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PERFORMANCE EVALUATION OF BASIC AND REHEATED COOLED GAS TURBINE CYCLE: EXERGY ANALYSIS

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Abstract: Present energy scenario depicts a continuous increase in gap between energy demand and energy supply with increasing electricity cost. Owing to increase in fuel prices electricity manufacturing companies have to adopt an energy conversion practice that must have higher performance side with lowest possible investment. Among the various techniques available, gas turbine with complex configuration for enhancing performance is the one. The gas turbine with blade cooling allows manufacturers to go for higher turbine inlet temperature as allowable blade material temperature is 1150K only. Advantage of gas turbine blade cooling over metallurgical constraints that doesn't allow the turbine inlet temperature (TIT) to increase beyond a particular temperature, gas turbine blade cooling has become a revolutionary area of research. Present work deals with the thermodynamic analysis of basic gas turbine (BGT) and reheated gas turbine (RHGT) cycle incorporating air film blade cooling technique.
Keywords: Thermodynamic analysis, Gas turbine cycle, Blade cooling, Air film cooling, reheated gas turbine

1. INTRODUCTION

Gas turbine blade cooling has been a prime area of research since decades. According to International Energy Agency report (Energy Outlook 2008) [1] the Global energy demand has been forecasted to increase continuously by approximately 1.6 % per year during the period 2006 to 2030. The regulation norms for cutting down the greenhouse gases emissions have forced the power producing units to adopt environment friendly (low emission) power producing techniques in the area of gas turbine and combined cycle. Various such techniques have been developed over the years, of which one is gas turbine blade cooling. Gas turbine blade cooling system now allows gas turbines operations at higher blade temperatures i.e. beyond the temperature constrained by metallurgical limits. The blade cooling includes blowing of compressed air from the internal passages of the blades over it. Numerous research articles have been published by authors over gas turbine blade cooling with thermodynamic as well as thermoeconomic analysis [2-11]. Mithilesh Kumar Sahu et al. [2-4] have thermodynamically analysed the cooled gas turbine cycles based on energy and exergy analysis. Anupam Kumari et al. [5] have reported the thermodynamic and emission analysis of basic and intercooled gas turbine cycles and also stated that the cooled gas turbines are more environment friendly compared to uncooled gas turbines and other thermal power plants. Mithilesh Kumar Sahu and Sanjay [6-11] have thermoeconomically analysed the complex cooled gas turbines with film air cooling technique and also reported the effect of film air cooling on various thermodynamic and thermoeconomic performance parameters. A comparative analysis of open and closed loop cooling methods has been carried out by Louis et al. [12] based on a mathematical model, using air and steam as cooling medium. Gas turbine cycle performance is majorly affected by the difference of temperature between turbine inlet temperature and blade temperature.

Chuan and Louis et al. [13] have developed a mathematical model for the determination of coolant mass flow rate. They also carried out a comparative study over the effect of various inlet air cooling systems on the performance of combined cycle. El-Masri [14] developed an Interactive computer code named GASCAN based on the mathematical modelling of various components of gas turbine describing the performance of cooled gas turbine. Brieshet et al. [15] have suggested the possibility of thermal efficiency to be 60%. The author suggested the closed loop steam cooling as the best alternative. A detailed comparative study of advanced combined cycle alternatives with bottoming cycle has been carried out by Bolland [16]. Young and Wilcock [17] described the basic thermodynamics related to an air- cooled gas turbine model. The author suggested that these cycles include varying composition of gas mixtures and hence a realistic modelling of the components must be done. He further advocates the importance of cooling losses while estimating the performance of a cooled cycle. Sanjay et al. [18] have done the thermodynamic modelling and simulation of advanced combined cycle for the assessment of enhanced performance. The result illustrates that increase in the TIT has a positive effect over the overall plant efficiency i.e. efficiency is increased with respect to the reference cycle. Sanjay [19] investigated a parametric study to outline the thermodynamic performance of gas- steam combined cycle. Author further carried out a detailed component–wise exergy analysis. The article highlights that with an increase in TIT and r_{pc} values, the exergy destruction values decreases. Sanjay et al. [20] carried out a detailed energy and exergy analysis of reheat gas- steam combined cycle. He observed the superiority of closed-loop-steam-cooling over air-film cooling. The author further reveals that the reheat gas-steam combined cycle shows an enhance thermal efficiency and plant specific output with closed-loop-steam-cooling in comparison to basic steam-gas combined cycle with air- film cooling.

