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Thermodynamic Analysis of Cooled Blade Gas Turbine Cycle With Inlet Fogging

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ABSTRACT:- Gas turbine's compactness, high power to weight ratio and ease of installation has made them a popular prime mover. Ambient temperature and turbine inlet temperature are effect the power output and efficiency of the turbine. In this study a cooled blade basic gas turbine cycle power plant have been consider and analyzed the effect of three ambient conditions hot day condition, winter day condition ,and rainy day condition (HDC, WDC & RDC) on efficiency ,power output, specific fuel consumption and mass flow rate of coolant .Ambient conditions have been considered , central zone of India.

Keywords:- Fogging, Psychometric chart, Basic gas turbine, Cooled blade

I. INTRODUCTION

Gas turbine has gained widespread acceptance in the power generation ,and mechanical drive market. Their compactness , high power to weight ratio , and ease of installation have made them a popular prime mover. Due to improvement of materials and cooling technologies allowed in increase of firing temperature, hence increase the power of gas turbine cycle.

In recent years, due to global warming temperature has increased consequently the demand for power during summer days has increased. Unfortunately the output of pick load gas turbine will be reduced by 0.5% to 0.9% consists of cooling air at the inlet of compressor by water through spraying nozzle operating at high pressure forming fog. Fog is produced when water atomized to micro scale droplets. The fog inlet cooling on the gas turbine increase output power with increased fog mass flow rate, where as the efficiency curve is flat for a wide range of compression ratio and turbine inlet temperature [3].

Chaker [4-6] presented the result of extensive experiment and theoretical studies of nearly 500 inlet fogging system on gas turbine ranging from 5 to 259 MW. Their studies cover 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.

Sexton [7] conduct a computational simulation to examine effect of water injection fogging and overspray .Their results included the compressor water temperature .Their results show that for 311^{o}K with an ambient relative humidity of 60% nearly 7.2^{o}Kof temperature depression and 11% of power augmentation can be realized by overspray 0.38% of water mist.

Jobaidur Rahman Khan and Ting Wang [8] have shown in their paper the fog and overspray cooling reduce the compressor work and increase the net output power, but not necessarily the cycle thermal efficiency. In the mean time fog/overspray also increased the total mass flow rate, which further increase the power output and result to a significant augmentation of the net power output. Based on fixed pressure ratio turbine inlet temperature (TIT) air fuel mass flow, and component efficiency their result of simulation show (a) Main compressor –in the main compressor power consumption increases with ambient temperature and humidity, but decreases with the water spray due to increased air density after cooling.

(b) In turbine – low calorific value (LCV) produce 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 temperature or relative humidity increases, the net power decreases.(c) Thermal efficiency – for LCV fuels ,the thermal efficiency is approximately 10-16% lower than using the natural gas, burning fuels leads to small change in performances for different conditions of ambient temperature pressure and humidity, flow rate overspray and 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 an incombustible gases. Fog/overspray could either slightly increase or decreases the thermal efficiency depending on the ambient conditions. In this study, cooled blade basic gas turbine cycle with inlet cooling by fogging have been considered and analyzed the effect of independent parameters (efficiency, specific fuel consumption, power output, mass flow rate of coolant) with variation of dependent parameters(compression ratio, turbine inlet temperature) at three ambient central zone conditions in India.

II. DESCRIPTION OF THE SYSTEM

Fig (1) shows a schematic arrangement of the inlet cooling of the simple gas turbine cycle power plant. The

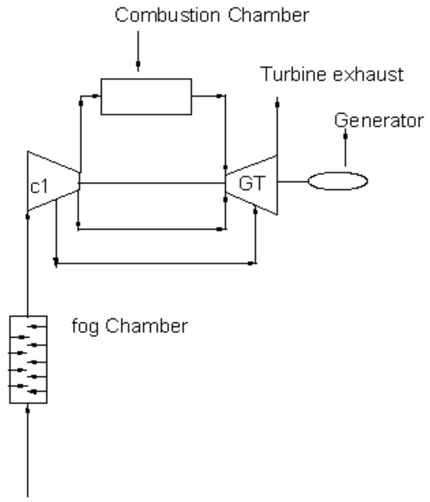

Fig. (1) Inlet fogging of BGT cycle

Unsaturated air passes from the fog chamber and absorb the sensible heat. The saturated air compressed in compressor and passes to combustion chamber, where heat is added. In combustion chamber fuel is burnt 200% excess air. High temperature flue gases expanded in the gas turbine. The gas turbine blades are cooled by air bleeded from appropriate compressor stage.

