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A NEW OVERALL APPROACH TO QUANTIFY WET COMPRESSION IN APPLICATIONS LIKE GAS TURBINES

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ABSTRACT

Wet compression is well known in the gas turbine industry. It may be called “high-fogging” or “over-fogging” too. The idea is to inject an evenly distributed cloud of fine water droplets into the compressor inlet of a gas turbine. There are many publications on the design and on the aerodynamic and thermodynamic effects in gas turbines with wet compression. The method is commercially used for augmenting the power output of open gas turbines and combined cycles.

A first step of the approach of this paper is the precise calculation of the water injection effects on the compression using the assumption of equilibrium (this means saturated gas phase and equal temperatures of droplets and gas) and maintained polytropic efficiency of the compressor. This assumption constitutes the best possible case. In a second step, a reduction of the polytropic efficiency dependent on the water injection rate is applied to calibrate the GT performance results to published power augmentation data of an OEM. The space between the two cases represents the opportunities to achieve with design improvements aimed at the wet compression process itself.

The above-mentioned precise calculation is a new method called “enthalpy based CDRA” for calculating the polytropic efficiency. CDRA stands for “Constant Dissipation Rate Algorithm”, which is applicable for any pressure ratio. The diagrams calculated in this paper aim at the typical performance changes of an H-class gas turbine and a derived combined cycle due to high fogging.

In most real applications, the over-fogging water to air ratio (by mass) is kept below 2%. Our diagrams show values up to 6%. The range above 2% is an extrapolation involving additional challenges.

We finally review some important additional aspects of wet compression in an extra chapter.

INTRODUCTION

The injection of fine water droplets into the compressor inlet of GT’s has a history since the 1940’s as a method to augment the GT power output. There are many publications describing this history and therefore only a few basic ones are mentioned here [1]…[7] without claim of completeness.

In the case of GT power augmentation the water to air ratio is kept typically up to around 2%. Already this leads to a considerable reduction of the compressor power and the compressor exit temperature. According to the authors own experience the development of these systems was initially driven by the fear that conditions far away from equilibrium will prevail up to the compressor discharge. However, the practical experience demonstrated a rather near equilibrium situation as shown in publications from Lecheler et al [3] or from Matz et al [5]. Comparing the real with the ideal power gain as done below supports this too. Nevertheless, in practical cases a near-equilibrium evaporation is fostered by atomization to very small droplet size. Such systems are described in a fundamental paper of Mee et al [1]. Our focus here is an overall assessment of high fogging in GT technology without going into details of the loss mechanisms.

In another context the author has developed a straightforward algorithm for calculating polytropic changes of state in turbomachinery blading with given efficiency (or with given exit state). This algorithm called CDRA works accurately for any fluid including two-phase mixtures assuming thermodynamic equilibrium among the phases. It is usable for any pressure ratio. Reference [9] shows it for wet steam and reference [10] for air and combustion gases. It is a nearby idea to calculate the thermodynamic cycle (GT in simple or combined cycle) with wet compression using the CDRA method.

This method allows determining the performance of a multistage compressor with high fogging assuming inlet conditions, polytropic efficiency and pressure ratio only. Assuming the same polytropic efficiency as determined before for the single-phase fluid results in the best possible performance for wet operation. Modifying the assumed polytropic efficiency and/or the turbine inlet temperature of a GT dependent on the water injection share allows calibrating the real cases based on real engine measurements.

Therefore, in a first version, equilibrium condition will be assumed and a liquid water injection rate of up to 6% will be considered. In a second version a realistic prediction for power augmentation in GT’s and GTCC’s is given based on OEM published achievable data from Lecheler et al. [3], Matz et al. [5] and Savic et al. [7].

For injecting (intentionally) more than 2% water, no published experience from land-based GT's was found. Therefore, the performance predictions for up to 6% water injection rate have to be considered as extrapolations. The geometric arrangement of the water nozzles for y>2% in the inlet plenum while ensuring an even distribution and sufficient atomization represents a challenge.

To illustrate the upper limits of water injection we may look at the experience with turbofans in heavy rainfall condition. The European Aviation Safety Agency EASA requests a validation test with 4% liquid water content (LWC, by mass) at the inlet as indicated by Staudacher [11]. This publication contains also other interesting investigations on the atomizing and distribution effects of large droplet impact on the fan blades and the corresponding walls. Dunham showed 1987 that even the (by then) strongest observed thunderstorm rain does not constitute more than around 3% LWC in the air [12]. The EASA requirement is therefore on the safe side, but it is also an indication that 4% LWC at the fan inlet does not really represent a problem for the typical axial compressor and combustor design although we must assume that the core engine sees less than 4% LWC due to the centrifugal separation effect.

There are also publications on cases with even higher water content. Examples are from Zhang [13], which includes spray intercooling or a test by Ferrara and Bakken without complete evaporation for up to 30% LWC [14]. We do not consider inter-stage fogging here because there are rarely commercial applications yet. However, in connection with or by wet intercooling this could become an interesting option too.

