International Journal of Engineering Research and Application www.ijera.com ISSN : 2248-9622, Vol. 8, Issue 2, (Part -3) February 2018, pp.67-75

RESEARCH ARTICLE OPEN ACCESS

Effect of Fogging Cooling On the Performance Parameters of Cooled Gas Turbine Cycle

Dr Alok Kumar Mohapetra[1], K S Raghuram[2], Amarquirti Sethi[3]
[1] *Professor, Department of Mechanical Engineering, Gandhi Institute For Technology (GIFT), Bhubaneswar*
[3]*Associate Professor, Department of Mechanical Engineering, Gandhi Institute For Technology (GIFT), Bhubaneswar*
[2] *Assistant Professor, Department of Mechanical Engineering, Gandhi Engineering College, Bhubaneswar*

ABSTRACT
The study provides a computational analysis of the effects of compressor pressure ratio, turbine inlet temperature, ambient relative humidity and ambient temperature on the performance parameters of an air cooled gas turbine cycle with evaporative cooling of inlet air. The blade cooling method selected is air film cooling. The analysis indicates that the mass of coolant required for blade cooling is reduced with increase in temperature drop across the humidifier. Both decrease in ambient temperature and ambient relative humidity results in an increase in plant efficiency and plant specific work. The highest efficiency is obtained at a turbine inlet temperature of 1500K for all range of ambient relative humidity and ambient temperature after which it decreases. The compressor pressure ratio corresponding to the maximum plant specific work however varies with both ambient relative humidity and ambient temperature. The increase in specific work due to drop in ambient relative humidity is more pronounced at higher pressure ratios. Similarly the increase in efficiency due to ambient temperature drop is prominent at higher turbine inlet temperatures. Finally a design monogram is presented which may be used to determine find out the design parameters corresponding to best efficiency and specific work desired.

I. INTRODUCTION

Gas turbines have gained widespread acceptance in the power generation, mechanical drive, and gas transmission markets. I.G. Wright, T.B. Gibbons [1] have thoroughly reviewed the recent developments in gas turbine materials and technologies. Consequently, thermal efficiencies are currently very attractive, with simple cycle efficiencies ranging between 32% and 42 % and combined cycle efficiencies reaching the 60% mark. The efficiency of the gas turbine cycle has been improved mainly due to enhanced gas turbine performance through advancements in materials and cooling methods in recent years.

The two important methods of improving the gas turbine performance are by inlet air cooling and gas turbine blade cooling.

By the addition of an air-cooling system at the compressor intake, the inlet air can be conditioned to lower temperatures than ambient, thus improving plant performance at high ambient temperatures. As the inlet air temperature drops compressor work decreases and so the net work and cycle efficiency increases. In addition to this, air density increases with drop in inlet air temperature, which results in an increase in mass flow rate of air entering the compressor and so the power output is enhanced**.** Work in this area has been done by De Lucia et al. [2], Bassily [3, 4], and Karakas et al. [5].

The search for a better performance of gas turbine engines has also led to higher turbine inlet temperatures. The objective of the blade cooling is to keep the blade temperature to a safe level, to ensure a long creep life, low oxidation rates, and low thermal stresses. The universal method of blade cooling is by air bled from compressor flowing through the internal passages in the blades. Work in this area has been done by Louis et al. [6], El-Masri [7], Bolland and Stadaas [8], and Sanjay et. al. [9-20]. The present work is an attempt in this direction dealing with the combined effect of turbine blade cooling and evaporative inlet air-cooling on the performance of basic gas turbine cycle. The effects of compressor pressure ratio, turbine inlet temperature, ambient relative humidity and ambient temperature have been analyzed on the thermodynamic performance parameters of the cycle. Figure 1 shows the schematic diagram of a basic gas turbine cycle with inlet air humidifier and is being called air humidifier integrated gas turbine (AHIGT).

