Power Enhancement of Gas Turbine Plant by Intake Air Fog Cooling

J.P. Yadav[1*], Bharat Raj Singh[2] and Onkar Singh[3]

ABSTRACT

Although gas turbines are known as constant volume machines, but its performance considerably depends upon the ambient air temperature and mass flow rate. During summer season the density of the air decreases which affects the mass flow rate and ultimately the power output of a gas turbine is reduced. In order to overcome this situation several techniques are already in the practice and one of the most effective and economical is adopting the inlet fog cooling, and this technique basically enhances the power output of the machine. The cooling of ambient air by fog cooling up to wet bulb temperature increases the mass flow rate on account of increase in air density, as a result it ultimately increases the power output of a gas turbine. Fogging is applied with consideration of relative humidity of ambient air not only during summer season but also during dry days of summer season in order to increase the power output of gas turbine. This paper describes the effect on percentage enhancement of power out adopting various fuel options with low and high humidity ambient conditions. The result indicates the potential increase in the power output up to 14%. It is also observed that the total cost of power production increases due to increase in fuel consumption on account of enhanced power output. Thus the best suitable selling cost of power should be selected to compensate the increased investment on fuel cost.

Keywords: *Inlet air cooling, fogging, gas turbine power*

1. INTRODUCTION

It is well established fact that the versatility of gas turbine is greatly degraded by ambient conditions such as high air temperature, humidty and dusty environment. The most favorable condition to operate a gas turbine is 288.6 K. Basically, there are two methods for improving the power output of gas turbine:-

- Reduction in compressor work.
- Increase in the gross turbine work output.

The reduction in compressor work is basically done by evaporative cooling me thod. This method is gaining popularity in air conditioning applications and gas turbine power augmentation .

In fog cooling the water is injected at inlet manifold under high pressure 14 -25 MPa [1] with specially designed nozzles whose basic function is to atomize the water into fog droplets so that they could be sprayed over a large area and evaporate quickly and effectively whereas in conventional evaporative cooling only 90% saturation can be achieved.

1*. JP Yadav, Associate Professor, Chandra Shekhar Azad University of Agriculture & Technology , Campus-Etawah (U.P.)-India, e-mail: jpyadav_caet@yahoo.com

2. Bharat Raj Singh, Professor, Mech. Engg. Deptt., School of Management Sciences, Technical Campus, Lucknow-227125, (U.P.) -India, e-mail: brsinghlko@yahoo.com

3. Onkar Singh, Professor, Mech. Engg. Deptt., Harcourt Butler Technological Institute, Kanpur (U.P.)- India, e-mail:onkpar@rediffmail.com

Nomenclature

CDT	Compressor discharge temperature (K)
CIT	Compressor inlet temperature (K)
C_p	Specific heat at constant pressure (kJ/kg-K)
D.A	Dry Air
DBT	Dry bulb temperature (K)
f'	Fuel Air ratio
h	Specific enthalpy (kJ/kg)
H	Enthalpy of reaction
K	Polytropic index
L	Latent heat of water (kJ/kg)
NG	Natural gas
A	Mass flow rate (kg/s)
OS	Over spray
p	Pressure (kPa)
Q_{in}	Heat added in combustion chamber (kJ)
R	Gas constant
RH	Relative humidity
s	Specific entropy (kJ/kg-K)
T	Temperature (K)
T_D	Dew point temperature (K)
TIT	Turbine inlet temperature (K)
v	Specific volume (m^3/kg)
WBT	Wet bulb temperature (K)
LCV	Lower calorific value (kJ)

Greek symbols

γ	Specific heat ratio
η	Thermal Efficiency
ø	Relative humidity
ω	Humidity ratio (m_v/m_g)
ρ	Density(kg/m^3)

Subscripts

1	Compressor inlet
2	Compressor exit
3	Turbine Inlet
4	Turbine exit
1f	Fuel in ambient condition
2f	Fuel in compressor discharge condition
a	Dry air
Amb	Ambient
c	Compressor
fc	Fuel compressor
f	Liquid water
g	Water vapour
I	Index for different element
Pr	Product
Re	Reactant
T	Turbine
Th	Thermal
DB	Dry bulb
WB	Wet bulb
ac	Air cooler

2. EFFECT OF ELEVATED AMBIENT TEMPERATURE ON GT PERFORMANCE

The effect of elevated tempera ture on GT power output and efficiency can be e xplained by analyzing the P-v and T-s diagrams. Path 1-2-3-4 in Fig.1 shows the ideal Brayton cycle at the reference ISO condition (15°C and 60% relative humidity) and 1'-2'-3-4 shows the processes on hot days.

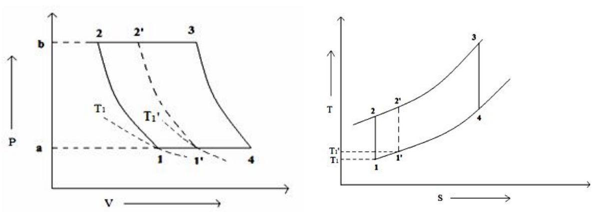

Fig. 1: Effect of increased ambient temperature on gas turbine efficiency and output power per unit mass flow rate.

