Proceedings of GT2006
ASME Turbo Expo 2006: Power for Land, Sea & Air
May 8-11, Barcelona, Spain

ASME GT2006-90396

Fog and Overspray Cooling for Gas Turbine Systems with Low Calorific Value Fuels

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ABSTRACT

During the summer, power output and the efficiency of gas turbines deteriorate significantly. Gas turbine inlet air fog cooling is considered a simple and cost-effective method to increase power output as well as, sometimes, thermal efficiency. During fog cooling, water is atomized to micro-scaled droplets and introduced into the inlet airflow. In addition to cooling the inlet air, overspray can further enhance output power by intercooling the compressor.

With continued increase of volatility of natural gas prices and concerns regarding national energy security, alternative fuels such as low calorific value (LCV) synthetic gases (syngas) derived from gasification of coal, petroleum coke, or biomass are considered as important common fuels in the future. The effect of fogging/overspray on LCV fuel fired gas turbine systems is not clear. This paper specifically investigates this issue by developing a wet compression thermodynamic model that considers additional water and LCV fuel mass flows, non-stoichiometric combustion, and the auxiliary fuel compressor power. An in-house computational program, FogGT, has been developed to study the theoretical gas turbine performance by fixing the pressure ratio and turbine inlet temperature (TIT) assuming the gas turbine has been designed or modified to take in the additional mass flow rates from overspray and LCV fuels. Two LCV fuels of approximately 8% and 15% of the NG heating values, are considered respectively. `Parametric studies have been performed to consider different ambient conditions and various overspray ratios with fuels of different low heating values.

The results show, when LCV fuels are burned, the fuel compressor consumes about 10-18% of the turbine output power in comparison with 2% when NG is burned. LCV fueled GT is about 10-16% less efficient than NG fueled GT and produces 10-24% of net output power even though LCV fuels significantly increase fuel compressor power. When LCV fuels are burned, saturated fogging can achieve a net output power increases 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. Fog/overspray could either slightly increase or decrease the thermal efficiency depending on the ambient conditions.

NOMENCLATURE

CDT	Compressor discharge temperature
CIT	Compressor inlet temperature
C_P	Specific heat at constant pressure (kJ/kg-K)
D.A.	Dry air
DBT	Dry bulb temperature
f'	Fuel Air ratio
h	Enthalpy (kJ/kg)
H	Enthalpy of reaction (kJ/kmol)
k	Polytropic index
L	Latent heat of water (kJ/kg)
NG	Natural gas
m	Mass flow rate(kg/s)
n	Stoichiometric coefficient
OS	Overspray
P	Pressure (kPa)
Q_{In}	Heat added in combustion chamber (kJ/kg)
R	Gas constant (kJ/kg-K)
RH	Relative humidity
s	Entropy (kJ/kg-K)
T	Temperature (K)
T_D	Dew point temperature
TIT	Turbine inlet temperature
v	Specific volume (m^3/kg)
WBT	Wet bulb temperature

Greek

γ	Specific Heat Ratio
ρ	Density (kg/m^3)
ϕ	Relative Humidity (%)
ω	Humidity ratio, specific humidity (m_v/m_g)
η	Thermal efficiency

Subscripts

1	Compressor Inlet
2	Compressor Exit
3	Turbine Inlet
4	Turbine Exit
1f	Fuel in ambient condition
2f	Fuel in fuel compressor discharge condition
a	Dry air
Amb	Ambient
c	Compressor
fc	Fuel compressor
f	Liquid water
g	water vapor
i	Index for different elements
Pr	Product
Re	Reactant
s	Isentropic process
t	Turbine
th	Thermal

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INTRODUCTION

Gas turbines (GT) suffer from both decreasing output power and efficiency as the ambient temperature increases because the air becomes less dense (which results in less mass flow rate), and the compressor works harder as ambient temperature increases. It has been found that every 1°F raise of ambient temperature reduces gas turbine efficiency by approximately 0.3-0.5% [1]. Gas turbine inlet air fog cooling is considered a simple and cost-effective method ($40-60/kW) to increase power output and often also increase thermal efficiency. During fog cooling, water is atomized to micro-scaled droplets and introduced into the inlet airflow. The water droplets remaining after the air flow reaching the wet bulb temperature is considered as the overspray, which can further cool the compressor. Fog cooling with overspray into entering air at the inlet of the compressor is gaining popularity due to its low initial and maintenance costs.

Fog/Overspray and Wet Compression

Zheng et al. [2] established a thermodynamic wet compression model to analyze the effect of fog inlet cooling on gas turbine performance. They conclude that the output power increases with increased fog mass flow rate, whereas the efficiency curve is flat for a wider range of compression pressure ratio and turbine inlet temperature. Later, Zheng et al. [3] expand their analysis to a regenerative gas turbine cycle.

Bhargava and Meher-Homji [4] presented the results of a comprehensive parametric analysis on the effect of inlet fogging on a wide range of existing gas turbines. Both evaporative and overspray fogging conditions were analyzed. It shows that the performance parameters indicative of inlet fogging effects have definitive correlation with the key gas turbine design parameters. In addition, they indicated that aero-derivative gas turbines, in comparison to the industrial machines, have higher performance improvement from inlet fogging.

Chaker et. al. [5-7] presented the results of extensive experimental and theoretical studies conducted over several years and coupled with practical aspects learned in the implementation of nearly 500 inlet fogging systems on gas turbines ranging from 5 to 250 MW. Their studies covered the underlying theory of droplet thermodynamics and heat transfer and provided practical points relating to the implementation and application of inlet fogging to gas turbine engines. They also described the different measurement techniques available to design nozzles. These papers collectively provided experimental data on different nozzles and recommended a standardized nozzle testing method for gas turbine inlet air fogging. The complex behavior of fog droplets in the inlet duct was addressed and experimental results from several wind tunnel studies were documented.

To investigate the mist transport in the entrance duct, Wang et. al. [8] conducted a computational fluid dynamic (CFD) study of different fundamental geometries including a straight tunnel, a diffuser, a contraction, and a 90^o bend. These geometries were used to investigate the separate effect of acceleration, deceleration, and centrifugal force on mist transport and cooling effectiveness, respectively. Lastly, a duct representing a real application was used for simulation. The effects of droplet size, droplet distribution, and humidity on cooled air temperature distribution were examined. Analysis on droplet history (trajectory and size) was employed to interpret the mechanism of droplet dynamics under influence of acceleration, diffusion, and body forces. They showed that in the contraction the acceleration significantly lowers the cooling effectiveness when large droplets of 50-μm were used. In the diffuser, the average temperature increases near the exit because the flow separates and the reverse flow entrains warmer airflow from downstream. Near exit of the diffuser, high evaporation ratio occurs due to the high average temperature as well as lower flow velocity (for longer residence time). In the 90-degree bend, centrifugal force and secondary flow move the droplets toward the outer wall and result in a non-uniform temperature distribution at the exit with cooler area near the outer wall. High cooling effectiveness is achieved due to the secondary flow mixing. Simulation with a complex duct similar to those used in real applications shows regions of large recirculation. The recirculation regions should be removed or minimized because they produce flow pulsation, induce aerodynamic losses, trap water droplets, and could allow water droplets to coalesce into larger ones that may be detrimental to the compressor blades.

Sexton et. al. [9] conduct a computational simulation to examine the concept of water injection, fogging and overspray. Their results included the compressor performances for different conditions of ambient temperature, pressure and humidity, flow rate, overspray and water temperature. Their results showed that for 100°F (311K) day, with an ambient relative humidity of 60%, nearly 13°F (7.2K) of temperature depression and 11% of power augmentation can be realized by overspraying 0.38% of water mist.

Bagnoli et. al. [10] investigated effects of interstage water injection on the performance of a 17-stage gas turbine using aero-thermodynamic modeling. They discussed the impact of interstage injection on the stage-by-stage compressor performance characteristics of the selected gas turbine to estimate the overall gas turbine performance. They found that the maximum power could be obtained if the water injection is realized upstream of the compressor compared to the other injection locations. Moreover, 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 increased amount of water injection causes the last compressor stage to operate with closer to surge line.

Hartel et. al. [11] studied the effects of high fogging on the work of compression. They used a droplet model where they took finite time of evaporation into account by introducing discrete droplets and modeling explicitly the heat and mass transfer between liquid and gaseous phase. Their results showed that the beneficial effects of wet compression diminish when the droplet diameter increases and hence, evaporation is slowed down. For compression rates typical of modern heavy-duty gas turbines, droplets need to be as small as about 1-μm in order to achieve evaporation under approximate thermodynamic equilibrium during compression.

White et. al. [12] evaluated the effect of water injection on compressor performance. They investigated the thermodynamic and aerodynamic aspects of wet compression by a numerical method applicable to very fine droplet sprays. The combination of thermodynamic losses and impaired aerodynamic efficiency result in the fractional work reduction due to evaporative intercooling being substantially less than that suggested by ideal wet compression calculations. On that basis, they suggested some redesign of the compressor to achieve the full benefits that are possible with water-injected cycles.

