac{p_{v_d}}{T_d} - \frac{p_{vap}}{T_a}\right) \quad (30)$$

Final Equations to Calculate the Temperature of the Droplet and the Air

By combining equations (4 and 30), the heat quantity Φ defined in equation **(3)** becomes

$$\Phi = S_d\left[h_{cv}(T_a - T_d) - \frac{\Delta t.M.Sh.Df_a}{R.D_d}\left(\frac{p_{v_d}}{T_d} - \frac{p_{vap}}{T_a}\right)\right] \quad (31)$$

The equation of the droplet temperature at the instant t+Δt may be written as follows, by replacing the heat quantity Φ (31) by its value in equation (2)

$$T_{d(t+\Delta t)} = T_{d_t} + \frac{\Delta t.S_d}{m_d Cp}\left[h_{cv}.\left(T_{a_t} - T_{d_t}\right) - L_v \phi_{evap}\right] \quad (32)$$

Knowing the droplet temperature at the instant t+Δt, we may now calculate the air temperature around the droplet by establishing a thermal balance with the air

$$m_a Cp_a\left(T_{a(t+\Delta t)} - T_{a_t}\right) = h_{cv}.S_d.\left(T_{d(t+\Delta t)} - T_{a_t}\right) \quad (33)$$

where, Cp_a is the specific heat of the air (J.kg^{-1}.K^{-1}), and is given for 273 K ≤ T_a ≤ 373 K, by

$$Cp_a = 981 + 0.08.T_a \quad (34)$$

Finally, the air temperature is equal to

$$T_{a(t+\Delta t)} = T_{a_t} + \frac{h_{cv}.S_d}{m_a.Cp_a}\left(T_{d(t+\Delta t)} - T_{a_t}\right) \quad (35)$$

Droplet In The Airflow (Forced Convection)

The above analysis was for an isolated droplet in still air. However, in gas turbine fogging applications, the droplets are injected into a moving airstream at a different velocity. Due to the small size of the droplets and low Reynolds number (<1), the drag forces are very large and the droplets attain the air stream velocity in a few milliseconds[5].

Depending on the droplet relative velocity, either natural or forced convection will occur. Natural convection occurs when the relative velocity of a droplet compared to the surrounding air is zero, and forced convection occurs when a relative velocity differential exists.

With forced convection, the Nusselt number is given by Ranz and Marshall [14] equation

$$Nu = 2 + 0.6 \times Re_d^{0.5} Pr^{0.33} \quad (36)$$

And for forced mass transfer, the Sherwood number is given by [14]

$$Sh = 2 + 0.6 Re_d^{0.5} Sc^{0.33} \quad (37)$$

Where Re_d is the Reynolds number of the droplet and given by

$$Re_d = \frac{D_d \times \Delta U \times \rho_a}{\mu_a} \quad (38)$$

[5] Less than 10 ms for the largest droplets developed in the MeeFog nozzles

The effect of velocity is non significant in our conditions (the effect is less than 10% for 50 microns droplet and further decreases when the droplet size diminishes) because droplets are very small and the velocity response time is lower than 10 ms. This means that the use of insolated droplet model gives a good estimation of the behavior of the droplets in the duct. This has been verified by visual observations taken in a wind tunnel experimental setup.

COMPUTATIONAL MODEL

Based on the equations derived above, a computational model was developed. The model works in iterative manner and provides the transient behavior of fog droplets in terms of the droplet diameter, change in relative humidity and temperature and time to attain saturation. The iterations stop ($I=I_{max}$) when the air in the volume around the droplet within Active Radius (RA) becomes saturated or when the droplet evaporates completely. The steps followed by the model shown in Figure 6.