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 the power at ISO condition. The turbine power output remains same in both the conditions, so the net power output (per unit mass flow rate) decreases.

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

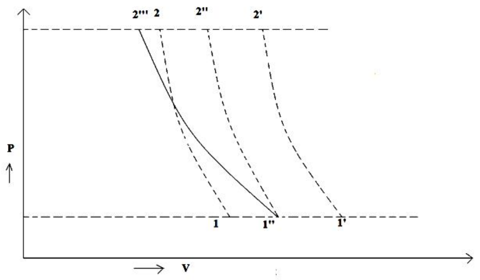

Fig. 2: Different fog/overspray cooling processes in the air–intake duct and in the compressor

The effects of fog cooling and overspray are shown in Figure 2 where 1-2 shows th e compression under the ISO condition; 1'-2' shows the compression in an elevated ambient temperature c ondition, 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?. It is also noticed 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 p rocess (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 can be seen by the curve 1?-2'? (moist compression with overspray). Therefore, fog and overspray cooling increases in both the cases ne t output power as well as the thermal efficiency of c ycle. In the mean time, fog/overspray further increases the total mass flow rate, which does not affect th e thermal efficiency but increases the power output. He nce, augmentation of the total power output is more pronounced than efficiency.

3. DEVELOPMENT OF WET COMPRESSION FORMULATION FOR FOG/OVERSPRAY COOLING GAS TURBINE SYSTEM

Fig. 1 is again referred, with a different representation of curves 1-2 and 1'-2' from earlier description. During the deriva tion of wet compression formulation, isobaric line 1'- 1 represents the inlet fog cooling where evaporation of water takes place to saturate the air. Polytropic l ine 1-2 represents either moist compression (saturated a ir without overspray) or wet compression (intercooli ng due to overspray). Isentropic line 1'-2' represen ts compression process of the main compressor without fog/overspray cooling. Assuming the fuel is supplied at the ambient temperature, line 1'-2' repres ents the compression of the fuel compressor although a t a different mass flow rate.

According to Gibb's equation:

$$Tds = dh - \frac{dp}{\rho}$$

Because,

$$Tds = dh - VdP \qquad (1)$$

As we know,

Mass = volume × density

If m= 1

Then,

1 = volume × density

$$\text{Volume} = \frac{1}{density} = \frac{1}{\rho}$$

For an ideal wet compression p rocess 'Evaporation heat equals to reversible heat '

Then

$$Tds = - L.dW \qquad (2)$$

Where dW = mass flow

L = latent heat

(-ve sign indicates 'Heat reje ction')

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

Here, $dh = C_p dT = \frac{\gamma R}{\gamma - 1} dT$

From equation of state,

$$P = \rho RT$$

$$\frac{1}{\rho} = \frac{RT}{P}$$

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

It becomes

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

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

$$\frac{dP}{p} = \left[\frac{\gamma}{\gamma - 1} + \frac{L}{R}\frac{dW}{dT}\right]\frac{dT}{T} \qquad (5)$$

Assuming that the evaporative rate varies linearly with the temperature

$\frac{dW}{dT}$ = constant, then equation PV^{γ} = constant

$$PT - \frac{K}{K-1} = C$$

(Where k = polytropic index for wet compression)

$$\frac{dP}{p} = \frac{K}{K-1}\frac{dT}{T} \qquad (6)$$

From equation (11) and (12)

$$\frac{K}{K-1}\frac{dT}{T} = \frac{\gamma}{\gamma - 1} + \frac{L}{R}\frac{dW}{dT} \qquad (7)$$

Equation (7) shows that the increase of evaporation rate decreases the polytropic index (k) of wet compression from isentropic process (k = γ) 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. 2 as a less steep curve (1"-2"' vs. 1"-2") requires 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 T_1 is obtained by applying energy balance via enthalpy.

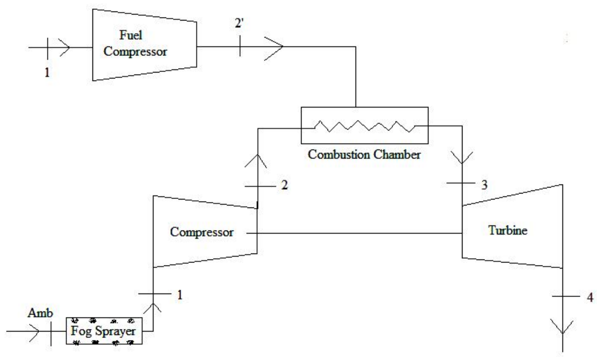

Fig. 3: Gas turbine system with fog spraying device

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

$$h_1 = \frac{\sum m_{f2} h_{f2}}{\sum m_{f2}} = \frac{m_{a2} h_{a2} + m_{f2} h_{f2} + m_{g2} h_{g2}}{m_{a2} + m_{f2} + m_{g2}} \qquad (8)$$