III. MODELING & GOVERNING EQUATIONS

Thermodynamic study of basic gas turbine cycle using inlet air cooling by fogging has been performed by modeling the various elements of gas turbine cycle.

3.1Gas model:

In the present work low temperature gases are modeled by assuming specific heat at constant pressure as a function of temperature alone [9] and high temperature gases fuel have been burnt in presence of 200% of excesses air.

For air at low temperature range of 200-800 K,

$$c_p(T) = a + bT + cT^2 + dT^3 + \cdots$$

--------- (1)

For high temperature gases range of 800-2200 K,

$$C_{pg} = A \times C_{pN2} + B \times C_{pO2} + C \times C_{pAr} + D \times C_{pH2o} + E \times C_{pCo2}$$

---------- (2)

Thus the enthalpy and entropy of air be expressed as

$$h = \int C_p(T)dT$$

---------- (3)

$$h = \int C_p(T)dT - R\int \frac{dp}{p}$$ --------

(4)

and for gases

$$h = \int C_{pg}(T)dT$$ -------

(5)

$$h = \int C_{pg}(T)dT - R\int \frac{dp}{p}$$ ----

(6)

3.2 Inlet air modeling:

The air before to pass compressor passes through the fog chamber. Air enter the compressor are in saturated conditions. Adiabatic cooling in fog chamber have been considered, mass flow rate and temperature after fogging are taken from psychometric chart.

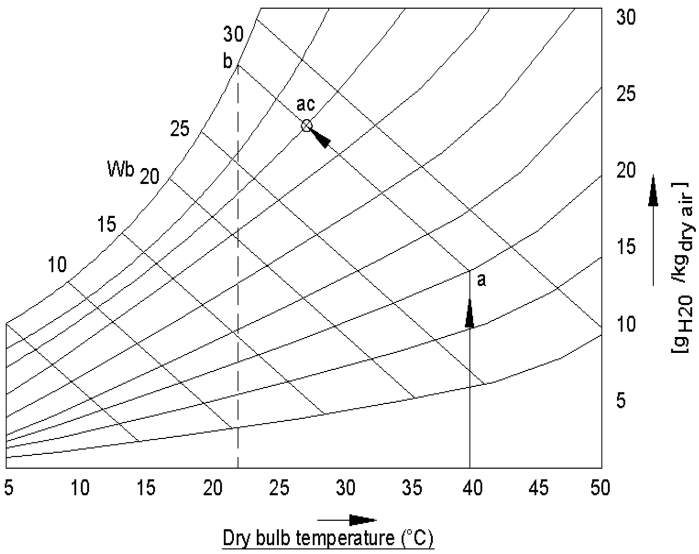

Fig (2) Psychometric chart

3.3 Compressor Model:

The concept of polytrophic efficiency thermodynamic loss in an axial compressor is incorporated in the model. The temperature and pressure relation in the compressor is

$$\frac{dT}{T} = \left(\frac{R}{\eta_c . C_{pa}}\right)\frac{dP}{P}$$ --------------- (7)

3.4 Combustor Model:

The lower heating value of fuel (natural gas) has been taken as 42MJ/kg. From mass and energy balance fuel flow are calculated at discharge temperature. Combustion efficiency & 2% pressure loss have been considered at combustion chamber.

$$m_{out} = m_a + m_f$$ ------- (8)

$$m_f \times LHV_f \times \eta_{cc} = m_{out} \times h_{out} - m_{in} \times m_{in}$$ -------- (9)

3.5 Gas Turbine Model:

The expansion in Turbine is assumed to be polytrophic to account the losses in the turbine. For a given polytrophic efficiency pressure ratio and the temperature are calculated by the relation of

$$\frac{dT}{T} = \left(\frac{R.\eta_{GT}}{C_{pg}}\right)\frac{dP}{P}$$ ----------- (10)

3.6 Cooled gas turbine model

In the present work, open loop convective cooling has been assumed. The cooled gas turbine blades model is base on the work of Horlock, et. [sj] that is

