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DISCUSSION OF SOME MYTHS/FEATURES ASSOCIATED WITH GAS TURBINE INLET FOGGING AND WET COMPRESSION

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ABSTRACT

Gas turbine inlet fogging and overspray (high-fogging) have been considered the most cost-effective means of boosting a gas turbine's total power output, especially under hot or dry weather conditions. The result of employing fogging or overspray is indisputably clear – total power output is increased; however, development of the theory and explanation of the phenomena associated with fogging and overspray are not always consistent and are sometimes misleading and incorrect. This paper focuses on reviewing several interesting features and commonly discussed topics, including (a) entropy production of water evaporation, (b) the effect of centrifugal force on water droplets, and (c) whether water droplets can survive the journey in the compressor and enter the combustor. Furthermore, three turbine myths: that fogging/overspray increases the air density in the compressor, reduces the compressor power consumption, and noticeably enhances the gas turbine efficiency, are examined and discussed.

Some common mistakes in describing the compressor work are identified and corrected. A newly constructed multiphase T-S diagram is used to explain the physics of water droplet evaporation process and corresponding entropy production during wet compression.

NOMENCLATURE

CET	Compressor exit temperature (K)
CIT	Compressor inlet temperature (K)
d	Droplet diameter (m)
D	Mean diameter of compressor shaft (m)
Cp	Specific heat (J/kg-K)
f	Mass fraction of liquid water
F	Force (N)
g	Gravitational acceleration (9.81 m/s²)
GT	Gas Turbine
h_{fg}	Latent heat of vaporization (kJ/kg)
N	RPM
OS	Overspray
q	Heat transfer (kJ/kg)
RH	Relative humidity (%)
s	Specific entropy (kJ/kg.K)
T	Temperature (K)
TIT	Turbine inlet temperature (K)
U	Translational velocity of blade (m/s)
V	Axial velocity (m/s)
v	Mass fraction of water vapor
w	Specific work (kJ/kg)

Greek

ρ	Density (kg/m³)
η	Efficiency

Subscript

a	Dry air
amb	Ambient
c	Compressor
f	Liquid water
fc	Fuel compressor
t	Turbine
w	Water

INTRODUCTION

A comprehensive thermodynamic wet compression model was first established by Hill [1] and was further expanded by many researchers. For example, Zheng et al. [2, 3] provided the relationship between the dry compression index and wet compression index. Ransom et al. [4] exhibited a calculation procedure of enthalpy and some of the limitations of its calculation with associated source of errors.

Using aero-thermodynamic modeling, Bagnoli et al. [5] investigated the effects of interstage water injection on the performance of a 17-stage gas turbine. To estimate the overall gas turbine performance, they discussed the impact of interstage injection by viewing the stage-by-stage compressor performance of selected gas turbines using the shape factor approach. They found that better power augmentation could be obtained when the water injection is applied upstream of the compressor as compared to other injection locations between stages. Since the maximum amount of water injection is limited by ambient conditions, maximum allowable gas turbine power output, and the compressor surge limit, they cautioned that increasing the amount of water injection may cause the last compressor stage to operate closer to the surge line.

Bagnoli et al. [6] further developed a calculation code by modeling the exchange of latent heat at the inlet and outlet stage, as well as the exchange of sensible heat at constant temperature and pressure in the middle of the stage. Using the model developed above, Bagnoli et al. [7] investigated the effects of interstage water injection on the performance of a GE Frame 7EA gas turbine (GT). They found that 1.6% of water injection at stator of the second compressor stage allows recovery of 15% of the lost ISO rated power for high ambient temperature conditions. They also found that the power boost per unit of water injection decreases as the injection point moves from the first to the fifth stage. The droplet residence time decreases as the injection point moves closer to the compressor inlet. In the case of water injection, the maximum reduction in compressor specific work, which reaches a value of about 97% of the specific work in ISO dry case, takes place with injecting in the inlet duct at high ambient temperature conditions.

White and Meacock [8] described the method of meanline calculation for wet compression. They applied the method to see the nature of the off-design condition due to evaporative cooling. In their calculation, they assumed that the polytropic efficiency is not affected by evaporation and that the entropy changes are due only to aerodynamic effects, not irreversible phase change.

Khan and Wang [9] developed a wet compression thermodynamic model for a gas turbine system (FogGT) with inlet fog cooling specifically for burning low calorific value (LCV) fuels. The results showed that in the combustor, as heating value decreases for LCV fuels, there are more incombustible gases in the fuel to absorb the energy and suppress the combustion temperature; therefore, more heat addition (23%-46%) is required to allow the combusted gas to reach the desired Turbine Inlet Temperature (TIT). In the turbine, the LCV fuels produce more net output power than natural gas, even though LCV fuels significantly increase fuel compressor power. When LCV fuels are burned, saturated fogging can achieve net output power increases of approximately 1-2%, while 2% overspray can achieve 20% net output enhancement. As the ambient temperature or relative humidity increases, the net output power decreases. For LCV fuels, the thermal efficiency is approximately 10-16% (3-5 percentage points) lower than when using natural gas. Burning LCV fuel leads to small changes in thermal efficiency irrespective of a large increase in net power output because the increased demand for additional heat input makes up the sensible heat required for an increased fuel flow rate and incombustible gases. Fog/overspray could either slightly increase or decrease the thermal efficiency depending on the ambient conditions.

Kim and Perez-Blanco [10] explained the theoretical limits of machines via a heat and mass transfer model. They modeled continuous compression cooling numerically via evaporation based on droplet evaporation analysis using a 1-D method. They found, initially, that the increase of the droplet temperature overrides the decrease in droplet mass, thus resulting in an internal energy increase in the droplet. The internal energy peaks and then decreases with time as the liquid evaporates.