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

$$S_1 = \frac{\sum m_{f2} S_{f2}}{\sum m_{f2}} = \frac{m_{a2} s_{a2} + m_{f2} s_{f2} + m_{g2} s_{g2}}{m_{a2} + m_{f2} + m_{g2}} \qquad (9)$$

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 an overspray. In this paper the overspray percentage is

S-JPSET : ISSN : 2229-7111, Vol. 3, Issue 1

defined as the ratio of over sprayed 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,

$$s_2 = \frac{\sum m_{f2} s_{f2}}{\sum m_{f2}} = \frac{m_{a2}\left\{s_{a2} - R\ln\left(\frac{v_2}{p_2}\right)\right\} + m_{g2} s_{g2} + m_{f2} s_{f2}}{m_{a2} + m_{g2} + m_{f2}} \quad (10)$$

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

$$s_2 = \frac{\sum m_{f2} s_{f2}}{\sum m_{f2}} = \frac{m_{a2}\left\{s_{a2} - 0.287\ln\left(r_p\right)\right\} + m_{g2} s_{g2}}{m_{a2} + m_{g2}} \quad (11)$$

T_{2s} can be determined by assuming $s_1 = s_2$. 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,

$$h_{2s} = \frac{\sum m_{f2} h_{f2s}}{\sum m_{f2}} = \frac{m_{a2} h_{a2s} + m_{g2} h_{g2s}}{m_{a2} + m_{g2}} \quad (12)$$

The compressor efficiency is obtained as,

$$\eta_c = \frac{h_{2s} - h_1}{h_2 - h_1} \quad (13)$$

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

$$h_2 = \frac{\sum m_{f2} h_{f2}}{\sum m_{f2}} = \frac{m_{a2} h_{a2} + m_{g2} h_{g2}}{m_{a2} + m_{g2}} \quad (14)$$

Iteration is needed to determine T_2 by satisfying all the property values as function 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,

$$h_3 = \frac{\sum m_{f3} h_{f3}}{\sum m_{f3}} = \frac{m_{a2}(1+f_2)h_{a3} + m_{g2} h_{g3}}{m_{a2}(1+f_2) + m_{g2}} \quad (15)$$

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

$$s_3 = \frac{\sum m_{f3} s_{f3}}{\sum m_{f3}} = \frac{m_{a2}(1+f_2)s_{a3} + m_{g2} s_{g3}}{m_{a2}(1+f_2) + m_{g2}} \quad (16)$$

The fuel mass flow rate is 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:

$$s_4 = \frac{\sum m_{f4} s_{f4}}{\sum m_{f4}} = \frac{m_{a4}(1+f_4)\left[s_{a4} - R\ln\left(\frac{p_4}{p_3}\right)\right] + m_{g4} s_{g4}}{m_{a4}(1+f_4) + m_{g4}}$$

$$s_4 = \frac{\sum m_{f4} s_{f4}}{\sum m_{f4}} = \frac{m_{a4}(1+f_4)\left[s_{a4} - 0.287\ln\left(r_p\right)\right] + m_{g4} s_{g4}}{m_{a4}(1+f_4) + m_{g4}} \quad (17)$$

s_3 is set to equal s_4 to determine T_{4s}. All the property values in these two expressions are the functions of T_3 (which is assigned as an operating parameter) and T_{4s} (which is determined by iteration). At state 4, the isentropic enthalpy of moist air is calculated as

$$h_4 = \frac{\sum m_{f4} h_{f4}}{\sum m_{f4}} = \frac{m_{a4}(1+f_4)h_{a4} + m_{g4} h_{g4}}{m_{a4}(1+f_4) + m_{g4}} \quad (18)$$

Turbine efficiency is determined as,

$$\eta_t = \frac{h_3 - h_4}{h_3 - h_{4s}} \quad (19)$$

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

Compressor work, $w_c = h_2 - h_1$ (20)

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. Assume the fuel behaves as an ideal gas, the power required for fuel compressor is calculated as,

$$w_f = \frac{\gamma}{\gamma-1}\left(\frac{P_2}{\rho_f} - \frac{P_t}{\rho_{tf}}\right)\Big/\eta_{fc} = \frac{\gamma R\left(T_2 - T_1\right)}{\eta_{fc}\left(\gamma-1\right)} \quad (22)$$

To ensure that 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:

$$w_{net} = w_t - w_c - w_f = (m_3 w_t - m_2 w_c - m_f w_f) \quad (23)$$

Equations from (1) to (23), have been solved by the software EES (Engineering Equation Solver). It can be solved by hand calculation also.

