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PERFORMANCE MODELING OF A POWER GENERATION GAS TURBINE WITH WET COMPRESSION

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Abstract

In the last 15 years more than 1000 power generation gas turbines have been modified with an OEM or aftermarket module to generate the wet compression phenomenon where "Hot Day" conditions are present on the site. This modification to the gas turbine increases power, but can produce performance problems including reduced compressor surge margin and possibly a shorter maintenance cycle because of resulting problems present in the compressor such as blade vibration and erosion with impingement of water droplets on the surface of the compressor blades[1].

In the last few years researchers in academia and the private sector have worked to understand the principles behind the wet compression process in order to know in depth how to use the application to best advantage with gas turbines. The main areas of the research on wet compression are thermodynamic analyses, computer fluid dynamic analysis, and the use of operational data.

Because present technology is unable to obtain detailed operational data on the evaporation process within the compressor, researchers rely on computer simulations based upon aerothermodynamics and physical measurements of the gas turbines, and assumptions based upon available information. These computer simulations are typically aimed toward explaining the performance data from a specific gas turbine model. Most of these computer simulations are cycle analyses of the gas turbine [2-7], although a few are CFD analyses for a specific compressor using either in-house computer programs or commercial CFD software [8-10]

CFD analysis takes into account the fact that an evaporation model should be used in order to predict how the evaporation of the water droplets occurs through the stages of the compressor. Many of the CFD simulations that have been performed for wet compression assume that the mixture of air, liquid water, and water vapor is at equilibrium throughout the compressor. Also, a single water droplet size is sometimes used for the simulation instead of a size distribution for the droplets. These assumptions simplify the calculations for the software.

The results of these simulations may over-forecast the effect of the wet compression and the power output of the gas turbine because of incorrect predictions of evaporation models, or because of the lack of a proper droplet size distribution affecting the calculation.

An analysis that properly forecasts the power output of a gas turbine with wet compression is important for design, performance prediction, and operation. The intention of this paper is to show how performance predictions for a power generation gas turbine is affected by applying several evaporation models [2, 4, 5, 7] in a gas turbine model with a detailed, stage-by-stage compressor model. Model predictions are compared with available operational performance data. Conclusions are provided regarding the best evaporation model assumptions for accurate predictions of gas turbine performance with wet compression.

Nomenclature

$B_{M,eq}$	Spalding Mass Number at equilibrium
$B_{M,neq}$	Spalding Mass Number at non-equilibrium
$\frac{dm_d}{dt}$	Evaporation mass rate in derivative form
$\frac{\Delta m}{\Delta t}$	Evaporation mass rate in numerical form
d	Droplet Diameter
H_M	Specific Mass Driving Potential
m_d	Mass of the droplet
P_{sat}	Saturation Pressure
P_{st}	Static Pressure
R_W	Gas Constant for Water
RH	Relative Humidity
Sc_G	Gas Carrier Schmidt Number
Sh	Sherwood Number
T_a	Dry Air Temperature
T_d	Droplet Temperature
T_f	Mean temperature between air and water droplet
Y_G	Mass fraction of vapor in the gas carrier
$Y_{v,eq}$	Mass fraction of vapor at equilibrium
$Y_{v,neq}$	Mass fraction of vapor at non-equilibrium
δ_f	Diffusivity constant at T_f
τ_d	Droplet Time Constant

Introduction

The loss of power output is an issue that is of common knowledge for power generation gas turbines, specifically during the summer when the "Hot Day" condition is present. This condition is present more often in regions such as Middle East, Central and South America, as well as Asia. The loss of power generated by the power generation plants can lead to partial or total blackouts in the populated areas, and loss of revenues for the power generation companies [5, 11-13].

The gas turbine is a thermal engine, and therefore temperature dependant, specifically upon the inlet temperature. Fig. 1 was generated based upon the results of computer program simulations without wet compression. It shows the influence of ambient temperature on the power output of a typical power generation gas turbine.

This sharp decrease of power output with increasing inlet temperature is the main reason for power generation companies and researchers in turbomachinery to search for technologies which will increase the power output of gas turbines which are operating during "Hot Day" events. Therefore, there is interest in technologies such as Evaporative Cooling, High Fogging, Chillers, and Wet Compression [14]. In the nomenclature of this paper, "wet compression" means the injection of water droplets into the compressor inlet immediately upstream of the IGV.

Evaporative Cooling, High Fogging and Chillers have their advantages and disadvantages which will not be discussed in this paper, although there is an extensive literature available [3, 11, 15-21]. Wet Compression is one of the most used technologies to improve power output during a "Hot Day" event. At this moment power generation companies around the world have reported that more than 1000 gas turbines with such technology installed.