NOMENCLATURE

A = ratio of mass of coolant to mass of gas flow
a,b,c = constants
c_p = specific heat…………..(kJ-kg^{-1}·K^{-1})
f_h = correction factor to account for vapor added in humidifier
F = factor
F_{sa} = correction factor to account actual blade surface
gt = gas turbine
h = specific enthalpy………(kJ·kg^{-1})
ΔH_r = lower heating value……(kJ.kg^{-1})
$\dot{m}$ = mass flow rate……….. (kg·s^{-1})
Q = heat energy transfer……(W)
r_p = cycle pressure ratio
R = gas constant……………(kJ-kg^{-1}·K^{-1})
p = total pressure………………..(bar)
S = blade perimeter……..(m)
$\overline{St}$ = average Stanton number $= \dfrac{\overline{h}_g}{c_{p,g} \cdot \rho_g \cdot C_g}$
t = pitch of blade …………(m)
T = temperature……………(K)
TIT = turbine inlet temperature (K) = combustor exit temperature
W = specific work………(kJ·kg^{-1})
t_a = air temperature………… (°C)

Greek symbols

ϕ = relative humidity ……………..(ratio)
ω = specific humidity ……………… (kg/kg)
α = gas flow discharge angle ………… (degree)
ε = effectiveness ………..(%) $= \dfrac{T_{c.e} - T_{c,i}}{T_b - T_{c,i}}$
η = efficiency………………(%)

Subscripts

a = air , ambient
av = average
b = blade
c = compressor
comb = combustor
dr = drop
e = exit
f = fuel
g = gas
alt = alternator
gt = gas turbine
h = humidifier
I = inlet, stage of compressor
in = inlet
inc = increase
j = coolant bleed points
net = difference between two values
p = pressure
plant = gas turbine plant
pt = polytropic

sat = saturation
vap = water vapor
v = volume ……..(m^3)
w = water

Acronym

AFC = Air film cooling
AHIGT = Air Humidifier Integrated Gas Turbine
C = Compressor
CC = Combustion chamber
CIT = Compressor inlet temperature
GT = Gas turbine
RH = relative humidity (ratio)
WBT = Wet Bulb Temperature
DBT = Dry Bulb Temperature

II. MODELING OF GOVERNING EQUATIONS

Parametric study of the AHIGT cycle has been carried out by modeling the various elements of a gas turbine cycle and using the governing equations. The following are the modeling details of various elements.

2.1 Gas model

The specific heat of real gas varies with temperature and also with pressure (at extreme high pressure levels). However, in the present model it is assumed that specific heat of gas varies with temperature only and is given in the form of polynomials as follows

$$c_p(T) = a + bT + cT^2 + dT^3 + ... \quad (1)$$

(1)

where a, b, c, and d are coefficient of polynomials, as taken from the work of Touloukian and Tadash [16]. A factor called humidity correction factor represented by f_h is introduced to account for the increase in specific humidity of ambient air across the air-humidifier and is calculated as

$$f_h = 1 + 0.05\phi_{h,e} \quad (2)$$

where $\phi_{h,e}$ is the relative humidity at the outlet of humidifier.

Thus, the enthalpy of gas is expressed as

$$h = \int_{T_a}^{T} c_p(T)dT \quad (3)$$

.. ..

The enthalpy of ambient air entering the air-humidifier is assigned zero value. In the gas model, natural gas (NG) is the fuel used in combustors and the composition and physical properties (such as $c_{p,g}$,etc.) of burnt gas composition depend upon the composition of NG that may vary from well-to-well (i.e. the source of NG). For thermodynamic study, the fuel composition is taken as CH_4 = 86.21 per cent, C_2H_6 = 7.20 per cent, and CO_2 = 5.56 per cent, and N_2 = 1.03 per cent by weight.

2.2 Humidifier Model

Cooling in hot, relatively dry climate can be accomplished by evaporative cooling. Evaporative cooling involves passing of air across a spray of water or forcing air through a water soaked pad that is kept replenished with water [22]. Owing to the low humidity of entering air, a part of the water injected evaporates. The energy required for evaporation is provided by the air stream, which is undergoes a reduction in temperature. The following assumptions are made in the humidifier model.

- The relative humidity at the humidifier outlet is 0.95 (95%)
- The pressure drop of air in the humidifier is 1% of the ambient air pressure.

Applying the mass balance equation across the humidifier control volume boundary gives

$$\omega_{a,e} = \omega_{a,i} + m_w \quad (4)$$

where ω is the specific humidity and is calculated at a certain temperature as

$$\omega = \frac{0.622 p_{vap}}{p - p_{vap}} \quad (5)$$

where $p_{vap} = \phi p_{sat}$ is the partial pressure of vapour, ϕ is the relative humidity and p_{sat} is the saturation pressure of air corresponding to the given temperature.