4. RESULT AND DISCUSSION

The theoretical results are obtained for all cases by keeping the values of following parameters fixed as: compression ratio (12), TIT (1400 K), air mass flow rate (1200 Kg/min), inlet pressure (1atm), compressor, fuel compressor and turbine isentropic efficiency is 90%.This analysis has been carried out with three types of fuel (i.e. natural gas, LCV 1 and LCV 2). Four different conditions are introduced: (i) Low temperature, low humidity, (ii) Low temperature high humidity, (iii) High temperature low humidity and, (iv) high temperature high humidity.

Four different fog cooling are analyzed including moist compression (unsaturated air), compression with saturated air (100% RH), 1% OS and 2% OS. More than 2% OS is not recommended. The theoretical results have been recorded and shown in table 1.

4.1 Fog/Overspray Effect on compressor discharge temperature

Figure 4 and 5 shows the compressor discharge temperature and compressor power under four different ambient conditions, respectively. In both of figures, a vertical saturation line is drawn to clearly separate under spray from over spray regions.

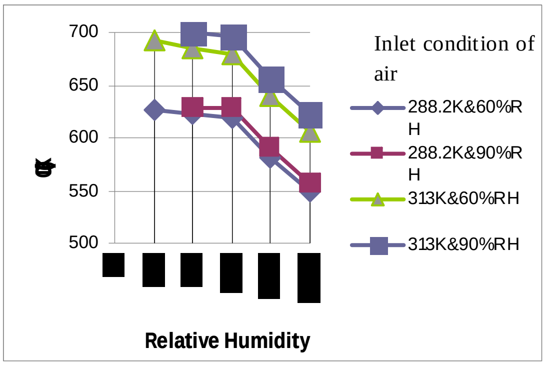

Fig. 4: Compressor Discharge Temperature under different conditions.

Figure 4 shows that the compressor discharge temperature reduces up to 2 % when fogging is done up to saturation. When overspray is being done then CDT reduces up to 7% - 8% and with 2% overspray, it reduces up to 13% - 15% in low ambient humidity condition, while in high ambient relative humidity, it reduces up to 10% - 20%. The reduction in compressor discharge temperature shows that when the rate of evaporation increases, the adiabatic compression tends towards isothermal compression.

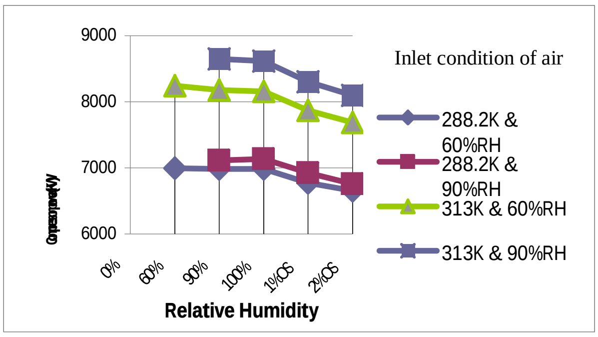

Fig.5: Compressor power under different conditions

S-JPSET : ISSN : 2229-7111, Vol. 3, Issue 1

4.2 Fog/overspray effect on compressor power

Figure 5 shows the compressor power (work input required to drive the compress or) under four different ambient conditions. According to output data and graph, it can be seen that the power input required to drive the compressor is reduce d up to 1% - 2% when fogging is done up to saturati on state, and it is reduced up to 4% -5% with 1% overspray and 6% - 8% with 2% overspray.

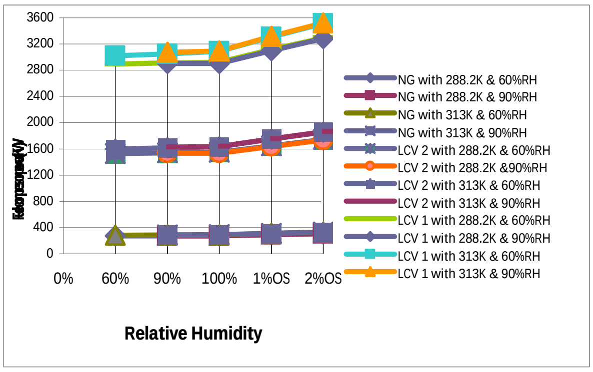

Fig. 6: Performance of fuel compressor under different conditions

4.3 Fog/Overspray effect on fuel compressor

Figure 6 shows that the fuel compressor power increases with the increase of overspray percentage because more overspray require s more fuel to achieve required TIT. \

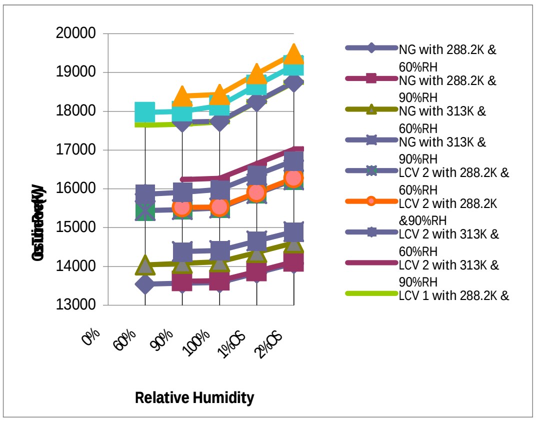

Fig.7: Gross Turbine Power under different conditions

Figure 6 also shows results of the fuel compressor work using both NG and LCV fuels. When natural gas is used, fuel compressor c onsumes about 4% of the main air compressor power (about 2% of the gross power produced by the turbine). It is also seen that the fuel compressor power incr eases 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.