Sanaye et al. [11] studied the effects of inlet fogging and wet compression on gas turbine performance using the shape factor approach. They modeled the evaporation of water droplets in the compressor inlet duct and estimated the diameter of water droplets at the end of the inlet duct. They compared their findings with the results from FLUENT software. They also predicted the compressor discharge air temperature due to the presence of un-evaporated water at the inlet duct. They found that the flow coefficient increases in first few stages due to the water spray. This increase in the flow coefficient leads to an increase in axial velocity at the first few stages, and the corrected speed increases due to the cooling of compressor inlet air. The result showed an increase in density and pressure and a decrease in the axial velocity at later compressor stages.

Many other related studies [12-20] deserve credit for contributing to the improvement of wet compression theory or calculation models. Although not all of them are cited in this paper, they all conclude with an indisputable result: fogging/overspray will increase the total net output power. However, development of the theory and explanation of the phenomena associated with fogging and overspray are not always consistent and are sometimes misleading and incorrect. The following three myths have prevailed in the open literature due to ambiguous descriptions of conditions (at inlet, boundary, or exit), misleading statements, or incorrect concepts. The three myths are that fogging or overspraying will

(a) increase the air density in the compressor,
(b) reduce the compressor power consumption, and
(c) increase the gas turbine efficiency.

The **objective** of this paper is to examine the myths, identify the sources of these misconceptions, and clarify the misunderstanding. In addition, this paper also reviews several interesting features and commonly discussed topics, including (a) entropy production of water evaporation, (b) the effect of centrifugal force on water droplet dynamics, and (c) whether water droplets can survive the journey in the compressor and enter the combustor.

Some common mistakes in describing the compressor work are identified and corrected. A multiphase T-S diagram is constructed to explain the physics of the water droplet evaporation process and the corresponding entropy production during wet compression.

THEORY OF WET COMPRESSION

Examining the theory of fogging and wet compression is the first step in finding the sources that lead to the myths. A standard explanation of fogging/wet compression theory follows below.

The effect of elevated temperature on GT power output and efficiency can be explained by analyzing the P-v and T-S diagrams. Path 1-2-3-4 in Fig. 1 shows the ideal Brayton cycle at the reference ISO condition (59°F and 60% relative humidity), and 1′-2′-3-4 shows the processes on a hot day.

In the ISO condition, the required compressor power is represented by the area 1-a-b-2; whereas, under elevated ambient temperature, the required compressor power is represented by area 1′-a-b-2′, which is larger than at ISO condition. The turbine output power remains the same for both conditions, so the net specific output power (i.e., the power per unit mass flow rate) decreases.

On the other hand, the rising isobaric curves (1-4 and 2-3) in the T-S diagram shows that heat addition in the combustion chamber at lower temperatures produces more useful energy. This can be explained by noticing that more heat will be rejected (area under curve 1-4) at higher inlet temperature $T_{1'}$ if the same amount of useful energy (e.g. area 1-2-2'-1') is to be harnessed.

The above analysis is on a per unit mass flow rate basis. Elevated ambient temperatures further make the air lighter and reduce the air mass flow rate. Since the gas turbine is a constant volume flow rate machine at a fixed rotational speed, a reduced mass flow rate results in a reduction of the total output power. Because the output power is influenced by both compressor power and mass flow rate, while the efficiency may not be affected by the mass flow rate, elevated ambient temperatures will affect the output power *more* than efficiency.

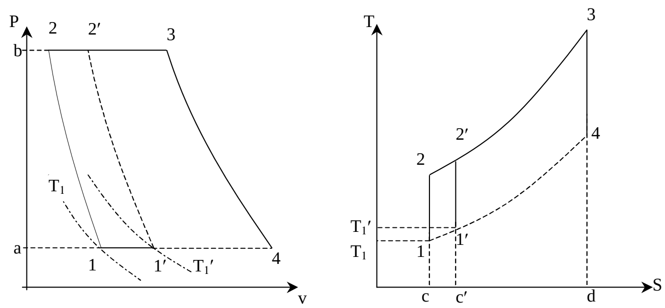

Figure 1 Effect of increased ambient temperature on GT efficiency and output power per unit mass flow rate (i.e., specific power)

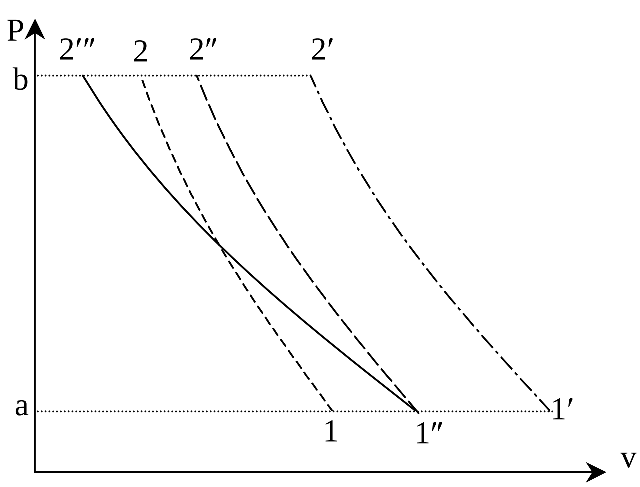

Figure 2 Different fog/overspray cooling processes in the air-intake duct and in the compressor