Effect of Low Calorific Value (LCV) Fuels

All the previous studies of fog/overspray cooling are applied to gas turbines fired with nature gas or oils. With continued increase of volatility of natural gas prices and concerns regarding national energy security, alternative fuels such as synthetic gas (syngas) derived from gasification of coal (a more stable energy source), petcoke, or biomass (more environmentally friendly) are considered as important common

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fuels in the future. Depending on the feedstock types and gasification process, the heating value could range from 35% of an oxygen-blown synthetic gas derived from coal to 10-15% of a producer gas derived from air-blown biomass gasification. When LCV fuels are used, more fuel mass flow rate (3 - 10 times) is needed to achieve the required heating value by adding an additional fuel compressor. This additional fuel mass flow rate will imposes heavier load to the main compressor due to increased backpressure in comparison when natural gas is used. Implementation of fog/overspray cooling will further strain the already overloaded compressor if existing commercially available GTs are used. To achieve a better GT performance, the existing GTs may need to be modified to accommodate the increased fuel flow rate in the combustor and turbine. In addition, the effect of fog/overspray on the performance of LCV fired gas turbine systems is not clear. Motivated by these two reasons, the objectives of this paper are (a) to develop a wet compression thermodynamic model by including the fuel compressor work and the additional fuel mass flow rate in calculating the turbine inlet temperature and (b) to specifically investigate the influence of fog/overspray on the output power and efficiency when LCV fuels are burned and compare the results with those fired with natural gas.

An in-house computational program, FogGT, has been developed to study the theoretical gas turbine performance by fixing the pressure ratio and turbine inlet temperature (TIT) assuming the gas turbine has been designed or modified to take in the additional mass flow rates from overspray and LCV fuels. The analysis has been performed on ideal gas turbine systems without considering the losses. No real gas turbine is employed in this study.

THERMODYNAMIC MODEL

The present study employs a hypothetical gas turbine system equipped with a fog sprayer device and a fuel compressor, as shown Fig. 1.

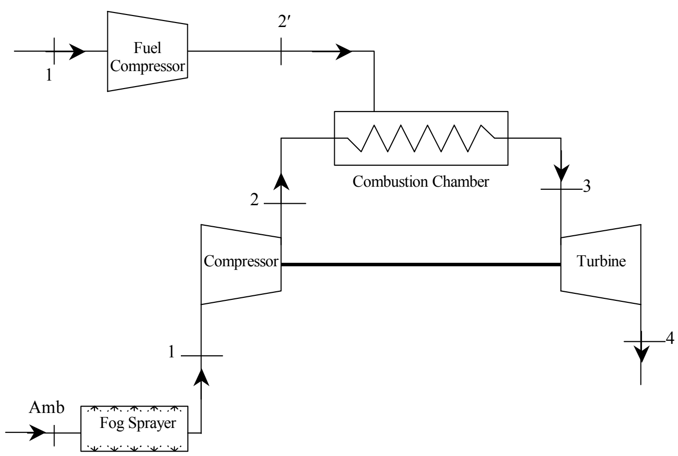

Figure 1 The gas turbine system with a fog spraying device and an additional fuel compressor

Effect of Elevated Ambient Temperature on GT Performance

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. 2 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.

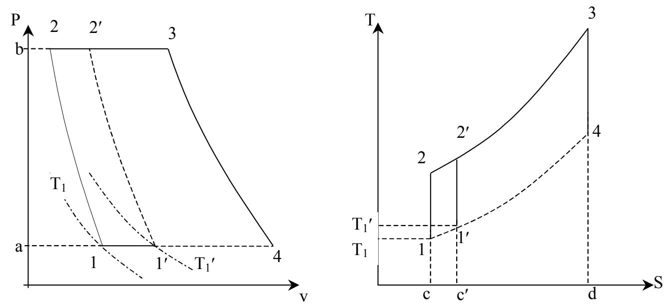

Figure 2 Effect of increased ambient temperature on gas turbine efficiency and output power per unit mass flow rate

In 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 same in both conditions, so the net output power (per unit mass flow rate) decreases.

On the other hand, the rising isobaric curves (1-4 and 2-3) in T-S diagram shows heat addition in the combustion chamber at lower temperatures produces more fraction of useful energy. This can be explained by noticing that more heat will be rejected (area under curve 1-4) at higher T1 if the same amount of useful energy (e.g. area 1-2-2'-1') is to be harnessed. Therefore, the GT efficiency will be reduced when the compressor inlet temperature T_1 increases.

Above analysis is based on per unit mass flow rate. Elevated ambient temperature further makes the air lighter and reduces 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 to a reduction of the total output power. Because the output power is influenced by both compressor power and mass flow rate, while efficiency is not affected by the mass flow rate, elevated ambient temperature will affect output power more than efficiency.

The effects of fog cooling and overspray are shown in Fig. 3. In Fig. 3, 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 **moist 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 web bulb temperature due to the heat transfer between the saturated air and remaining water droplets when the water is supplied at a temperature below the wet bulb temperature.) 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′. This is because wet compression reduces the polytropic index (k) of the compression work (Pv^k = Constant) from isentropic process (k = γ, specific heat ratio) to a k-value closer to the isothermal process (k = 1). 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 curve 1″-2‴ from the curve 1″-2″ (moist compression without overspray). Therefore, fog and overspray cooling increases both the net output power and the cycle thermal efficiency. In the mean time, fog/overspray further

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increases the total mass flow rate, which does not affect the thermal efficiency but increases the power output. Hence, augmentation of the total power output is more pronounced than efficiency.

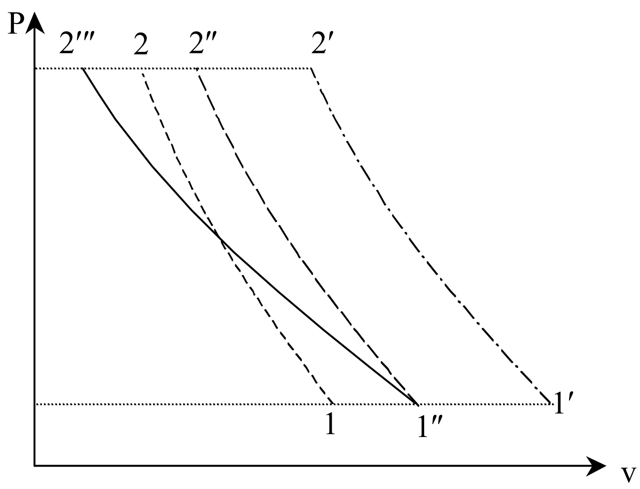

Figure 3 Different fog/overspray cooling processes in the air-intake duct and in the compressor

Development of Wet Compression Formulation for Fog/Overspray Cooling Gas Turbine System

Refer to Fig. 2 again, but with a different representation of curves 1-2 and 1'-2' from earlier description. During derivation of wet compression formulation, isobaric line 1′-1 represents the inlet fog cooling where evaporation of water takes place to saturate the air. Polytropic line 1-2 represents either moist compression (saturated air without overspray) or wet compression (intercooling due to overspray). Isentropic line 1′-2′ represents compression process of the main compressor without fog/overspray cooling. Assuming the fuel is supplied at the ambient temperature, line 1'-2' also represents the compression of the fuel compressor although at a different mass flow rate.

The development of the wet-compression formulation is similar to Zheng et. al. [2-3], but they did not consider the fuel compressor work nor the fuel mass flow which is essential in LCV fuel applications when the turbine inlet temperature (TIT) and excess air are calculated. In this paper, the formulation includes (a) the elaboration of **moist compression** (fogging without overspray) and **wet compression** (overspay), (b) discrimination of the different augmentation effects between thermodynamics and increased mass flow rate, (c) non-stoichiometric combustion reaction to calculate excess air and TIT including LCV fuel mass flow rate, (d) auxiliary power of the fuel compressor, and (c) efficiency and power enhancement comparisons . The dynamics of water droplet evaporation inside the compressor is not considered in this paper, but it will be considered in the future study.

According to the Gibbs equation,

$$\mathrm{Tds} = \mathrm{dh} - \frac{\mathrm{dP}}{\rho} \qquad (1)$$

For an ideal wet compression, assume the evaporative heat equals to the reversible heat,

$$\mathrm{Tds} = -\mathrm{LdW} \qquad (2)$$

From equations (1) and (2), we get,

$$-\mathrm{LdW} = \mathrm{dh} - \frac{\mathrm{dP}}{\rho} \qquad (3)$$

Here, $\mathrm{dh} = c_p \mathrm{dT} = \frac{\gamma R}{\gamma - 1}\mathrm{dT}$ and γ is the specific ratio.

From the equation of state, $P = \rho RT \Rightarrow \frac{1}{\rho} = \frac{RT}{P}$

Substituting the value of dh and $\frac{1}{\rho}$, equation (3) becomes,

$$-\mathrm{LdW} = \frac{\gamma R}{\gamma - 1}\mathrm{dT} - \frac{\mathrm{RTdP}}{P} \qquad (4)$$

$$\Rightarrow \quad \frac{dP}{P} = \frac{\gamma}{\gamma - 1}\frac{dT}{T} + \frac{L}{R}\frac{dW}{dT}\frac{dT}{T}$$

$$\Rightarrow \quad \frac{dP}{P} = \left(\frac{\gamma}{\gamma - 1} + \frac{L}{R}\frac{dW}{dT}\right)\frac{dT}{T} \qquad (5)$$

Assuming evaporative rate varies linearly with temperature, i.e. $\frac{dW}{dT} =$ Constant, the isentropic relation is obtained as, $Pv^k = C$ or

$$PT^{-\frac{k}{k-1}} = C \quad \text{[k = polytropic index of ideal wet compression]}$$

$$\Rightarrow \frac{dP}{P} = \frac{k}{k-1}\frac{dT}{T} \qquad (6)$$

Equations (5) and (6) give,

$$\frac{k}{k-1}\frac{dT}{T} = \left(\frac{\gamma}{\gamma - 1} + \frac{L}{R}\frac{dW}{dT}\right)\frac{dT}{T}$$

$$\Rightarrow \quad \frac{k}{k-1} = \frac{\gamma}{\gamma - 1} + \frac{L}{R}\frac{dW}{dT} \qquad (7)$$

Equation (7) shows that the increase of evaporation rate decreases the polytropic index (k) of wet compression from isentropic process ($k = \gamma$) towards the isothermal process (k=1), which results to a reduction of compression power. This can be seen in the P-v diagram in Fig. 3 as a less steeper curve (1"-2"' vs. 1"-2") requires a less compression power.

