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Parametric Analysis of Cooled Gas Turbine Cycle with Evaporative Inlet Air Cooling

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Abstract— The article investigates the effect of compressor pressure ratio, turbine inlet temperature and ambient temperature on the performance parameters of an air-cooled gas turbine cycle with evaporative cooling of inlet air. Air film cooling has been adopted as the cooling technique for gas turbine blades. The mass of coolant required for turbine blade cooling is calculated for a selected range of ambient conditions and found to vary with temperature drop achieved in the evaporator. The effect of ambient temperature on plant efficiency and plant specific work is computed at different TIT and $r_{p,c}$ and it was found that the rate of increase in these performance parameters are more pronounced at higher TIT and $r_{p,c}$. The results indicate that a maximum temperature drop of 21 ^oC is achieved in the evaporator. The inlet cooling is found to increase the efficiency by 4.1% and specific work by 9.44%. The optimum plant performance is obtained at a TIT of 1500K and $r_{p,c}$ of 20 for all values of ambient temperature .

Index Terms— Air-film blade cooling, Air-humidifier, Evaporative cooling, Gas turbine, Inlet-air cooling, Parametric analysis.

1 INTRODUCTION

Gas turbines have gained widespread acceptance in the power generation, mechanical drive, and gas transmission markets. Their compactness, high power-to-weight ratio, and ease of installation have made them a popular prime mover. Improvements in hot section materials, cooling technologies, and aerodynamics have allowed increases in firing temperatures. 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 % to 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.

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

The highest losses in gas turbine power output usually coincide with periods of high electricity demand. A gas turbine loses approximately 7% of its nominal power when the intake temperature increases from 15 ^oC, ISO conditions, to 25 ^oC, and in cases such as in summer when the ambient temperature increases above 25 ^oC, the losses are still bigger, reaching even 15% of the power rating with 36 ^oC. 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, air density increases with drop in inlet temperature, which results in an increase in mass flow rate of air entering the compressor and so the power output is enhanced**.** De Lucia et al. [2] have studied gas turbines with inlet air cooling and concluded that evaporative cooling could enhance the power produced by up to 4% per year. Hosseini et al. [3] have summarized that the output of a gas turbine of the Fars combined cycle power plant at 38 ^oC ambient temperature and 8% relative humidity is 11 MW more, and the temperature drop of the inlet air is about 19 ^0C with media evaporative cooling installations. Zadpoor and Nikooyan [4] concluded that evaporative inlet cooling systems do not work well in humid areas. Najjar [5] analyzed that the addition of absorption inlet cooling improves the power output by about 21%, Overall thermal efficiency by about 38 % and overall specific fuel consumption by 28 %. Bassily [6] carried out energy balance analysis of gas turbine cycles and reported that introducing indirect inlet air cooling, evaporative after cooling of the compressor discharge along with regeneration, intercooling, and reheat increase the performance significantly. Wang and Chiou [7] calculated the expected benefits from the inclusion of the inlet cooling feature is about 12% increase in power output and 5.16% increase in efficiency when the ambient temperature is cooled from 305 to 283 K. In addition to this investigation by other researchers [8],[9],[10],[11] have also been carried out in the field of inlet air cooling of gas turbine.
The search for a better performance of gas turbine engines has also led to higher turbine inlet temperatures. However, the maximum value of TIT is restricted by metallurgical limits of turbine blade material, which should be kept at 1123 K in order to protect the blades from damage. The TIT could be raised above this limit by using especial high temperature materials or cooling the hot turbine components with a suitable coolant. 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. [12], Wu and Louis [13], El-Masri [14],[15], Briesh et al. [16], Bolland and Stadaas [17], Bolland [18], Chiesa and Macchi [19],Dechamps [20], and Sanjay et. al. [21],[22],[23],[24],[25].