Another preliminary note is that the OEM's typically give performance guarantees for a "new and clean" condition of the GT or GTCC. After some operating time of a GT typically between 2 and 5% of power and thermal efficiency is lost due to fouling and ageing (i.e. because of clearance changes by wear).

ABRIDGEMENT OF THE ANALYSIS METHOD

The methods described below may find its application within condition monitoring or preliminary design as well as in considerations on new cycle concepts. Our method has a predecessor described by Hoffmann and Collins [15]. They calibrated their performance model with measured data from a GT operated with high fogging. Unfortunately, their paper suffered from the censoring practice of an OEM, which typically removes the numbers from the scales in the graphs. Here we avoid this problem by using a notional reference GT.

Methods for calculating performance data of GT's exist in many variants and with different levels of detail. There are many publications and commercial software on this. We use here a relatively simple but realistic model, which has been tested with commercial data and is described by the author for the evaluation of commercial GTCC's [16]. Compared to this model the calculation of the changes of state in compressor and turbine blading has been replaced by the Constant Dissipation Rate Algorithm (CDRA) as described by the author in [9]. This allows a convincing evaluation of wet compression assuming thermodynamic equilibrium between the liquid and the gaseous phase.

The CDRA requires a specific volume function v(p,T), which is based on a real gas formulation for the fluid in question. The author has described a first CDRA version in [10]. It is applicable for all kind of real gases (single phase mixture) and it is used here for the turbine blading. A second version [9] allows application to homogeneous mixtures of two phases in (thermodynamic) equilibrium. In its core, the specific volume function uses the specific enthalpy of the fluid mixture instead of the temperature v(p,h). Corresponding two-phase fluid libraries are available and no additional argument is needed to describe the thermodynamic state of the fluid. This second version, also called "enthalpy based CDRA" is used for the compression across both the wet and the dry sections.

Both CDRA versions use a calculation of the change of state in small increments. For the typical pressure ratio in question we used 100 steps (with equal pressure ratio in each step) according to the accuracy recommendations as described in the before mentioned publications.

From the air, the gases N2, O2, Ar, and CO2 have been considered in the fluid formulation according to the libraries of Zittau [17]. These formulations follow a guideline of the VDI [18] and the overall calculation has been programmed with Mathcad [19]. For water and steam, a steam table from Zittau is used as well. As fuel pure methane with a LHV of 50 MJ/kg is used following typical commercial performance catalogue indications for GT's and corresponding to the ISO standard for GT acceptance tests [21]. The methane gas data are taken from the mentioned Zittau libraries. The absolute water content (by mass) in the humid air at the compressor inlet is designated with x (%) and the injected over-fogging water related to the dry air inlet mass flow rate with y (%). The increase of the pressure ratio of the GT due to the change of the mass flow rate and composition by the water injection (y) is considered corresponding to the "Kegelgesetz", which is sometimes called Stodola law.

Additionally we apply a reduction of the turbine hot gas inlet temperature of 3K per % of injected high fogging water in both the ideal and in the calibrated case. The reason is the deteriorating effect of the steam in the flue gas to the materials used in the hot section. This is an arbitrary setting due to lack of published information, which influences the GT thermal efficiency marginally (-0.07%-points for y=0.06).

For the overall heat balance of the GTCC, also the HHV of methane is used with a value of 55.5 MJ/kg. The heat balance was checked for all calculated cases without further mentioning.

The bottoming steam cycle is not modelled in detail. Its net power is simply determined as 72% of the GT's exhaust gas exergy. This is an "exergetic efficiency" typically achieved by H-class steam bottoming cycles according to the author [16]. John Gulen gives a comprehensive overview of such exergetic efficiency data of bottoming cycles based on commercial catalogue indications i.e. in figure 4 of [22].

REFERENCE CASE IN H-CLASS

For demonstrating the calculation, we have chosen reference data for a notional large H-class GTCC. The performance relevant assumptions in Table 1 are based on the evaluation of commercial catalogue data by the author according to methods described in [15]. The reference case intentionally does not correspond to an existing commercial GTCC, because later not only relative but also absolute performance data are presented.

The yellow highlighted data in table 1 are derived based on the analysis with the given assumptions above.

Table 1: *Data of the reference GT, chosen based on commercial H-class performance level Reference GTCC data, evaluated at ISO ambient conditions*

GT pressure ratio (compressor exit plenum to ambient)	20
Compressor inlet mass flow rate at ISO cond. (kg/s)	600.0
GT combustor outlet temperature (hot gas) (°C)	1500.0
GT polytropic compressor blading efficiency	91.2%
GT polytropic turbine blading efficiency	87.0%
GT relative inlet pressure loss	1.00%