The effect of additional moisture on compressor performance due to overspray is analyzed below. At ambient temperature (T_{amb}) and relative humidity (ϕ), the following parameters can be obtained from the psychrometric chart: dew point (T_D), wet bulb temperature (WBT), the humidity ratio ω_0 (moisture content at DBT), and ω_1 (moisture content at WBT). The compressor inlet temperature T1 is obtained by applying energy balance via enthalpy.

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The moist air enthalpy at state 1, on the basis of mass fraction is,

$$h_1 = \frac{\sum m_{i_1} h_{i_1}}{\sum m_{i_1}} = \frac{m_{a_1} h_{a_1} + m_{f_1} h_{f_1} + m_{g_1} h_{g_1}}{m_{a_1} + m_{f_1} + m_{g_1}} \qquad (8)$$

where m_a, m_f, and m_g represent the mass of dry air, liquid water, and water vapor, respectively. And, the moist air entropy at state 1, on the basis of mass fraction,

$$s_1 = \frac{\sum m_{i_1} s_{i_1}}{\sum m_{i_1}} = \frac{m_{a_1} s_{a_1} + m_{f_1} s_{f_1} + m_{g_1} s_{g_1}}{m_{a_1} + m_{f_1} + m_{g_1}} \qquad (9)$$

Under fog/overspray cooling, the compressor inlet temperature is typically fully saturated at WBT. The inlet air will evaporate and absorb the moisture from the sprayed water as much as it needs to saturate itself; the rest of the water will be treated as overspray. In this paper the **overspray percentage** is defined as the ratio of oversprayed water mass over the total air mass flow rate.

To determine state 2, the isentropic temperature of compressor discharge, T_{2S} needs to be determined first. The moist air entropy at state 2, on the basis of mass fraction is,

$$s_2 = \frac{\sum m_{i_2} s_{i_2}}{\sum m_{i_2}}$$

$$= \frac{m_{a_2}\left\{s_{a_2} - R\ln\left(\frac{P_2}{P_1}\right)\right\} + m_{f_2} s_{f_2} + m_{g_2} s_{g_2}}{m_{a_2} + m_{f_2} + m_{g_2}} \qquad (10)$$

In practice, all the water droplets shall be evaporated at the compressor discharge (i.e. $f_2 = 0$), so the above expression becomes,

$$s_2 = \frac{\sum m_{i_2} s_{i_2}}{\sum m_{i_2}}$$

$$\Rightarrow s_2 = \frac{m_{a_2}\left[s_{a_2} - 0.287\ln(r_P)\right] + m_{g_2} s_{g_2}}{m_{a_2} + m_{g_2}} \qquad (11)$$

T_{2S} can be determined by letting $S_1 = S_2$. All the property values in these two expressions are the function of T_1 (which is already known) and T_{2S} (which is obtained by iteration). At state 2, the isentropic enthalpy of moist air is calculated as,

$$h_{2S} = \frac{\sum m_{i_2} h_{i_{2S}}}{\sum m_{i_2}} = \frac{m_{a_2} h_{a_{2S}} + m_{g_2} h_{g_{2S}}}{m_{a_2} + m_{g_2}} \qquad (12)$$

The Compressor Efficiency is defined as,

$$\eta_c = \frac{h_{2S} - h_1}{h_2 - h_1} \qquad (13)$$

Equation (13) gives the actual moist air enthalpy, which is,

$$h_2 = \frac{\sum m_{i_2} h_{i_2}}{\sum m_{i_2}} = \frac{m_{a_2} h_{a_2} + m_{g_2} h_{g_2}}{m_{a_2} + m_{g_2}} \qquad (14)$$

Iteration is needed to determine T_2 by satisfying all the property values as functions of T_2 in equation (14).

State 3 is known as the turbine inlet temperature (TIT), which is assigned as an operating parameter. The moist air enthalpy at state 3, on the basis of mass fraction is,

$$h_3 = \frac{\sum m_{i_3} h_{i_3}}{\sum m_{i_3}} = \frac{m_{a_3}(1+f')h_{a_3} + m_{g_3} h_{g_3}}{m_{a_3}(1+f') + m_{g_3}} \qquad (15)$$

The moist air entropy at state 3, on the basis of mass fraction is,

$$s_3 = \frac{\sum m_{i_3} s_{i_3}}{\sum m_{i_3}} = \frac{m_{a_3}(1+f')s_{a_3} + m_{g_3} s_{g_3}}{m_{a_3}(1+f') + m_{g_3}} \qquad (16)$$

The fuel mass flow rate in included in the gas flow in terms of (1+f '), where f ' is fuel/air ratio. To determine state 4, the isentropic state, $T_{4S,}$ needs to be determined first. The moist air entropy at state 4 on the basis of mass fraction is:

$$s_4 = \frac{\sum m_{i_4} s_{i_4}}{\sum m_{i_4}} = \frac{m_{a_4}(1+f')\left[s_{a_4} - R\ln\left(\frac{P_4}{P_3}\right)\right] + m_{g_4} s_{g_4}}{m_{a_4}(1+f') + m_{g_4}}$$

$$\Rightarrow s_4 = \frac{m_{a_4}(1+f')\left[s_{a_4} - 0.287\ln(r_P)\right] + m_{g_4} s_{g_4}}{m_{a_4}(1+f') + m_{g_4}} \qquad (17)$$

S_3 is set to equal S_4 to determine T_{4S}. All the property values in these two expressions are functions of T_3 (which is assigned as an operating parameter) and T_{4S} (which is determined by iterations). At state 4, the isentropic enthalpy of moist air is calculated as,

$$h_{4S} = \frac{\sum m_{i_4} h_{i_{4S}}}{\sum m_{i_4}} = \frac{m_{a_4}(1+f')h_{a_{4S}} + m_{g_4} h_{g_{4S}}}{m_{a_4}(1+f') + m_{g_4}} \qquad (18)$$

Turbine efficiency is defined as,

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{4S}} \qquad (19)$$

Equation (19) gives h_4 at the actual state 4.

Compressor Work, $w_c = h_2 - h_1 \qquad (20)$

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Turbine Work, $w_t = h_3 - h_4$ (21)

The fuel compressor needs a substantial amount of power to pump the fuel to the combustion chamber. Assuming the fuel behaves as an idea gas, the power required for fuel compressor is calculated as,

$$w_f = \frac{\gamma}{\gamma-1}\left(\frac{P_2'}{\rho_{2f}'} - \frac{P_1}{\rho_{1f}}\right)\Big/\eta_{fc} = \frac{\gamma R\left(T_2' - T_1\right)}{\eta_{fc}(\gamma-1)} \quad (22)$$

To ensure the fuel can be injected into the combustion chamber, the fuel compressor is assigned to deliver 25% higher pressure than the compressor discharge pressure in equation (22) by letting $P_2' = 1.25P_2$. η_{fc} is the fuel compressor efficiency. The net work is:

$$W_{net} = W_t - W_c - W_f = (m_3 w_t - m_2 w_c - m_f w_f) \quad (23)$$

Heat Input from Low Calorific Value (LCV) Fuels

Two LCV fuels derived from biomass gasification are used in this study. LCV1 is identical with LCV2 but is diluted with nitrogen. The compositions of the LCV fuels are given in Table 1.

Table 1 Studied LCV Fuels 9

Compound	LCV1 Vol (%)	LCV2 Vol (%)	NG Vol (%)
Methane (CH_4)	7.00	11.15	100
Ethane (C_2H_6)	0.08	0.13	
Ethylene (C_2H_4)	0.11	0.18	
Benzene (C_6H_6)	0.14	0.22	
Carbon-Dioxide (CO_2)	14.60	23.2	
Carbon-Monoxide (CO)	10.60	16.8	
Hydrogen (H_2)	7.30	11.62	
Oxygen (O_2)	0.05	0.08	
Water Vapor (H_2O)	22.92	36.62	
Nitrogen (N_2)	37.20	0	
Total	100.00	100.00	100.00
Low Heating Value (KJ/Kg)	4,358	7,405	50,046
High Heating Value(KJ/Kg)	5,238	8,735	55,532

The heat input obtained from chemical reaction of the fuel with excess air can be obtained by the following equation [13] using LCV-1 fuel as an example:

$$\begin{aligned}
&n_{H_2}H_2 + n_{O_2}O_2 + n_{H_2O}H_2O + n_{CO}CO + n_{CO_2}CO_2 \\
&+ n_{CH_4}CH_4 + n_{C_2H_6}C_2H_6 + n_{C_2H_4}C_2H_4 \\
&+ n_{C_6H_6}C_6H_6 + X\ \text{D.A.} + x\ H_2O \\
&\rightarrow \left(n_{CO} + n_{CO_2} + n_{CH_4} + 2n_{C_2H_6} + 2n_{C_2H_4}\right. \\
&\left.+ 6n_{C_6H_6}\right)CO_2 + \left(n_{H_2} + n_{H_2O} + 2n_{CH_4} + 3n_{C_2H_6}\right. \\
&\left.+ 2n_{C_2H_4} + 3n_{C_6H_6} + x\right)H_2O + X\ \text{D.A.} \\
&-\left(\tfrac{1}{2}n_{H_2} - n_{O_2} + \tfrac{1}{2}n_{CO} + 3n_{CH_4} + 3\tfrac{1}{2}n_{C_2H_6}\right. \\
&\left.+ 3n_{C_2H_4} + 7\tfrac{1}{2}n_{C_6H_6}\right)O_2 \qquad (24)
\end{aligned}$$

where D.A. is the Dry Air (O_2 +3.768 N_2) and n is the stoichiometric coefficients.