The effects of fog cooling and overspray are shown in Fig. 2. In Fig. 2, 1-2 shows the compression under the ISO condition; 1′-2′ shows the compression in an elevated ambient temperature condition; 1′-1″-2″ shows the **humid compression** with inlet cooling without overspray; and 1″-2‴ shows the **wet compression** with overspray cooling. 1′-1″ shows the effect of compressor inlet temperature drop due to inlet fog cooling to saturation without any overspray. Evaporation in 1′-1″ saturates the air and reduces the air temperature to the wet bulb temperature (WBT) at state 1″. (Note that the saturated air temperature could be slightly lower than the WBT due to the heat transfer between the saturated air and remaining water droplets when the water is supplied at a temperature below the WBT.) Typically, fog inlet cooling does not reduce the inlet temperature lower than the ISO condition, so 1″ is typically on the right of 1 on the P-v diagram. Notice that 1″-2‴ is not parallel to 1′-2′. The reason for this is that wet compression reduces the polytropic index (k) of the compression work (Pv^k=Constant) from isentropic process (k = γ, specific heat ratio). 1″-2‴ may or may not cross over the ISO path 1-2. The additional reduction of compressor work due to overspray is evident from the departure of the curve 1″-2‴ from the curve 1″-2″ (humid compression without overspray). Therefore, fog and overspray cooling reduce compression work, which leads to an increase in the net specific output power.

In the meantime, fogging/overspray further increases the total mass flow rate, which does not affect the thermal efficiency but does increase the power output. These two statements seem reasonable at first glance, but they are not precise and can cause some confusion, as will be discussed later. In this study, the term "fogging" indicates the action of generating the fog. Depending on the amount of the injected water, "saturated fogging" implies the process of saturating the air to 100% relative humidity and "overspray" or "high-fogging" implies the process of injecting more than the water amount required to achieve saturated air. Strictly speaking, a 1% overspray indicates that the amount of water that weighs 1% of the dry air flow is injected in addition to the amount required to saturate the air. However, for simplicity, the amount of liquid used in overspay fogging also includes the amount of liquid required in saturated fogging in this study. For example, 2% water overspray at an ambient condition of 300K and 60% RH implies that 0.245% water is needed to saturate the air and (2 – 0.245) = 1.755% is actually used for overspray. In this paper, “dry” air means no water vapor in the air (RH=0); "humid" air means air contains water vapor but not liquid water droplets (RH>0); and “wet” air means that the air contains water droplets. The term “dry compression” has been used by industry to indicate compression of dry or humid air with no water droplets. Although it is a misnomer because the air is not completely dry, this paper adopts it nonetheless to follow industry practice.

Construction of a multiphase T-s diagram to examine entropy changes in the wet-compression process

It has been a general concern that the entropy production during wet compression can affect the compressor efficiency and that the change in compressor efficiency cannot be easily factored into analytically analysis without experimental data. To increase understanding of the entropy changing process during wet compression, the multiphase T-S diagram is specifically constructed in Fig. 3 to illustrate the process of fogging/wet compression in detail. It is obvious that the cooling effect of evaporation enhances compression efficiency, but due to the irreversibility produced through evaporation, the generated entropy inevitably degrades the benefit gained from the cooling effect of evaporation. Figure 3 qualitatively shows the specific entropies of different species (air, saturated/superheated steam, and liquid water) during wet compression.

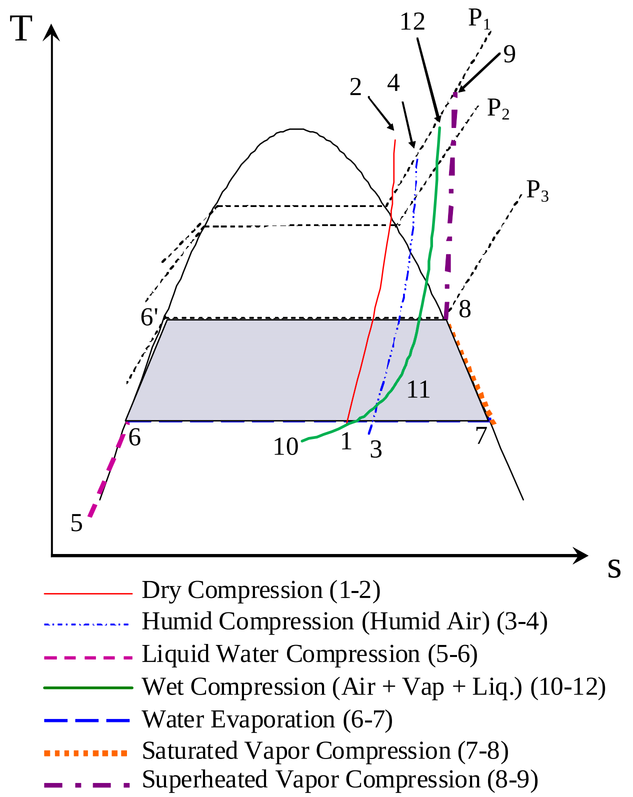

Figure 3 A qualitative T-s diagram illustration of different phases during a wet compression (Not to scale).

