Effect of Compressor Inlet Temperature & Relative Humidity on Gas Turbine Cycle Performance

Anoop Kumar Shukla & Onkar Singh

Abstract- Gas turbine cycle power plants are being extensively used for power generation across the world. The variation in temperature and humidity of atmospheric air significantly affects the plant performance of naturally aspirated compressors in gas turbine based installations. There occurs the deviation in temperature from less than 5°C to more than 45°C and relative humidity variation from less than 24% to more than 81% in tropical countries. Gas turbine output is a strong function of the ambient air temperature with its' power output dropping by 0.5-0.9% for every 1°C rise in ambient temperature due to air density becoming smaller on hot days. Inlet air cooling is one of the options for maintaining the temperature of admitting air in desired limits. There are various options for inlet air conditioning such as the cooling by evaporative coolers, absorption cooling etc. In this paper a study is undertaken for the analysis of the effect of humidity and compressor inlet temperature on the performance of the gas turbine cycle power plant. Study involves thermodynamic modeling of gas turbine cycle considering the effect of humidity & inlet air temperature on the cycle performance based on the first law of thermodynamics. The results obtained on the basis of modeling have been presented and analyzed in the form of graphical patterns through variations in efficiency, specific work output, cycle pressure ratio, inlet air temperature & density variation, turbine inlet temperature, specific fuel consumption etc. Conclusions obtained are of immense utility to the power sector professionals, especially in those countries where there is significant variation in ambient air conditions.

Index Terms- Gas turbines, relative humidity, pressure ratio, compressor inlet temperature (CIT)

1. Introduction

Gas turbines are playing key role and gaining widespread acceptance in the field of mechanical drive, gas transmission and power generation. Gas turbines become very popular prime mover due to attractive properties like high power to weight ratio, compactness and ease of installation. Performance of the gas turbine cycle power plant mainly depends on the cycle pressure ratio, turbine inlet temperature and ambient air conditions.

Mohanty et al [6] investigated the performance of gas turbine cycle by intake air cooling using absorption chiller in the weather data of Bangkok. Investigation reported, as reducing the cycle inlet temperature from ambient air condition to 15°C increases the instantaneous power output between 8 and 13% as result 11% additional electricity generated from the same gas turbine power plant.

Zadpoor et al [9] studied 16.6 MW gas turbine power plant, the power output of the plant increased by 11.3 % with decreasing the ambient temperature from 34.2 °C to ISO design condition. There is 0.74 % decrease in the power output for increment of each 1 °C ambient air temperature.

Alhazmy et al [1] evaluated in their study of the performance of ABB-11D5 gas turbine operating in hot humid conditions of Jeddah Saudi Arabiathat the average power output of the plant increases by 0.57 % for each 1 °C drop in ambient air temperature. The power output is increased by 10 percent during cold humid conditions and by 18 percent during hot humid condition. The purpose of this study is to analyses the effect of humidity and compressor inlet temperature on the performance of the gas turbine cycle power plant. Results obtained from the analysis presented and discussed in the terms of power output, efficiency and fuel input.

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2. Thermodynamic modeling

Fig.1 depicts the simple gas turbine cycle with inlet air cooler, where ambient air goes into the air cooler in which it is cooled up to certain temperature and has different relative humidity at inlet to compressor. After compression, the air goes into the combustion chamber where compressed air is mixed with fuel and burnt, so that the temperature of burnt gases increases up to the level of turbine inlet temperature (TIT). Burnt gases are further sent to turbine for expansion and producing positive work and subsequently exhausted through stack. Thermodynamic evaluation of gas turbine cycle has been carried out by modeling the different component of the cycle based on energy and mass balance upon them. The details of the modeling of different component are given ahead.

2.1 Gas Model

The specific heat of air and burnt gases is considered as the function of temperature, the ratio of specific heats (γ) for burnt gases also vary with temperature. The logarithmic polynomial functions given by Lanzafame and Messina [4] have been used to evaluate the enthalpy and specific heats of working fluids at different stages of the gas turbine.

$$c_p(T) = a + bT + cT^2 + dT^3 + \cdots \tag{1}$$

Where a, b, c & d are the polynomial coefficient.