4.4 Fog/Overspray effect on turbine

Figure 7 shows that the gross turbine power increases as the fog/overspray percentage increases. As an increase of 2% ove rspray, the turbine power output increases up to 4% for natural gas and 6% for LCV fuels.

4.5 Fog/overspray effect on Net Power Output

Figure 8 shows that the fog/ o verspray affect on the net power output. The net power output is calculated by deducting the air compressor power and fuel compressor power from the gross turbine power. It can be seen from out put data table.

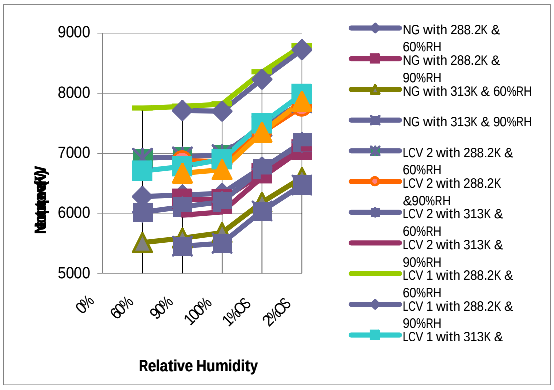

Fig. 8: Net output power under different conditions

When fogging is done in winter season (low temperature and low ambient hu midity), the net power output increases up to 0.87% in case of saturation; 7.77% and 13.33% increases in case of 1%OS and 2%OS respectively. But it can be seen also that at low temperature and high humid ity ambient condition, the net power output decreases up to 0.16% with saturation because in this case compressor power does not decrease sufficiently. It means fogging up to saturation is not beneficial against low temperature and high humidity ambient condition. But in case of 1% and 2% OS, the compressor power reduces sufficiently so net power increases up to 6.78% and 13.12% with 1%OS and 2% OS respectively.

In summer season, (high temperature and high humidity ambient condition), the net power output increases up to 2.9% with saturation and 12.20% and 19% with 1%OS and 2% OS respectively because in this case compressor power reduces sufficiently. It means fogging is more beneficial in high temperature ambient condition.

4.6 Fog/Overspray effect on thermal efficiency

Figure 9 shows the effect of relative humidity on thermal efficiency at different conditions. 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.

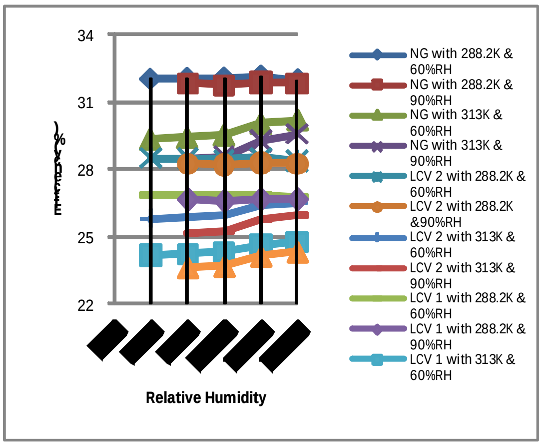

Fig.9: Thermal Efficiency under different conditions

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. The uncertainty of the current ideal model, fog/overspray should be considered as a means to augment power output, but not necessarily efficiency.

4.7 Economical Analysis

The economical analysis of gas turbine power plant without and with inlet air cooling by fogging system is carried out with the assumption that the plant is operating at full capacity for 365 days. Also, assuming the tariff of power is at the rate of Rs 3.70 per unit (kWh). The analysis shows that [see table 2] when fogging is done in low temperature and low relative humidity ambient condition, the increase in tariff up to 0.9% with fogging up to saturation of 7.20% and 11.8% against 1%OS and 2%OS respectively. It is important to note that when fogging is done up to saturation in low temperature high humidity ambient condition, there is economical loss around 0.15% but overspray of 1% or 2% is economically beneficial in this case. The most beneficial and economical conditions would be when fogging is done at high temperature and low humidity ambient condition exists while using natural gas as fuel.

5. CONCLUSION

From the above results it is concluded that:

- The compressor power (work required to move the compressor) is reduced up to 10% when fogging system is adopted and simultaneously the net work output increases up to 20% with fogging.
- Under some boundary conditions, efficiency is going to decrease while net work output is going to increase.
- Net work output is more pronounced than efficiency in some application of gas turbine system.
- The decrease in compressor discharge temperature shows that when fogging is adopted, the adiabatic compression process tend towards isothermal compression in the way to minimize the compression work.

- The result of economical analy sis shows that fogging is beneficial in terms of cost also.