By equating the enthalpy of reaction on both sides, the value of X (the mole of Dry Air) can be obtained. The mole of moisture (x) can be calculated from the psychrometric chart. The mole numbers of all the reactants are basically the volumetric percentage of the gases.

The enthalpy of the reactants is,

$$H_{Re} = \sum\left(n_i \bar{h}_{i,T_2}\right) \quad (25)$$

The enthalpy of the products is,

$$H_{Pr} = \sum\left(n_i \bar{h}_{i,T_3}\right) \quad (26)$$

From the value of X, the excess air percentage and fuel-air ratio (f') can be calculated. Hence, the heat addition into the combustion chamber can be obtained as:

$$q_{in} = f'(LHV) \quad (27)$$

and the thermal efficiency is obtained as,

$$\eta_{th} = \frac{w_{Net}}{q_{in}} \quad \text{(independent of mass flow rate)} \quad (28)$$

Development of the Computer Program, FogGT

An original computer program, FogGT, has been developed in-house and used for this study. The properties are created digitally including steam tables, psychrometric relations, thermal-fluid properties of various gases (e.g. CO_2, N_2, O_2, water vapor, etc.), and the air property table. The SI units are used in all the property databases. Conversion to English units is automatically executed when the option of using English units is selected.

In this study, the program is tailored to receive the following information as input:

[ambient temperature, relative humidity, ambient pressure, compression ratio, compressor isentropic efficiency, type of fuel, TIT, isentropic efficiency of turbine, mass flow rate of air, and percentage ratio of water flow rate with respect to the air flow rate.]

For the user's convenience, the program calculates and prompts the water flow rate required to saturate the air as well as the corresponding WBT once the ambient condition is provided. If the water mass specified is not enough to saturate the air, this is called **underspray**. If the supplied water mass is more than needed for saturating the air, it is called **overspray**. The percentage ratio of the water mass (that is left after saturating the air) to the dry air mass is defined as the **overspray percentage**, which is automatically calculated by the program. Up to 2% of overspray was considered in this paper. Cataldi et. al. [14] showed that overspray as high as 1-2% could be accepted without major changes to the engine design although specific engine adjustments and protections are needed.

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The FogGT program provides a database of high and low heating values (HHV and LHV) for many general fuels. Combination of fuels can also be specified and the appropriate heating values be calculated. The program provides output including, compressor power, turbine power, fuel compressor power, heat addition in combustion chamber, net output power, percentage of overspray, fuel flow rate, specific fuel consumption, etc. The iteration process is shown in the flow chart in Fig. 4. The detailed descriptions of FogGT program, the associated property database, and the source code are documented in the report by Khan and Wang [15].

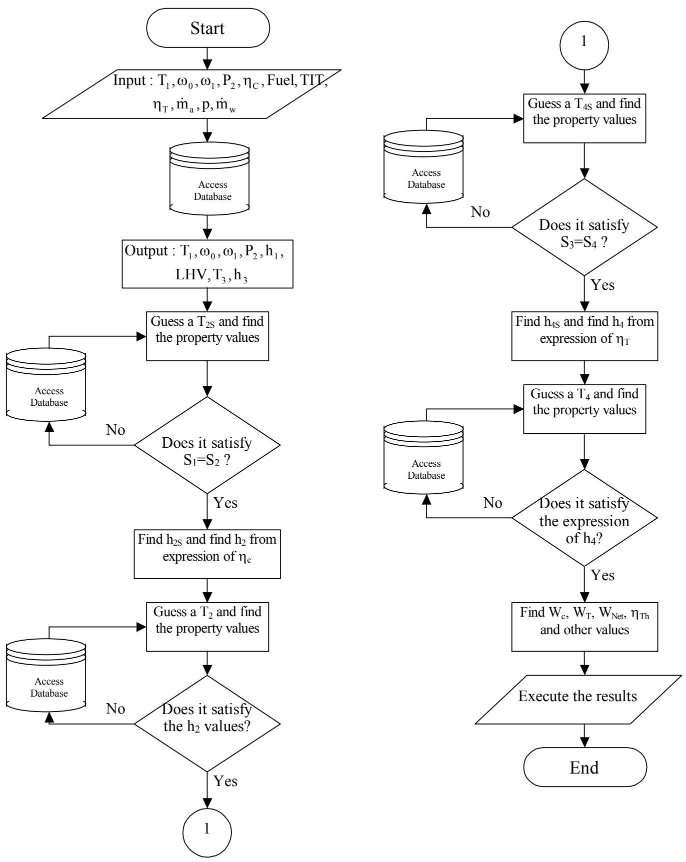

Figure 4 Flow chart of the computer program FogGT

RESULTS AND DISCUSSIONS

All the cases in the first batch are calculated by keeping the values of the following parameter fixed: compression ratio (12), TIT (1400K), air mass flow rate (20Kg/s), inlet pressure (1atm), compressor isentropic efficiency (88% for both the main and the fuel compressors), and turbine isentropic efficiency (88%). Methane is used as the fuel for the reference case. Two different LCV fuels are used, and the performances are compared with the reference case. Four different ambient conditions are considered: low temperature low humidity (ISO condition, 288.2K and 60% Rh), low temperature high humidity (288.2K and 90% Rh), high temperature low humidity (313K and 60% Rh), and high temperature high humidity (313K and 90% Rh). Although the condition of 313K (40°C) and 90% Rh is extremely rare to occur, it presents an upper limit of the hot-and-humid condition that shows the minimum augmentation fog/overspray can achieve at a hot environment. These ambient conditions are also applied as the inlet condition for the fuel compressor. Although the fuel is usually preheated in practice, the heating energy is also paid by some means. Therefore, using the ambient condition as the fuel compressor inlet condition implicitly include all the energy required to compress the fuel to 25% above the pressure in the combustor.

Four different fog cooling are analyzed including moist compression (unsaturated air), compression with saturated air, 1% overspray and 2% overspray. More than 2% overspray is not recommended [14]. The results of the first batch are shown in Table A1 in the appendix.

The simulations are performed by implicitly assuming that each case is matched with a gas turbine that is specifically designed to meet the specs of air mass flow rate, fuel flow rate, pressure ratio, and TIT. This means that the sizes of the compressor, turbine, and combustor, will be different from case to case if the same operating considerations are imposed such as the surge margin of the compressor, the inlet guide vane angle, the component efficiency of compressor, turbine, and combustor, the total pressure losses, and the blade cooling air, etc.

In the second batch (see Table A2 in the Appendix), the TIT for the cases of burning LCV fuels is reduced to match the net power output of NG fueled GT. With reduced TIT, the fuel flow rate is reduced and the size of the gas turbine will become comparable to the NG fueled GT, so only modest modification of the NG-fired GT is needed to burn the LCV fuels.

Fog/Overspray effect on compressor

Figures 5 and 6 show the compressor discharge temperature and compressor power under four different ambient conditions, respectively. In both figures, a vertical saturation line is drawn to clearly separate underspray from overspray regions. Figure 5 shows that the compressor discharge temperature decreases with an increase of water spray but with a decrease of ambient temperature or humidity. Increasing ambient relative humidity allows less water spray to achieve saturation, so the compressor discharge temperature increases. Figure 6 shows the compressor power increases with an increase of ambient temperature or humidity, but decreases with the increase of water spray due to increased air density as previously explained in the theory.

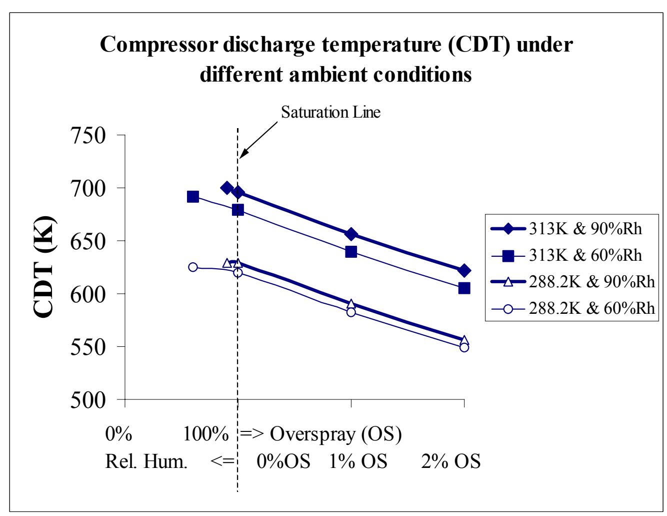

Figure 5 Compressor discharge temperatures under different ambient conditions

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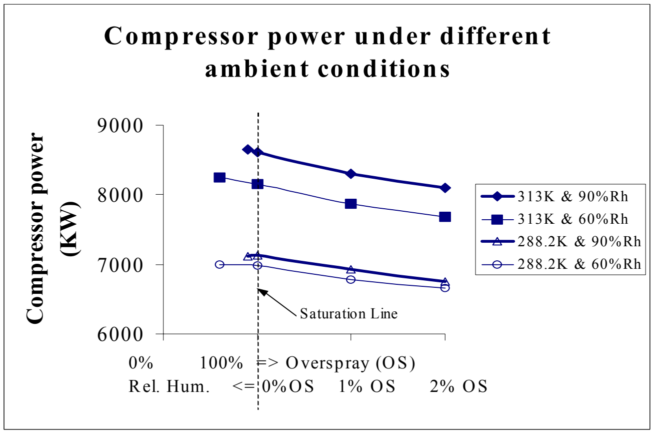

Figure 6 Compressor power under different ambient conditions

Fog/Overspray effect on fuel compressor

Figure 7 shows the fuel compressor work using both NG and LCV fuels. Fuel compressor work is significant for LCV fuels. As these fuels have less heating values, more mass flow rates are needed to achieve the required heat input in the combustion chamber. When natural gas is used, fuel compressor consumes about 4% of the main air compressor power (about 2% of the gross power produced by the turbine). The effect of fog/overspray is negligible on fuel compressor power in NG fired GT, as shown by overlapped curves in Fig. 7. The fuel compressor power increases to 20-40% of the main air compressor power (or 10-16% of the gross turbine power) when LCV-2 and LCV-1 fuels are burned respectively.