It is important to mention that the Wet Compression technology was implemented first, and only later did researchers begin working to explain the details of operation and the effects of such technology. Employing thermodynamic cycle analysis, researchers now agree that wet compression involves a benefit in terms of the power output. [4, 5, 7]

Researchers have focused their efforts on explaining the evaporation of water droplets with a more detailed analysis of the compressor which is the device in which the wet compression water evaporation takes place.[2, 6, 22-24] The present investigation is focused on the use of evaporation models with a stage-by-stage compressor analysis, since such an analysis using the compressor design method helps to better explain the effects of wet compression in the compressor. The issue in using the compressor design method is the selection of the best evaporation model to fit the data with the help of sparse operational data that is available.

The objective of this paper is to investigate the effect of using three different evaporation models with an in-house computer program that simulates a power generation gas turbine of gas generator and free power turbine design under wet compression conditions Power predictions are compared with available experimental data.[25]

Wet Compression Evaporation Models

The detailed analysis of a compressor operating under wet compression conditions involves the use of an evaporation model. The evaporation model is important in describing the effect of the evaporation of water in the compressor in terms of performance criteria such as pressure rise, work of the stage, total temperature, static temperature, and the stage velocity triangle angles.

In the present paper, three droplet evaporation models are considered. This section reviews the evaporation models investigated, and their apparent advantages and disadvantages.

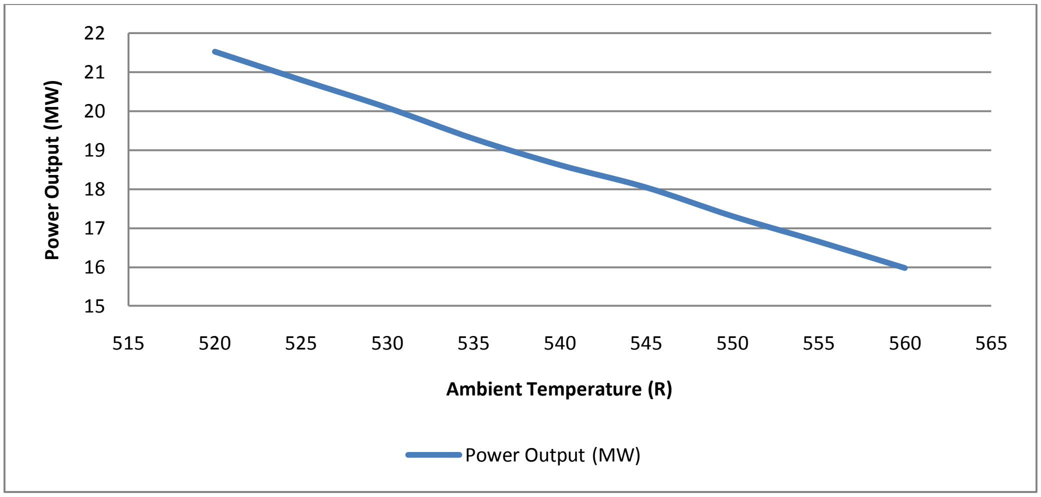

Fig.1. Variation of typical turbine engine power output with increasing ambient temperature

Selection of Evaporation Models

A recent paper by Miller, et al. [26] examined six water evaporation models and conducted experiments to calibrate and evaluate them. From these six, two evaporation models were selected for use in the present investigation. In addition to being calibrated, two of these models incorporate non-equilibrium models for the evaporation of the water, which is thought to be desirable for high fidelity representation of the compression-evaporation process because of the high velocities and short residence times of the water droplets in the compressor.

An additional evaporation model was incorporated in our stage-by-stage compressor model, the model of Beard and Pruppacher [27], which allowed for a wide range of droplet size (20 to 600 µm).

The following models were incorporated in the present stage-by-stage compressor-free power turbine model.

Classical Rapid Mixing evaporation model

The Classical Rapid Mixing evaporation model was derived by Spalding and Godsave [28, 29]. This model was derived for the evaporation of fuel droplets, but works well for water droplets according to Miller et al. [26]. The Classical Rapid Mixing model assumes that the air-water mixture is at quasi-steady state; the derivation of the model relies on the Spalding Mass Transfer number, which is function of the mass fraction of the vapor in the air and in the droplet.

Equation (1) shows the evaporation equation of the model used, while the mass specific driving mass potential H_M uses the mass fraction of the vapor at equilibrium.

$$\frac{dm_d}{dt} = -\frac{Sh}{3Sc_G}\left(\frac{m_d}{\tau_d}\right)H_M \qquad (1)$$

Equations (2) and (3) show the equations for the specific driving mass potential as well the Spalding mass number at equilibrium.

$$H_M = Ln\left(1 + B_{M,eq}\right) \qquad (2)$$

$$B_{M,eq} = \frac{Y_{v,eq} - Y_G}{1 - Y_{v,eq}} \qquad (3)$$

This model assumes that the system is at equilibrium, which may be true if the droplet size is larger than 50 µm where non-equilibrium effects are almost negligible. However, in the case of droplets smaller than 50 µm, non-equilibrium effects are going to be significant. Therefore, there is going to be some difference in the evaporation rate of the model compared to the experimental data as Miller, et al., proved previously.