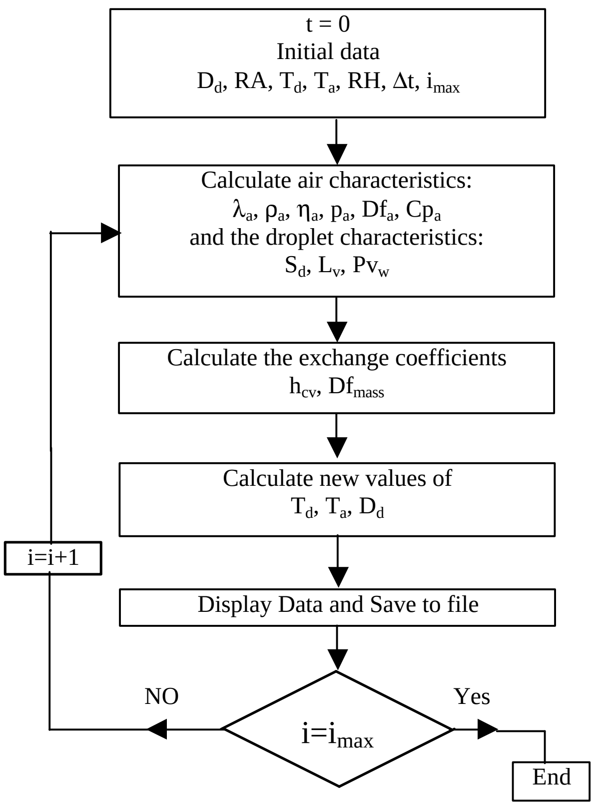

Figure 6. Computational Model for Droplet Evaporation

MODEL RESULTS RELATING TO FOG BEHAVIOR

From the model developed above, a quantification can be made of fog droplet behavior in gas turbine inlet ducts. As soon as a droplet touches the air, and if the vapor pressure near the droplet surface is higher than the vapor pressure of the air far from the droplet (i.e. the droplet is in unsaturated air), evaporation of the droplet starts to occur. In order to balance the evaporation (mass transfer), the droplet has to loose a quantity of energy (heat transfer), which reduces the temperature of the droplet and of the air surrounding it.

An application of the model developed in the earlier section is provided here. Figures 7 and 8 show the time related behavior of droplets for two droplet sizes. Figure 7 depicts the behavior of a 14-microns droplet and Figure 8 for a 30-microns droplet. The dry bulb temperature of the air around the droplet is 35°C (94°F), the RH is 30% and consequently the wet bulb temperature is 21.4°C (70.5°F).

In these examples, we used two different temperatures for the water droplet. In the first the temperature of the droplet was taken as equal to 20°C (68°F) and in the second to as 35°C (94°F).

Analysis of the figures indicate the following:

- The temperature of the droplet in the two cases converges very quickly to 21.4°C (70.5°F), i.e. to the WBT.
- The 14-micron droplet evaporated, and brought the surrounding air to the wet bulb temperature, in about 1.5 seconds, while the 30-micron droplet took around 5 seconds. The importance of droplet diameter to affect rapid evaporation is evident from this observation.
- The smaller droplet reached a diameter of 3-microns in 1.5 seconds (the residence time in a typical inlet air duct), while the larger droplet was still almost 15-micons in the same time frame.
- The effect of the temperature of the water droplet is negligible during the evaporation process. The higher temperature lowers the surface tension of the droplet, and increases the vapor pressure at the droplet surface, and therefore the evaporation is marginally but not significantly quicker.

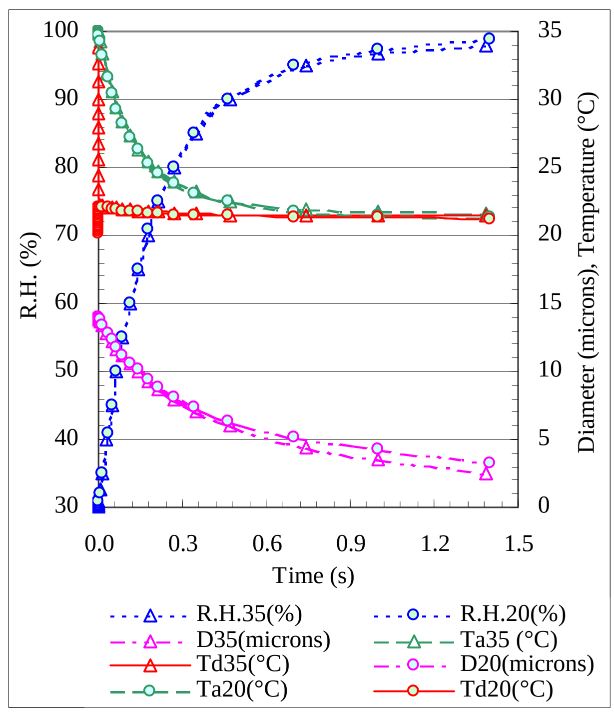

Figure 7. Transient behavior of a two 14-micron droplets, one at 35°C (94°F) and one at 30°C (86°F), starting air conditions are 35°C and 30% RH