It is observed from a study in the above-cited literatures that none of the previous work has investigated the simultaneous

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effect of compressor inlet air-cooling and gas turbine blade cooling on the performance of gas turbine cycle. It is interesting to note that the mass of coolant required for gas turbine blade cooling is a function of inlet air temperature and reduces with increase in temperature drop.

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 observed on the thermodynamic performance parameters of the cycle.

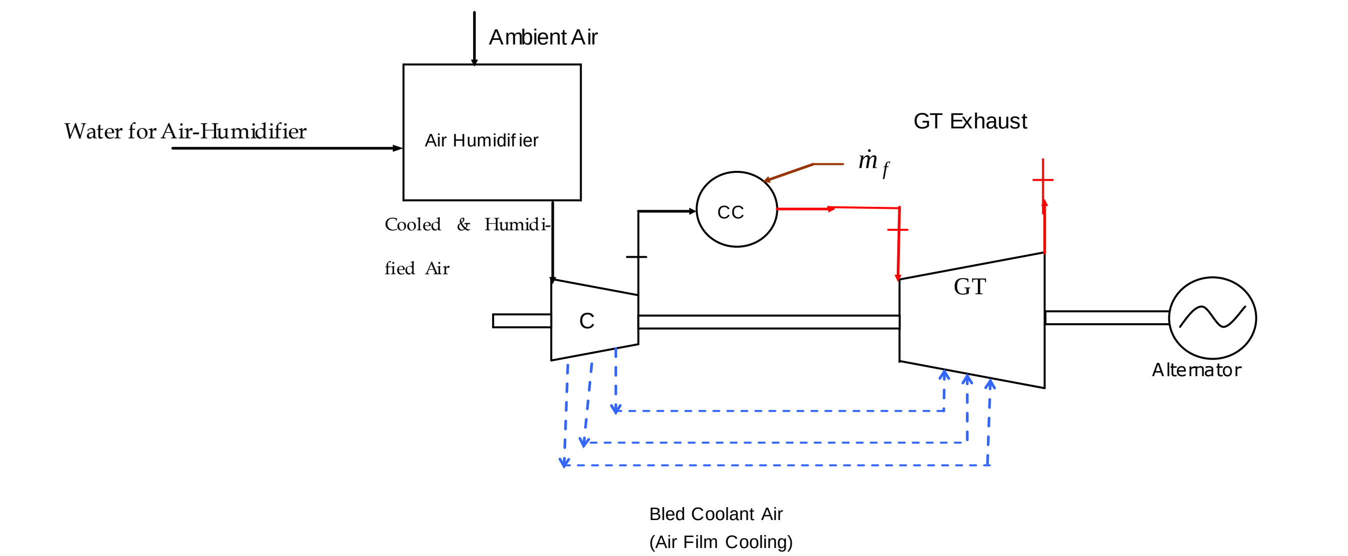

Fig. 1. Schematic of Air humidifier integrated gas turbine.

2 MODELING OF GOVERNING EQUATIONS

Parametric study of the combined cycle using different means of cooling 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. 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).

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 only with temperature in the form of polynomials as follows

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

where a, b, c, and d are coefficient of polynomials, as taken from the work of Touloukian and Tadash [26]. A factor called humidity correction factor represented by f_h is introduced to take into account the increase in specific humidity of ambient air across the air-humidifier and it is calculated as

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

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

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 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 taken into account is CH_4 = 86.21 %, C_2H_6 = 7.20 %, and CO_2 = 5.56 %, and N_2 = 1.03 % by weight.

2.2 Humidifier Model

Cooling in hot, relatively dry climate can be accomplished by evaporative cooling. Evaporative cooling involves passing air across a spray of water or forcing air through a soaked pad that is kept replenished with water [27]. 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 95%

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- 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

$$\omega_{a,e} = \omega_{a,i} + m_w \tag{4}$$

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

$$\omega = \frac{0.622\, p_{vap}}{p - p_{vap}} \tag{5}$$

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 desired temperature.