The energy balance equation for the humidifier is given by

$$h_{a,e} = h_{a,i} + (\omega_{a,e} - \omega_{a,i})h_w \quad (6)$$

where $h_{a,e}$ and $h_{a,i}$ are the enthalpy of moist air at outlet and inlet of the air humidifier respectively and are calculated as follows

$$h_{a,e} = c_{p,a,in}t_{a,e} + (2500 + 1.88t_{a,e})\omega_{a,e} \quad (7a)$$

$$h_{a,i} = c_{p,a,in}t_{a,i} + (2500 + 1.88t_{a,i})\omega_{a,i} \quad (7b)$$

$$T_{a,e} = t_{a,e} + 273 \quad (7c)$$

The equations (4–7) can be solved to determine the value of $T_{a,e}$, $\omega_{a,e}$ and m_w.

2.3 Compressor Model

The compressor used in gas turbine power plant is of axial flow type. The thermodynamic losses in an axial flow compressor are incorporated in the model by introducing the concept of polytropic efficiency. The temperature and pressure of air at any section of compressor are related by the expression

$$\frac{dT}{T} = \left[\frac{R_c}{\eta_{pt,c}c_{p,c}}\right]\frac{dp}{p} \quad (8)$$

where $\eta_{pt,c}$ is the compressor polytropic efficiency and $c_{p,c}$ and R_c are the specific heat at constant pressure and the gas constant across the compressor respectively. R_c is given by

$$R_c = c_{p,c} - c_{v,c} \quad (9)$$

where

$$c_{p,c} = c_{p,a} + \omega_{a,i}c_{p,vap} \quad (10)$$

$$c_{v,c} = c_{v,a} + \omega_{ai}c_{v,vap} \quad (11)$$

where $c_{p,a}$ and $c_{v,a}$ are the specific heats of air at constant pressure and at constant volume respectively, both in kJ/kg K, and are evaluated at the average temperature across the compressor from the following relations [22].

$$c_{p,a} = \frac{8.314}{28.97}\left(\begin{array}{l}3.653 - 1.337\times10^{-3}T_{av} + 3.294\times10^{-6}T_{av}^2 - \\ 1.913\times10^{-9}T_{av}^3 + 2.763\times10^{-13}T_{av}^4\end{array}\right) \quad (12)$$

$$c_{v,a} = c_{p,a} - 0.287 \quad (13)$$

where $c_{p,vap}$ and $c_{v,vap}$ are the specific heats of water-vapor at constant pressure and at constant volume respectively, both in kJ/kg K, and are evaluated at the average temperature across the compressor from the following relations [22].

$$c_{p,vap} = \frac{8.314}{18.02}\left(\begin{array}{l}4.07 - 1.108\times10^{-3}T_{av} + 4.152\times10^{-6}T_{av}^2 \\ -2.964\times10^{-9}T_{av}^3 + 8.07\times10^{-13}T_{av}^4\end{array}\right) \quad (14)$$

$$c_{v,vap} = c_{p,vap} - 0.4614 \quad (15)$$

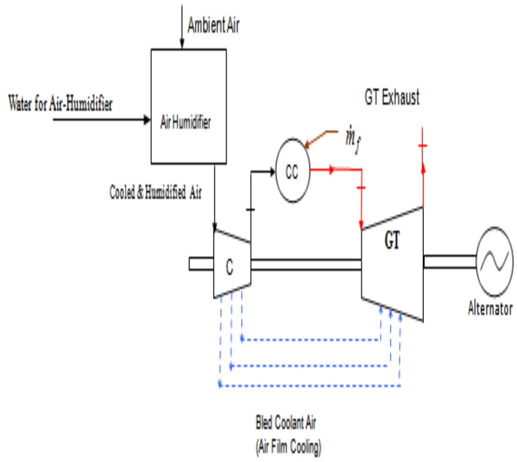

Fig. 1. Schematic of Air humidifier integrated gas turbine

The enthalpy at any polytropic stage of compressor may be calculated using equations (1) , (3), and (8).Using mass and energy balance across control volume of compressor, the compressor work is calculated as follows:-

$$\dot{m}_{c,i} = \dot{m}_{c.e} + \sum \dot{m}_{coolant,j} \quad (16)$$

$$W_c = \dot{m}_{c,e}h_{c,e} + \sum \dot{m}_{coolant,j}h_{coolant,j} - \dot{m}_{c,i}h_{c,i} \quad (17)$$