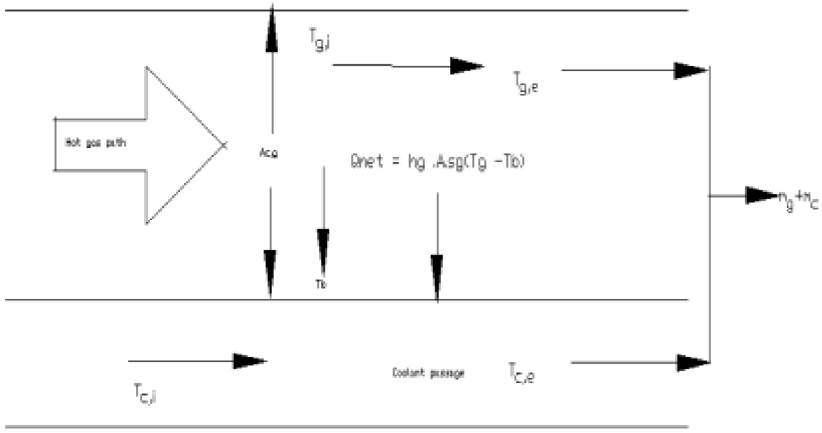

Fig. (3) Convection turbine blade cooling

A simple cooling model for internal convection cooling of blades is shown in fig.(3).The cooling factor 'R_c ' is expressed as below

$$R_c = \frac{(T_{gi}-T_b).c_{pg}}{\varepsilon(T_b-T_{ci}).c_{pc}} \qquad \text{------------ (11)}$$

A simple enthalpy balance of internally cooled blade row is given by

$$Q_{net} = \dot{m}_c \cdot c_{pc} \cdot (T_{ce} - T_{ci}) = \dot{m}_g \cdot c_{pg}\left(T_{gi} - T_{ge}\right) = \overline{h_g} \cdot A_{sg}\left(T_{gi} - T_b\right) \qquad \text{------------ (12)}$$

Using 'ε ' we have

$$Q_{net} = \dot{m}_c \cdot c_{pc} \cdot \varepsilon \cdot (T_b - T_{ci}) = \dot{m}_g \cdot c_{pg}\left(T_{gi} - T_{ge}\right) = \overline{h_g} \cdot A_{sg}\left(T_{gi} - T_b\right) \qquad \text{--------- (13)}$$

The exposed area of heat transfer $\left(A_{sg}\right)$ is give by

$$A_{sg} = \left(A_{sg}\right)_{pl} \cdot F_{sa} = \lambda \cdot F_{sa} \cdot A_{sg} = \lambda \cdot F_{sa} \cdot \dot{m}_g / \rho_g \cdot c_{pg} \qquad \text{---------- (14)}$$

Where, F_{sa} is a correction factor to account blade surface, and λ is a ratio.

Replacing A_{sg} Eq. becomes

$$\frac{m_c}{m_g} = \lambda . F_{sa} \cdot \left(\frac{c_{pg}}{c_{pc}}\right) . \overline{st_{in}} \cdot \left[\frac{(T_{gi}-T_{gb})}{\varepsilon \cdot (T_b - T_{ci})}\right] \qquad \text{---------- (15)}$$

Where $\overline{st_{in}}$ is the mean Stanton number based on the condition at cascade inlet. The ratio f wall heat transfer rate to mass heat flow rate and λ is defined as

$$\lambda = \frac{A_{sg}}{A_g} = \frac{2Hc}{H \cdot tcos\alpha} = \frac{2c}{tcos\alpha} = \frac{S_g}{tcos\alpha} \qquad \text{------------- (16)}$$

Where S_g is nearly equal to 2c.combining Esq. (8), (11) and (12) ,it would be

$$\frac{m_c}{m_g} = [\overline{St_{in}}] . \left[\frac{S_g}{tcos\alpha} . F_{sa}\right] . [R_c] \qquad \text{-------------- (17)}$$

Eq. (17) Shows that the cooling requirement in a blade row depend upon average Stanton number (St_{in}), turbine blade geometry and cooling factor. In general,

$St_{in} = 0.005$, $S_g/t \ \cos\alpha = 3.0$ and if $F_{sa} = 1.05$, Eqs. (14) takes the form as

$$\frac{m_c}{m_g} = 0.0156\,FR_c$$ -----

(18)

3.6 Gas Turbine Performance Parameters:

$$\eta_{GT} = \frac{w_{neat}}{Q_{GT}}$$ ----------

(19)

Where

$$Q_{GT} = \dot{m}_f \times LHV_f \times \eta$$ ------------

(20)

$$w_{neat} = w_{GT} \times \eta_{GT} - \frac{w_c}{\eta_c}$$ ----------

(21)

$$sfc == \frac{\dot{m}_f \times 3600}{w_{neat}}$$ ------------

(22)

IV. RESULTS AND DISCUSSION

Based on the above modeling and governing equations, in house FOGGT10 are coded in C language and the following results have been plotted using given input data.