Enthalpy of air at any temperature is given by [10]

$$h = c_{pair}T + (2500 + c_{pH_2O}T) \tag{2}$$

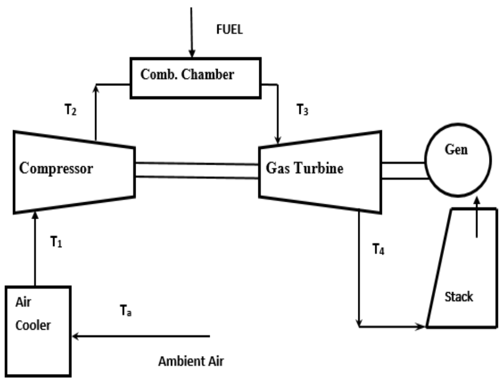

Fig.1 Schematic diagram of simple gas turbine power plant with inlet air cooler

The enthalpy of combustion gases at any temperature is given by the [10]

$$h = \int_{T_0}^{T} [\, c_{pCO2}(T) + c_{pH2O}(T) + c_{pO2}(T) + \, c_{pN2}(T)] dT \tag{3}$$

The fuel used in the combustion chamber is natural gas whose composition by volume is taken as the

CH_4 = 90.00%; C_2H_6 = 4.50%; N_2 = 4.00%; CO_2 = 1.50%. The enthalpy is given zero value at the ambient pressure of 1 bar and temperature 288 K.

2.2 Air Cooler

Fig.2 shows the air cooler in which air is passing over the outer surface of the coil and experiences a drop in temperature and specific humidity. The temperature of coil is adjusted to allow air to reach certain desired temperature.

Applying mass and energy balance on air cooler gives

$$c_{pa}(T_1 - T_a) = \omega_1 h_{fg1} + \omega_a(h_{ga} - h_1) \tag{4}$$

$$\omega_a = \frac{0.622 P_{ga}}{P_a - P_{ga}} \tag{5}$$

2.3 Compressor Model

An axial flow compressor is considered with the suitable value of polytropic efficiency to take care of various thermodynamic losses occurring in it. The relation between temperature and pressure of air at any section of compressor is given by the expression

$$\frac{dT}{T} = \left(\frac{R_c}{\eta_{pt} \cdot c_{pc}}\right)\frac{dP}{P} \tag{6}$$

Where η_{pt} is the polytropic efficiency and c_{pc} and R_c specific heat at constant pressure and gas constant across the compressor respectively.

Compressor work per unit mass is calculated by using mass and energy balance across the compressor and is given by

$$w_c = h_{c,e} - h_{c,i} \tag{7}$$

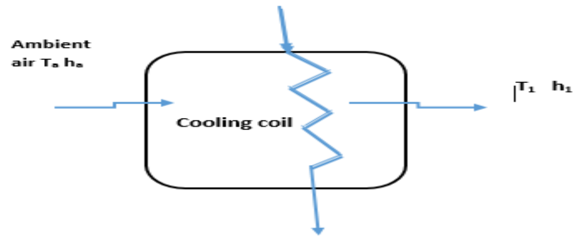

Fig.2 Air cooler

2.4 Combustion Chamber Model

Application of the mass and energy balance across the control volume of combustion chamber gives the mass of fuel required to attain a specified exit temperature of combustion chamber i.e. turbine inlet temperature(TIT). Mass and energy balance yields

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \tag{8}$$

$$\dot{m}_f . LCV . \eta_{cc} = \dot{m}_e h_e - \dot{m}_i h_i \tag{9}$$

2.5 Gas Turbine

Considering polytropic expansion in the gas turbine the relation between the temperature and pressure at any stage of expansion is given by

$$\frac{dT}{T} = \left(\frac{dP}{P}\right)^{\eta_{pt}\left(\frac{\gamma-1}{\gamma}\right)} \tag{10}$$

Gas turbine work per unit mass is calculated by using mass and energy balance across the turbine and is given by

$$w_{gt} = h_{gt,e} - h_{gt,i} \tag{11}$$

The performance parameters of the gas turbine cycle power plant are specific work output, cycle efficiency and specific fuel consumption and are calculated from the following governing equations

$$w_{net} = w_{gt} - w_c \tag{12}$$

$$\eta_{th} = \frac{w_{net}}{\dot{m}_f . LCV} \tag{13}$$

$$sfc = \frac{3600 f}{w_{net}} \tag{14}$$

Thermodynamic modeling and governing equations of the above proposed cycle is coded using C++ for carrying out the analysis.