The energy balance equation for the humidifier is given by

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

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

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

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

$$T_{a,e} = t_{a,e} + 273 \tag{7c}$$

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

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

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

$$R_c = c_{p,c} - c_{v,c} \tag{9}$$

where

$$c_{p,c} = c_{p,a} + \omega_{ai} c_{p,vap} \tag{10}$$

$$c_{v,c} = c_{v,a} + \omega_{ai} c_{v,vap} \tag{11}$$

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 [27]:

$$c_{p,a} = \frac{8.314}{28.97}\begin{pmatrix} 3.653 - 1.337\times10^{-3} T_{av} + 3.294\times10^{-6} \\ T_{av}^2 - 1.913\times10^{-9} T_{av}^3 + 2.763\times10^{-13} T_{av}^4 \end{pmatrix} \tag{12}$$

$$c_{v,a} = c_{p,a} - 0.287 \tag{13}$$

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 [27]:

$$c_{p,vap} = \frac{8.314}{18.02}\begin{pmatrix} 4.07 - 1.108\times10^{-3} T_{av} + 4.152\times10^{-6} \\ T_{av}^2 - 2.964\times10^{-9} T_{av}^3 + 8.07\times10^{-13} T_{av}^4 \end{pmatrix} \tag{14}$$

$$c_{v,vap} = c_{p,vap} - 0.4614 \tag{15}$$

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:

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

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

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 yield the mass of fuel required to attain a specified exit temperature of combustor which is taken as turbine inlet temperature (TIT), given by,

$$\dot{m}_e = \dot{m}_i + \dot{m}_f \tag{18}$$

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

2.5 Cooled Gas Turbine Model

Unlike steam turbine blading, gas turbine blading 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

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be cooled by air-film cooling (AFC) method. The cooling model used for cooled turbine is the refined version of that by Louis et al [12]. The mass flow rate of coolant required in a blade row is expressed as [25]:

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

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), $St_{in} = 0.005$.

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 at the compressor inlet. With a drop in temperature of air at the inlet of compressor achieved in the humidifier, 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.

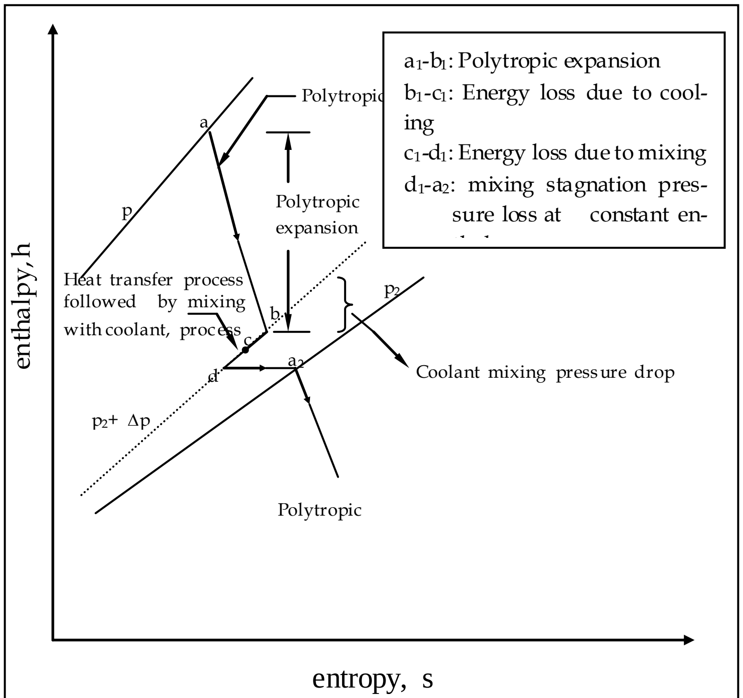

Fig. 2. Expansion path in cooled gas turbine row

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% [25]. Turbine work and exergy destruction are given by the mass, energy and exergy balance of gas turbine as under:

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

3 PERFORMANCE PARAMETERS

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

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

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

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

Modeling of cycle components and governing equations developed for cycles 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 [25]. The input data used in the analysis is given in Table 1.