Alok and Sanjay [21] carried out a detailed study investigating thermodynamic assessment of the performance of gas turbine and combined cycle incorporating inlet cooling techniques. The article focuses the comparative study of impact of two different inlet air cooling methods namely vapour compression and vapour absorption. The author advocates the use of vapour compression cooling method in combined cycle for higher plant performance. A detailed report over thermodynamic performance of combined cycle power plant has been submitted incorporating seven different methods of blade cooling in a technical paper by Sanjay et al. [22] The study further reveals the possibility of highest plant work and efficiency in case of closed loop steam cooling while internal convection cooling has been found as the least effective cooling method.

A cogeneration cycle based on gas turbine has been taken for study by Sanjay et al. [23]. The author investigated the effect of different gas turbine blade cooling on the discussed cycle. The maximum power to heat ratio has been observed for steam cooled internal convection while minimum has been observed for internal convection cooling taking air as the coolant. A. K. Mohapatra and Sanjay [24] studied the parametric study of variation in performance parameter such as TIT, compressor pressure ratio on a cooled gas turbine plant with two inlet air cooling techniques. The authors suggest that integration of two techniques have a positive effect over the plant performance. Anupam Kumari and Sanjay [25] studied the effect of parameters affecting the exergetic and emission performance of basic and intercooled gas turbine cycles. The author advocates that inter-cooled gas turbine cycle should be preferred over basic cooled gas turbine cycle as it delivers higher specific power output and plant efficiency. The emission performance of IcGT is better than basic cooled gas turbine cycle.

The present paper deals with the exergetic performance evaluation of reheated cooled gas turbine cycle with basic cooled gas turbine cycle incorporating air film cooling technique. The compressed air has been taken as the coolant in the present study. Film cooling is the widely accepted cooling technology. In film cooling the surface is covered with a thin film of cool air bled from the compressor. This film separates the blade surface to make a direct contact with hot gases coming out of combustion chamber [26].

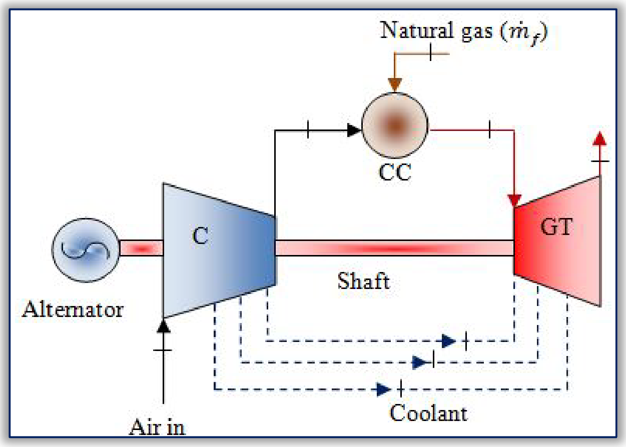

Figure 1. Schematic diagram of basic gas turbine cycle with air film blade cooling

2. SYSTEM CONFIGURATION

The schematic diagram of basic and reheated gas turbine based power plant is shown in Figure 1 and Figure 2. Air enters the compressor, which compresses air to a required pressure level. In BGT compressed air after passing through the CC expands in GT, while in RHGT the burnt flue gases after expansion enters to the second CC where it reheated up to the required turbine inlet temperature. Thereafter the expansion in the comparatively lower pressure gas turbine takes place. During expansion the blade material of the expansion turbine gets exposed to high temperature gases (above 1150 K), which is likely to lead to high temperature oxidation of blade material. This phenomenon has to be controlled to extend the working life of turbine blades. The same is achieved by internally cooling the blades and maintaining its temperature below 1150 K. Coolant air is bled from the compressor at an appropriate pressure level and is allowed to enter the hollow gas turbine blades from the blade root. The coolant air passes through a serpentine path before exiting the blade at its leading edge and creating a film of air as a thermal barrier such as to avoid direct contact of main hot gas with blade material. The coolant air thereafter mixes with the main flow gases for further expansion in subsequent stages. Other aspects of the cycle are similar to conventional gas turbine cycle.