3. Results & Discussion

Present paper analyzed the effect of operating parameters like turbine inlet temperature, pressure ratio, relative humidity and ambient air conditions on the performance of the gas turbine cycle. To analyze the effect of compressor inlet temperature and relative humidity on the performance of gas turbine power cycle various input parameter is taken from the Table 1 given in the appendix.

Fig.3 depicts the variation of thermal efficiency with respect to relative humidity on various compressor inlet temperature of gas turbine cycle at 1150K TIT and CPR equal to 12. It is seen that the thermal efficiency increases with increase in relative humidity from 0 to 100 percent for particular CIT. There is 0.65 percent increase in thermal efficiency for every 15 percent increase of relative humidity. It is evident that GT power cycle is more efficient for compressor inlet temperature of 282K.

In Fig.4 the specific work output variation is shown with respect to relative humidity for various compressor inlet temperature of GT power cycle at TIT and CPR of 1150K and 12 respectively. Here the specific work output slightly increases with increase in relative humidity from 0 to 100 percent for varying compressor inlet temperature. As the relative humidity increases the compressor work increases and this increment is taken care by increase in gas turbine work due to high specific heat capacity of combustion gas. Effectively the GT cycle produces more work for lesser temperature at compressor inlet. There is 0.77 percent increase in the specific power output for every 15 percent increase of relative humidity of GT cycle power plants.

Fig.5 shows the variation of specific fuel consumption for GT cycle with respect to relative humidity on various compressor inlet temperatures at 1150K TIT and CPR equals to 12. Specific fuel consumption decreases with increase in relative humidity from 0 to 100 percent and increases as increasing compressor inlet temperature. There is 0.75 percent fuel reduction for every 15 percent increment of RH. The lesser fuel is consumed for 282K temperature at compressor inlet.

Fig.6 presents the variation of specific work output of the GT cycle with different TIT on various compressor inlet temperature of gas turbine cycle at CPR and RH of 16 and 60 percent respectively. It is evident that the specific work output increases with increase in gas turbine inlet temperature for particular CIT. There is 10.3 percent increment of specific work for every 50K increment of TIT. Graphical presentation shows that GT power cycle gives more work for lower compressor inlet temperatures.

Fig.7 shows the variation of thermal efficiency with different TIT on various compressor inlet temperature of gas turbine cycle CPR and RH of 16 and 60 percent respectively. Increasing the TIT of the GT cycle gives more turbine work for particular compressor inlet temperature this cause the increase in the thermal efficiency of the power cycle. There is 0.7 percent increment of thermal efficiency for every 50K increment of TIT. Figure shows that GT power cycle is more efficient for 282K compressor inlet temperature.

Fig.8 depicts the variation of specific fuel consumption for GT cycle with different TIT on various compressor inlet temperatures CPR and RH equal to 16 and 60 percent respectively. Specific fuel consumption decreases with increase in turbine inlet temperature and decreases as decreasing the compressor inlet temperature. There is 0.77 percent reduction in the sfc for increment of 50K in the TIT level. There is average 0.968 percent fuel consumption increases for increment of every 9K in the compressor inlet temperature.

The effect of the pressure ratio on the specific work output of the GT cycle at TIT of 1150 K and RH equal to 60% on various compressor inlet temperature of gas turbine cycle is shown in Fig 9. It shows that the specific work output increases with increase in gas turbine cycle pressure ratio for particular CIT. It is evident from the variation initially specific work increases at a higher rate by increasing the CPR by 8 to 16, after that gain in specific work become slower due to the increasing component of compression work. It is seen that GT power cycle gives more work for lower compressor inlet temperatures.