Curve 1-2 in Fig. 3 shows the dry compression path (the air contains no water vapor). The air state at 1 is the ambient condition with a relative humidity typically less than 100%. 3-4 shows the T-s curve for only the humid air component of a wet compression process. The air entropy increase is less in wet compression than in dry compression; but this picture is different when liquid water droplets are involved. As wet compression progresses, the liquid water entropy increases, as shown in 5-6 on the saturated liquid side of the vapor dome (i.e., left curve) of the T-s diagram. When the liquid water evaporates, the entropy increase is higher as the entropy in the water vapor side is higher than the liquid side of the saturation dome (6-7). For example, at 20°C, the entropy of the saturated water is 0.2966 kJ/kg•K, while it is 8.6672 kJ/kg•K for the saturated steam. This fact is revealed as the curve 10-11 in Fig. 3 for the mixture in wet compression. Entropy increases significantly through 6-7 as long as evaporation takes place. Due to compression, the evaporation process actually takes place at different pressures along the path in the compressor, as shown by the shaded region between evaporation lines 6-7 and 6'-8. Line 6-7 represents evaporation near the compressor inlet and line 6'-8 indicates the evaporation line of the last drop of liquid. Once all the liquid water evaporates to vapor, the saturated vapor might lose entropy as compression progresses as shown by 7-8 on the saturated vapor side (right curve) of the vapor dome. This entropy-reduction process along the saturated vapor line with increased pressure could occur due to the cooling effect of liquid evaporation. However, it is not clear (without experiment) what is the exact path of the process between 7 and 8. Although the saturated steam loses entropy, the overall change of the mixture's entropy still increases through 10-11 and up to point 12. When the superheated water vapor (steam) is compressed, the entropy increases (rate of increase is not same as the rate for 10-11), so does the air-steam mixture entropy (11-12). Figure 3 is qualitatively drawn for illustration purposes and is not to scale; the actual wet compression path in Fig. 4 is not as distinctive as shown in Fig. 3.

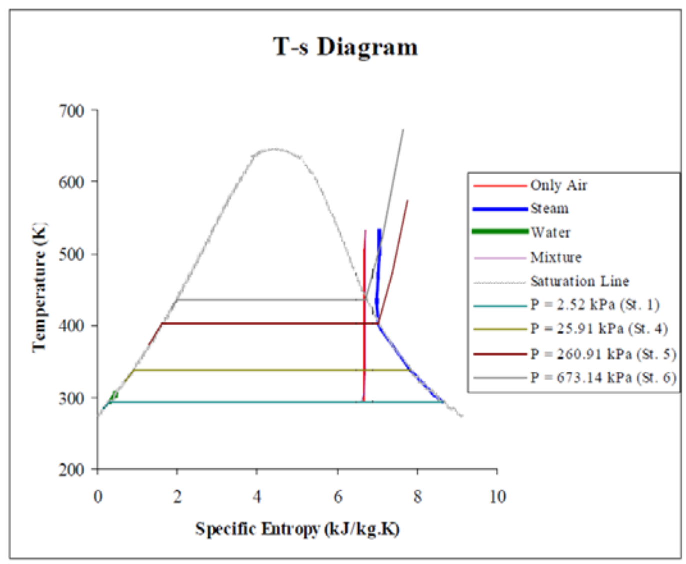

Figure 4 T-s Diagram with actual data for all the species. Note: The air-only line almost coincides with the mixture line and cannot be seen clearly in this figure

Figures 4 show the real data for a wet compression with the ambient temperature of 300K, 60% relative humidity, and 2% overspray of water, where the data was taken from Wang and Khan [21, 22]. Some quantitative data is presented below.

Taking hot and 2% overspray cases from Wang and Khan [21, 22]:

For the hot case without fogging,
T_{amb} = 300K, RH = 60%, T_1 = 300K, T_2 = 557.9K
P_1 = 101.35 kPa, P_2 = 748.2 kPa,
$f_1 = 0$, v_1 = 0.01354 (from 60% RH)
$f_2 = 0$, v_2 = 0.0134,

$$s_1 = \frac{s^0_{a_1} + f_1 s_{f_1} + v_1 s_{v_1}}{1 + f_1 + v_1} = 6.7293 \text{ kJ/kg}\bullet\text{K},$$

$$s_2 = \frac{\left[s^0_{a_2} - R\ln\left(\frac{P_2}{P_1}\right)\right] + f_2 s_{f_2} + v_2 s_{v_2}}{1 + f_2 + v_2}$$

= 6.7686 kJ/kg.K,

The change in entropy is $\Delta s_h = s_2 - s_1 = 0.0393$ kJ/kg•K.

For the hot case with overspray, the conditions are:

T_{amb} = 300K, RH = 60%, 2% overspray,

T_1 = 294.2K, T_2 = 533.4K, P_1 = 101.35 kPa, P_2 = 796.9 kPa,

f_1 = 0.01755, v_1 = 0.01599, f_2 = 0, v_2 = 0.0334.

Use the similar equations as above,

s_1 = 6.6076 kJ/kg•K and s_2 = 6.7103 kJ/kg•K,

The change in entropy, $\Delta s_{OS} = s_2 - s_1 = 0.1027$ kJ/kg•K.

The entropy change for the overspray case is 161% more than the hot case without fogging, which is significant. The increased entropy due to droplet evaporation would degrade the “equivalent" polytropic efficiency of aerodynamic performance, as proposed by White and Meacock [8] and is shown in Eq. (1):

$$\eta_a = \frac{1}{1 + \Delta s_a / [R\ln(P_2/P_1)]} \tag{1}$$

Indeed, Sanaye et. al. [11], Abdelwahab [15], Roumeliotis and Mathioudakis [18, 23], and Wang and Khan [22] have all shown that the polytropic efficiency of the compressor during wet compression decreases. Roumeliotis and Mathioudakis [18] specifically showed that the compressor power increased by water injection and that the increased compressor power varied linearly with the quantity of water entering the stage. As a result, the compressor isentropic efficiency decreases linearly with the amount of water injected.

MYTHS AND EXPLANATION

To examine the myths, the data from Wang and Khan [22] is used for illustrative purposes. In [22], a stage-stacking method was used to calculate the compressor performance under fogging/overspray condition of a compressor operated at constant RPM and a pressure ratio between 8-9. The water droplets were provided at 288K. The thermal equilibrium method was used, which means that the saturated condition will be reached at the end of each stage as long as sufficient liquid water is available. Six cases were simulated:

Case 1: Designed baseline case at ISO condition (288K and 60% RH).

Case 2: Under hot weather at 300K and 60% RH

Case 2S: Saturated (0.245%) spray at the 1st rotor inlet at 300K and 60% RH.