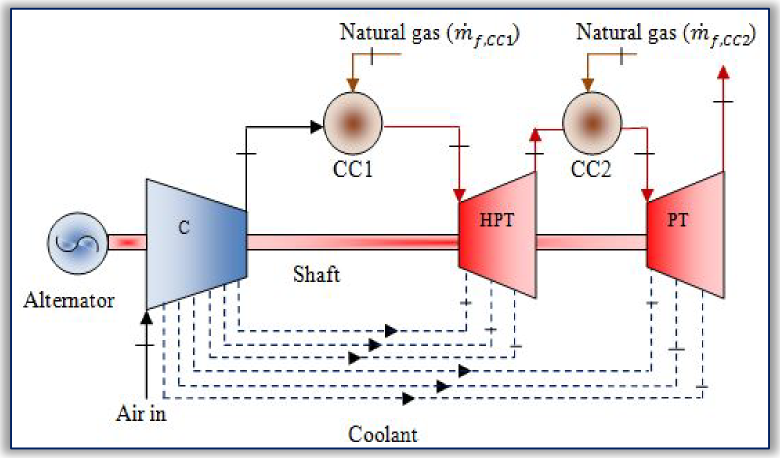

Figure 2. Schematic diagram of reheated gas turbine cycle with air film blade cooling

3. MODELLING AND GOVERNING EQUATIONS

The thermodynamic model has been developed using energy and mass balance equations which are utilised to determine unknown state points in the cycle components.

— Air/gas Model

The specific heat of air has been modelled based on polynomial given by Touloukian and Tadash [27].

$$c_{pa} = 1.023204 - 1.76021 * 10^{-4}T + 4.0205 * 10^{-7}T^2 - 4.87272 * 10^{-11}T^3 \quad (5)$$

The specific heat of flue gas has been modelled based on polynomial given by Touloukian and Tadash [27]

$$c_{pg} = \begin{bmatrix} 15.276826 + 0.01005T - 3.19216 * 10^{-6}T^2 + 3.48619 * 10^{-10}T^3 \\ +x_0(0.104826 + 5.54150 * 10^{-5}T - 1.67585 * 10^{-8}T^2 + 1.18266 * 10^{-12}T^3) \end{bmatrix} /V \quad (6)$$

Thus, the enthalpy, entropy and exergy of the flue gas and air can be calculated as under:

$$h = \int_{T_0}^{T} c_p(T)\, dT \quad (7)$$

$$\varphi = \int_{T_0}^{T} c_p(T) \frac{dT}{T} \quad (8)$$

$$s = \varphi - R\ln(\frac{p}{p_0}) \quad (9)$$

$$E = h - T_0.s \quad (10)$$

— Compressor Model

$$p_2 = p_1 * r_{pc} \quad (11)$$

$$T_2 = T_1\{1 + \frac{1}{\eta_{AC}}[(\frac{p_2}{p_1})^{\frac{\gamma_a - 1}{\gamma_a}} - 1]\} \quad (12)$$

$$\dot{W}_C = \dot{m}_e \cdot h_e + \sum \dot{m}_{cool,j} \cdot h_{cool,j} - \dot{m}_i \cdot h_i \quad (13)$$

— Combustion Chamber Model

$$\dot{m}_3 = \dot{m}_2 + \dot{m}_f \quad (14)$$

$$\dot{m}_f \cdot LHV \cdot \eta_{cc} = \dot{m}_3 \cdot h_3 - \dot{m}_2 \cdot h_2 \quad (15)$$

Mass of fuel required has been calculated using mass and energy balance of the combustion chamber:

$$\dot{m}_f = \frac{[\dot{m}_2 \cdot h_3 - \dot{m}_2 \cdot h_2]}{[\eta_{cc} \cdot LHV - h_3]} \quad (16)$$

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \quad (17)$$

$$p_3 = p_2(1 - \Delta p_{cc}) \quad (18)$$

where $\Delta p_{cc} = 0.02$

— Cooled Gas Turbine Model

Air Film cooling: A simple model of film cooling is shown in Figure 3. The hot gases ($\dot{m}_g$) passes over the bladesurface, while the coolant ($\dot{m}_{cool}$) passing internally through the blade channels is ejected out from the leading edge which forms a film over the blade surface and finally mixes with hot gas at the trailing edge. The film so formed reduces heat transfer from hot gas to the blades.