2.4 Combustor Model

Losses inside the combustor, which arise due to incomplete combustion and pressure losses are taken into account by introducing the concept of combustion efficiency and percentage pressure drop of compressor exit pressure [Table 1]. The mass and energy balances across the control volume of combustor yields the mass of fuel required to attain a specified exit temperature of combustor, which is taken as turbine inlet temperature (TIT), given by,

$$\dot{m}_e = \dot{m}_i + \dot{m}_f \quad (18)$$

$$\dot{m}_f \cdot \Delta H_r \cdot \eta_{comb} = \dot{m}_e \cdot h_e - \dot{m}_i \cdot h_i \quad (19)$$

2.5 Cooled Gas Turbine

Unlike steam turbine blading, gas turbine bladings need cooling. The objective of the blade cooling is to keep the "blade temperature to a safe level, to ensure a long creep life, low oxidation rates, and low thermal stresses. The universal method of blade cooling is by air bled from compressor flowing through the internal passages in the blades. In the case of film cooling, the coolant exits from the leading edge of blade and a film is formed over the blade surface, which reduces the heat transfer from the hot gas to the blade surface.

In this work, the gas turbine blades have been modeled to be cooled by air-film cooling (AFC) method[9-15]. The cooling model used for cooled turbine is the refined version of that by Louis et al [6].The mass flow rate of coolant required in a blade row is expressed as [15]

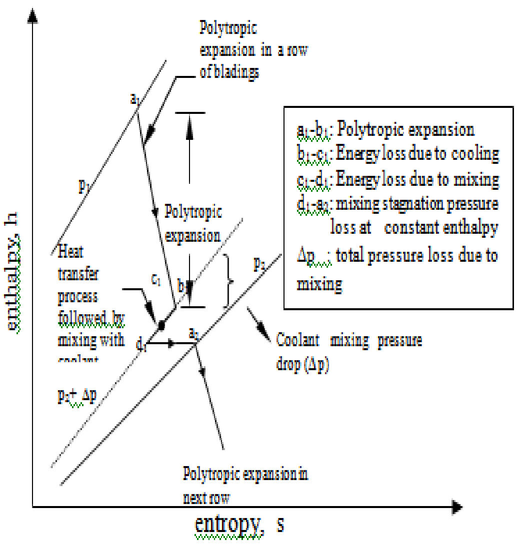

Fig. 2. Expansion path in cooled gas turbine row

$$a_{coolant} = \frac{\dot{m}_{coolant}}{\dot{m}_g} = \left[\frac{St_{in} \cdot c_{p,g}}{\varepsilon \cdot c_{p,coolant}}\right] \times \left[\frac{S_g \cdot F_{sa}}{t \cdot \cos\alpha}\right] \times \left[\frac{T_{g,i} - T_b}{T_b - T_{coolant,i}}\right] \quad (20)$$

where $S_g \cong 2c$, $S_g/t\cos\alpha = 3.0$, $F_{s,a} = 1.05$, $\alpha = 45°$ (for stator), $\alpha = 48°$ (for rotor), $St_{in} = 0.005$

Table 1. Input data for analysis [9-20, 23]

PARAMETER	SYMBOL	UNIT
Gas Properties:	$C_p = f(T)$ Enthalpy $h = \int c_p(T)\, dT$	kJ/kg K KJ/kg
Air-Humidifier	Pressure drop across humidifier = 1 Wetted Pad, Cross flow type	% -
Compressor	Polytropic efficiency($\eta_{p,c}$)=92.0 Mechanical efficiency(η_m)=98.5 Inlet plenum loss= 0.5% of entry pr.	% % bar
Combustor	Combustor efficiency (η_{comb})=99.5 Pressure loss (p_{loss})=2.0% of entry pressure Lower heating value (LHV)= 42.0	% bar MJ/kg
Gas turbine	Polytropic efficiency (η_{pt})=92.0 Exhaust pressure=1.08 Exhaust hood loss=4 Turbine Blade Temperature= 1123	% bar K K
Alternator	Alternator efficiency=98.5	%

Also blade coolant requirement is dependent on the temperature of coolant air at the bleed points, which in turn is dependent upon the temperature of air achieved in the humidifier, at the compressor inlet. With a drop in temperature of air at the inlet of compressor, there is a proportionate drop in the temperature of bled coolant due to more effective blade cooling achieved by lower temperature bled coolant and hence lesser coolant requirement. Also, as the mass of bled coolant is less, hence the quantum of pumping and mixing loss associated with the mixing of coolant stream with main gas stream is also less.