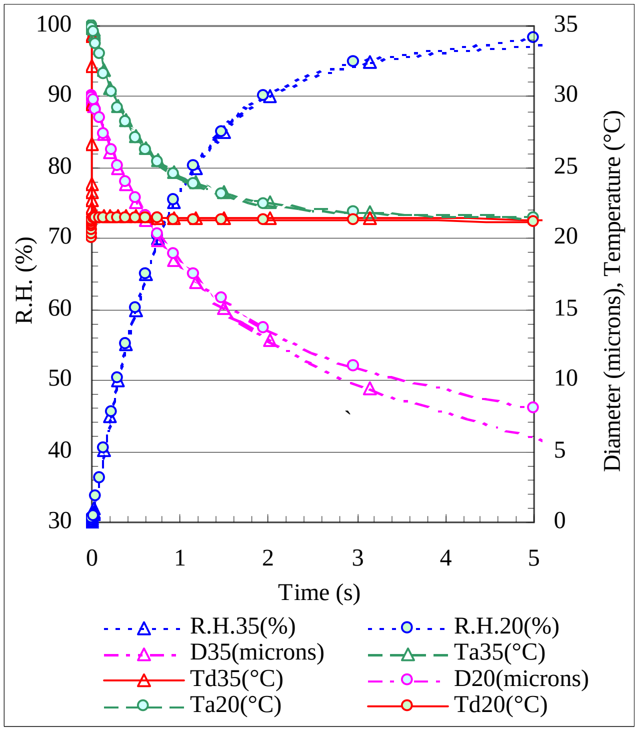

Figure 8. Transient behavior of two 30-micron droplets, one at 35°C and one at 20°C, starting air conditions are 35°C and 30% RH

Figure 9 shows data derived from duct measurements corroborating the general pattern of the temperature fall and relative humidity rise as predicted by model. The abscissa (i.e., the timescale) is a function of droplet size and starting air conditions. However, experience with gas turbines has indicated that temperature reduction to the web bulb temperature can be obtained with a residence time from 0.8 to 2.0 seconds.

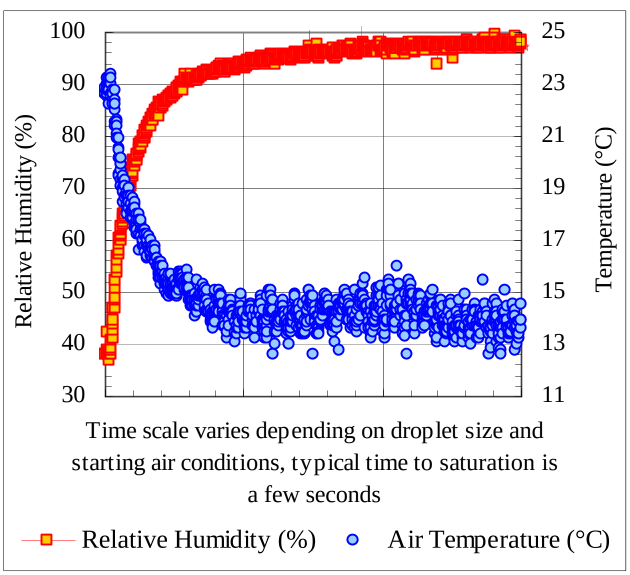

Figure 9. Qualitative behavior measured in a duct showing the increase in relative humidity from a starting condition of 35% RH and the corresponding drop in air temperature in the duct

USE OF THE MODEL TO SHOW THE INTERACTION OF DROPLET WITH THE SURROUNDING AIR

In reality, the fog evaporative cooling process for gas turbines involves trillions (10^{12}) of droplets[6] interacting with the ambient air as they progress down the duct. Therefore, considering only one droplet in isolation can be highly misleading.

To examine the real-world situation more closely we use the concept of active radius (RA), which defines the volume around the droplet for which saturation conditions are calculated. Varying RA can give one an idea of how the evaporation process changes as a function of the how well the fog is mixed with the inlet air. Water vapor will diffuse rapidly across the entire RA volume so that a large droplet will evaporate faster in a larger RA volume. In the real world, droplets are not evenly distributed in the airflow. This is especially true when larger orifice nozzles (with high flow rates) are used because the spacing between the nozzles is greater, leaving large sections of un-fogged air.

The model calculates the temperature and relative humidity conditions over time for different RA values, providing results as shown in Figures 10 and 11. Figure 10 shows results of the evaporation of a droplet of 50 microns at 20% relative humidity (Figure 10 a, b, and c; in the left column) and 60% (Figure 10 d, e, and f, in the right column), both at an initial ambient temperature of 30°C. The effect of RA size on droplet evaporation rate can be seen by comparing Figures 10 a. (RA 81) and 10 c. (RA 500). Note that with the larger RA the droplet size decreases rapidly but the temperature and humidity remain nearly constant. With actual, real-world conditions RA tends to be smaller, because the droplets are in reality very close together, particularly inside the initial spray plumes. A smaller RA means that the temperature and humidity in the RA volume change quickly but the droplet size remains quite large. For example, in the time frame allowed in real gas turbine ducts (about two seconds), the 50-micron droplet would be approximately 30-microns when it reached the compressor inlet.

The situation is further exacerbated by the fact that, in real-world installations, the fog-nozzle plumes do not completely cover the duct cross section, so that a substantial portion of the air passes the nozzle manifolds without mixing into the fog plumes (especially true for large-orifice, high-flow nozzles). Water vapor from the spray plumes must diffuse into the space between the spray plumes and, more significantly, heat must flow from the un-cooled spaces between the plumes to the cool air inside the plumes. Neither of these processes are very rapid. The information presented here shows that the real-world evaporation process, where droplet density is very high, is far slower than when a droplet is considered in isolation, with a large RA.