Beard and Pruppacher evaporation model

The Beard and Pruppacher evaporation model was derived in order to explain the falling of water droplets at terminal velocity when they are suspended in air [27], however, this model is also suitable for the evaporation of the water droplets for applications such as high fogging and wet compression. This evaporation model was used by Kollar et al. [30] to describe the evaporation of a cloud of water droplets in a duct, which would be the situation in the inlet duct of a power generation turbine.

The Beard and Pruppacher model describes the evaporation of droplets for a wide range of sizes from 20 up to 600 µm, and therefore is flexible enough to explain the evaporation of water droplets in equilibrium and non-equilibrium conditions. However, the problem is that since the commercial nozzles mentioned by Chaker et al. [31] produce water droplets between 5 and 35 µm, the Beard and Pruppacher model may have a certain percentage of error, which will not be known until some experiments are performed to quantify this margin of error.

Equation (4) shows the equation that Kollar et al. used to calculate the mass rate of the evaporated liquid.

$$\frac{\Delta m}{\Delta t} = -\frac{\pi d\delta_f\left(P_{sat}(T_d) - RH_a P_{sat}(T_a)\right)}{R_w T_f\left(1 - \frac{P_{sat}(T_f)}{P_{st}}\right)} * Sh \qquad (4)$$

Kollar et al. performed experiments on droplets in the range of 40 – 250 µm and therefore their results are not useful in assessing whether the evaporation model is suitable for non-equilibrium wet compression or the margin of error. It is important to mention that although this evaporation model covers only part of the droplet size range, it is fair to assume that the margin of error is not large enough that the difference between the evaporation rate calculated from the model and the experimental data is negligible for droplet sizes smaller than 20 μm.

It is worth to mention that Beard and Pruppacher [27], use a different nomenclature than Miller et al. [26], for the L-K and CRM evaporation models.

Langmuir-Knudsen evaporation model

Miller et al. [26] derived their evaporation model from the Langmuir-Knudsen law for non-equilibrium evaporation. The Langmuir-Knudsen law allows the assumption that the droplet temperature is not uniform. However, they developed their evaporation model assuming that the droplet conductivity is infinite, so in this case the droplet temperature remains uniform.

It is important to mention that the evaporation equation used in this model is the same as the equation derived for the Classical Rapid Mixing model. However, the Spalding mass transfer number is modified, because instead of including the equilibrium mass fraction of vapor, it includes the non-equilibrium mass fraction of vapor which then adds a new set of equations.

Equation (5) is the same to equation (1); however, the difference lies in the equations (6) and (7), because the specific driving mass potential and Spalding mass number include the non-equilibrium effect. A more detailed description of the equations can be found in Miller et al. [23].

$$\frac{dm_d}{dt} = -\frac{Sh}{3Sc_G}\left(\frac{m_d}{\tau_d}\right)H_M \quad (5)$$

$$H_M = Ln\left(1 + B_{M,neq}\right) \quad (6)$$

$$B_{M,neq} = \frac{Y_{s,neq} - Y_G}{1 - Y_{s,neq}} \quad (7)$$

This model has been compared with experimental data for droplet sizes both under and over 50 μm, thus potentially having a minimal error for this application in comparison with other evaporation models available. Therefore, this model is more flexible than the other models, including the one of Beard and Pruppacher.

These three evaporation models were selected for study, since they encompass the spectrum of current evaporation models. Other evaporation models that have been used [2], have similarities with the models described in this paper.

Turbine Performance Analysis Computer Program

The computer program for the present paper is an in-house turbine performance analysis incorporating a stage-by-stage compressor model computer program, where some assumptions were made in order to focus on assessment of wet compression effects. The compressor model follows the design of the compressor reported by Sexton, et al. [24], but different evaporation models are used. The assumptions are as follows:

- The compressor has two sections where the first section uses NACA-65-(12)10 blades, and the second section uses NACA-65-(8)10 blades
- The combustor is simulated as a heat exchanger to reach the turbine inlet temperature, which is an input.
- The evaporation of the droplet occurs in the compressor rotor section of the stage.
- The compressor is a 16 stage compressor.
- The analysis of the velocity triangles is a mean radius analysis.
- The turbine is divided into two sections: the compressor-turbine and the power turbine.
- The turbine is not analyzed with velocity triangles but as a black box using the change of enthalpy on the inlet and the outlet of each section of the turbine.
- The droplet distribution is between 3 – 35 μm, using the same distribution of Sexton et al. [24]

The computer program also includes numerical methods to accommodate the non-linearity of the equations used to solve for the flow conditions in the different sections of the compressor and the turbine. Fig. 2 shows the flow diagram of the computer program that was designed and used for this research. Since the program is not coupled to any specific gas turbine model, the computer program is flexible enough to simulate a generic gas turbine. If it were used for a specific gas turbine model, appropriate modifications would have to be made to the program in terms of the mass flow and compressor blade profiles.