Fig. 2 gives the details of expansion process for a cooled turbine stage. Process b_1-c_1 in Fig. 2 depicts cooling due to heat transfer between hot gas and coolant, which takes place at constant pressure line due to which exergy decreases, while process c_1-d_1 depicts drop in temperature due to mixing of coolant with gas which is an irreversible process and also takes place along constant pressure line, which leads to drop in entropy. Process d_1-a_2 in the model denotes a process similar to throttling.

The deviation between actual and theoretical value is driven by the amount of coolant and coolant temperature used for cooling of blades and the actual value varies with blade cooling requirements. At TIT 1700K for air-film cooling, its maximum value is ±6% [15]. Turbine work and exergy destruction are given by the mass, energy and exergy balance of gas turbine as under:

$$W_{gt} = \left[\dot{m}_{g,i} \cdot (h_{g,i} - h_{g,e})\right] + \left[\sum \dot{m}_{coolant} \cdot (h_{coolant,i} - h_{coolant,e})\right]$$
(21)

3. Performance Parameters

The performance parameters $W_{gt,net}$, W_{plant}, η_{plant} and are expressed as follows:

$$W_{gt,net} = W_{gt} - \frac{|W_c|}{\eta_m}$$
(22)

$$W_{plant} = \lfloor W_{gt,net} \rfloor \cdot \eta_{alt}$$
(23)

$$\eta_{plant} = \frac{W_{plant}}{Q} = \frac{W_{plant}}{\dot{m}_f \cdot \Delta H_r}$$
(24)

Modeling of cycle components and governing equations developed for the cycle proposed above have been coded using C++ and results obtained. A flowchart of the programme code 'Simucomb' illustrating the method of solution is detailed in the author's earlier article [15]. The input data used in the analysis is given in Table 1.

III. RESULT AND DISCUSSION

The influence of evaporative inlet air cooling on gas turbine performance has been shown through the performance curves, plotted using modeling, governing equations and input parameters (Table1).

Fig. 3 shows the effect of inlet cooling on variation of coolant mass required for blade cooling with respect to ambient temperature. It is evident from the graph that the coolant mass increases with increase in ambient temperature for both the cases. However as the ambient temperature increases for a given RH, the effectiveness of cooling is also increased due to higher saturation pressure and as a result, the increase in the mass of coolant required for turbine blade cooling is comparatively less for a given rise in temperature in case of inlet cooled gas turbine.

Fig 4 shows the benefit of inlet cooled GT cycle over the basic GT cycle without inlet cooling in terms of enhancement in plant efficiency. It is observed that when the relative humidity is decreases for a given ambient temperature, the efficiency of inlet cooled GT cycle is increased due to higher drop in temperature achieved in the evaporator, where as the efficiency of the GT cycle without inlet cooling almost remains unaltered. It can also be concluded from the graph that for a given ambient relative humidity, as the ambient temperature increases, though the efficiency is reduced for both the cases, diminution in efficiency is comparatively less for inlet cooled GT cycle. This is due to the fact that at higher ambient temperature the difference between wet and dry bulb temperature is higher resulting in more effective cooling leading to larger temperature drop in the humidifier and lesser amount of coolant requirement as discussed for Fig.3.

Fig. 5 shows the effect of TIT on plant efficiency at different values of ambient temperatures. It is observed that the efficiency reduces with increase in temperature. This is because at higher ambient temperature though the drop in temperature is higher owing to higher difference between WBT and DBT, the compressor inlet temperature is also high. A drop of 20^o C at an ambient temperature of 318K results in a CIT of 298K against a CIT of 280K at an ambient temperature of 288K where the temperature drop is only 8^oC. However the reduction in efficiency at higher ambient temperature becomes less pronounced compared to basic gas turbine due to inlet air cooling. It is observed that the plant efficiency increases with increase in TIT upto 1500K beyond which the performance is limited by cooling penalties.The inlet air cooling boosts the efficiency by 3.51 % at a TIT of 1300K ($r_{p,c}$=25, RH_a=0.2) when the ambient temperature drops by 30^0 C. This enhancement increases to 4.01% for a

TIT of 1500K at the same value of $r_{p,c}$, RH_a and ambient temperature drop.