It also points out another key factor, if there are a mix of droplet sizes present, with some droplets above 30 microns, the smaller droplets will evaporate first increasing the local relative humidity and so making it even more difficult for the larger droplets to evaporate. Therefore the large droplets pose an even bigger risk for erosion should they be ingested into the compressor and are more likely to impact or fall out and cause excessive wetting of the duct floor.

In a gas turbine, the residence time available is 1-2 seconds, and hence droplets larger than about 30 microns will not evaporate and will, in fact, be quite large when they enter the compressor.

[6] One impaction pin nozzle operating at 138 barg can create over three billion droplets per second and a large gas turbine may have as many as one thousand fog nozzles.

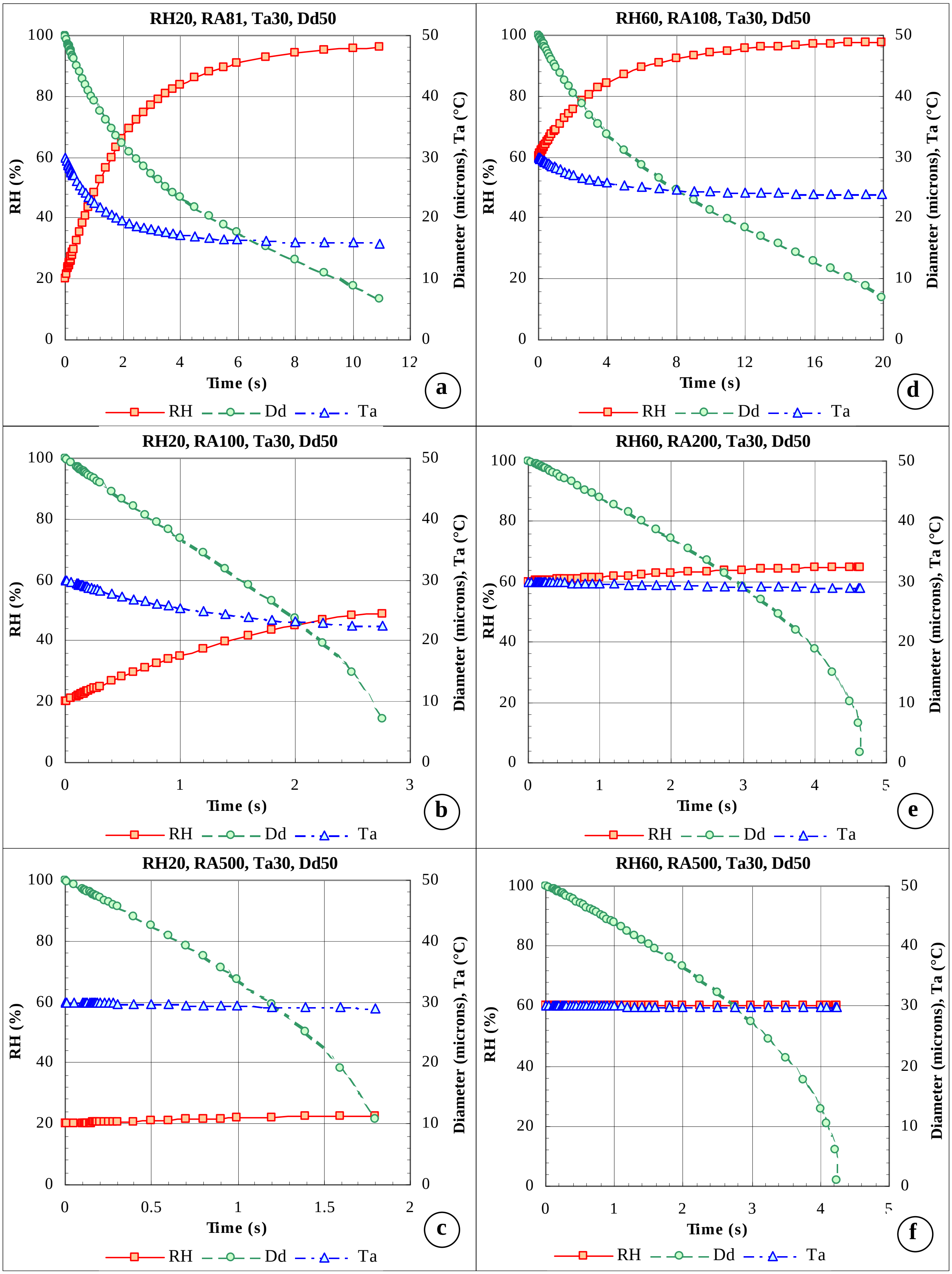

Figure 10. Interaction of droplet to surrounding air conditions, Droplet diameter: 50 microns