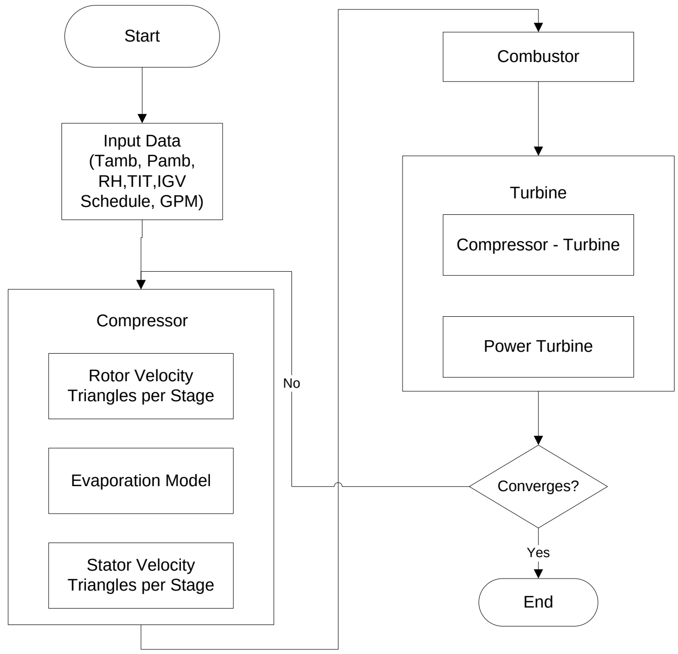

Fig. 2 - Computer Program Flow Diagram

Results

By definition, wet compression forces the compressor of a gas turbine to operate under off-design conditions. In order to compare the computer program results against experimental performance data for an operating turbine, the work of Jolly, et al. [25] was selected. This work provides experimental data from gas turbine model LM2500PE under wet compression. The present computer program was designed to approximate the design of the LM2500 gas turbine, but does not incorporate the actual design. The data obtained from Jolly, et al. is summarized in Table 1. The table gives important information about the effect of ambient conditions such as pressure, temperature and relative humidity on the power output with and without wet compression.

Table 1 – Conditions of the gas turbine LM2500PE from Jolly, et al [25]

Conditions	Without Wet Compression	With Wet Compression
Ambient Temperature (Dry Bulb) °F	100.6	101.4
Ambient Temperature (Wet Bulb) °F	66.1	65.9
Ambient Relative Humidity	~0.13	~0.13
Power Output (MW)	23.46	25.02
Shaft Speed (rpm)	9511	10027
Exhaust Temperature (°F)	1032	1019

The power output is not the only parameter that is published by Jolly, et al., but also the exhaust temperature, since it is a parameter which permits knowing the inlet and outlet conditions of the gas turbine. An important and necessary additional parameter for calculation of the cycle is the turbine inlet temperature.

This parameter is not present in the listed data, since it is considered proprietary information. In order to perform the simulation, the turbine inlet temperature was assumed within reasonable limits; an educated guess was made.

Jolly, et al., mention that the inlet duct of the gas turbine was modified to accommodate a wet compression kit. The kit was able to inject water in increments of 5 GPM each, although the study provides results only for 22 GPM of water injected into the compressor. Thus a very limited data set is provided for making a statistically significant benchmark. Despite this handicap, using the three evaporation models presented in the study, it was possible to simulate the gas turbine operation with different compressor evaporation models to assess which model compared best with the available experimental data. Table 2 summarizes the results of the different models and compares the results with the experimental data.

Table 2 – Comparison of the simulation results with the different evaporation models with available experimental data

Variable	Operational Data	Simulation Data w/o Wet Compression	Langmuir-Knudsen	Beard & Pruppacher	Classical Rapid Mixing
Turbine Inlet Temperature (°R)	--	2697.96	2698.9	2695.8	2695.83
Exhaust Temperature (°F)	1019	963.41	968.1	971.2	971.2
Power Output (MW)	25.02	23.46	25.06	24.47	24.47
Cycle Efficiency	--	38.45%	36.67%	36.32%	36.32%
Heat Rate (BTU/kW-hr)	--	9361.79	9818.2	9911.9	9911.8

Table 2 shows that the Langmuir-Knudsen, Beard and Pruppacher, and the Classical Rapid Mixing all produce similar results which are close to the experimental data; although, each one has considerations that could recommend it, depending on the characteristics of an experiment. The Classical Rapid Mixing is a model derived by Spalding and Godsave as previously mentioned.