4 RESULTS AND DISCUSSIONS

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 a plot illustrating the nature of variation of specific heat of air as its temperature rises due to compression process through the compressor It is clear from the graph the specific heat of air is a function of temperature and specific heat of air is higher for higher values of temperature at the inlet of compressor

Fig. 4 depicts the variation in mass of turbine blade coolant required with respect to ambient temperature at various values of TIT in an air-humidifier integrated gas turbine. It can be seen that as the ambient temperature increases, though the drop in temperature achieved in the humidifier is more (due to higher difference between WBT and DBT) the temperature of air entering the compressor is also higher than design ISO condition. This results in higher amount of coolant bleed from the compressor as per discussions detailed in section 2.5 above. It is worthwhile to note that the coolant requirement in air-humidifier integrated gas turbine will always be lesser than in a gas turbine without inlet-air humidifier.

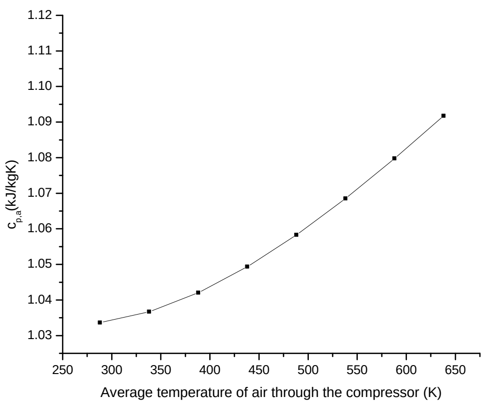

Fig. 3. Variation of specific heat of air w.r.t. average air temperature through the compressor

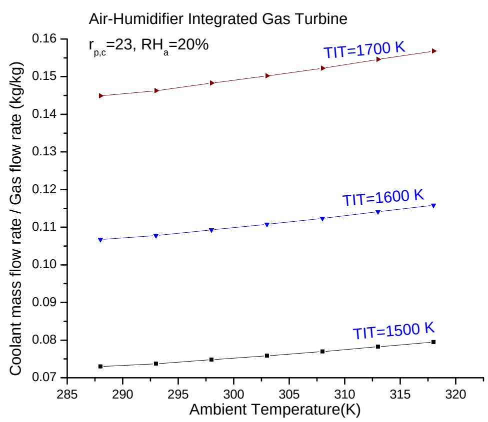

Fig. 4. Effect of ambient temperature on coolant flow rate at various values of TIT in an AHIGT.

Fig. 5 depicts the variation in mass of blade coolant required with respect to ambient temperature at different compressor pressure ratio. The coolant requirement increases with increase in ambient temperature and is higher at higher value of $r_{p,c}$. The enhanced coolant requirement is due to higher temperature of bleed air at respective bleed points, which achieve lower level of blade cooling effectiveness. It can also be observed that the cooling requirement at higher values of $r_{p,c}$ is also higher at higher values of temperature as per discussions detailed in section 2.5 above.

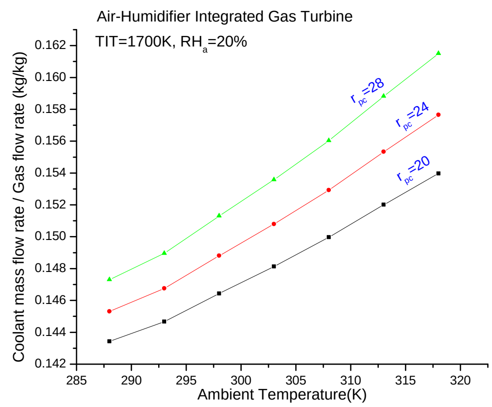

Fig.5. Effect of ambient temperature on coolant flow rate at various $r_{p,c}$ in an AHIGT.