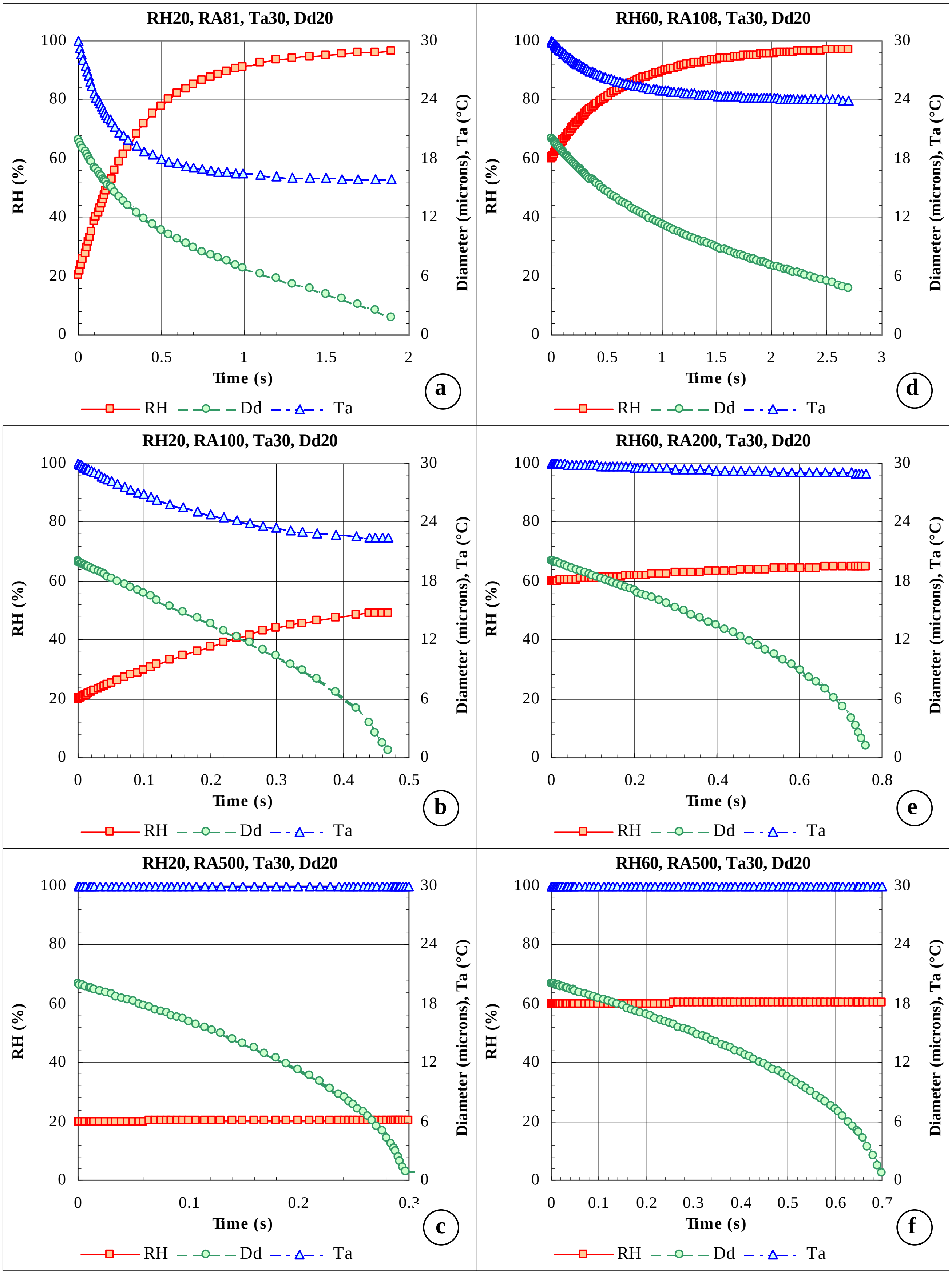

Figure 11. Interaction of droplet to surrounding air conditions, Droplet diameter: 20 microns

PRACTICAL CONSIDERATIONS

When specifying or evaluating an inlet fog system, the following engineering factors must be considered:

- Methods for avoiding FOD (Foreign Object Damage).
- Methods for avoiding Icing at the inlet.
- The existence and proper design of Duct Drains.
- Potential for compressor surge and methods to mitigate risks (applies more to compressors that are being fog intercooled).
- Methods for ensuring intake temperature uniformity to avoid potential stall or blade vibration.
- Fouling and gas turbine performance deterioration.
- Droplet size of fog produced and the potential for compressor erosion or blade coating distress.

Foreign Object Damage (FOD)

As the nozzle manifolds exist in the air stream, care must be taken to avoid any chance of foreign object damage (FOD) that might result from loose nozzles or nozzle array components. It should be noted that wind tunnel tests for MeeFog nozzles have shown that even at airflow velocities of 3000 fpm (15.2 m/sec), the nozzle will fall to the floor very quickly, although normally the nozzles are located in the much lower velocity area after the air filtration. Safety wiring of nozzles (Figure 12) and analysis of the fog nozzle array for airflow-induced vibration ensures that the structure is strong and cannot fatigue. Details of the airflow induced vibration design of the nozzle array along with a chart that users can utilized to evaluate their designs have been provided in Part C of this paper.