GT relative pressure loss between compressor and turbine blading	6.50%
GT relative outlet pressure loss (includes loss of heat recovery steam generator)	3.80%
GT lower heating value of fuel gas (MJ/kg)	50.000
GT fuel gas to GT delivery temperature in °C	15.0
GT cooling and leakage air ratio of turbine vane 1	9.00%
GT cooling and leakage air ratio downstream of vane 1	11.00%
GTCC assumed net use factor of the GT exhaust exergy	72%
GT mechanical efficiency	99.5%
GT generator electric efficiency	98.6%
GT power share of auxiliary losses (simple cycle)	0.220%
GT ambient temperature at ISO condition [20] (°C)	15.0
GT ambient pressure at ISO condition [20] (bar)	1.01325
GT ambient relative humidity at compressor inlet at ISO condition [20]	60%
GT thermodynamic power at ISO cond. (MW)	284.0
GT thermodynamic efficiency at ISO cond.	41%
CC net combined cycle efficiency at ISO cond.	60.00%
CC net combined cycle power at ISO cond. (MW)	416.6
Compressor exit temperature (°C)	449.1
GT ISO mixed turbine inlet temperature (°C)	1321.5
GT vane outlet temperature (°C)	1393.8
GT average exhaust temperature (°C)	610.7
GT compressor thermodynamic power (MW)	270.0
GT turbine thermodynamic power (MW)	550.5
GT exhaust exergy export to the steam water cycle (MW)	197.3

WATER EVAPORATION ASSUMING EQUILIBRIUM

The figure 1 shows how the liquid water content in the compressor flow drops as a function of the achieved pressure ratio. This applies for a compressor according to the reference data assuming thermodynamic equilibrium and maintained polytropic efficiency along the compressor blading.

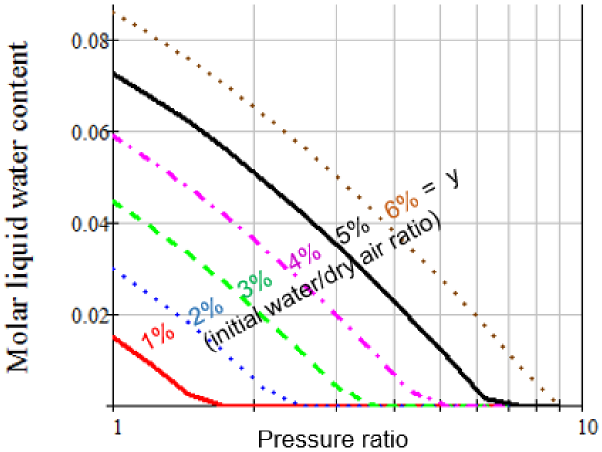

Figure 1: *Molar liquid water content for wet compression starting at 100% relative humidity and 1.013 bar pressure as a function of the pressure ratio with ηpol=0.912.*

Where the curves hit the zero line is the first possible pressure at which all water is evaporated. The curve parameter is the water injection rate y increasing from bottom to top from 1% in steps of 1% to 6% (by mass). In reality, water droplets proceed further downstream in a non-equilibrium condition. In existing GT's with 2 % high fogging water injection the last traces of liquid water were typically observed in the 7th stage. This corresponds to a pressure ratio of close to 5 and the total water content there corresponds to a relative humidity of around 7%. For our calibrated case with 6% high fogging water injection (y=6%, maximal used value) the relative humidity at the compressor discharge (pressure ratio = 20) was evaluated with 7.4%. Figure 1 represents the equilibrium case, which is the limit of the earliest possible evaporation during compression. Causes for then non-equilibrium case like droplet coagulation, film formation and general material exchange theory are subject of numerous specialized publications. We intend to avoid this discussion in favour of the overall calibration shown below.

RELATIVE RESULTS ASSUMING EQUILIBRIUM

Figure 2 shows the equilibrium results. The first three calculated points in this and all later figures until figure 11 correspond to the following inlet conditions with all at 15°C ambient temperature and stagnation condition:

1. Pure air with zero humidity
2. ISO conditions (60% relative humidity)
3. 100% relative humidity

The other points apply for the water injection rate y as indicated on the x-axis. The kink in the curves indicates the transition to over fogging, characterized by water evaporation within the compression process.

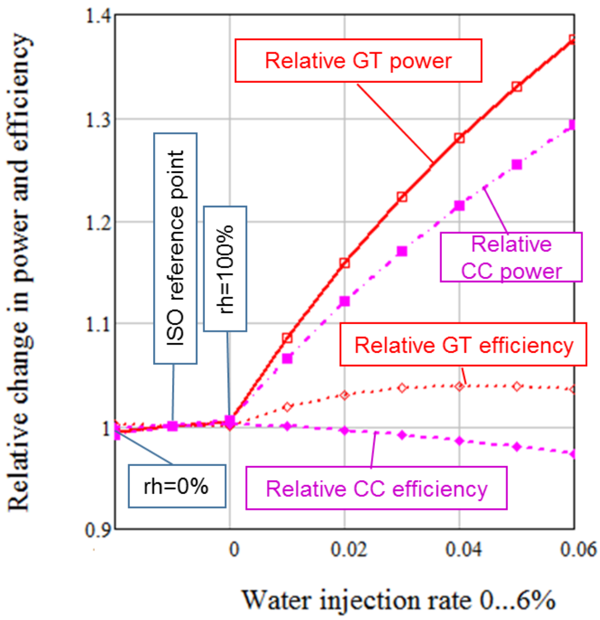

Figure 2: *Relative performance changes with wet compression of GT and GTCC for equilibrium conditions and maintained polytropic efficiencies (best case).*

EMPIRIC CALIBRATION TO MEASURED ENGINE CONDITIONS

Wet compression in axial compressors is combined in reality with some evaporation delay, with droplet coagulation and potential forming of liquid films at the walls. Many studies aim at this issue. It influences the aerodynamic performance of the blading. Eisfeld mentions a drop of aerodynamic efficiency in a subsonic range [6] and Zhang claims a gain in aerodynamic efficiency in a transsonic condition [20].