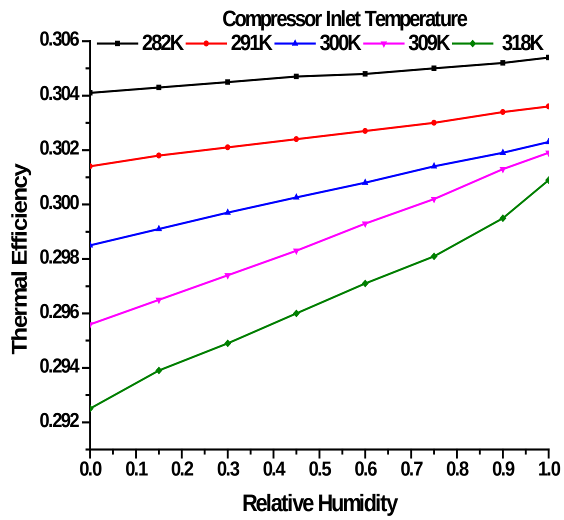

Fig.3 Variation of thermal efficiency of GT power cycle with different relative humidity at TIT 1100K & CPR 8

Fig.10 shows the variation of thermal efficiency with different cycle pressure ratio on various compressor inlet temperature of gas turbine cycle at TIT of 1150 K and RH equal to 60%. Increasing the cycle pressure ratio of the GT cycle gives more turbine work for particular compressor inlet temperature this cause the increase in the thermal efficiency of the power cycle. Figure shows that the GT power cycle is more efficient for 282K compressor inlet temperature.

Fig.11 depicts the variation of specific fuel consumption for GT cycle with different cycle pressure ratio on various compressor inlet temperatures at TIT of 1150 K and RH equal to 60%. Specific fuel consumption decreases with increase in cycle pressure ratio and decreases with decreasing compressor inlet temperature.

Fig.12 shows the variation of specific work output of the GT power cycle with different compressor inlet temperature at TIT equal to 1200K and RH equal to 60 percent.

In this figure the specific work output increases with increase in gas turbine cycle pressure ratio from 8 to 16 at particular CIT. There is 10.12 percent gain in the specific work output by decreasing the compressor inlet temperature from 318K to 282K. It is seen from the figure that GT power cycle gives more work for 282K compressor inlet temperature.

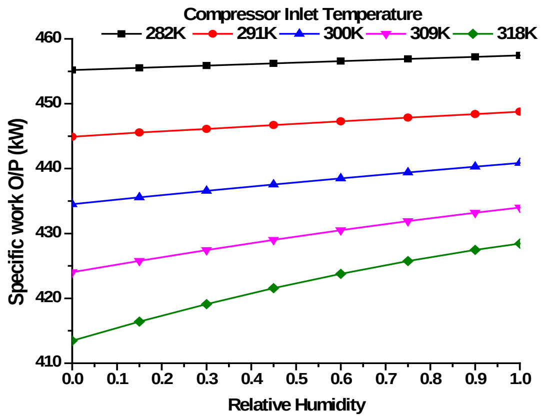

Fig.4 Variation of specific work output of GT power cycle with different relative humidity at TIT 1100 K & CPR 8

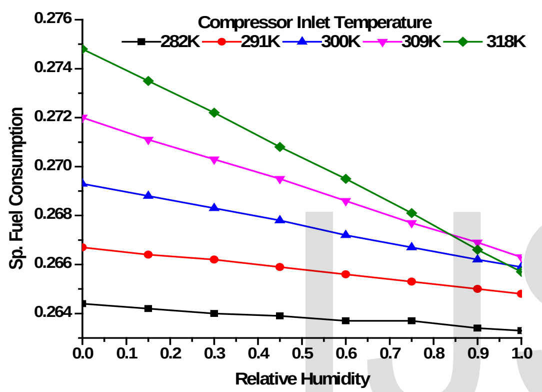

Fig.5 Variation of specific fuel consumption of GT power cycle with different relative humidity at TIT 1100 K & CPR 8