Case 3: 2% overspray at the 1st rotor inlet at 300K and 60% RH

Case 4: 2% over spray at stage 1 rotor inlet at 300K and 60% RH and 1% at stage 3 stator inlet

Case 5: 2% spray at stage 1 rotor inlet at 300K and 60% RH and 1% at stage 4 rotor inlet

Myth 1: Fogging or overspray will increase air density in the compressor

The main reason for reduced compressor work under fogging condition is due to the increase of air density (or reduction of specific volume, v), as shown in Fig. 2. This idea is intuitive because the air density increases as fogging or overspray cools the air. However, it is unexpected to see from the analysis that air density actually decreases in part of the compressor for overspray cooling, although the saturated fogging (without overspray) does increase air density throughout the compressor. For example, the data shows that the density unexpectedly decreases in interstage overspray cases. A further investigation reveals the following reason: When overspray is applied, the temperature drops significantly due to water evaporation. This excessive temperature reduction results in a significant reduction in pressure. Pressure usually reduces more than the temperature as can be seen in the polytropic relation that $P^{-1}T^{k/(k-1)}$ = Constant, i.e. $P \propto T^{k/(k-1)}$. For example, take k = 1.38 for humid air; k/(k-1) = 3.63, which means that, if the temperature reduces 10%, the pressure will reduce about 30%. Based on the ideal gas law $\rho \sim P/RT$, the density is, therefore, reduced instead of increased. Although the air receives more water vapor when water droplets vaporize, the slightly increased density due to water evaporation is not large enough to compensate for the density reduction due to temperature-induced pressure reduction. This density variation trend due to overspray fogging is also in agreement with the compressor loading behavior shown in the results of White and Meacock [8] and Roumeliotis and Mathioudakis [23] although no density data is shown on each stage. White and Meacock [8] showed that increase in mass flow at the new operating condition results in an unloading (i.e., higher flow coefficient) for the first stage. Consequently, the pressure and density increase are less than their design values, thereby giving an initial increase in axial velocity and the observed rise in the flow coefficient over the first few stages. Eventually, however, the effect of evaporative cooling takes over in such a way that the density rises and the flow coefficient falls below the design value in the latter stages, which is consistent with the density change trend shown in Fig. 5. Therefore, the myth is true for saturated fogging, but not always valid for overspray.

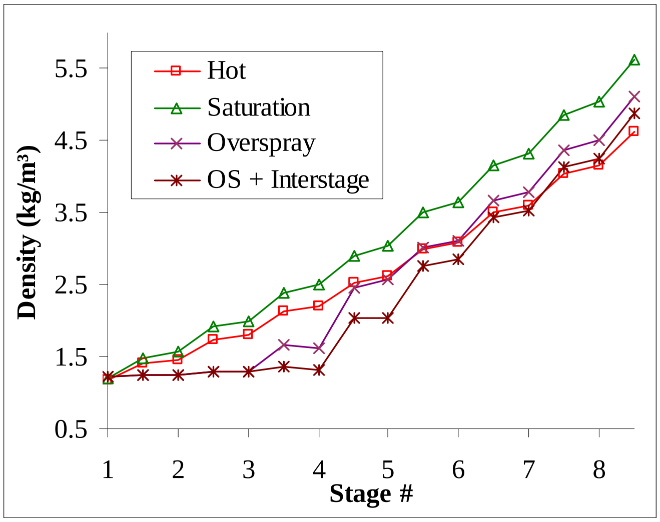

Figure 5 Density variation in a 8-stage axial compressor with fogging/overspray [22].

Myth 2: Fogging or overspray will reduce the compressor power consumption.

Increased compressor power consumption is the most important issue when the ambient air becomes hot. Assuming an adiabatic condition, the specific compressor work can be calculated as

$w_c = h_2 - h_1 \approx Cp_a(T_2 - T_1)$ (kJ/kg) [If Cp is constant] (2)

$W_c = m\, w_c$ (kW),

Where h_1 = air enthalpy at compressor inlet

h_2 = air enthalpy at compressor exit

T_1 = CIT and T_2 = CET.

Following a typical analysis (e.g., Jolly [19]), it has been stated that the compressor specific work in Eq. (2) reduces because the compressor discharge temperature (CDT), T_2, decreases. This simple reasoning has the following flaws:

(1) When fogging is applied, T_1 is also reduced; so stating only that T_2 is reduced does not indicate whether w_c is reduced or not. The expression needs to be converted to $w_c = C_p T_1 [(p_2/p_1)^{(k-1)/k} -1]$ using the isentropic process.

(2) Only air (not a moisture-air mixture) is considered. A more precise equation is developed below to prove that more power is needed for humid-air compression.

(3) T_2 is not always reduced because the exit pressure usually increases after fogging, so T_2 could be either reduced or increased. This increased compression pressure ratio requires more specific compressor work.

More discussions of flaws 2 and 3 are presented below.

Flaw (2) – only consider air compression.