Large H-class GT's have a transsonic front portion of the compressor and subsonic conditions in the rear stages. The water is injected into the first stage and may be partly or completely evaporated in the downstream subsonic part.

In order to predict the achievable performance data in front of this complex situation we simplify it by reducing the polytropic compressor efficiency to meet the power augmentation promises given for commercial GT systems. Lecheler [3], Matz [5] and Savic [7] indicate such numbers. They indicate concordantly a relative power augmentation of 7% per % of injected high fogging water (y=1%) for the GT.

In order to match this 7% power gain per % injected water the empiric correction of the polytropic efficiency of the compressor blading results in a reduction by 0.68% points for every % injected water with linear dependency up to 6% water injection. This is used below. The linear dependency was chosen as a first extrapolation into the "unknown". In other words, the calculation model assumes again compression in equilibrium condition, but with a reduced polytropic efficiency.

The polytropic turbine blading efficiency is kept constant independent of the water injection rate. We assume that the turbine design point is selected within the mass flow rate and composition range of high fogging.

The results gained are best for the GT's measured, but also applicable for any comparable GT. If a user of the described method has access to measured data of his own GT, he can calibrate with his own data.

RELATIVE RESULTS WITH EMPIRIC CALIBRATION

Figure 3 shows the calibrated results. The difference between Figures 2 and 3 represents the cumulated effects of non-equilibrium evaporation, aerodynamic effects of the wet compression and others. This difference is subject of most publications dealing with wet compression.

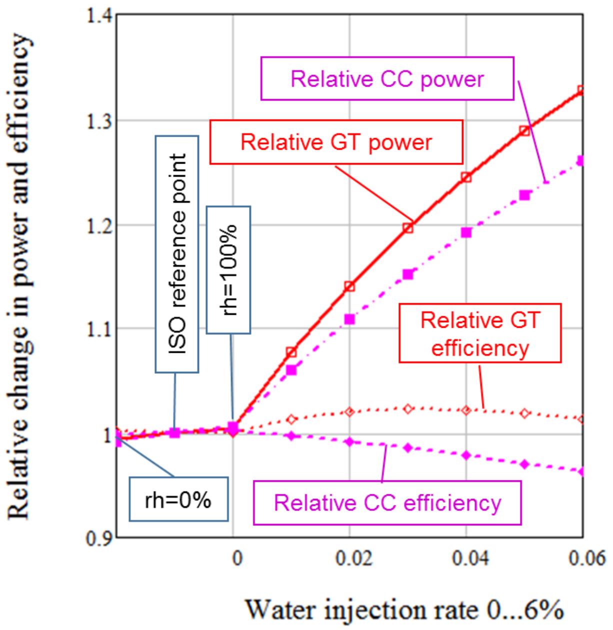

Figure 3: *Relative performance changes with wet compression of GT and GTCC assuming empiric calibration.*

COMPARISON IDEAL CASE WITH EMPIRIC CALIBRATION

The following diagrams all show absolute results for the ideal case (with maintained polytropic efficiencies) and based on the above described empiric calibration. The calibrated results for high fogging water injection rates above 2% are extrapolations without an experimental backing. They give a hint for quantifying the main effects. The GT's used for the empiric calibration have compressors with their design point in dry operation. This evolved because high fogging was a typical later side application of GT's. All similar figures from 4 to 11 indicate between the "ideal curve" for the respective quantity and the empirically calibrated curve the space where more specific design solutions could end up. However, no one can be better than the indicated "ideal curve".

As explained before the first three calculated points of each diagram (figures 4 until 11) correspond to the normal operation while the others apply for the water injection rate y as indicated on the x-axis.

Figure 4 shows the comparison of the compressor power in the ideal case with maintained polytropic efficiency with the empirically calibrated case.

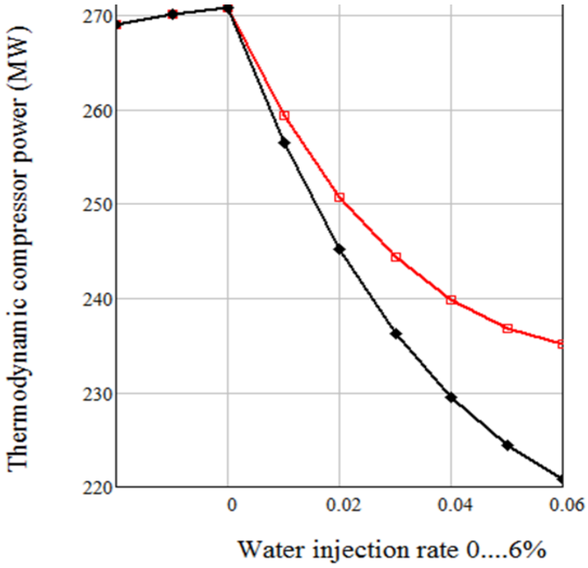

Figure 4: *Compressor power in the ideal case compared with the calibrated version as implemented in all other figures below. (Black diamond = ideal; red square = empirically calibrated).*