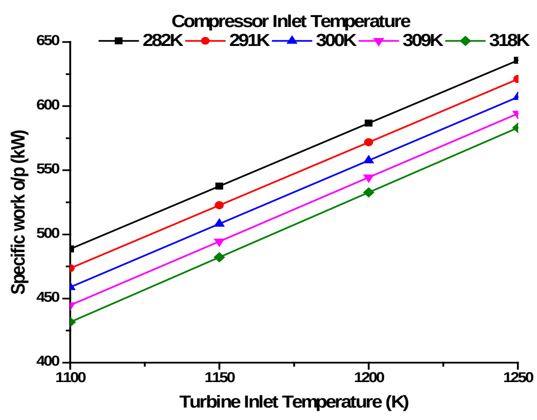

Fig.6 Variation of specific work output of GT power cycle with different turbine inlet temperature at CPR 16

The variation of thermal efficiency with different compressor inlet temperature at TIT equal to 1200K and RH equal to 60 percent is shown in fig.13. Increasing the cycle pressure ratio of the GT cycle gives more turbine work for particular compressor inlet temperature this cause the increase in the thermal efficiency of the power cycle. There is 3.45 percent gain in the thermal efficiency by decreasing the compressor inlet temperature from 318K to 282K. It shows that GT power cycle is more efficient for 282K compressor inlet temperature.

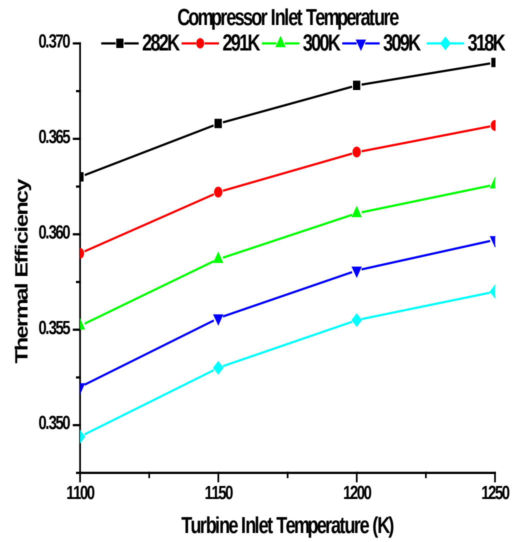

Fig.7 Variation of thermal efficiency of GT power cycle with different turbine inlet temperature at CPR 16

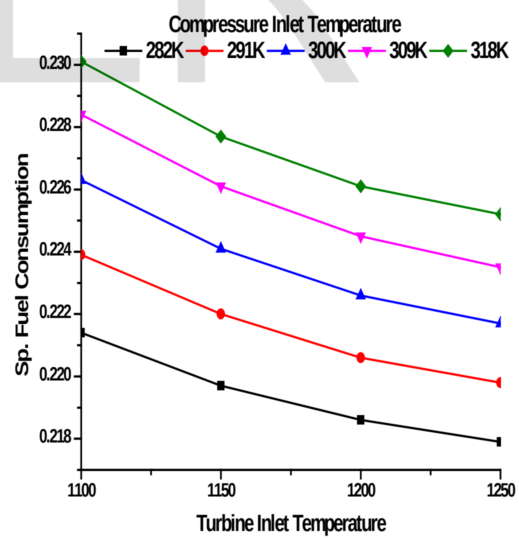

Fig.8 Variation of specific fuel consumption of GT power cycle with different turbine inlet temperature at CPR 16

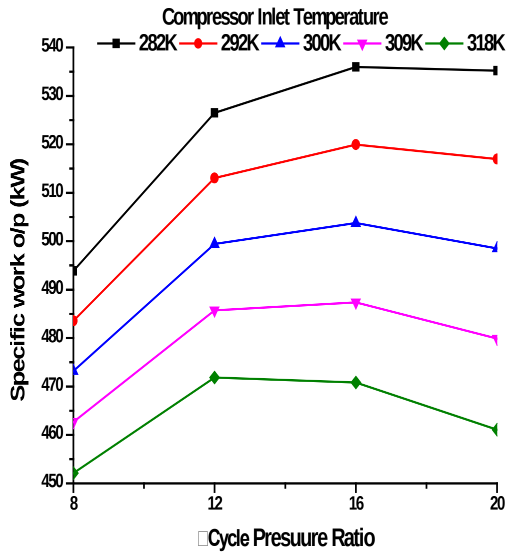

Fig.9 Variation of specific work output of GT power cycle with different cycle pressure ratio at TIT 1150K