Assuming the compression work is dominantly consumed for compressing humid air and that the work for compressing liquid droplets is ignored, the compression work can then be derived by drawing a control volume surrounding only the humid air. Since there are millions of droplets in the compressor, all the liquid droplets can be considered collectively into one big droplet conceptually for the convenience of illustration as shown in Fig. 6. Applying the energy equation on the humid air control volume, the compressor power can be represented as Eq. (2). Since the control volume is applied on the humid air only, the heat transfer across the boundary between the droplet and humid air needs to be included. The detailed heat transfer between liquid droplets and the air is quite complicated, involving three typical steps: (a) The droplet is heated up by the air; (b) a thin layer of liquid evaporates; (c) both the air and liquid is cooled by supplying evaporation energy (latent heat); (d) the saturated water vapor near the droplet is transported away from the liquid surface; and (d) this cycle continues until all liquid evaporates.

$$W_c = (h_2 - h_1)_{\text{humid air}} - q_{in} + q_{out} \quad (3)$$

$$W_c = (h_{a2} - h_{a1}) + v(h_{v2} - h_{v1}) - \Delta f(h_2 - h_{sat}) - \Delta f\, h_{fg}$$
$$\approx Cp_a (T_2 - T_1) + v\, Cp_v (T_2 - T_1) - \Delta f\, Cp_v (T_2 - T_{sat}) - \Delta f\, h_{fg} \quad (4)$$

where Δf = change in liquid water (due to evaporation) fraction.

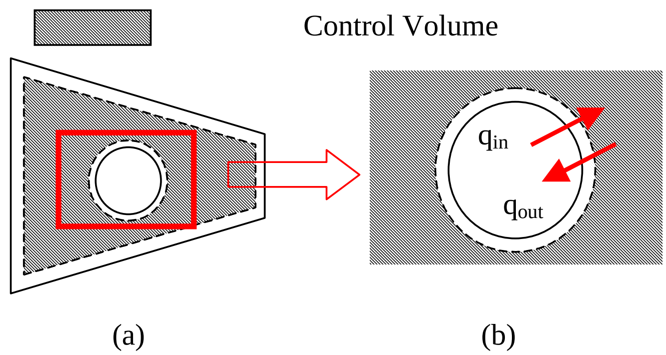

Figure 6 Control volume for (a) dry compression and (b) wet compression.

On the right hand side, the 1st and 2nd terms indicate the increase of sensible heats of dry air and existing water vapor, respectively. The 3rd term is the sensible heat of newly evaporated water vapor, and the last term is the energy extracted from the humid air to supply evaporation heat. All the terms on the right hand side are positive. Note that the 3rd and 4th terms are also both positive because Δf is negative due to evaporation. Therefore, the specific compressor work calculated by Eq. (3) is higher than that obtained from Eq. (2). An equation similar to Eq. (4) has been shown in many papers, such as in [8]; but the hand-wave explanation of wet compression using Eq. (2) has also appeared often in papers and brochures.

Flaw (3) – Compression ratio usually increased with fogging/overspray.

Figure 1 shows a traditional p-v diagram for an ideal Brayton cycle. If the inlet temperature is cooled from state 1" to 1, the compressor specific work can be qualitatively shown as the area enclosed by the curve and the ordinate axis and is reduced from area 1'-2'-b'-a' to area 1-2-b-a. Indeed many papers have shown that fogging/overspray reduces compressor specific work [24, 6, 7, and 19]. However, the analysis of [22] showed that (a) under saturated fogging conditions, the compressor specific work (kJ/kg) does decrease, but the compressor power (kW) increases; and (b) under overspray condition, both the compressor specific work and compressor power increase (not decrease). The increased compressor power could be explained by the increased mass flow rate due to fogging, but the question of what causes the increase in specific work, which is independent of mass flow rate, remains. Thus, there appears to be a discrepancy.

This discrepancy can be explained in the following way: the theoretical GT cycle diagram shown in Fig. 2 is plotted by assuming the compression pressure ratio maintains at a constant value, whereas pressure ratio always increase in the fogging condition. To reflect this increased compression pressure ratio under fogging, Figure 2 is revised as Fig. 7 to show a theoretical representation of the process of dry and wet compression on a p-v diagram. It shows that compressor specific work (Total power is divided by total mass, where total mass = air mass + water mass) actually increases due to the extra area enclosed by the additional compressor discharge pressure for wet compression. If this additional power for pressure increase is more than the savings due to overspray, the total compressor power will increase. This happens for wet compression cases where there is overspray (i.e., liquid water is carried through the compression after inlet evaporation), but the compressor power decreases if there is no liquid water left after saturation at the inlet. Note that the p-v diagram shows the specific work, which is independent of the mass flow rate, so the added mass due to fogging does not enter into the discussion here.

Overspray causes an increase in work, and an interstage spray requires more specific compression work. An interstage spray reduces the temperature and hence the pressure (path c-d shown in Fig. 7), which introduces recompression and increases the specific compressor work. This recompression area is shown by the trapezoid in Fig. 7. The difference of specific work and power is revealed in Table 1. For the saturation fogging case, the specific work (270.2 kJ/kg) is less than that of the hot case (272.6 kJ/kg), whereas the total power for saturation fogging case (9.172 MW) is more than that of hot case (7.936 MW) due to increased mass flow rate. Since more power is used to produce more pressure ratio, a fair way to compare the compressor performance is to compare the compressor power per unit increase of pressure ratio (W_c/P_r) in Table 1: the smaller the better. As it can be seen, saturation fogging case predicts more efficient compression with W_c/P_r = 825 kW than the hot case with W_c/P_r = 972 kW and the overspray case with W_c/P_r = 944 kW. This fact reveals that the saturation case, which needs less power for high compression ratio, is the most effective one.

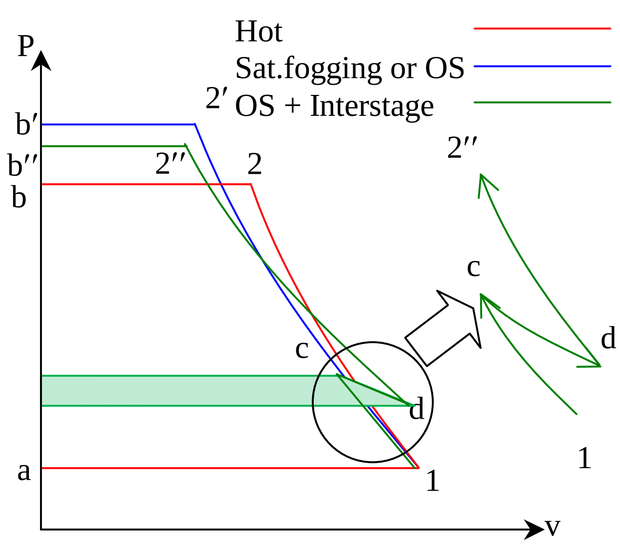

Figure 7 Theoretical representation of overspray and/or Interstage spray cases. The shaded area represents the double compression work due to the interstage spray.