According to the results data, it can also be seen that the increase in tariff could b e:
i) 0.7%- 2.8% with fogging up to saturation,
ii) 6.3% - 10% with 1%OS and
iii) 11.5% -16% with 2% OS.

REFERENCES

[1] Mc Craken, C.D. 1991-"Off peak air conditioning : a major energy saver, ASHRAE J.(12-13)

[2] MEE Industries inc. 2006 "Gas Turbine Inlet Air Cooling: Las Vegas Cogeneration".

[3] Mustapha Chaker, Mehar Homji and Thomas Mee, 2001 "Inlet fogging of gas turbine engines detailed climatic analysis of gas turbine evaporative cooling potential " ASME-2001-GT-0526.

[4] Jobaidur Rahman Khan And Ting wang "Fog Overspray Cooling for gas turbine system with low calorific values fuels"ASME-2006-GT-90396.

[5] Williams J. "Further effects of water ingestion on axial flow compressors and aero-engines at part speed"ASME-2008-GT-50620.

[6] Wang, T. and Khan, J.R .2008, "Over spray and Interstage fog cooling in compressor using stage staking scheme part-1 and 2"ASME -2008-GT-50323.

[7] Wang, T and Pinninti "Simulation of mist transport for gas turbine inlet air cooling"ASME-2004-GT-60133.

[8] Sexton, W.R., Sexton, M.R. "The effects of wet compression on gas turbine engine operating performance" ASME-2003-GT-38045.

[9] White A.J. Meacock A.J. "An evaluation of the effects of water injection on compressor performance" ASME-2003-GT-38237.

[10] Bagnoli, M. Bianchi, M. Melino and Peretto, "A parametric study of interstage injection on gas turbine" 2004.

[11] Mehar-Homji, C.B. 1995, "Blading Vibration And Failures In Gas Turbine Part A : Blading Dynamics And The Operating Environment; Part –B Compressor And Turbine Airfoil Distress;

[12] Chaker, M., Mehar –Homji, C. B., T. R. "Inlet fogging of gas turbine engines Part-A" ASME-2002-GT-30562, 30563, 30564

Table 1: Output Data for different cases

A. In Winter Season :

i). When Natural Gas as fuel

Case 1: Low humidity ambient condition

S. no	Description	CDT (K)	Comp power (kW)	Fuel Comp Power (kW)	Fuel Flow (kg/s)	Heat Add. (kW)	Turb. Power (kW)	Net Output power (kW)	**Net power increase (%)**	Therm Eff. (%)	Eff. Increase (%)	Remarks
1	NG-288.2K & 60%RH	625.5	6997	271	0.391	19544	13546	6278	----	32.12	---	Without fogging
2	NG-288.2K &60%RH,Sat.	619.7	6982	274	0.395	19746	13588	6333	**0.87**	32.07	-0.17	Fogging up to saturation (RH 100%)
3	NG-288.2K &60%RH,1% OS	581.8	6775	292	0.420	21042	13833	6766	**7.77**	32.16	0.10	1% Overspray
4	NG-288.2K &60%RH,2% OS	549.0	6653	308	0.444	22242	14077	7115	**13.33**	31.99	-0.42	2% Overspray

Case 2: Low humidity ambient condition

5	NG-288.2K &90%RH	629.0	7113	272	0.391	19588	13621	6237	**---**	31.84	---	Without fogging in rainy season
6	NG-288.2K &90%RH,Sat.	628.7	7133	272	0.392	19613	13631	6227	**-0.28**	31.75	-0.16	Fogging up to saturation in rainy season
7	NG-288.2K &90%RH,1% OS	590.9	6926	290	0.418	20905	13876	6660	**0.06**	31.86	6.78	1% overspray in rainy season
8	NG-288.2K &90%RH,2% OS	556.2	6758	307	0.443	22153	14120	7055	**0.03**	31.85	13.12	2% overspray in rainy season

ii). When LCV 1 as fuel

Case 3: Low humidity ambient condition

S. no	Description	CDT (K)	Comp power (kW)	Fuel Comp. Power (kW)	Fuel Flow (kg/s)	Heat Add. (kW)	Turb. Power (kW)	Net Output power (kW)	**Net power increase (%)**	Therm Eff. (%)	Eff. Incre--ase (%)	Remarks
9	LCV 1-288.2K &60%RH	625.5	6997	2894	6.619	28847	17639	7748	---	26.86	---	Without fogging
10	LCV 1-288.2K &60%RH,Sat.	619.7	6982	2924	6.687	29146	17724	7817	**0.90**	26.82	-0.14	Fogging up to saturation (RH 100%)
11	LCV 1-288.2K &60%RH,1% OS	581.8	6775	3116	7.126	31058	18240	8348	**7.75**	26.88	0.08	1% Overspray
12	LCV 1-288.2K &60%RH,2% OS	549.0	6653	3294	7.532	32829	18735	8787	**13.42**	26.77	-0.34	2% Overspray