Lecheler [3] describes measurements of the compressor internal pressure and temperature pattern for y up to 1.2%. With high fogging, the pressure increase is typically shifted from the front stages to the rear stages of the compressor resulting in lower pressure of the intermediate bleeds. This means that the mass flow rate to compressor internal bleeds for the secondary air system (SAS) drops. Lecheler therefore compares variable and fixed SAS hardware. Hoffmann [15] gives more data from the same engine fleet. For higher injection rates, such effects intensify and must be considered.

Figure 5 shows the comparison of the net power output of the reference GTCC. High fogging is indeed an interesting business case because temporarily up to 10% excess power is available with y=2%.

According to figure 6 the thermal efficiency of a GTCC drops somewhat if the excess power is called by applying high fogging. At 2%, injection rate the efficiency difference between the ideal case and the empiric calibration is around 0.5%-points.

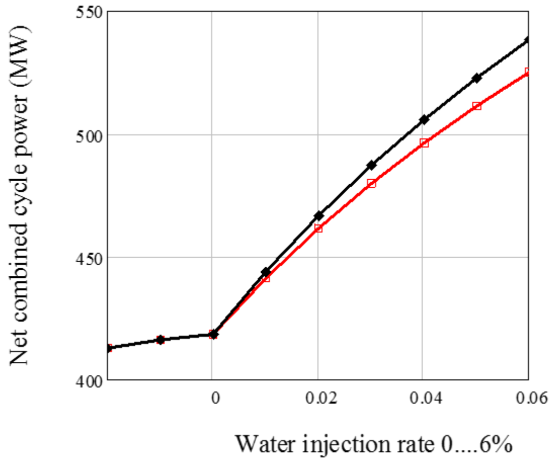

***Figure 5:** Net power output of a reference GTCC with wet compression (black diamond = ideal; red square = empirically calibrated).*

According to Figures 7 and 8, both the GT and the steam bottoming cycle contribute to the power augmentation by high fogging. However, the GT contributes more. Dependent on the GTCC arrangement either as a single shaft or with an extra steam turbine shaft the generators and transformers have to be checked for compliance with the power excess for the site-specific conditions.

Figure 8 shows that the power of the steam bottoming cycle does not depend on the GT compressor efficiency. It is the increased exergy export of the additional steam in the GT exhaust gas contributing to the power augmentation effect in the bottoming steam cycle.

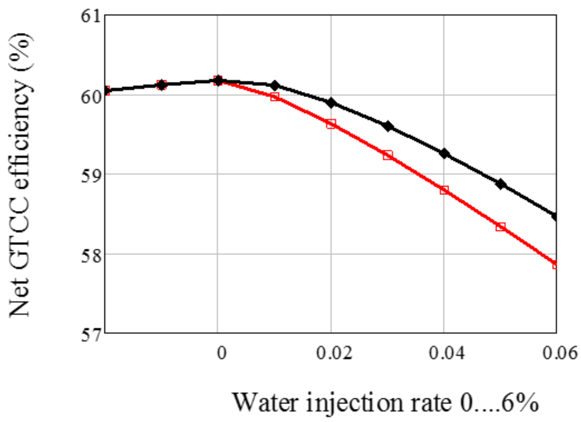

***Figure 6:** Thermal efficiency of a reference GTCC with wet compression (black diamond = ideal; red square = empirically calibrated).*

Figure 9 demonstrates a gain of the thermal GT efficiency for high fogging injection rates up to 3% even in the empirically calibrated case.

Even in the ideal case (with constant polytropic efficiency), the efficiency drops above 4% injection rate. This is the cumulated effect of the (favourable) compressor power drop, the increasing exergy loss by combustion due to the (adverse) lower combustor inlet temperature and the reduction of the turbine hot gas inlet temperature of 3K per % of injected high fogging water, which was also applied in the ideal case. The latter influence is marginal and does not prevent the thermal efficiency drop caused by the lower combustor inlet temperature for higher injection rates.