4.1 **Input data for analysis:**

Components	Parameters and Values
Atmospheric Conditions	**HDC = 40 ^{o} C, 55% R.H.** **WDC = 25 ^{o} C,90%R.H** **RDC = 30 ^{o} C,95%R.H** **Po = 1bar**
Compressor	$\eta = 91\%$
Combustor	$\eta_{cc} = 99\%$ **Excess air = 200%** **Pressure loss = 2% of entry pressure** **LHV = 4200kJ/kg**
Gas turbine	$\eta_{gt} = 91\%$,$T_b = 1122$, **Exhaust pressure=1.05 bar**

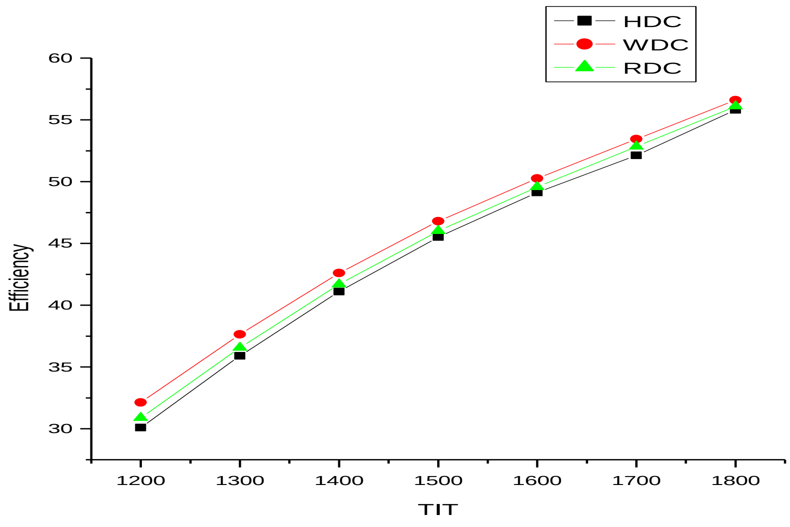

Fig.(4) Variation of efficiency with TIT

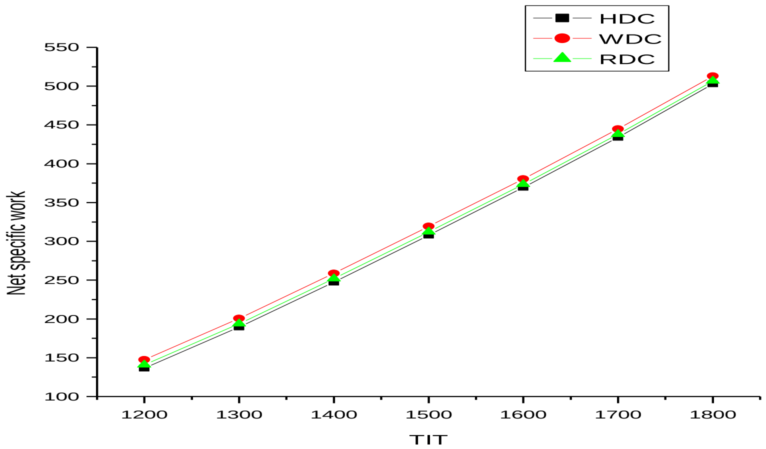

Fig. (5) Variation of net specific work with TIT

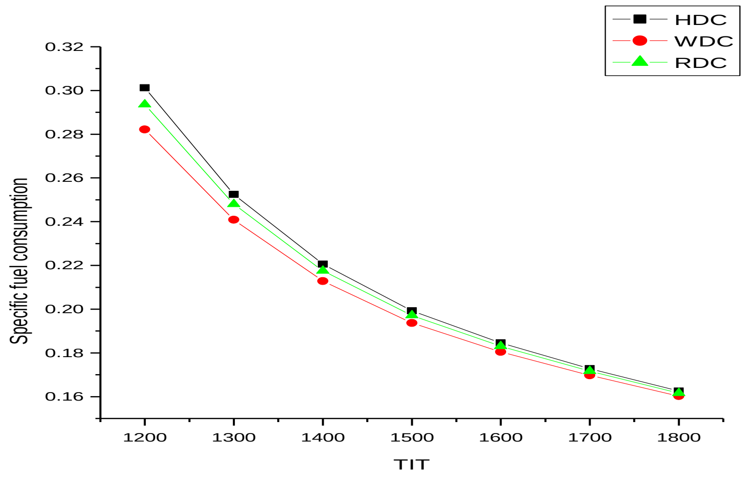

Fig. (5) Variation of net specific fuel consumption TIT

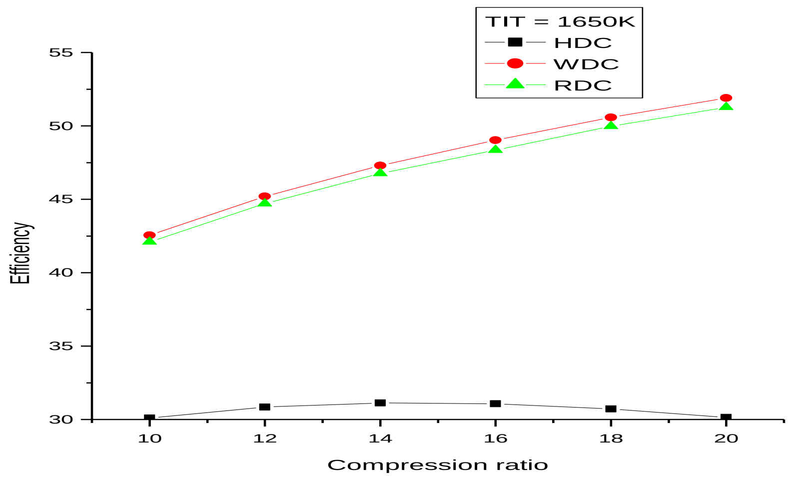

Fig. (5) Variation of efficiency with TIT

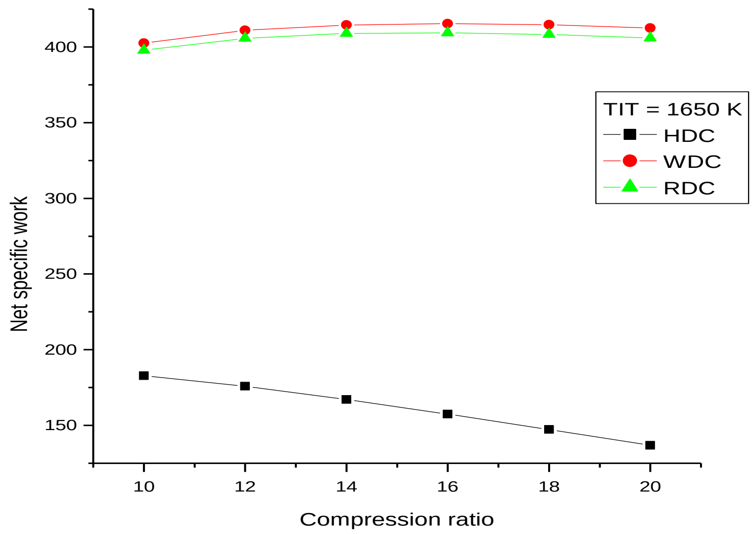

Fig. (5) Variation of net specific work with TIT