Figure 12. Lock wire arrangements

Gas Turbine Inlet Icing

The fog control system should automatically terminate fogging whenever there is any chance of inlet icing due to the static temperature depression that occurs in the bellmouth, which is caused by the acceleration of the air to Mach numbers of about 0.5 for heavy-duty gas turbines and 0.8 for aeroderivative machines.

The MeeFog system control software includes a user-definable set point called “inlet temp minimum.” This parameter can be set between 10°C and 27°C (50°F and 80°F) but it is set at the factory to 15.5°C (60°F). The fog system is staged such that the calculated after-fogging temperature will not fall below the value of the inlet temperature minimum parameter. Furthermore, once the after-fog temperature reaches this value any and all active over-fogging component of the system is disabled.

Several OEMs publish a combination of relative humidity and temperatures at which anti-icing measures are turned on. With fogging applications where the ending relative humidity is close to 100%, temperatures as low as 10°C can be utilized[7].

Duct Drainage

This is an important subject and there are a lot of practical issues involved that a fog supplier should have learned from experience. Drains should be strategically located both near the nozzle manifolds and also in the intake bell mouth region. Drains should be carefully designed and must be continuously operating. The number of drains should be determined based on experience, and the configuration of the duct and obstructions that might result in water collection. Special shaped channel sections can be located on the duct walls and floors to channel collected water to the drains. One such channel is shown in Figure 13. Appropriate sealing systems must be used to prevent the flow of ambient air into the duct. In the case of P trap seals, it is imperative that water be supplied to these to ensure that they do not run dry due to evaporation, which will allow the ingress of air. We have found greater success with stainless steel swing-check valves. Experience has shown that drain flows are typically 2-6% of the total water flow, though in some cases with restrictive duct configurations, as high as 8% may be attained under full flow conditions.

Drain flow should be monitored and logged as a function of ambient conditions and the number of stages. This can be done simply by measuring the flow for 5 minutes and then calculating the flow rate as a percent of the overall water flow.

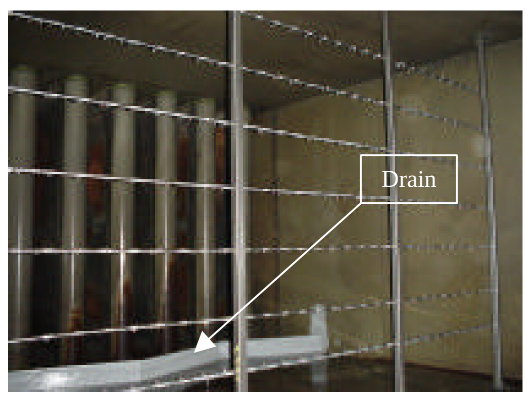

Figure 13. Special channel system for water drain

A leak in the high-pressure lines inside the duct would cause an increase in drain flow rate but a well-designed fog system has pressure switches on the outlet of the pumps, which immediately terminate water flow in the event of a leak. More details relating to drains are provided in Part C of this paper.

[7] There are several considerations other than just calculating the intake temperature static depression caused by air acceleration to Mach numbers of 0.5 to 0.8. There is also some heating (on the order of 1°C) due to the condensation that occurs and also due to heat transfer from the number 1 bearing etc.

Compressor Surge

This is mostly only a concern on turbines that are being fog intercooled (i.e., those in which a quantity of overspray is allowed). The effect of overspray is to cause the operating points on the compressor map to move towards the surge line. Ingistov [17] has reported on four GT-7EA Units that have been operating with overspray for over two years. No surge related problems have been reported with over-fogging levels as high as 0.6% to 1% percent of the air mass flow.

Compressor Intake Temperature (CIT) Uniformity

Axial compressors have stringent inlet temperature and pressure uniformity criteria. This issue relates to blading vibration that can be induced due to extensive variations in air temperature at the face of the compressor. Details of blading issues may be found in Meher-Homji [18].

Fogging systems are designed in multiple stages and each stage has manifolds distributed within the inlet duct to provide a relatively uniform intake temperature. This is an important consideration as the susceptibility of the compressor to stall or surge could also be affected by severe temperature distortion. Even though an attempt is made to evenly distribute the nozzles on each stage, it may be desirable to avoid operating a fogging system at low staging levels for extended periods of time.

The accurate measurement of temperature distortion requires specialized instrumentation. However the following equation provides an idea of the criteria for a typical high Mach number aeroderivative machine. Different machines would have different criteria.

$$Distortion = \frac{\Delta T_{\max}}{T_{avg}} = \frac{(T_{avg,\max})60^\circ - (T_{ave,\min})60^\circ}{T_{avg,\,face}} \le 0.03 \qquad (42)$$

where,

$T_{avg,max}$ = Maximum area weighted average total temperature (°R) in the warmest 60° sector of the annulus.