Initially this model was used to describe the evaporation of fuel droplets for combustion mixing where the droplet size is greater of 50 microns. In addition, the Classical Rapid Mixing model assumes that the system is at thermal equilibrium.

The Beard and Pruppacher model was initially used to describe the evaporation of water droplets in a quasi-stationary media, besides the experiments done by Beard and Pruppacher had a range of droplet size from 20 to 600 microns, therefore the model works for equilibrium and non-equilibrium systems, although is not explicitly described by the authors. This assumption is made based on discussions by Miller, et al. Meanwhile, the Langmuir-Knudsen model is able to describe the evaporation of water droplets taking into account the non-equilibrium effects of a droplet with a size of under 50 microns. This feature helps to better describe the problems of evaporation due to the size of the droplets. Miller, et al. has mentioned that this model is more accurate than other available models. The effects of wet compression are mostly found in the compressor, as a cooling and mass addition effect.

The working fluid becomes denser so that even though the compressor continues to raise the temperature and the pressure, the pressure increases more with wet compression, especially in the last stages. The effect is similar to intercooling. However, the property changes in the working fluid produce an operating mismatch in the multistage compressor.

The computer program delivered a detailed analysis of the compressor, since it calculates parameters such as the static and total temperature, static and total pressure, the angles of the velocity triangles, as well the angles of attack for both rotor and stator. The program delivers a detailed analysis of the compressor, calculating parameters such as the static and total temperature, static and total pressure, the angles of the velocity triangles, as well the angles of attack for both rotor and stator. With operational information available, the comparison could be much more detailed. Such information would be the heat rate, although in the present case Jolly, et al. do not present any heat rate information, this would give an indirect information of the thermal efficiency of the gas turbine.

Fig. 3 shows the volumetric flow of water evaporated in the compressor when the gas turbine is under wet compression conditions based in the results of the computer program. Notice that Langmuir-Knudsen evaporation model simulates a faster evaporation process, though the volumetric flow evaporated is not as much as might be expected due to the rise in temperature. The water evaporated in the compressor and the energy removal due to the evaporation produces an increase of the pressure rise through the compressor stages. The other evaporation models predict a much slower evaporation through the compressor stages, because of a reduced evaporation rate.

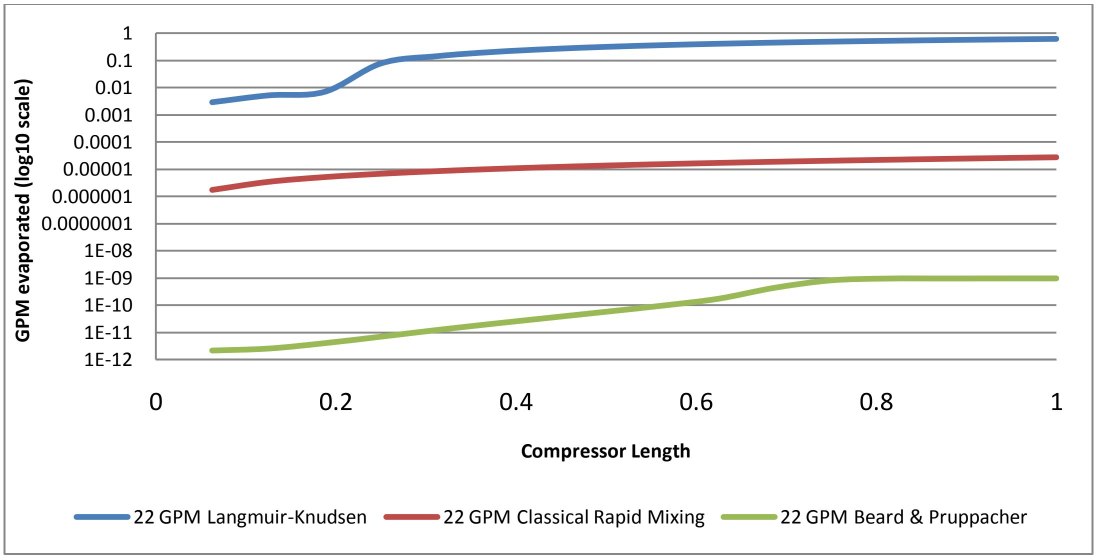

Fig. 3 - Water evaporated due to wet compression through the compressor stages

Since the water evaporates throughout the compressor, the total and static pressure conditions are going to be modified as well. Figure 4 shows the comparison of the results of the computer program in terms of the static pressure of the compressor with and without wet compression (dry conditions) per stage.