The Fig. 6 (a) and (b) shows a histogram of the sensitivity of plant efficiency and plant specific work to ambient air temperature for fixed values of TIT and $r_{p,c}$ in the case of air-humidifier integrated gas turbine. The histogram illustrates the advantage of inlet air-cooled GT cycle over the basic GT cycle without inlet air-cooling in terms of enhancement in plant efficiency and plant specific work. It is observed that both plant efficiency and specific work of inlet air-cooled GT cycle are higher than that without inlet air cooling for a given ambient temperature. For a given relative humidity, as the ambient temperature increases, an improvement in plant efficiency and specific work is observed due to inlet cooling (though the efficiency is reduced for both the cases). This is because at higher ambient temperature the difference between wet and dry bulb temperature is higher resulting in more effective cooling of inlet air, leading to larger temperature drop in the humidifier and hence lesser coolant requirement. The values of plant specific work and plant efficiency compares well with published works and existing gas turbines with variation in the range of 2.5 to 3% as shown in table. 1

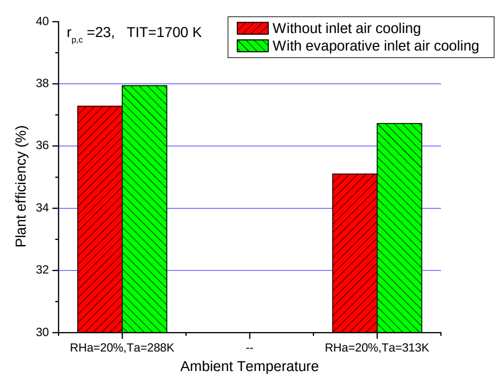

Fig 6 a. Effect of ambient temperature and evaporative inlet air cooling on plant efficiency

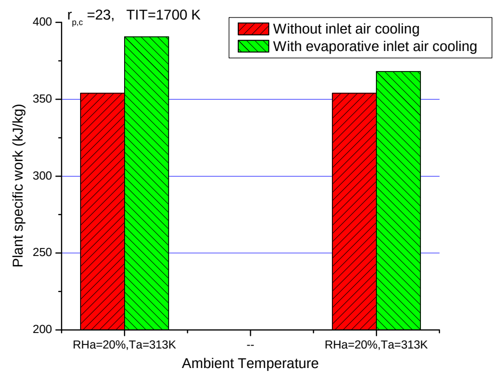

Fig 6 b. Effect of ambient temperature and evaporative inlet air cooling on plant specific work

Fig. 7 depicts the effect of TIT and drop in temperature achieved in the evaporator (due to increase in ambient temperature) on turbine efficiency and specific work. It is interesting to note that the efficiency and specific work reduces even when there is a drop in the temperature of air. This is because at higher air ambient temperature, though the drop in temperature is higher, owing to higher difference between WBT and DBT, the compressor inlet temperature is also high. At an ambient temperature of 318 K the humidifier produces a drop of 20 K resulting in a compressor inlet temperature of 298 K. In comparison at 293 K ambient temperature though the temperature drop across the humidifier is only 11 K, the compressor inlet temperature is also lower (284 K). However, as the ambient temperature increases due to higher drop in temperature achieved in the evaporator, the diminution in efficiency and specific work is not as pronounced as in the case of a gas turbine cycle without inlet cooling as per discussions detailed for Fig. 6(a) above.