It can be said that if the OS case and the “dry” cases are compared for the same pressure ratio the specific work decreases with OS. Therefore, the second myth that fogging or overspray will reduce the compressor power consumption is not always valid in terms of compressor power or specific work, but <u>it is always true in terms of compressor power per unit pressure ratio.</u>

Myth 3: Fogging or overspray will noticeably increase the gas turbine efficiency.

By ignoring many non-essential parasitic loads, the GT thermal efficiency can be defined as

$$\eta = \frac{W_{net}}{Q_{in}}, \tag{5}$$

where $W_{net} = W_t - W_c - W_{fc}$ and Q_{in} = Heat input in combustion chamber.

Table 1 Overall compressor performance and net gas turbine output power

Cases	Mass flow rate (kg/s)	Specific Work (kJ/kg)	Comp. Power, W_c (MW)	Press. Ratio (Pr)	W_c/Pr (kW)	Net GT Output (MW)
Hot	25.77	272.6	7.026	7.23	972	7.936
Sat.	26.19	270.2	7.093	8.59	825	9.172
OS	26.19	297.6	7.948	8.42	944	8.674

It is clear from Eq. (5) that the thermal efficiency of a gas turbine system is dependent on 4 major parameters, including power produced by turbine, power consumed by compressor, power consumed by fuel compressor, and heat input in combustion chamber. All these parameters are dependent on the CIT. During the fog cooling, CIT decreases, which raises the W_c (due to increased mass flow rate, see Table 1) but reduces CET, which needs more fuel; therefore, Q_{in} and W_{fc} increase. For overall fog cooling, W_t also increases. Because all parameters increase without linear proportionality, the qualitative efficiency prediction contains uncertainty without plugging in all real numbers.

Utamura et. al. [25] showed that 1.84% efficiency increase to a 0.65% water (Overspray) injection was observed under the condition of 29.7°C, 70% R.H. using F9E simple cycle machine (130MW). Lecheler and Hoffman [26] found 1.8% efficiency increase to a 1% overspray @ ISO condition using GT24/26 simple cycle. Deneve et. al. [27] showed 0.9% efficiency increase to 2% overspray water injection using Swirlflash™ wet compression system on a 150MW heavy-duty gas turbines.

Taking the data from Khan and Wang [9], a natural gas fired gas turbine for different ambient conditions is plotted in Fig. 8, where pressure ratio (=12) and TIT (=1400K) are fixed. Figure 8 shows that the efficiency monotonously decreases slightly as overspray increases at T_{amb} = 288.2K, whereas, when T_{amb} increases to 313K, the thermal efficiency increases slightly instead of decreasing as fog overspray increases. This reversing trend of thermal efficiency indicates that applying overspray is more efficient on hotter days. Since the thermal efficiency may slightly increase or decrease under fog/overspray conditions, considering the uncertainty of the current ideal model, fog/overspray should be considered as a means to considerably augment power output, but not necessarily efficiency.

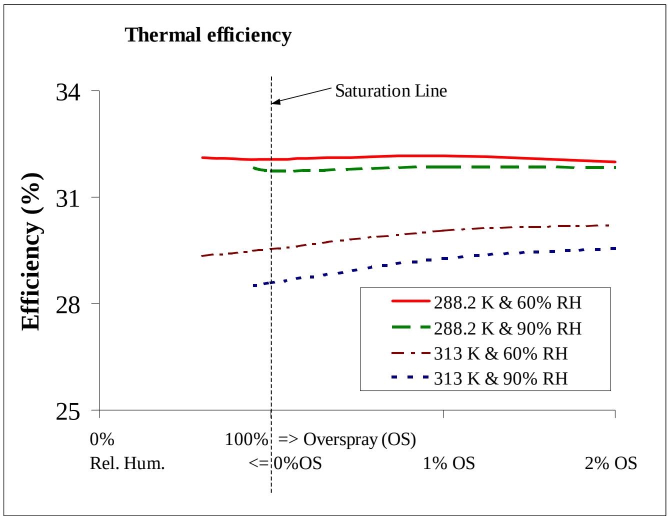

Figure 8 Thermal efficiency variations under different ambient conditions

Discussion of effect of centrifugal force on droplet evaporation

One common question related to wet compression is whether the centrifugal force or weight of the droplet affects the droplet evaporation.

An estimate of the importance of centrifugal force or gravity can be analyzed by conducting the order of magnitude analysis in comparison with the axial momentum of the droplet.