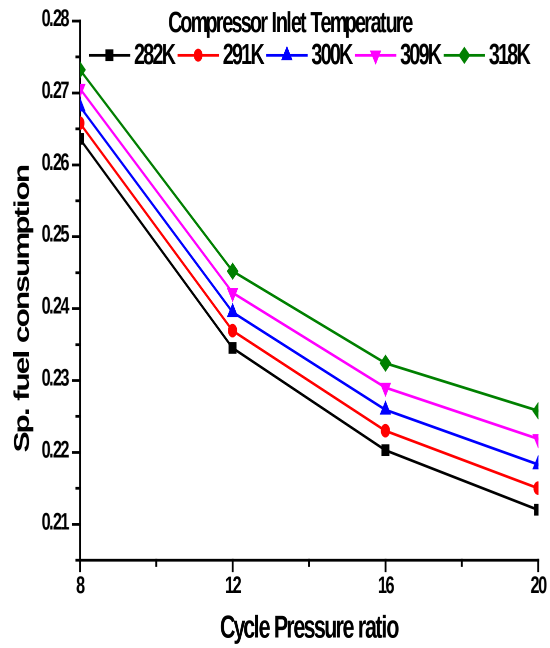

Fig.11 Variation of specific fuel consumption of GT power cycle with different cycle pressure ratio at TIT 1150K

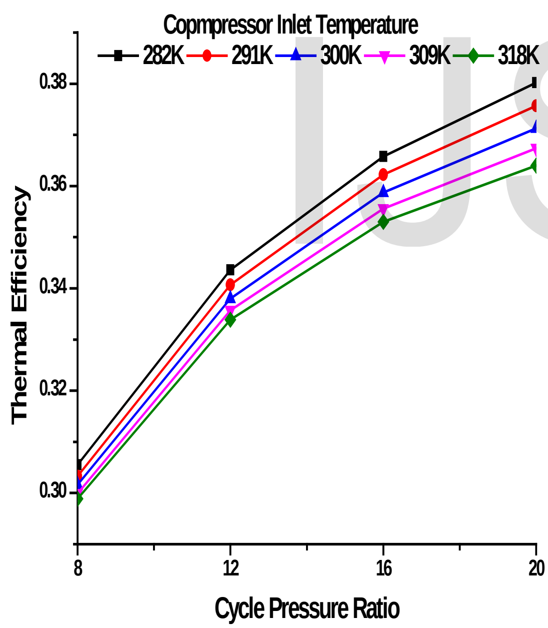

Fig.10 Variation of thermal efficiency of GT power cycle with different cycle pressure ratio at TIT 1150K

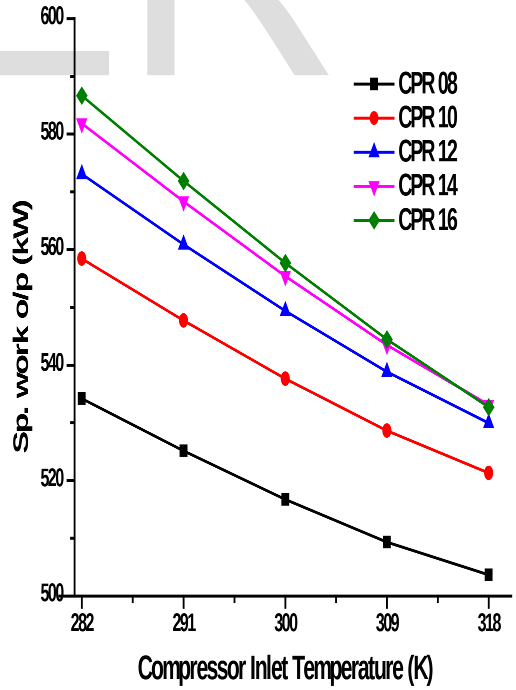

Fig.12 Variation of specific work output of GT power cycle with different compressor inlet temperature at TIT 1200K

Fig.14 depicts the variation of specific fuel consumption for GT cycle with different compressor inlet temperatures at TIT equal to 1200K and RH equal to 60 percent. Specific fuel consumption decreases with increase in cycle pressure ratio and decreases as decreasing the compressor inlet temperature. By lowering the CIT from 318K to 282K there is reduction of fuel consumption by 3.43 percent.