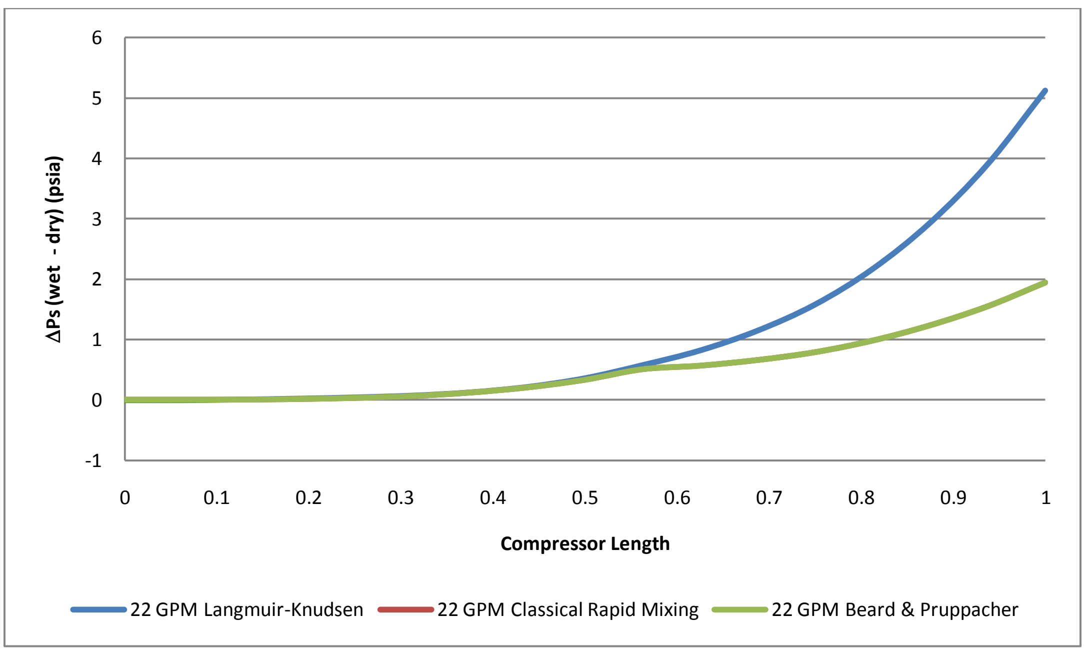

Fig. 4 – Stage static pressure rise difference due to wet compression along the compressor stages

From Fig. 4 it is possible to assess that the evaporation models stimulate a steep increase in loading in the last 4 stages of the compressor because of the prediction of more rapid evaporation. The three evaporation models predict an increase in pressure loading of a shape corresponding to the experiments reported by Alstom [32], but actual pressure increase numbers were not provided in the reference. While Alstom created a plot similar to Fig. 4, theirs was for the high fogging module that has designed specifically for their gas turbine models. Also from Fig. 4 it can be seen that the increased loading predicted by the Classical Rapid Mixing and Beard & Pruppacher models are similar, therefore since are similar both models overlap each other, on the other hand the loading prediction of the Langmuir-Knudsen model is somewhat higher. The "bump" that shows at the location 0.58 is due to the change in the blade profile, changing the angles of attack of the velocity triangles.

Since the results of the evaporation models shown in Fig. 4 look closer to each other because of the resolution of Fig. 4. Fig. 5 shows the static pressure rise difference for the last two stages of the compressor, as a closer look of the difference of the results of the models. It is important to mention that Jolly, et al. does not provide any experimental data along the compressor, and this is under the consideration of the compressor as a black box, while the in-house computer program generated for this paper makes an analysis of the compressor stage-by-stage. It is also worth mentioning that Fig. 5 shows that the Beard and Pruppacher (BP) and Classical Rapid Mixing (CRM) models have similar results for the pressure rise, while the Langmuir-Knudsen (L-K) model has a larger pressure rise. The BP and CRM models are models that assume the air-water-vapor mixture at equilibrium, while the L-K model is based upon the assumption that the mixture is at non-equilibrium. The assessment is that wet compression produces an air-water-vapor mixture that alters the flow properties to increase the pressure per stage and slightly increases the exit temperature compared with the exit pressure and temperature of the compressor without wet compression, although this increase is in the order of 3 – 5 °F depending of the evaporation model that is used.. So, increased fuel is added for the fixed turbine inlet temperature, and the power turbine delivers more power output. The heat rate actually increases, because the gas turbine needs more fuel flow into the system to reach the fixed turbine inlet temperature with a lowered compressor discharge temperature.