The rate change of axial inertia momentum of the single droplet is

$$F_I = \rho V^2 d^2 \tag{6}$$

Weight of a single droplet is given by

$$F_W = mg = \pi\rho d^3 g/6 \tag{7}$$

Centrifugal force exerted by a single droplet is obtained from

$$F_C = mU^2/(D/2) = \pi\rho d^3 U^2/(3D) \tag{8}$$

The translational velocity, $U = \pi DN/60$, putting this value into Eq. (8) yields

$$F_C = \pi^3\rho d^3 N^2 D/10800 \tag{9}$$

The ratio of weight to momentum is

$$\frac{F_W}{F_I} = \frac{\pi\rho d^3 g}{6\rho V^2 d^2} = \frac{\pi d g}{6V^2}\ . \tag{10}$$

Taking some typical values, d = 10μm, V = 90m/s and putting these values into Eq. (10), the ratio between the weight of the droplet over axial flow momentum is small, indicating a negligible effect of gravity.

$$\frac{F_W}{F_I} = \frac{\pi \times 10^{-5} \times 9.81}{6 \times 90^2} = 6.34 \times 10^{-9}$$

The ratio of centrifugal force to axial momentum is

$$\frac{F_C}{F_I} = \frac{\pi^3 \rho d^3 N^2 D}{10800 \rho V^2 d^2} = \frac{\pi^3 d N^2 D}{10800 V^2} \tag{11}$$

Taking some typical values as d = 10μm, N = 3600RPM, D = 0.81m, V = 90m/s and putting these values into Eq. (11), the ratio of centrifugal force to axial momentum is very small, indicating that the effect of centrifugal force if negligible.

$$\frac{F_C}{F_I} = \frac{\pi^3 \times 10^{-5} \times 3600^2 \times 0.81}{10800 \times 90^2} = 3.72 \times 10^{-5}$$

The conclusion of this estimate is consistent with CFD animation associated with [28]. No clear trace of droplets excursing toward the tip of blades is observed in the rotating frame simulation.

Discussion of whether water droplets would survive in the compressor and enter the combustor

The question of whether water droplets would survive in the compressor and enter the combustor has been a concern. The answer has always been straightforward in a qualitative way: "Small droplets all evaporate, but large droplets may not." However, it has been uncertain for quantifying the droplet initial sizes that would enter the combustor. In reality, there shouldn't be a uniform answer to this question because the droplet evaporation rate depends on many factors, including compressor pressure ratio, stage number, loading factor, airfoil geometry, droplet initial temperature, surrounding air temperature, droplet size, RH of surrounding air, etc. The state of the air inside the compressor is dynamic, not static, as the temperature and pressure are constantly changing during compression. In general, if the time required for water droplet evaporation is shorter than the droplet residence time in the compressor, the wet compression ensures complete evaporation of droplets; otherwise, the droplet would not evaporate completely.

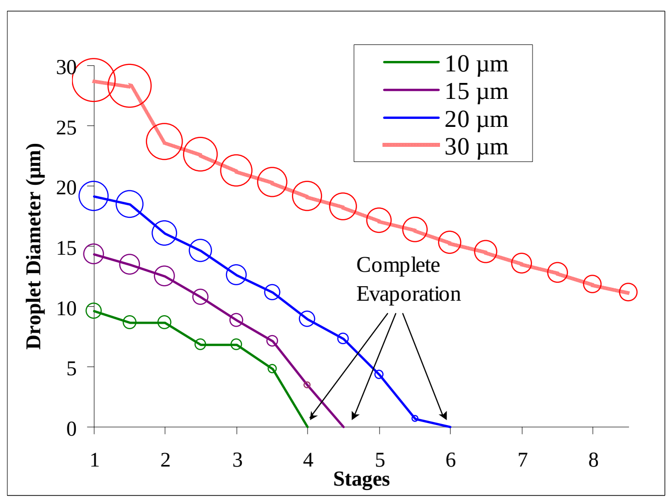

Figure 9 Reduction of droplet diameter for non-equilibrium cases

For example, White and Meacock [28] evaluated that 55% of the water will leave the compressor unevaporated assuming spray of a droplets with Sauter mean diameter of 12um. As another example, Khan and Wang [29] employed a non-thermal-equilibrium stage-stack method through an 8-stage axial compressor with a dry-compression pressure ratio of 8 to investigate at what initial droplet size a droplet can survive the wet-compression journey in the compressor. The ambient condition is taken as 300K and 60% relative humidity. The initial droplet temperature is 288K. Four different droplet sizes were studied: 10μm, 15μm, 20μm and 30μm. The results in Fig. 9 show that the droplets of 20μm or smaller complete evaporation, but the large 30μm droplet escapes from the stage compressor at about 12 μm in diameter. Although the results can vary depending on initial and boundary conditions, most of the fogging devices produce droplets with an average size less than 10μm. Therefore, it is fair to conclude that the majority of the droplets will evaporate in a typical fogging and GT operating condition.

SUMMARY

Some myths and specific features associated with gas turbine inlet fogging and wet compression were discussed. A common mistake in deriving the compressor work was briefly discussed. The following epitomizes the key points:

1. A comprehensive multiphase T-s diagram showing wet compression process is created.
2. Entropy change for fogging and overspray due to liquid evaporation is about 2½ times that for the hot case, which can degrade the aerodynamic performance during wet compression and counteracts a portion of the advantage of lowered temperature compression.
3. Myth 1, which states that fogging or overspray will increase air density in the compressor, is true for saturated fogging; but it is not always valid for overspray.
4. Myth 2, which states that fogging or overspray will reduce the compressor power consumption, is not always valid in terms of compressor power or specific work; but it is always true in terms of <u>compressor power per unit pressure ratio.</u>
5. Myth 3, which states that fogging or overspray will noticeably increase the gas turbine efficiency, is not always true. Therefore, fogging or overspray should be considered as a means to augment power output, but not necessarily efficiency.
6. A simple, common explanation of reduced compressor work due to fogging is misleading, and a corrected formulation and explanation have been presented.
7. The centrifugal force has a negligible effect on droplet dynamics or wet compression efficiency.
8. There is no definite answer to whether the water droplets would survive the wet compression in the compressor and enter the combustor. A rule of thumb shows droplets less than 20μm in diameter will completely evaporate during a compression with a pressure ratio above 8.

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