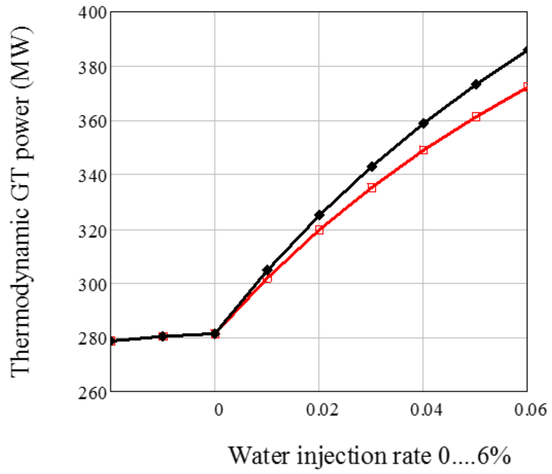

***Figure 7:** Thermodynamic GT power (black diamond = ideal; red square = empirically calibrated).*

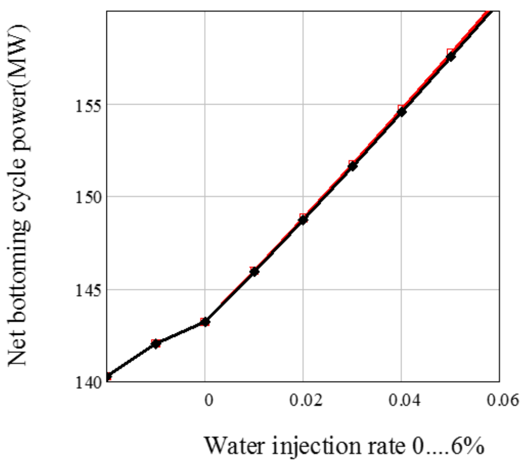

***Figure 8:** Net power of the steam bottoming cycle as a function of the water injection rate (black diamond = ideal; red square = empirically calibrated, both nearly identical).*

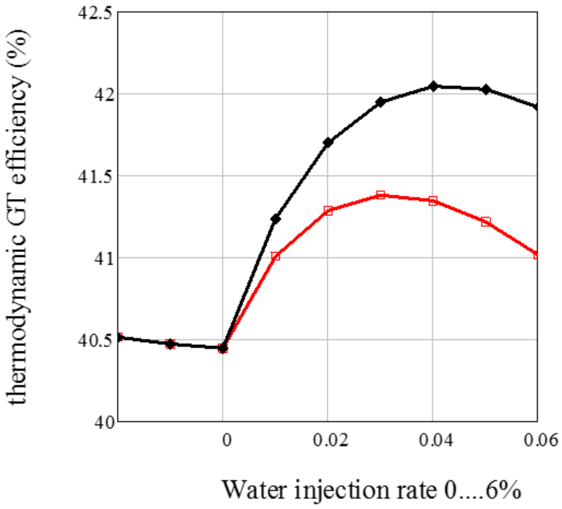

***Figure 9:** Thermal GT efficiency (black diamond = ideal; red square = empirically calibrated).*

The figure 10 shows the reduction of the compressor exit temperature by high fogging. This could become an opportunity for expanding the life of the rotor and other hot parts. However, the author does not know any OEM considering this in its hot part life predictions.

The lower compressor power consumption together with the increased pressure ratio (Stodola law) and with the additional steam content in the hot gas causing higher turbine power are the reasons for the GT power increase. In order to achieve nearly the same turbine inlet temperature also more fuel is required.

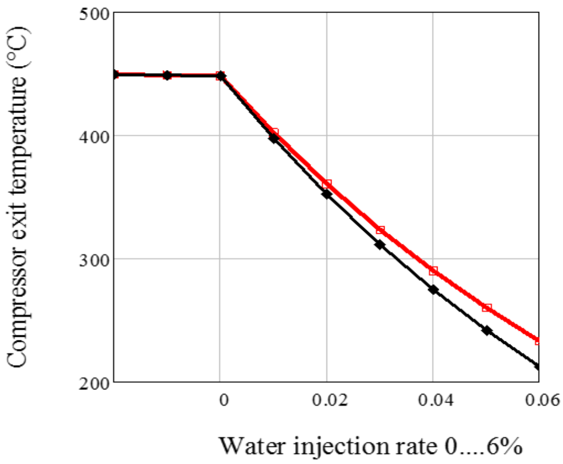

Figure 10: *Compressor exit temperature as a function of the water injection rate. (Black diamond = ideal; red square = empirically calibrated)*

High fogging reduces the GT exhaust temperature marginally according to figure 11. This is mainly due to the increase of the pressure ratio due to the higher mass flow rate while the polytropic efficiency of the turbine blading is assumed unchanged. In addition, the changed fluid composition is considered here. Additionally the correction of the hot gas temperature has been applied to both the ideal case and the calibrated one. This explains the marginal difference of the two curves.

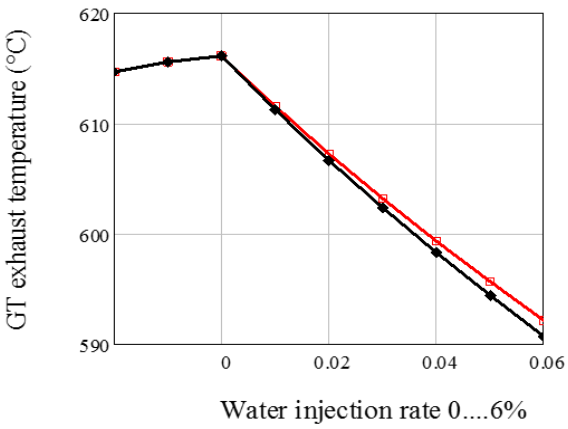

Figure 11: *GT exhaust temperature as a function of the water injection rate (black diamond = ideal; red square = empirically calibrated)*

The effect of the higher mass flow rate and steam content in the flue gas also outbalances the exergy loss due to the lower GT exit temperature. The figure 8 shows the corresponding increase of the steam bottoming cycle output.