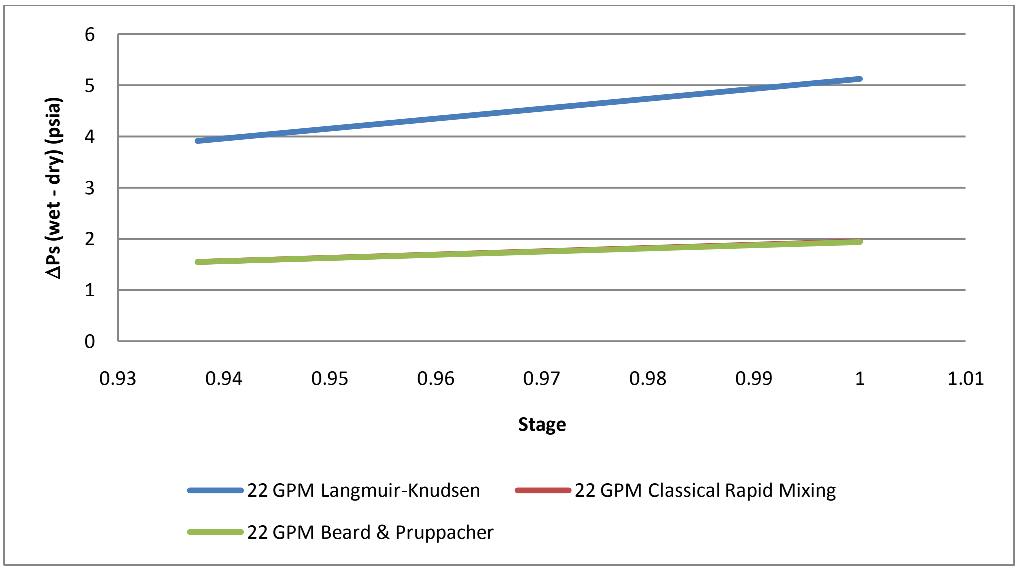

Fig. 5 – Static pressure rise difference due to wet compression through stages 15 and 16

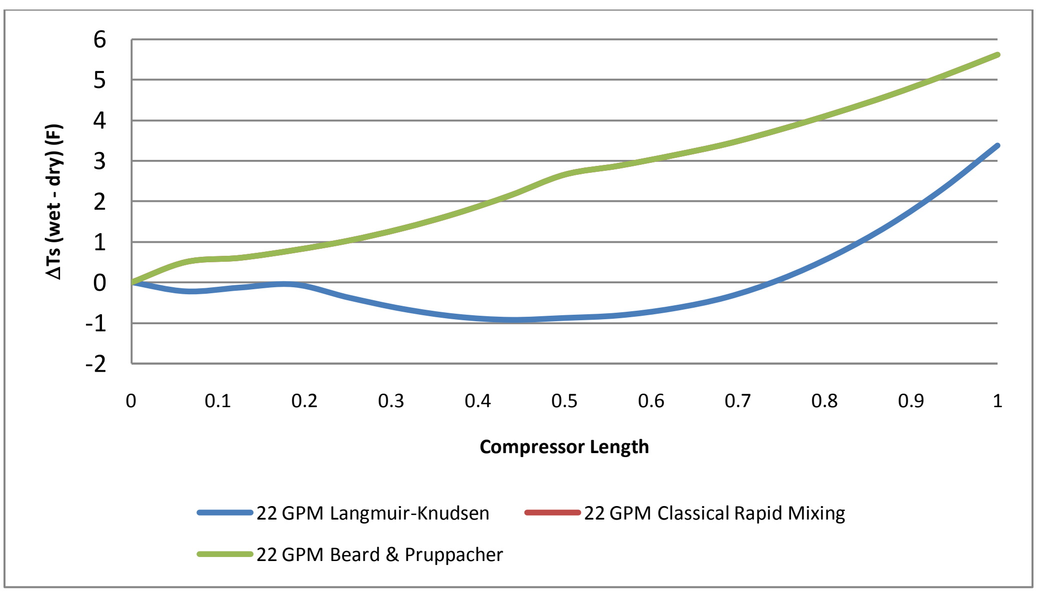

Fig. 6 – Stage static temperature difference due to wet compression along the compressor length