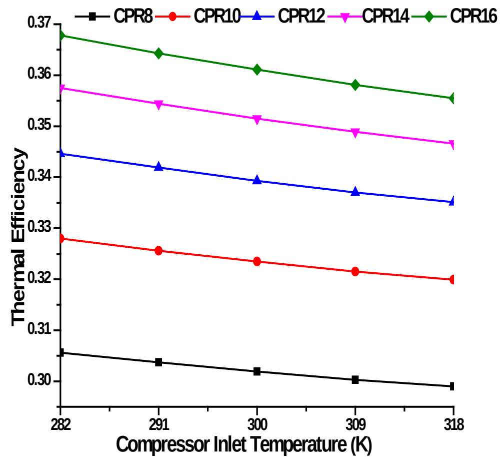

Fig.13 Variation of thermal efficiency of GT power cycle with different compressor inlet temperature at TIT 1200K

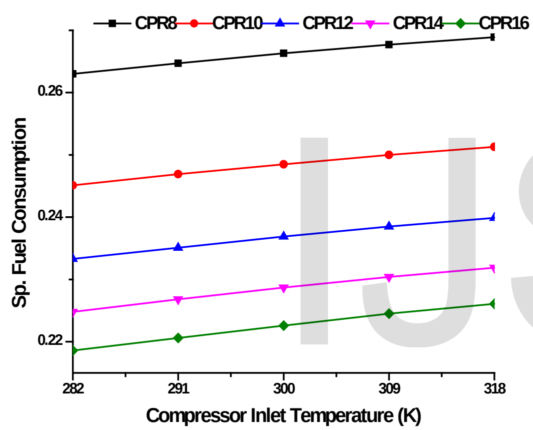

Fig.14 Variation of specific fuel consumption of GT power cycle with different compressor inlet temperature at TIT 1200K

4. Conclusions

It is evident from the present study that the performance parameters having strong dependency on the climatic conditions. Decreasing the compressor inlet temperature is useful tool for increasing the net power generation capabilities of GT cycle power plants. Other conclusions drawn from the present analysis are presented below:
Reducing the inlet air temperature to the gas turbines increases the mass flow rate and enhances its net power and thermal efficiency.
There is 0.77 percent increase in the specific power output and 0.65 percent increase in thermal efficiency for every 15 percent increase of relative humidity of GT cycle power plants.
There is 10.12 percent gain in the specific work output, 3.45 percent in thermal efficiency and 3.43 percent saving of fuel by decreasing the compressor inlet temperature from 318K to 282K.

Acknowledgments

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Appendix

Nomenclature

P	Pressure (atm)
T	Temperature (K)
RH	Relative humidity
TIT	Turbine inlet temperature
CIT	Compressor inlet temperature
GT	Gas turbine
CPR	Cycle pressure ratio

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c_p	Specific heat at constant pressure (kJ/kgK)
h	Enthalpy (kJ//kg)
w	Specific work (kJ/kg)
η	Efficiency
sfc	Specific fuel consumption
ω	Humidity Ratio
ϕ	Relative humidity
Υ	Adiabatic exponent
LCV	Lower calorific value (kJ/kg)

Subscripts

a	Air
c	Compressor
cc	Combustion chamber
pt	Polytropic
gt	Gas turbine
e	Exit
i	Inlet
g	Combustion gas
f	Fuel
N_2	Nitrogen
CO_2	Carbon di oxide
O_2	Oxygen
H_2O	Water vapor

Table.1 Input Parameters for Gas Turbine Performance Analysis [7, 8]

Compressor polytropic efficiency = 92%
Ambient Air temperature = 318K
Compressor inlet pressure = 1atm
Pressure loss in combustion chamber = 3% of entry
Gas turbine polytropic efficiency = 92%
Gas turbine inlet temperature = 1100K, 1150K, 1200K, 1250K
Cycle pressure ratio 8, 10, 12, 14, 16, 20
Combustion Chamber Efficiency = 99.5%
Natural gas composition CH_4 = 90.00%, C_2H_6 = 4.50%,
LCV of natural gas = 44769 kJ/kg