MORE ASPECTS OF WET COMPRESSION

In the following chapters, we indicate some practical features of wet compression by referring to corresponding references containing more details.

COMPRESSOR ISSUES

Uneven circumferential distribution of the water fog injection can cause undesired compressor blade excitation and/or influences the surge limit. The allowable limits need being respected, which may require in some cases validation measurements and continuous monitoring. Lecheler [3] and later publications comment on details. The experience record of the involved OEM plays an important role for these issues.

DROPLET EROSION AND CORROSION

Erosion and corrosion is an issue in some cases. Some compressor designs suffered from corrosion fatigue with occasional blade failures. Droplet erosion attacks especially the high stressed leading edges of the airfoils and this limits the application of protective coatings. The OEM's responded differently to these challenges. While one OEM simply banned high fogging operation, others allowed it for long-term operation with some limitations for the compressor airfoil lifetime [7]. A point to consider is that the "primary" droplets are small enough to limit erosion, but secondary droplets formed by water films leaving the trailing edges of the airfoils are larger and cause more erosion. However, we also learn from Staudacher [11] that large droplets are atomized in a compressor blading during their next impact to small droplets as desired. There are many also theoretical publications on erosion as shown with many references by Kamkar et al. [23].

PROCUREMENT OF MAKE-UP WATER

High fogging requires demineralized water and this should be available at low price. The pumping power for the atomization is rather limited. Lecheler [3] and other publications describe corresponding systems.

In arid areas, high fogging of GT's may not be economic due to the procurement of the required water. However, technologies like Closed Cycles described by Frutschi [24] or the Semi-Closed Recuperated Cycle described by the author [25] condense (combustion) water out of the flue gas. Developing such systems represents an opportunity for future.

EXPANSION OF POWER AUGMENTATION TO HIGHER WATER INJECTION RATIO

The main obstacle to a higher LWC than two % in existing GT's is not the evaporation but the rear shift of the pressure gain in the compressor ("hanging" pressure build up"). This affects the discharge flow rates into the secondary air systems (SAS), which supply the open cooling systems of the turbine blading. The surge limit is influenced too. This is explained by Lecheler [3] and Hoffmann [15] based on their GT with design point in dry operation.

However with a design point of compressor and the SAS around LWC=2% the operating range could be expanded to i.e. LWC = 0…4%. If considered already in the design state of the compressor and the GT, the assumed deterioration of the polytropic compressor efficiency with the high fogging water injection rate can be limited. Thus, an even higher thermal efficiency than indicated in figure 9 can be expected in the whole range of high fogging operation. However, of course the ideal case cannot be exceeded. However, such GT's are not yet announced. This may change in future due to the growing requirement for temporary electric grid stabilizing power.

In addition, the NOx emission level will benefit from a higher steam content in the flue gas.

A practical challenge will be the arrangement of the injection water nozzles for a sufficiently fine and even droplet distribution with a LWC above 2%. Commercial systems are not yet available.

In cases where the make-up water is available at low cost base load operation with high fogging could result in lower $/kW. The maintenance cost for the compressor blades could be higher because of erosion but the positive long term operating experience reported by Matz based on his experience in an OEM company [5] demonstrates that this compensates by the improved flexibility.

ISOTHERMAL WET COMPRESSION

For completeness, it is attractive to think about the "other end" with the liquid content ratio up to nearly one (by volume). Indeed the high density of water would support compression. However, in rotating equipment undesired demixing due to centrifugal effects is inevitable. A way to avoid this is applying straight streamlines and replacing deviation forces by gravity.

This idea ends up with a device, which we may call a **rainfall compressor**. Such a device is a vertical tunnel with a well-distributed intense rain in it, which drives the air down for being compressed. At the lower end, compressed air is separated from the water in a cavity. Due to the intense heat transfer between the air and the water (with its high heat capacity) such a compressor works nearly ideally isothermal.

One successfully operated example of this kind has no rotor and no blades. Woodbridge describes a system built in 1907 in which compression of the air is caused by falling water droplets in a vertical tunnel [26]. He claimed a measured isothermal compression efficiency of 82% for compression to around 8bar. The system was operated for a mine. In spite of the poor quality of the more than 100-year-old picture, the figure 12 gives an evidence.