Fig. 6 shows the effect of variation of $r_{p,c}$ and ambient RH on plant specific work. A lower ambient RH corresponds to a higher increase in RH achieved in the humidifier. It is clearly seen that the specific work is highest when the RH of inlet air is lowest (0.2). This condition of inlet air results in maximum enhancement in relative humidity of inlet air resulting in maximized inlet air cooling and hence lowest compression work thus maximum specific work of the cycle. This enhancement is higher at higher value of $r_{p,c}$. The specific work increases by 11.62% at $r_{p,c}$=28 against an increase of 8.4% at $r_{p,c}$=16 for the same rise in RH of air achieved in the humidifier. The effect of variation of $r_{p,c}$ on turbine specific work suggests that specific work slightly increases with increase in pressure ratio for all range of specific humidity after which it decreases. It is also observed that the $r_{p,c}$ corresponding to maximum specific work increases with decrease in ambient relative humidity. This suggests that for a higher value of plant specific work there exist an optimum $r_{p,c}$ (for a given increase in RH achieved in the humidifier) and the $r_{p,c}$ needs to be chosen.

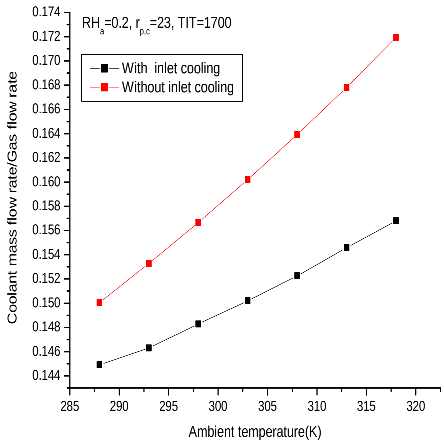

Fig. 3. Effect of ambient temperature on coolant mass for basic and inlet cooled gas turbine

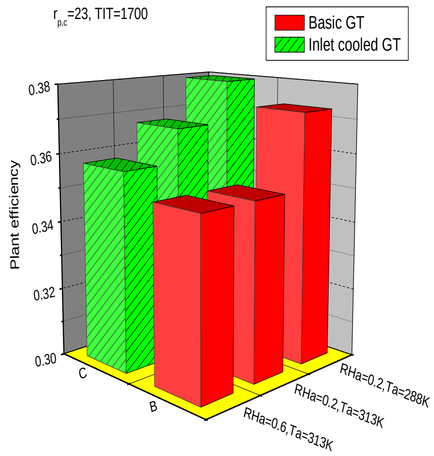

Fig. 4.Effect of ambient condition on plant efficiency of gas turbine with and without inlet cooling

Fig. 7 shows the variation of $r_{p,c}$ and T_a on specific fuel consumption and plant specific work of AHIGT cycle. An increase in the specific work and decrease in specific fuel consumption is observed with decrease in T_a. This is because at higher temperature though the temperature drop in humidifier is higher the compressor inlet temperature is also high. The specific fuel consumption decreases by 4.29% at $r_{p,c}$=28 against a decrease of 3.66% at $r_{p,c}$=16 for the same range of ambient temperature drop. The effect of variation of $r_{p,c}$ on turbine specific work suggests that specific work increases with increase in pressure ratio for all values of ambient temperature after which it decreases. It is also observed that the $r_{p,c}$ corresponding to maximum specific work is higher at lower ambient temperature. This suggests that for a higher value of plant specific work there exist an optimum $r_{p,c}$ (corresponding to a given ambient temperature) and the same needs to be chosen as per discussions detailed in above section.

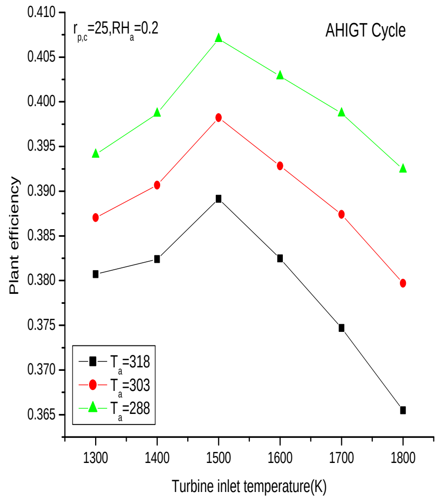

Fig. 5. Effect of turbine inlet temperature on plant efficiency at different ambient temperature