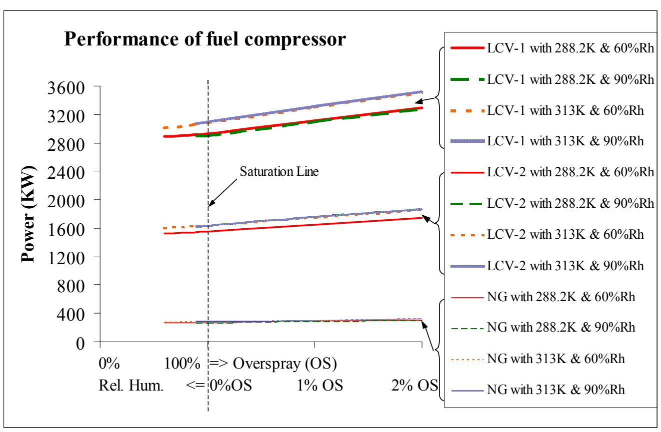

Figure 7 Fuel compressor power under different ambient conditions

Figure 7 also shows that the fuel compressor power increases with the increase of overspray percentage because more overspray requires more fuel (see Fig. 8) to achieve the TIT value at 1400K. This is contrary to the descending power consumption trend of the main air compressor when overspray is increased in Fig. 6. Effect of ambient temperature and relative humidity on the required fuel compressor power is not significant because the ambient pressure does not change. The only change takes place due to the ambient condition is air density, which is considered in Eq. 22. The effect of fuel heating value is predominant.

As more mass flow rate of LCV fuels are needed to provide sufficient energy as in the NG cases, additional energy is needed to heat up the inert gases in the LCV fuels as shown in Figure 8. The heating value of LCV-1 fuel is less than one tenth of the natural gas, and the required fuel mass flow rate is about 17 times more than NG to achieve TIT = 1400K. The heating value of LCV-2 fuel is about one seventh of the NG and the required fuel mass flow rate is about 8 times more than NG to achieve required TIT.

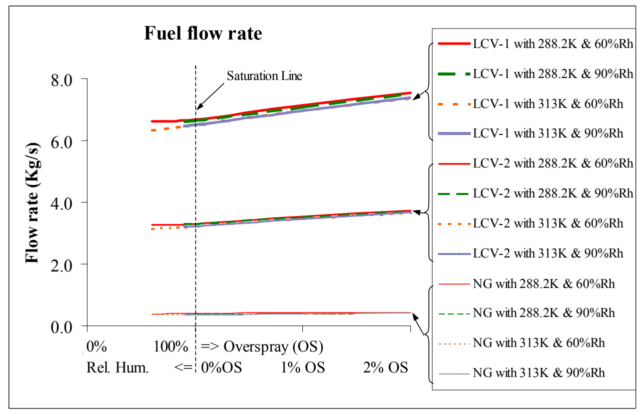

Figure 8 Fuel mass flow rate under different ambient conditions for various fuels

Fog/Overspray effect on combustor

Figure 9a shows that the heat added into the combustion chamber from LCV-1 and LCV-2 fuels is 46% and 23% more than NG, respectively. As heating value decreases for LCV fuels, there are more non-combustible gases in the fuel to absorb the energy and suppress the combustion temperature, so more heat addition is required to allow the combusted gas to reach the desired TIT. LCV-1 consists of 37% N_2, 23% water vapor and 11% CO_2, and LCV-2 consists of 37% water vapor and 17% CO_2, which are all non-combustible gases. When the overspray percentage is increased, more non-combustible water vapor is in the combustion gas to absorb heat, so the required heat addition is further increased as overspray ratio increases (see Fig. 9a).

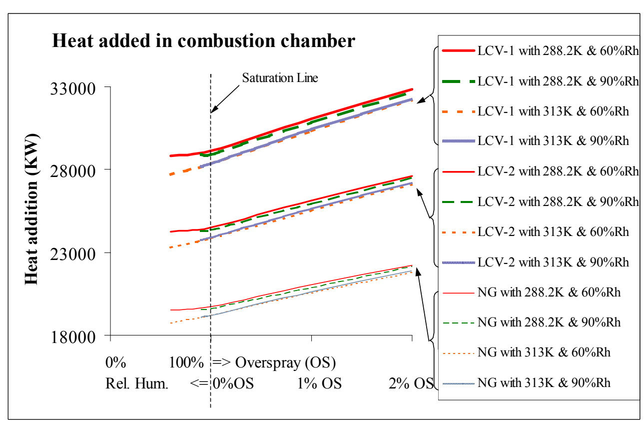

Figure 9a Heat added in the combustor under different ambient conditions.

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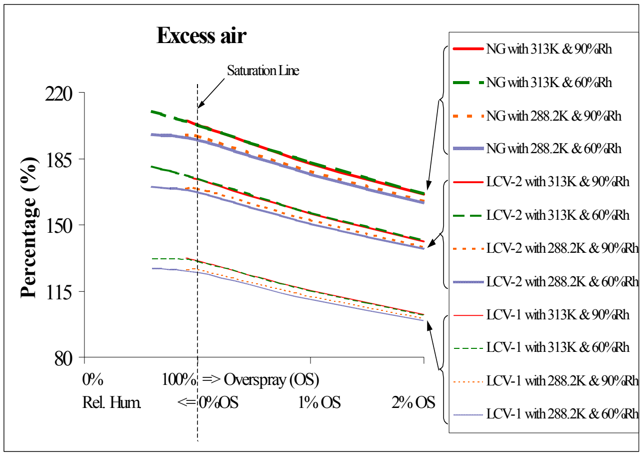

Figure 9b Percentage of Excess air under different ambient conditions

Heat addition is not very much affected by the ambient condition; lower ambient temperature obviously requires more heat addition. Excess air reduces as overspray increases (Fig. 9b) due to increased water vapor acting as a temperature-suppressing diluent.

Fog/Overspray effect on turbine

Figure 10 shows turbine gross power increases up to 30% for using LCV-1 fuel and up to 15% for using LCV-2 fuel from the NG fueled output because fuel mass flow rates are significantly increased for using LCV fuels. Notice again, as previously discussed, the LCV fired GT size will be different from the NG-fired GT if the same operating condition (surge margin, total pressure loss, etc.) and component efficiencies are imposed. The gross turbine power also increases as the overspray percentage increases. For example, an overspray of 2% increases turbine power up to 4% for natural gas and 6% for using LCV fuels.

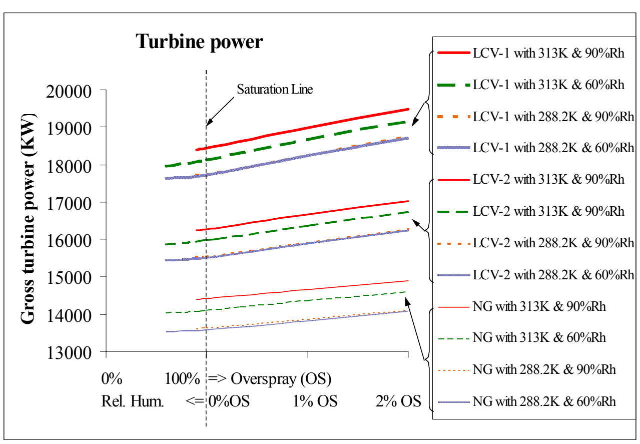

Figure 10 Gross turbine power under different ambient conditions

The net output power is calculated by deducting the air compressor power and fuel compressor power from the gross turbine power. Figure 11 shows that LCV fuels produce more net output power than natural gas even though LCV fuels significantly increases fuel compressor power 11 times for using LCV-1 and 6 times for using LCV-2 (see Fig. 7). When LCV fuels are burned, fog/overspray cooling seems as effective in achieving net power enhancement as when natural gas is burned. With saturated fogging, the net output power increases approximately 1-2%. With 2% of overspray, the net output power increases as high as 20%. As the ambient temperature increases, the net output power decreases; likewise increase of relative humidity lowers the net output power but with less impact than from the increased ambient temperature. Judging from the slopes of the curves in Fig. 11, rate of increase of net output power for overspray is higher for higher temperature and higher humidity when either NG or LCV fuels are burned.

Figure 10 shows that higher ambient temperature or humidity actually increases the gross turbine power; in these conditions, however, both the air and fuel compressor powers consumed are also increased (Fig. 2 & 3). As a result, the net effect favors larger net power output at low ambient and humidity conditions.