$T_{avg,min}$ = Minimum area weighted average total temperature (°R) in the coldest 60° sector of the annulus.

$T_{avg,face}$ = Average area weighted average total temperature (°R) over the full face of the annulus.

Local temperatures within the sectors must be within 20% of the face average. There are also criteria relating to rapid changes in the inlet temperature but fogging cannot exceed these criteria. Even in the event of an emergency shutdown of the fog system, the residual fog in the duct will take some time to evaporate, so the result is never an instantaneous shock temperature change. The location of gas turbine CIT sensors, typically do not provide a means to evaluate temperature distortion. In several engines with complicated ducts, considerable temperature and pressure distortions exist even without fogging and it is not uncommon to find temperature difference between the CIT sensors of as much as 2°C (3.6°F). In some gas turbines, the CIT sensors may be located in the intake duct in a location that is partially starved of airflow. These sorts of pre-existing distortions can be made more evident by inlet fogging. It is a good idea to map the compressor inlet sensor readings with different ambient conditions and operating parameters and different numbers of fog stages in operation. This can form a useful baseline for analysis.

Intake Duct Considerations

There are several issues relating to the intake duct itself that should be considered. For instance, when installing nozzle manifolds on existing turbines, there is often not an ideal amount of space in the inlet ducts, hence careful compromises need to be made. Items such as bleed heating pipes (often present for DLN machines where air flow is bled for controlling the combustor conditions under certain operating regimes) or silencers may present obstructions, which the fog can impact on, resulting in excessive pooling of water on the duct floors. Duct supports as shown in Figure 14 must also be considered in fogging system design. Further, most engines utilize trash screens and these will act as fog agglomerators causing some larger droplets. For the intake bell region of cold end drives that have intake cones, cone wetting may occur and some water may progress towards the compressor. However due to the Weber number effect, the high air velocities result in droplet shattering and consequently the extent of this problem is minimized. The fog system supplier should be knowledgeable of these practical considerations and design to minimize any problems. Much of the choice of selecting and designing a system is an art and involves CFD analysis coupled with past experience and visual observations.

Figure 14. Duct support structures like those shown here can collect fog and result in excessive pooling of water. Such obstructions must be taken into account when designing the nozzle manifolds

Axial Compressor Fouling

It is important to distinguish between the problems of natural climatic fog and the fog generated by the fogging system.

High **natural** humidity and climatic fog, as often occurs during nights and early mornings, can cause high filter delta P trips and sometimes the heavily fouled filters tend to unload (leaching effect through the filter), thus causing compressor fouling. However, if the air filtration system is working well, the increased humidity caused by the fogging system does not inherently increase fouling. Fouling is so site specific that it is very difficult to predict. If the No.1 bearing is leaking oil this may combine with the high humidity (caused by inlet fogging) to create some fouling.

An important issue with inlet fogging is to wash the silencers and duct surfaces thoroughly to avoid dirt that has been accumulated here being washed into the compressor by the fogging system. This is particularly important when fogging is being installed as a retrofit on older machines. On retrofit applications, it may be necessary to perform several crank washes before the problem resolves. More details on compressor fouling may be found in Meher-Homji [19] and Meher-Homji et al [20].

Compressor Erosion

For pure evaporative cooling systems, system parameters can be adjusted to ensure evaporation of fog prior to the compressor inlet. Yet, it is always possible that some droplets do enter the compressor. For droplet sizes that are relatively small (less than 15-20 microns) CFD studies have shown that the flow will tend to follow the air stream. There is the possibility of larger water particles forming on the trash screen and inlet cone of the gas turbine but with proper design and drainage approaches this can be minimized.

Corrosion in the Inlet Duct

The use of demineralized water can further deteriorate inlet ducts that are already in a deteriorated state. The increased humidity, as well as water pooling on the duct floor, are clearly corrosion factors. With proper maintenance and painting this problem can be mitigated. The use of Stainless Steel 316L as the duct material is gaining in popularity as life cycle studies have indicated that while the first cost is a little higher, the life cycle costs are significantly lower. It is important to note that ducts, the world over, often operate in a distressed condition such as seen in Figure 15. It is important that such problems are addressed and corrected prior to the implementation of fogging.

Figure 15. Corroded inlet air duct floor

Compressor Coating Distress

Some gas turbines that have undergone overspray have experienced coating distress in the first few stages of the axial flow compressor. In most cases, this can be minimized by using nozzles with the smallest possible droplet size, careful location of fogging nozzles, avoidance of excessive water accumulation on ducts walls and floors and on the inlet cone and by several other proprietary approaches. Corrosive ambient conditions such as chlorides or even trace amounts of HCl will cause acidity and hence coating damage. Airborne contaminants in even the ppb range can often create very acidic environments for the axial compressor blading. This is an issue that must be resolved by proper inlet air filtration, especially in aggressive industrial environments. Special coatings can help protect blading in areas with high levels of ambient contaminants. For overspray application, the situation should be studied on a case-by-case basis by carefully evaluating the blading material and coating technologies available. Haskell [21] has described the criticality of ambient air quality.