Case 4: High humidity ambient condition

13	LCV 1-288.2K &90%RH	629.0	7113	2901	6.634	28913	17723	7709	---	26.66	---	Without fogging in rainy season
14	LCV 1-288.2K &90%RH,Sat.	628.7	7133	2904	6.642	28949	17738	7701	**-0.11**	26.60	-0.23	Fogging up to saturation in rainy season
15	LCV 1-288.2K &90%RH,1% OS	590.9	6926	3096	7.080	30856	18254	8231	**6.77**	26.68	0.05	1% overspray in rainy season
16	LCV 1-288.2K &90%RH,2% OS	556.2	6758	3281	7.502	32698	18759	8721	**13.12**	26.67	0.02	2% overspray in rainy season

iii). When LCV 2 as fuel

Case 5: Low humidity ambient condition

S. no	Description	CDT (K)	Comp power (kW)	Fuel Comp. Power (kW)	Fuel Flow (kg/s)	Heat Add. (kW)	Turb. Power (kW)	Net Output power (kW)	**Net power increase (%)**	Therm Eff. (%)	Eff. Incre--ase (%)	Remarks
17	LCV 2-288.2K &60%RH	625.5	6997	1531	3.277	24266	15443	6915	**---**	28.50	---	Without fogging
18	LCV 2-288.2K &60%RH,Sat.	619.7	6982	1547	3.311	24517	15505	6976	**0.88**	28.45	0.15	Fogging up to saturation (RH 100%)
19	LCV 2-288.2K &60%RH,1% OS	581.8	6775	1648	3.528	26126	15875	7452	**7.76**	28.52	0.09	1% Overspray
20	LCV 2-288.2K &60%RH,2% OS	549.0	6653	1742	3.729	27616	16235	7840	**13.37**	28.39	-0.38	2% Overspray

Case 6: High humidity ambient condition

21	LCV 2-288.2K &90%RH	629.0	7113	1534	3.284	24322	15522	6875	---	28.27	---	Without fogging in rainy season
22	LCV 2-288.2K &90%RH,Sat.	628.7	7133	1536	3.288	24352	15534	6866	**-0.13**	28.19	-0.26	Fogging up to saturation in rainy season
23	LCV 2-288.2K &90%RH,1% OS	590.9	6926	1637	3.505	25956	15905	7341	**6.78**	28.28	0.05	1% overspray in rainy season
24	LCV 2-288.2K &90%RH,2% OS	556.2	6758	1735	3.714	27506	16270	7777	**13.12**	28.27	0.02	2% overspray in rainy season

A. In Summer season :

i). When Natural Gas as fuel

Case 7: Low humidity ambient c ondition

S. no	Description	CDT (K)	Comp power (kW)	Fuel Comp Power (kW)	Fuel Flow (kg/s)	Heat Add. (kW)	Turb. Power (kW)	Net Output power (kW)	**Net Power increase (%)**	Therm Eff. (%)	Eff. Increase (%)	Remarks
25	NG-313K &60%RH	691.6	8244	283	0.375	18775	14037	5510	----	29.35	----	Without fogging
26	NG-313K &60%RH,Sat.	678.8	8155	289	0.384	19213	14118	5674	**2.98**	29.53	0.64	Fogging up to saturation (RH 100%)
27	NG-313K &60%RH,1%OS	639.4	7872	310	0.411	20572	14363	6182	**12.20**	30.05	2.40	1% Overspray
28	NG-313K &60%RH,2%OS	605.2	7685	329	0.436	21815	14607	6594	**19.08**	30.21	2.96	2% Overspray

Case 8: High humidity ambient condition

29	NG-313K &90%RH	699.6	8648	288	0.382	19119	14388	5452	----	28.51	----	Without fogging in rainy season
30	NG-313K &90%RH,Sat.	696.2	8616	290	0.384	19237	14408	5503	**0.94**	28.60	0.32	Fogging up to saturation in rainy season
31	NG-313K &90%RH,1%OS	656.5	8305	310	0.412	20621	14654	6039	**10.77**	29.28	2.70	1% overspray in rainy season
32	NG-313K &90%RH,2%OS	622.2	8096	330	0.437	21892	14898	6472	**18.73**	29.57	3.69	2% overspray in rainy season

ii). When LCV 1 as fuel

Case 9: Low humidity ambient c ondition

S. no	Description	CDT(K)	Comp power (kW)	Fuel Comp Power (kW)	Fuel Flow (kg/s)	Heat Add. (kW)	Turb. Power (kW)	Net Output power (kW)	**Net Power increase (%)**	Therm Eff. (%)	Eff. Increase (%)	Remarks
33	LCV 1-313K &60%RH	691.6	8244	3020	6.358	27713	17969	6705	---	24.19	----	Without fogging
34	LCV 1-313K &60%RH,Sat.	678.8	8155	3090	6.507	28358	18141	6897	**2.86**	24.32	0.52	Fogging up to aturation (RH 100%)
35	LCV 1-313K &60%RH,1%OS	639.4	7872	3309	6.967	30365	18672	7491	**11.73**	24,67	1.97	1% Overspray
36	LCV 1-313K &60%RH,2%OS	605.2	7685	3510	7.391	32213	19178	7983	**19.06**	24.78	2.43	2% Overspray