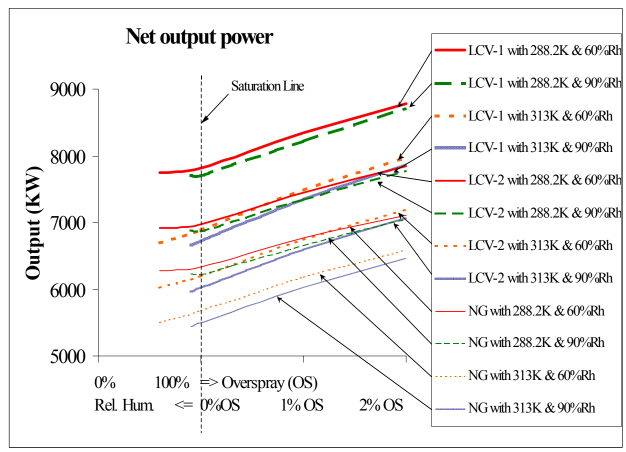

Figure 11 Net output power under different ambient conditions

Fog/Overspray effect on thermal efficiency

For cases using LCV fuels, the thermal efficiency is approximately 10~16% (or 3 ~ 5 percentage points) less than using NG because the fuel compressor consumes a significant auxiliary power. In the previous discussions, the influence of fog/overspray is either monotonously decreasing (such as compressor power and excess air) or monotonously increasing (such as fuel compressor power and net power output). The trend of efficiency variation is not so straightforward. Taking natural gas in Fig. 12 for example, 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 at hotter days. Since the thermal efficiency may slightly decrease or increase under fog/overspray conditions, considering the uncertainty of the current ideal model, fog/overspray should be considered as a means to augment power output, but not necessarily efficiency.

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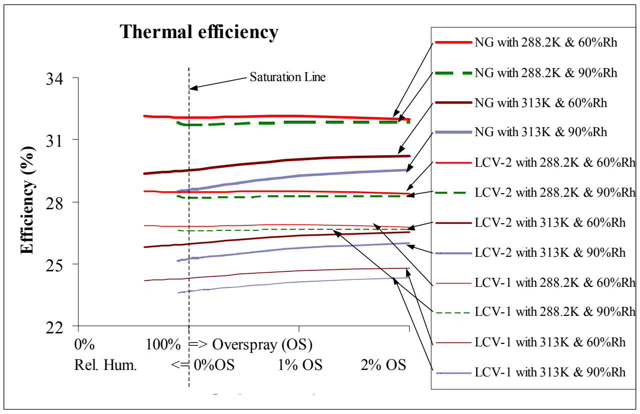

Figure 12 Thermal efficiency under different ambient conditions

Modifications of NG-fired GT for burning LCV fuels

Although the above results show that using the LCV fuels produces more net output power, as previously discussed, the present simulation requires to use bigger GT to burn LCV fuels if the same operating condition (surge margin, total pressure loss, etc.) and component efficiencies are imposed as the NG-fired GT.

This simulation treats the combustor as a black box and assumes that the combustor is functional when LCV fuels are burned. The actual combustion mechanisms are not modeled.

When lCV fue In this study, the pressure ratio is maintained at a fixed value of 12 by using the same compressor but different turbines. The turbine could be modified by (a) increasing the tip/hub ratio, (b) reducing the solidity (i.e. reducing the turbine blade numbers) but increasing the loading factor of each blade, or (c) using the same airfoils but with different staggering angle and incidence angle. Method (a) will increase the radius of the turbine. Method (b) can maintain the same size of the turbine but needs to redesign the turbine airfoils in order to achieve the same turbine performance. Method (c) keeps the same turbine airfoils but with degraded turbine performance because the incidence angle and lift coefficient will be altered as the staggering angle is compromised.

In the real application when a commercial GT is used, the turbine nozzle area needs to be opened (method C above) to maintain this fixed pressure ratio when LCV fuels are burned. Otherwise, the flow may be choked at the first-stage turbine nozzle. Consequently, the pressure will increase due to this choking condition as well as the increased friction in combustor and in the non-modified turbine stages. When the compressor is working against a higher than designed back pressure, it will operate off the design point with a reduced stability margin. Brun, et. al. [16] specifically discussed a simplified method to evaluate the principal factors that affect the aerodynamic stability of a single shaft gas turbine's axial compressor. Their analysis showed that when inlet and interstage water injection is combined with other factors such as LCV fuels and combustor steam injection, gas turbine compressor aerodynamic stability problems such as rotating stall and flutter will likely occur. These aerodynamic instabilities can be directly linked to blade high-cycle fatigue and possible catastrophic gas turbine failure. Furthermore, any water injection into a gas turbine may affect the hot-section turbine parts life. Therefore, care must be taken to employ inlet fog cooling when LCV fuels are burned. A companion paper by Roy and Wang [17] assesses the option of changing pressure and TIT to optimize a commercial GT output power and efficiency when producer gases are burned.

If the NG-fired GT is to be used to burn LCV fuels and it is desired to minimize modifications, it would be interested in studying the effect of reducing TIT on the GT performance. Reducing TIT will decrease the work output and hence will reduce the load on each blade. This approach will allow method (c) to be used by changing the stagger angle without changing the airfoils. Figures 13a&b show the results of varying TIT (see data in Table A2). An upper limit of 125% output power is drawn in Fig. 13 to represent the two limiting factors: the maximum shaft power rating and the capacity of electric generators. Both are designed with accepting 20-25% additional power output. (Note, some OEMs do not recommend operating over 15% of the rated capacity.) The lower broken line is the designed power output of the NG-fired GT.

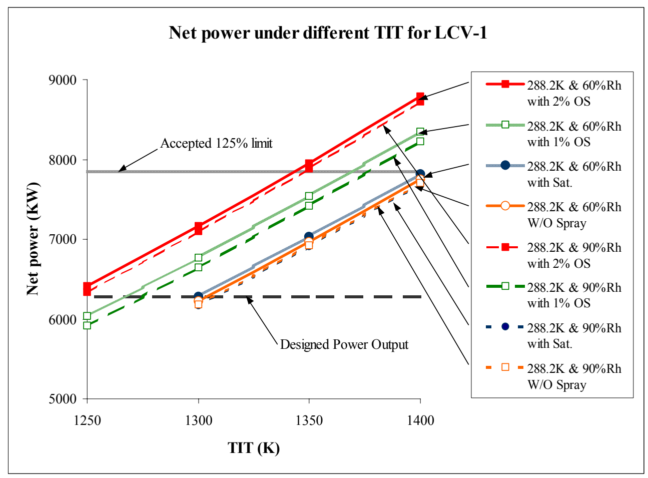

Figure 13a Net Output power under different TIT conditions for LCV-1

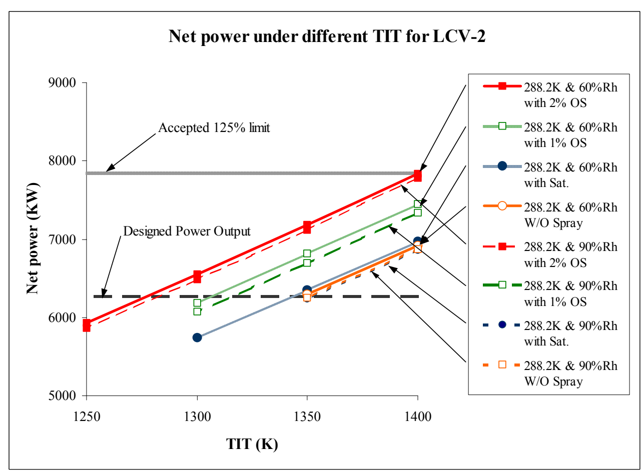

Figure 13b Net Output power under different TIT conditions for LCV-2

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In each case, the performance curves for dry compression and saturation compression almost coincide. When LCV fuels are burned, all of the cases are within the maximum limit except three LCV1 case are above the 125% limit. Take the Case of 288K, 60%RH with 2% overspray for example, the net output power is unacceptable high (over 125%) to the NG-fired GT. However, if the TIT is reduced to 1250K, the output power is comparable to the NG-fired GT. In this case, it is interesting to see that the net output power of LCV1-fired GT degrades profoundly 27% as TIT decreases 150K, but the thermal efficiency only reduces less than 0.5 percentage point (Table A2).

SUMMARY

A wet compression thermodynamic model of gas turbine system (FogGT) with inlet fog cooling specifically for burning LCV fuels has been developed in this paper. The theory of wet compression with fog and overspray cooling is introduced. The fog and overspray cooling reduces the compressor work and increases the net output power, but not necessarily the cycle thermal efficiency. In the mean time, fog/overspray also increases the total mass flow rate, which further increases the power output and result to a significant augmentation of the net power output.

Based on fixed pressure ratio, TIT, air-fuel mass flow, and component efficiencies, the results of simulations show:

(a) In the main compressor --- the compressor power consumption increases with an increase of ambient temperature or humidity, but decreases with the increase of water spray due to increased air density after cooling.

(b) In fuel compressor --- When LCV fuels are burned, the fuel compressor consumes about 10-16% of the gross turbine output power in comparison with 2% when NG is burned. The fuel compressor power increases with the increase of overspray percentage because more overspray requires more fuel to achieve the TIT value. This is contrary to the descending power consumption trend of the main air compressor with increased overspray. Effect of ambient temperature and relative humidity on the required fuel compressor power is not significant. The effect of heating value is predominant.

(c) In combustor --- As heating value decreases for LCV fuels, there are more uncombustible gases in the fuel to absorb the energy and suppress the combustion temperature, so more heat addition (23%-46%) is required to allow the combusted gas to reach the desired TIT.

(d) In turbine --- 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 a net output power increases 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.

(e) Thermal efficiency --- For LCV fuels, the thermal efficiency is approximately 10~16% (3~5 percentage points) lower than using the natural gas. Burning LCV fuels leads to small change in thermal efficiency irrespective of a large increase in net power output, due to increased demand of additional heat input to make up the sensible heat required for increased fuel flow rate including uncombustible gases. Fog/overspray could either slightly increase or decrease the thermal efficiency depending on the ambient conditions.

If NG-fired GT is to be used for LCV fuels, modifications of turbine through-flow passage or airfoils are required. The limits imposed by the capacity ratings of electric generator and the shaft material strength need to be considered. It must be noted that both the fog/overspray devices and fuel compressor will increase the operating and maintenance costs, which are not included in the analysis in this paper.