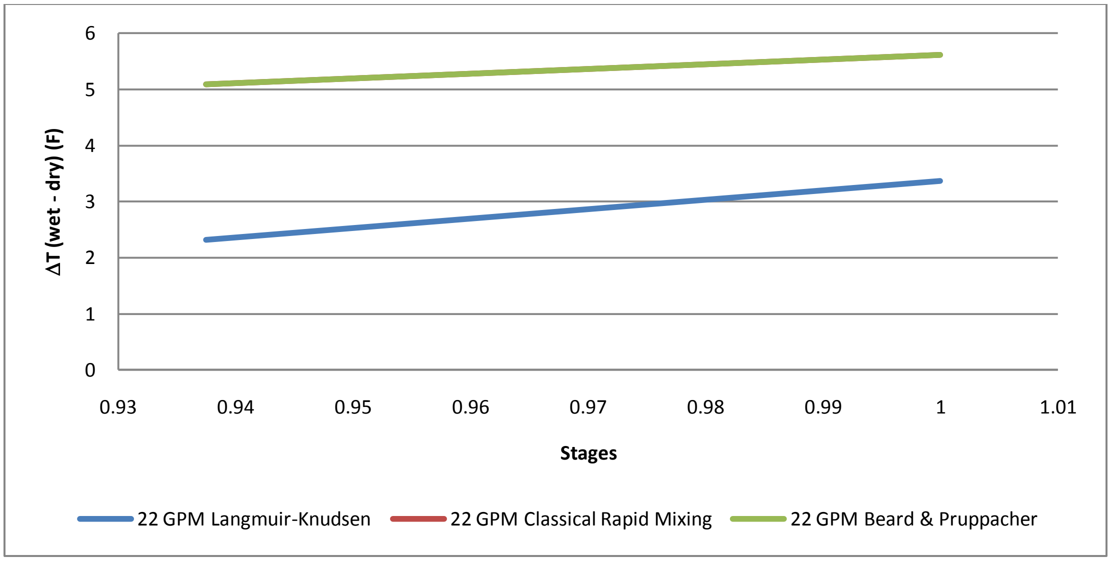

Fig. 7 – Stage static temperature difference due to wet compression through stages 15 and 16

Fig. 6 and 7 shows the same characteristic as Fig. 4 with additional resolution.

Analyzing Figs. 3, 4, 5, 6, and 7, it is possible to conclude that the L – K model makes a close prediction the result of the operational data of Jolly, et al., while CRM and BP models they fall short by ~0.5 MW. Additionally the L – K shows a larger pressure load in the last stages than CRM and BP.

It is important to mention that the CRM and BP models produce a large decrease of temperature, while the L – K model behaves almost as an isothermal compressor over the first half of the compressor length.

Fig. 7 shows that the thermal load is similar to the pressure load of Fig. 5, as an overall analysis, it is possible to conclude that the three evaporation models predict a decrease in the compressor work that is shown in the pressure and thermal loads, which result in an increase of the work of the power turbine. As previously mentioned, there is also a predicted increase in the gas turbine heat rate.

Conclusions

A gas turbine performance simulation model was designed and utilized to examine the effects of wet compression on the performance of the compressor and the associated gas turbine. The model was constructed as a free power gas turbine with properties similar to the General Electric LM2500 gas turbine. Three evaporation models were incorporated in the gas turbine model for evaluation.

The selection of the evaporation model has to take into account the droplet distribution of the nozzles that are injecting the water into the compressor, as well as the evaporation rate. The evaporation models that were described in the present paper cover the range of droplet distribution and size currently used in wet compression. The models used had been previously calibrated by others, it is important to mention that the present paper and the references used are based upon assumptions to explain the phenomenon of wet compression; since there is very little detailed information of a compressor under wet compression.

The available operational data showed that the increase in power output due to wet compression under the tested conditions was approximately 1.6 MW, which means that the wet compression increased the power output, but not at the scale that has been predicted by previous models [20]. The evaporation of the water droplets predicted by our models does not occur as rapidly as many models assume.

The simulation of the computer program using the different evaporation models mentioned in the present paper shows that the predicted power output increases only approximately 1.0 MW using the BP and CRM models, while using the L – K model the simulation of the computer program predicts an increase of ~ 1.6 MW, that matches with the operational data available; this is assuming the turbine inlet temperature of the gas turbine is kept constant. Therefore, the computer program can simulate a gas turbine operating with and without wet compression operation, but since the compressor modeling uses blade profiles that are not the actual compressor blade profiles, then the predicted power output increase is only representative of the actual LM2500 data.

In summary, wet compression is a technology that brings an increase in the power output, but it is not reflected in an increase on the cycle efficiency since there is an increase in the heat addition to the combustor. The correct evaporation model must cover the evaporation of the droplets including the droplet size range The computer program created for this paper is flexible enough to simulate a variety of power generation gas turbines, as well as being able to simulate stage-by–stage compressor operation with and without wet compression over a wide range of volumetric flow of water. The differences that may show with respect to the operational data are at least partly due the lack of detailed design information for modeling. As additional operational data become available, the model can be used for additional sturdy of the wet compression process, and calibrated for more accurate predictions.

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