Case 10: High humidity ambient condition

37	LCV 1-313K &90%RH	699.6	8648	3075	6.475	28220	18391	6668	---	23.63	---	Without fogging in rainy season
38	LCV 1-313K &90%RH,Sat.	696.2	8616	3094	6.515	28394	18436	6727	**0.88**	23.69	0.26	Fogging up to saturation in rainy season
39	LCV 1-313K &90%RH,1%OS	656.5	8305	3317	6.983	30437	18972	7351	**10.24**	24.15	2.20	1% overspray in rainy season
40	LCV 1-313K &90%RH,2%OS	622.2	8096	3521	7.414	32313	19483	7866	**17.66**	24.34	3.01	2% overspray in rainy season

iii). When LCV 2 as fuel

Case 11: Low humidity ambient condition

S. no	Description	CDT (K)	Comp power (kW)	Fuel Comp Power (kW)	Fuel Flow (kg/s)	Heat Add. (kW)	Turb Power (kW)	Net Output power (kW)	**Net Power increase (%)**	Therm Eff. (%)	Eff. Increase (%)	Remarks
41	LCV 2-313K &60%RH	691.6	8244	1597	3.148	23312	15859	6018	---	25.81	---	Without fogging
42	LCV 2-313K &60%RH,Sat.	678.8	8155	1634	3.221	23855	15982	6194	**2.92**	25.96	0.58	Fogging up to saturation (RH 100%)
43	LCV 2-313K &60%RH,1%OS	639.4	7872	1750	3.449	25543	16360	6738	**11.98**	26.38	2.20	1% Overspray
44	LCV 2-313K &60%RH,2%OS	605.2	7685	1856	3.659	27098	16725	7184	**19.39**	26.51	2.71	2% Overspray

Case 12: High humidity ambient condition

45	LCV 2-313K &90%RH	699.6	8648	1626	3.206	23739	16423	5969	---	25.14	---	Without fogging in rainy season
46	LCV 2-313K &90%RH,Sat.	696.2	8616	1636	3.225	23885	16275	6023	**0.91**	25.22	0.29	Fogging up to saturation in rainy season
47	LCV 2-313K &90%RH,1%OS	656.5	8305	1754	3.457	25604	16655	6596	**10.51**	25.76	2.46	1% overspray in rainy season
48	LCV 2-313K &90%RH,2%OS	622.2	8096	1862	3.670	27182	17023	7065	**18.36**	25.99	3.37	2% overspray in rainy season

Table 2: Total tariff in different cond ition

AMBIENT AIR CONDITION	TARIFF IN 24 hrs. (Rs.)						
	Without fogging	Fogging at saturation	1% OS	2%OS	Net gain in saturation (%)	Net gain in 1%OS (%)	Net gain in 2%OS (%)
NG-288.2 K &60% RH	557486.40	562370.40	600820.80	631812.00	0.86	7.21	11.76
NG-288.2 K & 90% RH	553845.60	552957.60	591408.00	626484.00	-0.16	6.35	11.59
LCV 1-288.2 K & 60% RH	688022.40	694149.60	741302.40	780285.60	0.88	7.19	11.82
LCV 1-288.2 K & 90% RH	684559.20	683848.80	730912.80	774424.80	-0.10	6.34	11.60
LCV 2-288.2 K & 60% RH	614052.00	619468.00	661737.60	696192.00	0.87	7.21	11.80
LCV 2-288.2 K & 90% RH	610500.00	609700.80	651880.80	690597.60	-0.13	6.34	11.60
NG-313 K & 60% RH	489288.00	503851.20	548961.60	585547.20	2.89	10.87	16.44
NG-313 K & 90% RH	483137.60	488666.40	536263.20	574713.60	1.13	9.91	15.93
LCV 1-313 K & 60% RH	595404.00	612453.60	665200.80	708890.40	2.78	10.50	16.01
LCV 1-313 K & 90% RH	592118.40	597357.60	652768.80	698500.80	0.87	9.29	15.23
LCV 2-313 K & 60% RH	534398.40	550027.20	593334.40	637939.20	2.84	9.93	16.23
LCV 2-313 K & 90% RH	530047.20	534842.40	585724.80	627372.00	0.90	9.51	15.51

Appendix 1: Composition of Fuels

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 di oxide(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 vapour(H_2O)	22.92	36.62	-
Nitrogen (N_2)	37.20	0	-
Total	**100.0**	**100.0**	**100.0**
Low Heating Value (kJ/kg)	4,358	7,405	50,046
High Heating Value (kJ/kg)	5,238	8,735	55,532

S-JPSET : ISSN : 2229-7111, Vol. 3, Issue 1