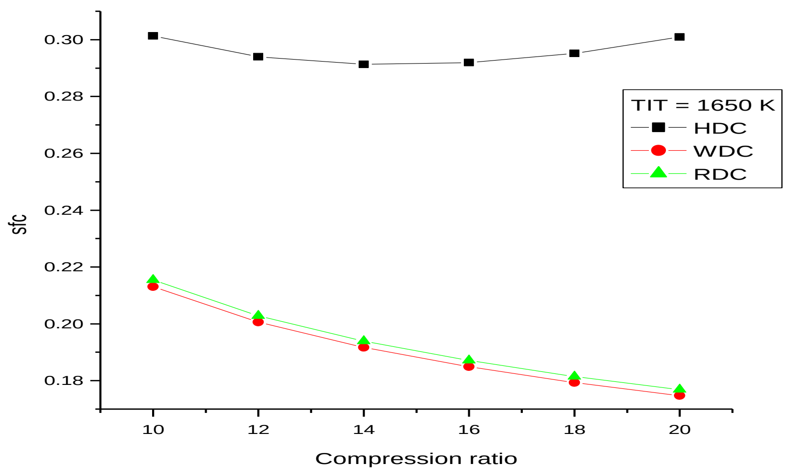

Fig. (5) Variation of sfc with TIT

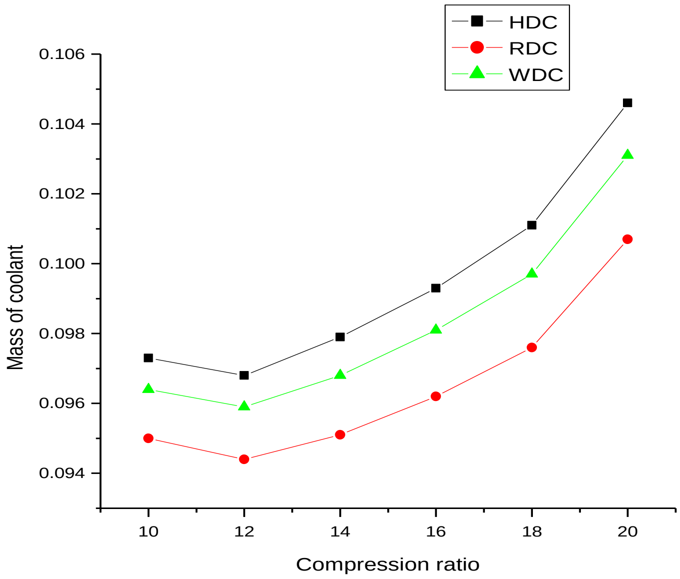

Fig. (5) Variation of mass of coolant with compression ratio

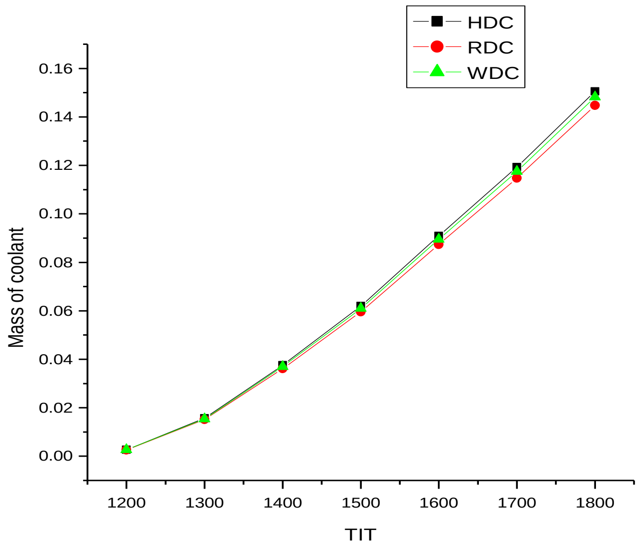

Fig. (5) Variation of mass of coolant with TIT

4.1 Effect of turbine inlet temperature:-

Fig. (4), fig. (5) &fig. (6) shows the effect of TIT on BGT performance. Increase the turbine inlet temperature increases the efficiency due to increase average heat additions. WDC ambient conditions show better efficiency at turbine inlet temperature. In BGT cycle net specific fuel consumption, net specific work at any given temperature are better in WDC compare to other ambient conditions. This shows ambient temperature play a vital role in turbine performance .

4.2 Effect of compression ratio:-

Fig.(7),fig.(8),&fig(9) shows the efficiency, specific net power and specific fuel consumption are better in WDC condition due to atmospheric temperature conditions.

Fig. (10) and (11) show the mass of bleed coolant from compressor required to cool BGT blade increases with increases compression ratio. Mass of coolant required to cool blade is more in HDC condition then other ambient condition.

V. CONCLUSION

In this parametric study following conclusion are concluded:-

(1) Net power output is increased with increased compression ratio and turbine inlet temperature.
(2) Efficiency, specific fuel consumption and net specific work is better in WDC condition
(3) Mass of coolant required to cool turbine blade is more in HDC.
(4) Fogging system is suitable in peak load period and HDC condition.

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