In spite of these precautions, overspray may, over time, create some coating distress but the benefit of increased output far outweighs the cost of more frequent recoating of blading.

In a few rare cases, there have been some reports of coating distress on units that are undergoing evaporative fogging only. There can be several causes of this:

- Improper settings of fog system control parameters resulting in overspray conditions.
- Improper orientation or location of fog nozzles resulting in fog impaction on obstructions in the duct and the creation of very large droplets, or of excessive pooling on the duct floor.
- Lack of drains or lack of continuously operating drains or inappropriately located drains.
- Corrosive ambient air contaminants that cause acidity and hence coating damage. This is an issue that may be resolved by proper inlet air filtration, especially in aggressive industrial environments, or by selecting corrosion resistant blade coatings.
- In some cases, if some leading edge coating distress already exists prior to fogging this might progress faster with fogging due to the reasons mentioned above.

Practical Considerations During System Design and Implementation

Some important practical considerations to be made in the implementation of any fogging system are provided below.

- Check vendors design calculations with regards to water flow requirements, particularly under off design conditions, and evaluate the design under different climatic conditions.
- Ensure that you provide the fogging system vendor with detailed sketches and photos of the inlet system. OEM's drawings may not show the exact configuration of duct internals, so up-to-date photographs are the best way of ensuring proper nozzle manifold design and location.
- Evaluate the amount of overspray required (i.e., intercooling effect) and ensure that the compressor can accommodate this. Surge margin should be evaluated.

- Evaluate generator capacity and lube oil coolers, etc. to ensure that the balance-of-plant equipment can tolerate higher turbine output on hot days. Often fog can be applied to cool these.
- Specify all stainless steel wetted components. Even the water supply lines should be stainless steel or some other non-corroding material. Iron or galvanized supply water lines can result in excessive plugging of inlet filters, or even plugging of the nozzles themselves. The presence of even trace amounts of certain minerals in the supply water can result in the growth of anaerobic bacteria, which can be a source of frequent nozzle plugging.
- In the event of a problem, does the design permit rapid isolation so operation can proceed without the fogging system?
- Does the design optimize the location of the fogging manifolds with respect to air filters and inlet systems. Insist on design justification from suppliers.
- Check that sequential fogging capability (cooling stages) is adequate to meet your demand profile and turndown profile. Uncertainties in the actual air mass flow of the axial compressor can be on the order of several degrees Celsius. Therefore, unless these uncertainties are mitigated in some way, very small stage increments are not meaningful.
- What design features have the vendor provided to avoid potential FOD?
- Check shedding frequency and rigidity of the manifolds to avoid flow-induced vibration. Vendor calculations should be reviewed.
- Review vendors proposed manifold design for structural rigidity and strength.
- Review proposed tie-in of the fog pump skid PLC (Programmable Logic Controller) with plant DCS (Distributed Control System).
- Ensure that the design does not impose a large pressure drop.
- Evaluate the proposed installation for maintainability and accessibility.
- Evaluate the fog droplet size. Close approximations of droplet size can be made using empirical formulae, given the type of nozzle and the system operating pressure.
- Evaluate the number of fog nozzles. Nozzles that flow more water, at a given pressure, generally make bigger droplets. Fewer nozzles may have a benefit in reduced maintenance or upfront cost but, as discussed above, they have the disadvantages of creating larger droplets and of providing fewer points of emission and, therefore, less coverage of the inlet airflow.
- Ensure that appropriate drain lines will be installed in the inlet system.
- Installation of the weather sensor can be at the skid location, if this is representative of inlet conditions to the gas turbine. Ensure that there is no secondary effect causing wrong measurements (e.g. close proximity to an air cooled heat exchanger or source of radiant heat).
- Demineralized water supply lines must be thoroughly flushed to ensure that no dirt has accumulated during installation.
- Thoroughly wash the inlet duct and silencers. Dirt in the inlet system can be carried into the compressor causing axial compressor fouling and creating performance deterioration or compressor damage. This is of special importance on retrofit applications.

CLOSURE

This paper has provided a comprehensive analysis of the thermodynamics and heat transfer phenomena for gas turbine inlet fogging and has presented a model that can be used to evaluate the behavior of fog droplets in inlet ducts. The importance of the correct use of droplet diameters and measurement approaches has been covered and the important effect of flow velocity on fog droplet behavior has been described. Several practical aspects relating to inlet fogging have been covered. Chaker et al [21, 22] provide further details on droplet sizing and behavior in inlet ducts.

ACKNOWLEDGMENTS

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