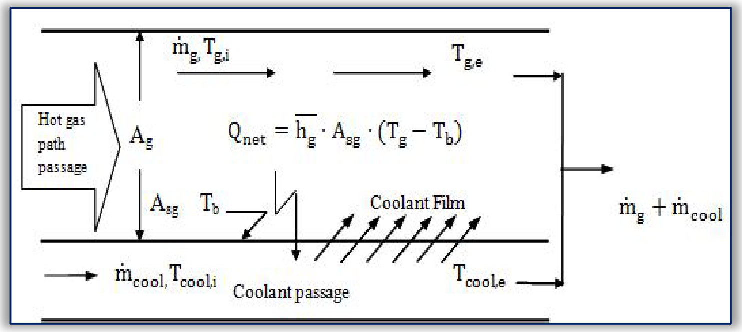

Figure 3. Model of air film cooling of turbine blade

The mass flow rate of coolant air bled (ζ) is given by:

$$\zeta = \frac{\dot{m}_{cool}}{\dot{m}_g} = \left(1 - \eta_{iso,air}\right)\frac{St_i \cdot S_g}{\varepsilon_{cool} \cdot t.\cos\alpha} * \frac{c_{pg}(T_{g,i} - T_b)}{c_{p,cool}(T_b - T_{cool,i})} * F_{sa} \quad (19)$$

Pressure at the inlet of gas turbine is given as:

$$p_3 = p_2(1 - \Delta p_{cc}) \quad (20)$$

The temperature and pressure of gas turbine exhaust stream and power output is given by the relation:

$$T_4 = T_3\{1 - \eta_{GT}[1 - (\frac{p_3}{p_4})^{\frac{1-\gamma_g}{\gamma_g}}]\} \quad (21)$$

where $p_4 = 1.08$ bar

$$\dot{W}_{GT} = \left(\dot{m}_i h_i + \sum \dot{m}_{cool,j} h_{cool,j} - \dot{m}_e h_e\right) * \eta_{mech} \quad (22)$$

$$\dot{W}_{net} = \dot{W}_{GT} - \dot{W}_C \quad (23)$$

4. RESULT AND DISCUSSION

Based on above detailed modelling, a computer code has been developed in MATLAB [28] to obtain the performance data for the basic and reheated cooled gas turbine cycle incorporating air film blade cooling technique. The operating conditions for the cycle have been detailed in Table 1. The results obtained have been discussed systematically with the help of various illustrating graphs.

Table 1. Operating and design parameters for analysis of proposed cycle [7, 9, 18 and 20]

Parameter	Symbol	Adopted value	Unit
Gas Properties	$c_p = f(T)$		kJ/kgK
	$h = \int_{T_0}^{T} c_p(T)\, dT$		kJ/kg
Compressor	Isentropic efficiency (η_{AC})	88	%
	Mechanical efficiency (η_{mech})	98.5	%
	Compressor pressure ratio	20-30	-
Combustion Chamber	CC efficiency (η_{CC})	99.5	%
	Pressure loss (p_{loss})	2.0% of p_{entry}	bar
	Lower heating value (LHV)	42.0	MJ/kg
	Fuel line pressure	1.5*p_{cc}	bar
	Isentropic efficiency (η_{GT})	90	%
	Exhaust pressure	1.08	bar
	Turbine blade temperature (T_b)	1150	K
	Turbine inlet temperature (TIT)	1500-1800	K

Figure 4 depicts the variation of plant specific work output of basic gas turbine cycle with coolant mass fraction at different TIT and r_{pc}. It can be clearly seen that coolant mass flow rate increases, as the compressor pressure ratio increases at a fixed TIT value while plant specific work output decreases at the same parameters. The increase in coolant requirement is driven by the fact that as we move on for higher r_{pc} it also results increase in temperature of coolant air available at compressor bleed points and of course higher r_{pc} needs more compressor work which ultimately results decrease in plant specific work. Coolant mass fraction and plant specific work both shows an increasing trend for fixed r_{pc} value with increasing TIT values. The rising tendencies of curve for both parameters are in line with the concept of heat transfer (for higher temperature more amount of coolant required) and thermodynamics (energy at higher temperature has more potential).