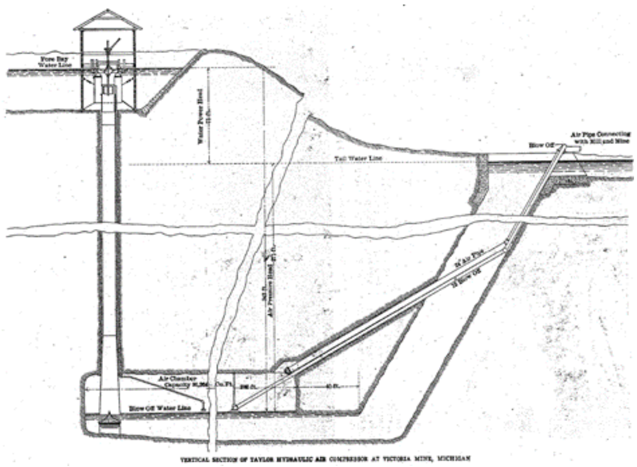

***Figure 12:** The rainfall compressor of Woodbridge, scanned from [26]*

A surprizing feature of such arrangements is that the sonic speed in air-water mixtures can drop to considerably below 100m/s. Figure 13 shows such "bath-tube curves" where on the left side droplets in air and on the right side bubbles in water are present. Therefore, transsonic effects could occur in such arrangements at flow speed considerably below 100m/s.

As a function of the water to air ratio (by mass) such curves are unsymmetrical. GT's use the range of y below 0.02 and there the drop of the sonic speed remains marginal. Attempts of the author and others to realize isothermal wet compression for recuperated gas turbine applications within Alstom with straight streamlines by using the injector principle have so far failed because of too low compression efficiency. The last sign of life for this idea is a patent application of Ni [27]. The old waterfall solution has a better efficiency potential than the injector principle because of the lower velocity level.

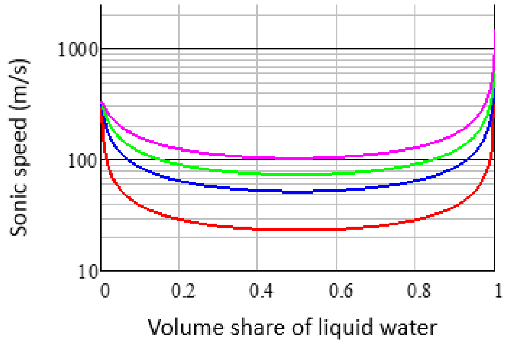

***Figure 13:** Sonic speed of even air-water mixtures at 15°C. Lowest curve for 1bar, above 5, 10 and 20 bar (highest curve).*

OTHER POTENTIAL APPLICATIONS

For GT's with intercooled compressor there is an opportunity to apply high fogging both at the compressor inlet and at the HP compressor inlet downstream of the intercooler. It is also possible to apply an intercooler with water spray cooling. Zhang describes this in [13]. This could result finally with up to around 6% steam content (by mass) at the compressor exit. It also opens the door to apply high fogging at low ambient temperature within the high-pressure compressor only. However, commercial applications of this have not been found yet. In case of ambient temperature below a certain limit, high fogging is not applicable due to the risk of icing. Lecheler [3] gives details that are more specific on this.

The window for recuperated GT's could be expanded with wet compression due to the lower compressor discharge temperature. Kim described such applications [28]. The limit and best case for recuperation is reached with a fully isothermal compressor. In this case, all the exhaust heat can be transferred to the pressurized combustion air. Whether this could be achieved with an improved waterfall compressor as described above is still speculation. However, it is worth to consider because a high thermal efficiency above 60% without a bottoming cycle is still a dream.

CONCLUSIONS

The potential of power augmentation by high fogging of GT's and derived combined cycles is firmly limited by the equilibrium assumption. Examples for an H-class GT are shown.

The practically achievable results of high fogging can be calibrated based on real engine measurements. Examples for an H-class GT are shown.

The opportunity of increasing the fogging water injection ratio to above 2% is attractive in spite of the fact that no commercial systems are available yet. The key challenges will be the arrangement of the injection water nozzles for a sufficiently fine and even droplet distribution as well as the GT system design point within the fogging water injection range. However, this requires considering high fogging already in the state of designing a new gas turbine.

GT concepts with (near) isotherm compression and recuperation earn reconsideration because of their potential for highest thermal efficiencies without an extra bottoming cycle.

ABBREVIATIONS AND NOMENCLATURES

CDRA	"Constant Dissipation Rate Algorithm" for calculating the polytropic efficiency [9]
GT	Gas turbine (open cycle)

GTCC	Gas turbine combined cycle, open GT with bottoming steam cycle
h(p,T)	Specific enthalpy as a function of pressure/temperature
K	Kelvin
LHV	Lower Heating Value (kJ/kg) of a fuel
HHV	Higher Heating Value (kJ/kg) of a fuel
LWC	Liquid Water content (see y)
NOx	Mixture of NO and NO_2 in exhaust gas of a GT
SAS	Secondary air system of a GT (cooling air supply)
T, T_a	Temperature, Ambient temperature
x	Absolute steam content at compressor inlet (by mass)
y	Liquid water injection rate at compressor inlet (by mass) related to dry air mass flow rate
v(p,h)	Specific volume as a function of pressure and enthalpy (m^3/kg, applicable to two or single phase fluids in equilibrium)
v(p,T)	Specific volume as a function of pressure and temperature (m^3/kg, applicable to single phase fluids only)
η	Efficiency (if polytropic it is meant for compressor or turbine blading only, not including inlet or diffuser losses)

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