ACKNOWLEDGEMENT

This study is supported by the Louisiana Governor’s Energy Initiative via the Clean Power and Energy Research Consortium (CPERC) and administered by the Louisiana Board of Regents.

REFERENCES

1. Meher-Homji, C.B. and Mee, T.R., 1999, "Gas Turbine Power Augmentation by Fogging of Inlet Air", Proceedings of 28th Turbomachinery Symposium, Houston, Texas, USA, September 1999.
2. Zheng, Q., Sun, Y., Li, S., Wang, Y, 2002, "Thermodynamic Analysis of Wet Compression Process in the Compressor of Gas Turbine", Proc. of ASME Turbo Expo 2002, Amsterdam, The Netherlands, June 3-6, ASME Paper No: GT-2002-30590.
3. Zheng, Q., Li, M., Sun, Y., 2003, "Thermodynamic Analysis of Wet Compression and Regenerative (WCR) Gas Turbine", Proceedings of ASME Turbo Expo 2003, Atlanta, Georgia, USA, June 16-19, ASME Paper No: GT-2003-38517.
4. Bhargava, R. and Meher-Homji, C.B., 2002, "Parametric Analysis of Existing Gas Turbines with Inlet Evaporative and Overspray Fogging", Proceedings of ASME Turbo Expo 2002, Amsterdam, The Netherlands, June 3-6, 2002, ASME Paper No: GT-2002-30560
5. Chaker, M., Meher-Homji, C.B., Mee, T.R., 2002, "Inlet Fogging of Gas Turbine Engines - Part A: Fog Droplet Thermodynamics, Heat Transfer and Practical Considerations", Proceedings of ASME Turbo Expo 2002, Amsterdam, The Netherlands, June 3-6, 2002, ASME Paper No: GT-2002-30562.
6. Chaker, M., Meher-Homji, C.B., Mee, T.R., 2002, "Inlet Fogging of Gas Turbine Engines - Part B: Fog Droplet Sizing Analysis, Nozzle Types, Measurement and Testing", Proceedings of ASME Turbo Expo 2002, Amsterdam, The Netherlands, June 3-6, 2002, ASME Paper No: GT-2002-30563.
7. Chaker, M., Meher-Homji, C.B., Mee, T.R., 2002, "Inlet Fogging of Gas Turbine Engines - Part C: Fog Behavior in Inlet Ducts, CFD Analysis and Wind Tunnel Experiments", Proceedings of ASME Turbo Expo 2002, Amsterdam, The Netherlands, June 3-6, 2002, ASME Paper No: GT-2002-30564.
8. Wang, T., Li, X. and Pinninti, V., 2004, "Simulation of Mist Transport for Gas Turbine Inlet Air Cooling", Proceedings of ASME International Mechanical Engineering Congress, 2004, Anaheim, California, USA, November 13-19, 2004, ASME Paper No: IMEGE-2004-60133.
9. Sexton, W.R., Sexton, M.R., 2003, "The Effects of Wet Compression on Gas Turbine Engine Operating Performance", Proceedings of ASME Turbo Expo 2003, Atlanta, Georgia, USA, June 16-19, 2003, ASME Paper No: GT-2003-38045.
10. Bagnoli, M., Bianchi, M., Melino, F., Peretto, A., Spina, P.R., Bhargava, R. and Ingistov S., 2004, "A Parametric Study of Interstage Injection on GE Frame 7EA Gas Turbine", Proceedings of ASME Turbo Expo 2004, Vienna, Austria, June 14-17, ASME Paper No: GT-2004-53042.
11. Härtel, C., Pfeiffer, P., 2003, "Model Analysis of High-Fogging Effects on the Work of Compression", Proceedings of ASME Turbo Expo 2003, Atlanta, USA, June 16-19, 2003, ASME Paper No: GT-2003-38117.

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12. White, A.J. and Meacock, A.J., 2003, "An Evaluation of the Effects of Water Injection on Compressor Performance", Proceedings of ASME Turbo Expo 2003, Atlanta, USA, June 16-19, 2003, ASME Paper No: GT-2003-38237
13. Bathie, W. W., 1998, Fundamentals of Gas Turbines, 2nd Ed., Chapter 4, John Wiley & Sons, Inc., 1998.
14. Cataldi, G., Güntner, H., Matz, C., McKay, T., Hoffmann, J., Nemet, A., Lecheler, S. and Braun, J., 2004, "Influence of High Fogging Systems on Gas Turbine Engine Operation and Performance", Proc. of ASME Turbo Expo 2004, Vienna, Austria, June 14-17, ASME Paper No: GT-2004-53788.
15. Khan, J. R. and Wang, T., 2005, "Development of the Computational Program FogGT for Wet Compression via Fog/Overspray Gas Turbine Inlet Cooling," ECCC Report 2005-07, Energy Conversion and Conservation Center, University of New Orleans.
16. Brun, Klaus, Kurz, R., and Simmons, H. R., 2005 "Aerodynamic Instability and Life Limiting Effects of Inlet And Interstage Water Injection Into Gas Turbines," Proceedings of ASME Turbo Expo 2005, Reno, Nevada, USA, June 6-9, ASME Paper No: GT-2005-68007.
[17] Yap, M. R. and Wang, T., " Simulation of Producer Gas Fired Power Plants with Inlet Fog Cooling and Steam Injection, " ASME Paper GT-2006-90164, manuscript submitted to the ASME Turbo Expo2006, Barcelona, Spain, May 8-11, 2006.

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APPENDIX

Table A1 Data for different cases. All cases are conducted with a pressure ratio 12, compressor and turbine adiabatic efficiencies of 88%, and 20Kg/s of air flow rate. The power and thermal efficiency increases are compared with the first case (no fogging) in each category separated with alternating gray shade. "Dry" means no fogging, but the air flow contains moisture from the ambient.