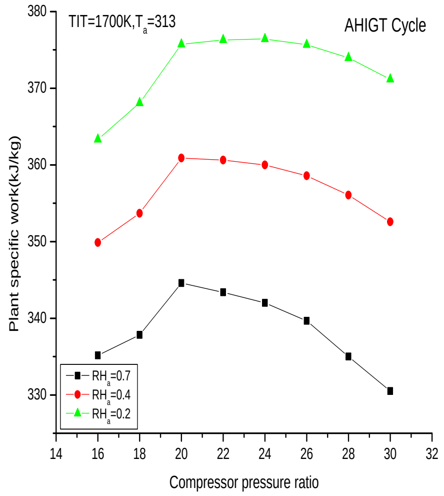

Fig. 6. Effect of compresson ratio on plant specific work at different ambient relative humidity

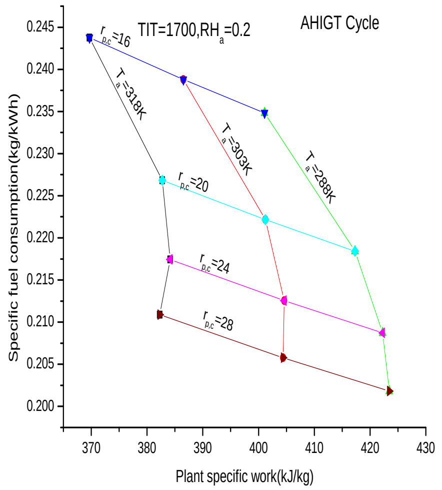

Fig. 7. Specific fuel consumption versus plant specific work for different $r_{p,c}$ and T_a

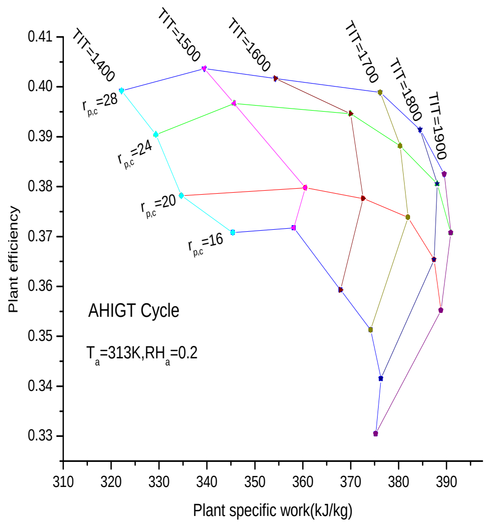

Fig. 8. Plant efficiency versus plant specific work for different $r_{p,c}$ and TIT

Fig. 8 also called design monogram, is helpful in selecting the design parameters such as $r_{p,c}$ TIT and cooling means for the best plant efficiency and specific work. The results show that for all pressure ratios, there exist an optimum TIT at which plant efficiency is the maximum. However, plant specific work continues to increase with increase in TIT and decreases with increase in $r_{p,c}$. The existence of optimum $r_{p,c}$ at any TIT with reference to the maximum plant efficiency is due to the combined effect of many factors. With increasing $r_{p,c}$ and TIT, the compressor work input, the fuel and coolant air requirements increase, however the gas turbine work also increases but is restricted by the increasing pumping, cooling and mixing losses.

It is proposed to experimentally verify the obtained analytical results presented in this paper in about a year time and publish the same subsequently.

IV. CONCLUSIONS

Based on the analysis of air-humidifier-integrated-gas turbine cycle presented above following conclusions have been drawn:

1. The mass of coolant required for turbine blade cooling increases with increase in ambient temperature for an air humidifier integrated gas turbine.