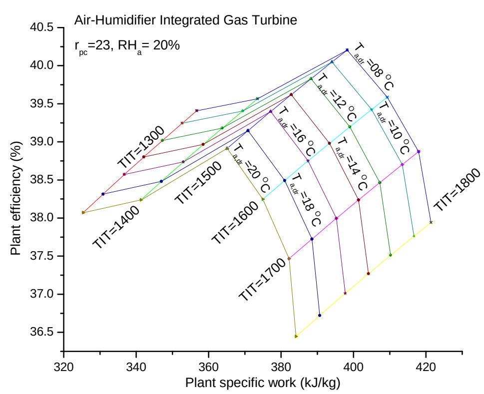

Fig 7. A design monogram of efficiency versus specific work in an AHIGT for different TIT and drop in temperature of air in the humidifier due to rise in ambient temperature

As the compressor inlet temperature increases, the compressor work is increased and the mass of coolant required increases correspondingly as per discussions detailed in section 2.5 above. Although the fuel consumption is reduced for a given TIT, the drop in plant specific work due to rise in mixing and cooling losses owing to higher amount of coolant required and the increase in compressor work due to higher compressor inlet temperature together results in a net reduction in plant specific work and plant efficiency. It is also found from the graph that the rate of increase in efficiency is more pronounced at higher TIT.

Fig. 8 depicts the effect variation of $r_{p,c}$ and drop in temperature achieved in the evaporator , on plant specific work and plant efficiency. It is observed that the specific work and efficiency increases with increase in temperature drop of air achieved in the humidifier, the enhancement being higher in higher range of $r_{p,c}$. The inlet air cooling boosts the efficiency by 3.44 % at $r_{p,c}$=23, RH_a=20% and a TIT of 1300 K when the ambient temperature drops by 30^0C. This enhancement increases to 4.1% for a TIT of 1800 K at the same value of $r_{p,c}$, RH_a and ambient temperature drop. The effect of variation of $r_{p,c}$ on plant 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 increase in temperature drop achieved in the humidifier. This suggests that for a higher value of plant specific work there exists an optimum value of $r_{p,c}$ (for a given temperature drop achieved in the humidifier) and the $r_{p,c}$ needs to be chosen judiciously.

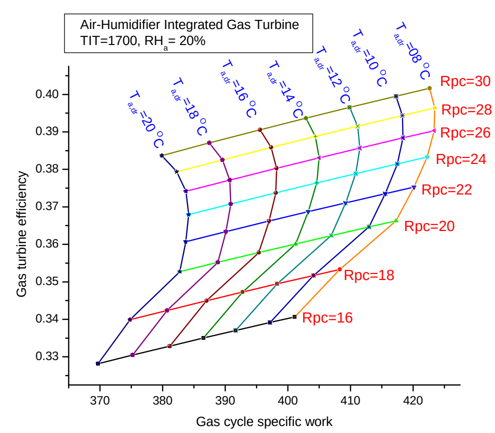

Fig 8. A design monogram of plant efficiency versus plant specific work in an AHIGT for different TIT and and drop in temperature of air in the humidifier

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TABLE 1
INPUT DATA FOR ANALYSIS [25, 28]

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 Air inlet temperature= 288 Inlet plenum loss= 0.5% of entry pressure	% % K 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	%

J = joule, kg = kilogram, K = kelvin.

5 CONCLUSION

Based on the analysis of air-humidifier-integrated-gas turbine cycle 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 coolant mass tends to increase with rise in TIT and $r_{p,c}$ for a given ambient condition.
3. The plant performance parameters e.g. plant efficiency and plant specific work increases because of inlet cooling using air-humidifier integrated to a gas turbine plant.
4. The enhancement in efficiency and specific work due to inlet-air cooling is higher at higher ambient temperature.
5. Gas turbine plant efficiency increases upto TIT of 1500 K where after it decreases.
6. The rate of increase in plant efficiency due to reduction in compressor inlet temperature is more pronounced at higher TIT.
7. Optimum plant performance is observed at TIT of 1500 K for all values of drop in temperature ($T_{a,dr}$ = 08 ^oC to 20 ^oC)
8. The compressor pressure ratio corresponding to maximum specific work increases with increase in temperature drop achieved in the humidifier.
9. The design monograms presented above can be used by power plant designer in selecting the optimum operating parameters as per the site condition.

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