Case	Description	CIT (K)	CDT (K)	Comp. Power (KW)	Fuel Comp. Power (KW)	Fuel Flow (Kg/s)	Heat Add. (KW)	Excess Air (%)	Turb. Power (KW)	Net Output Power (KW)	Therm Eff (%)	Net Power Increase (%)	Eff. Increase (%)
01	NG-288.2K w. 60%Rh	288.2	625.5	6997	271	0.391	19544	198.0	13546	6278	32.12	----	----
02	NG-288.2K w. 60%Rh, Sat.	283.9	619.7	6982	274	0.395	19746	194.9	13588	6333	32.07	0.87%	-0.17%
03	NG-288.2K w. 60%Rh, 1% OS	283.9	581.8	6775	292	0.420	21042	176.8	13833	6766	32.16	7.77%	0.10%
04	NG-288.2K w. 60%Rh, 2% OS	283.9	549.0	6653	308	0.444	22242	161.8	14077	7115	31.99	13.33%	-0.42%
05	NG-288.2K w. 90%Rh	288.2	629.0	7113	272	0.391	19588	197.3	13621	6237	31.84	----	----
06	NG-288.2K w. 90%Rh, Sat.	287.2	628.7	7133	272	0.392	19613	196.9	13631	6227	31.75	-0.16%	-0.28%
07	NG-288.2K w. 90%Rh, 1% OS	287.2	590.9	6926	290	0.418	20905	178.6	13876	6660	31.86	6.78%	0.06%
08	NG-288.2K w. 90%Rh, 2% OS	287.2	556.2	6758	307	0.443	22153	162.9	14120	7055	31.85	13.12%	0.03%
09	NG-313K w. 60%Rh	313.0	691.6	8244	283	0.375	18775	210.2	14037	5510	29.35	----	----
10	NG-313K w. 60%Rh, Sat.	305.4	678.8	8155	289	0.384	19213	203.1	14118	5674	29.53	2.98%	0.64%
11	NG-313K w. 60%Rh, 1% OS	305.4	639.4	7872	310	0.411	20572	183.1	14363	6182	30.05	12.20%	2.40%
12	NG-313K w. 60%Rh, 2% OS	305.4	605.2	7685	329	0.436	21825	166.8	14607	6594	30.21	19.68%	2.96%
13	NG-313K w. 90%Rh	313.0	699.6	8648	288	0.382	19119	204.6	14388	5452	28.51	----	----
14	NG-313K w. 90%Rh, Sat.	311.2	696.2	8616	290	0.384	19237	202.7	14408	5503	28.60	0.94%	0.32%
15	NG-313K w. 90%Rh, 1% OS	311.2	656.5	8305	310	0.412	20621	182.4	14654	6039	29.28	10.77%	2.70%
16	NG-313K w. 90%Rh, 2% OS	311.2	622.2	8096	330	0.437	21892	166.0	14898	6472	29.57	18.73%	3.69%
17	LCV 1-288.2K w. 60%Rh	288.2	625.5	6997	2894	6.619	28847	127.1	17639	7748	26.86	----	----
18	LCV 1-288.2K w. 60%Rh, Sat.	283.9	619.7	6982	2924	6.687	29146	124.8	17724	7817	26.82	0.90%	-0.14%
19	LCV 1-288.2K w. 60%Rh, 1% OS	283.9	581.8	6775	3116	7.126	31058	111.0	18240	8348	26.88	7.75%	0.08%
20	LCV 1-288.2K w. 60%Rh, 2% OS	283.9	549.0	6653	3294	7.532	32829	99.6	18735	8787	26.77	13.42%	-0.34%
21	LCV 1-288.2K w. 90%Rh	288.2	629.0	7113	2901	6.634	28913	126.6	17723	7709	26.66	----	----
22	LCV 1-288.2K w. 90%Rh, Sat.	287.2	628.7	7133	2904	6.642	28949	126.3	17738	7701	26.60	-0.11%	-0.23%
23	LCV 1-288.2K w. 90%Rh, 1% OS	287.2	590.9	6926	3096	7.080	30856	112.3	18254	8231	26.68	6.77%	0.05%
24	LCV 1-288.2K w. 90%Rh, 2% OS	287.2	556.2	6758	3281	7.502	32698	100.4	18759	8721	26.67	13.12%	0.02%
25	LCV 1-313K w. 60%Rh	313.0	691.6	8244	3020	6.358	27713	136.4	17969	6705	24.19	----	----
26	LCV 1-313K w. 60%Rh, Sat.	305.4	678.8	8155	3090	6.507	28358	131.0	18141	6897	24.32	2.86%	0.52%
27	LCV 1-313K w. 60%Rh, 1% OS	305.4	639.4	7872	3309	6.967	30365	115.8	18672	7491	24.67	11.73%	1.97%
28	LCV 1-313K w. 60%Rh, 2% OS	305.4	605.2	7685	3510	7.391	32213	103.4	19178	7983	24.78	19.06%	2.43%
29	LCV 1-313K w. 90%Rh	313.0	699.6	8648	3075	6.475	28220	132.2	18391	6668	23.63	----	----
30	LCV 1-313K w. 90%Rh, Sat.	311.2	696.2	8616	3094	6.515	28394	130.7	18436	6727	23.69	0.88%	0.26%
31	LCV 1-313K w. 90%Rh, 1% OS	311.2	656.5	8305	3317	6.983	30437	115.3	18972	7351	24.15	10.24%	2.20%
32	LCV 1-313K w. 90%Rh, 2% OS	311.2	622.2	8096	3521	7.414	32313	102.8	19483	7866	24.34	17.96%	3.01%
33	LCV 2-288.2K w. 60%Rh	288.2	625.5	6997	1531	3.277	24266	169.9	15443	6915	28.50	----	----
34	LCV 2-288.2K w. 60%Rh, Sat.	283.9	619.7	6982	1547	3.311	24517	167.2	15505	6976	28.45	0.88%	-0.15%
35	LCV 2-288.2K w. 60%Rh, 1% OS	283.9	581.8	6775	1648	3.528	26126	150.7	15875	7452	28.52	7.76%	0.09%
36	LCV 2-288.2K w. 60%Rh, 2% OS	283.9	549.0	6653	1742	3.729	27616	137.2	16235	7840	28.39	13.37%	-0.38%
37	LCV 2-288.2K w. 90%Rh	288.2	629.0	7113	1534	3.284	24322	169.3	15522	6875	28.27	----	----
38	LCV 2-288.2K w. 90%Rh, Sat.	287.2	628.7	7133	1536	3.288	24352	169.0	15534	6866	28.19	-0.13%	-0.26%
39	LCV 2-288.2K w. 90%Rh, 1% OS	287.2	590.9	6926	1637	3.505	25956	152.3	15905	7341	28.28	6.78%	0.05%
40	LCV 2-288.2K w. 90%Rh, 2% OS	287.2	556.2	6758	1735	3.714	27506	138.1	16270	7777	28.27	13.12%	0.02%
41	LCV 2-313K w. 60%Rh	313.0	691.6	8244	1597	3.148	23312	181.0	15859	6018	25.81	----	----
42	LCV 2-313K w. 60%Rh, Sat.	305.4	678.8	8155	1634	3.221	23855	174.6	15982	6194	25.96	2.92%	0.58%
43	LCV 2-313K w. 60%Rh, 1% OS	305.4	639.4	7872	1750	3.449	25543	156.4	16360	6738	26.38	11.98%	2.20%
44	LCV 2-313K w. 60%Rh, 2% OS	305.4	605.2	7685	1856	3.659	27098	141.7	16725	7184	26.51	19.39%	2.71%
45	LCV 2-313K w. 90%Rh	313.0	699.6	8648	1626	3.206	23739	175.9	16243	5969	25.14	----	----
46	LCV 2-313K w. 90%Rh, Sat.	311.2	696.2	8616	1636	3.225	23885	174.2	16275	6023	25.22	0.91%	0.29%
47	LCV 2-313K w. 90%Rh, 1% OS	311.2	656.5	8305	1754	3.457	25604	155.8	16655	6596	25.76	10.51%	2.46%
48	LCV 2-313K w. 90%Rh, 2% OS	311.2	622.2	8096	1862	3.670	27182	141.0	17023	7065	25.99	18.36%	3.37%

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Table A2 Design data for varying TIT. All cases are conducted with a pressure ratio 12, compressor and turbine adiabatic efficiencies of 88%, and 20Kg/s of air flow rate. The net power of each case is graphed in Figures 13a and 13b to show the results of changing TIT. The power and thermal efficiencies are compared with the first case in each category separated with alternating gray shade.

Amb. Temp.	Rel. Hum.	TIT	Fogging	Fuel	Net Power	Efficiency	Fuel	Net Power	Efficiency
(K)	(%)	(K)			(KW)	(%)		(KW)	(%)
288.2	60	1400	Dry	NG	6278	30.00	NG	6278	30.00
288.2	60	1400	Dry	LCV 1	7748	26.86	LCV 2	6915	28.50
288.2	60	1350	Dry	LCV 1	6966	26.74	LCV 2	6295	28.36
288.2	60	1300	Dry	LCV 1	6223	26.54	LCV 2	-------	-------
288.2	60	1400	Sat	LCV 1	7817	26.82	LCV 2	6976	28.45
288.2	60	1350	Sat	LCV 1	7032	26.70	LCV 2	6353	28.31
288.2	60	1300	Sat	LCV 1	6285	26.50	LCV 2	5744	28.10
288.2	60	1400	1% OS	LCV 1	8348	26.88	LCV 2	7452	28.52
288.2	60	1350	1% OS	LCV 1	7539	26.78	LCV 2	6813	28.41
288.2	60	1300	1% OS	LCV 1	6769	26.61	LCV 2	6190	28.23
288.2	60	1250	1% OS	LCV 1	6038	26.37	LCV 2	-------	-------
288.2	60	1400	2% OS	LCV 1	8787	26.77	LCV 2	7840	28.39
288.2	60	1350	2% OS	LCV 1	7955	26.67	LCV 2	7185	28.28
288.2	60	1300	2% OS	LCV 1	7164	26.51	LCV 2	6548	28.11
288.2	60	1250	2% OS	LCV 1	6412	26.27	LCV 2	5927	27.86
288.2	90	1400	Dry	LCV 1	7709	26.66	LCV 2	6875	28.27
288.2	90	1350	Dry	LCV 1	6924	26.52	LCV 2	6251	28.11
288.2	90	1300	Dry	LCV 1	6177	26.30	LCV 2	-------	-------
288.2	90	1400	Sat	LCV 1	7701	26.60	LCV 2	6866	28.19
288.2	90	1350	Sat	LCV 1	6915	26.45	LCV 2	6241	28.03
288.2	90	1300	Sat	LCV 1	6167	26.23	LCV 2	-------	-------
288.2	90	1400	1% OS	LCV 1	8231	26.68	LCV 2	7341	28.28
288.2	90	1350	1% OS	LCV 1	7421	26.55	LCV 2	6700	28.14
288.2	90	1300	1% OS	LCV 1	6651	26.36	LCV 2	6076	27.93
288.2	90	1250	1% OS	LCV 1	5919	26.08	LCV 2	-------	-------
288.2	90	1400	2% OS	LCV 1	8721	26.67	LCV 2	7777	28.27
288.2	90	1350	2% OS	LCV 1	7887	26.56	LCV 2	7121	28.15
288.2	90	1300	2% OS	LCV 1	7095	26.39	LCV 2	6482	27.96
288.2	90	1250	2% OS	LCV 1	6342	26.14	LCV 2	5859	27.70
313	60	1400	Dry	LCV 1	6705	24.19	LCV 2	6018	25.81
313	60	1350	Dry	LCV 1	5932	23.85	LCV 2	-------	-------
313	60	1400	Sat	LCV 1	6897	24.32	LCV 2	6194	25.96
313	60	1350	Sat	LCV 1	6116	24.00	LCV 2	-------	-------
313	60	1400	1% OS	LCV 1	7491	24.67	LCV 2	6738	26.38
313	60	1350	1% OS	LCV 1	6686	24.41	LCV 2	6090	26.10
313	60	1300	1% OS	LCV 1	5922	24.07	LCV 2	-------	-------
313	60	1400	2% OS	LCV 1	7983	24.78	LCV 2	7184	26.51
313	60	1350	2% OS	LCV 1	7155	24.55	LCV 2	6520	26.27
313	60	1300	2% OS	LCV 1	6369	24.25	LCV 2	5874	25.94
313	90	1400	Dry	LCV 1	6668	23.63	LCV 2	5969	25.14
313	90	1350	Dry	LCV 1	5876	23.22	LCV 2	-------	-------
313	90	1400	Sat	LCV 1	6727	23.69	LCV 2	6023	25.22
313	90	1350	Sat	LCV 1	5932	23.30	LCV 2	-------	-------
313	90	1400	1% OS	LCV 1	7351	24.15	LCV 2	6596	25.76
313	90	1350	1% OS	LCV 1	6532	23.83	LCV 3	5935	25.42
313	90	1300	1% OS	LCV 1	5755	23.42	LCV 4	-------	-------
313	90	1400	2% OS	LCV 1	7866	24.34	LCV 2	7065	25.99
313	90	1350	2% OS	LCV 1	7023	24.07	LCV 2	6387	25.69
313	90	1300	2% OS	LCV 1	6225	23.70	LCV 2	5729	25.31

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