2. The plant efficiency increases because of inlet cooling using air-humidifier integrated to a gas turbine plant.
3. The enhancement in efficiency due to inlet-air cooling is higher at higher ambient temperature.
4. The rate of increase in efficiency due to ambient temperature drop is higher at higher value of TIT.
5. specific work increases with increase in specific humidity of air achieved in the humidifier and this enhancement is higher at higher value of $r_{p,c}$
6. It is also observed that the $r_{p,c}$ corresponding to maximum specific work increases with decrease In ambient relative humidity.
7. An increase in the specific work and decrease in specific fuel consumption is observed with decrease in ambient temperature. It is also observed that the $r_{p,c}$ corresponding to maximum specific work is higher at lower ambient temperature
8. The design monogram shows that for all pressure ratios, there exist an optimum TIT at which plant efficiency is the maximum

REFERENCES

[1]. I.G. Wright, T.B. Gibbons, "Recent developments in gas turbine materials and technology and their implications for syngas firing" :International Journal of Hydrogen Energy 32 , (2007) 3610 – 3621._

[2]. De Lucia M , Lanfranchi C, and Boggio V," Benefits of compressor inlet air cooling for gas turbine cogeneration plants" : In Proceedings of the International Gas Turbine and Aero engine Congress and Exposition, Houston, Texas, 5–8 June 1995.

[3]. A M Bassily, "Performance improvements of the intercooled reheat regenerative gas turbine cycles using indirect evaporative cooling of the inlet air and evaporative cooling of the compressor discharge " : Proceedings of the Institution of Mechanical Engineers Part A Journal of Power And Energy (2001) Vol. 215, , pp: 545-557

[4]. A M Bassily, "Effects of evaporative inlet and aftercooling on the recuperated gas cycle": Applied Thermal Engineering, 21(2001) 1875-1890

[5]. E. Kakaras, A. Doukelis, S. Karellas, "Compressor intake-air cooling in gas turbine plants" : Energy 29 (2004) 2347–2358

[6]. Louis JF, Hiraoka K, and El-Masri MA, "A comparative study of influence of different means of turbine cooling on gas turbine performance" : ASME Paper no. 83-GT-180.

[7]. El-Masri MA, GASCAN- "An interactive code for thermal analysis of gas turbine systems" : Trans. of ASME, Journal of Engineering for Gas Turbines and Power,; 110(1988): pp 201-209

[8]. Bolland O and Stadaas JF, "Comparative Evaluation of combined cycles and gas turbine systems with injections, steam injection and recuperation" : ASME Journal of Engg. For Gas Turbine and Power, 117(1995):pp. 138-145.

[9]. Sanjay, Onkar Singh, and B. N. Prasad, "Thermodynamic Performance of Complex Gas Turbine Cycles" :ASME Conf. Proc. 2002, 529 (2002), DOI:10.1115/IJPGC2002-26109.

[10]. Sanjay, Onkar Singh, and B. N. Prasad "Performance Enhancement of Advanced Combined Cycles":ASME Conf. Proc. 2003, 523 (2003), DOI:10.1115 IJPG 2003-40117.

[11]. Sanjay, Onkar Singh, and B. N. Prasad, "Thermodynamic Evaluation of Advanced Combined Cycle Using Latest Gas Turbine" :ASME Conf. Proc. 2003, 95 (2003), DOI:10.1115 GT2003-38096

[12]. Sanjay, Onkar Singh, and B. N. Prasad, "Performance of Integrated Combined and Cogeneration Cycles Using Latest Gas Turbines":ASME Conf. Proc. 2004, 529 (2004), DOI:10.1115/GT2004-53312

[13]. Sanjay, Onkar Singh, and B. N. Prasad, "Thermodynamic Evaluation of Combined Cycle Using Different Methods of Steam Cooling" :ASME Conf. Proc. 2004, 361 (2004), DOI:10.1115/POWER2004-52152

[14]. Sanjay, Onkar Singh, B.N. Prasad, "Comparative Performance Analysis of Cogeneration Gas Turbine Cycle for Different Blade Cooling Means" International Journal of Thermal Sciences, Volume 48, Issue 7, July 2009, 1432-1440.

[15]. Sanjay, "Investigation of effect of variation of cycle parameters on thermodynamic performance of gas/ steam combined cycle" , Energy, 36 (2011) pp. 157-167.

[16]. Y. Sanjay, Onkar Singh, B.N. Prasad, "Energy and exergy analysis of steam cooled reheat gas–steam combined cycle" , Applied Thermal Engineering 27 (2007) 2779–2790.

[17]. Sanjay, Onkar Singh, B.N. Prasad, "Influence of different means of turbine blade cooling on the thermodynamic performance of combined cycle" , Applied Thermal Engineering 28 (2008) 2315–2326.

Dr Alok Kumar Mohapetra " Effect of Fogging Cooling C Cooled Gas Turbine Cycle" *international journal of engi* *(ijera)*, vol.8(2), 2018, pp 67-75.