Figure 5 represents a variation of coolant mass fraction with plant specific output of reheated cooled gas turbine with varying TIT and r_{pc} values. The graph predicts the similar trend with the previous literatures as the values of TIT is increased from 1500K to 1800K keeping r_{pc} value constant, the coolant mass fraction value increases because higher TIT causes more amount of coolant to be bled from compressor for the cooling of stages while the plant specific work output initially increases sharply and again starts decreasing after a certain TIT value. The diagram further describes that as r_{pc} values are increased along a fixed TIT value the coolant mass fraction ratio shows slight increment in the coolant mass fraction up to an r_{pc} while again it shows decline behaviour. It occurs because of increase in r_{pc} value causes more number of turbine stages to be cooled. The diagram describes that on increasing the r_{pc} value along a fixed TIT values the plant specific work output increases for RHGT cycle.

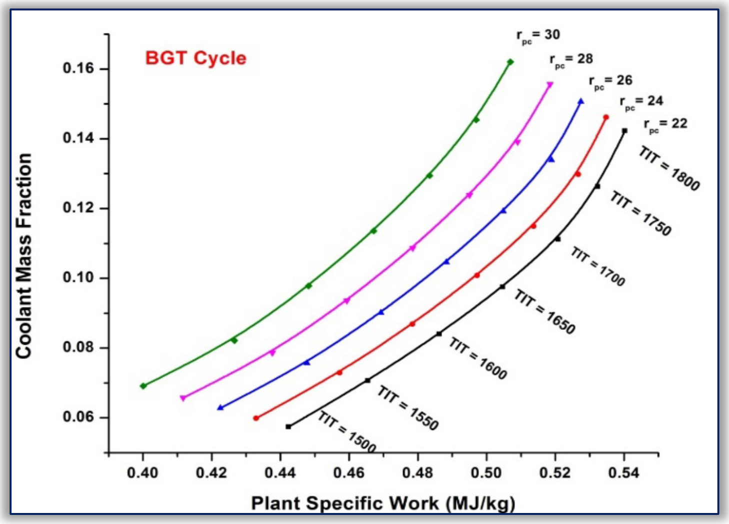

Figure 4. Effect of operating parameters on coolant mass fraction and plant specific work for BGT cycle

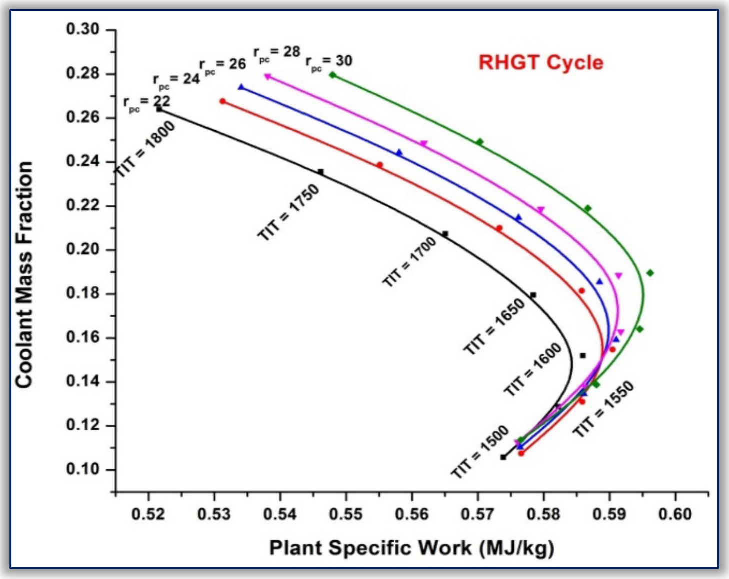

Figure 5. Effect of operating parameters on coolant mass fraction and plant specific work for RHGT cycle

In Figure 6 behaviour of coolant mass fraction with second law plant efficiency for the basic cooled gas turbine cycle is depicted. It is observed from Figure 6 that as TIT values increase from 1500K to 1800K at the same r_{pc}, the curve of coolant mass fraction shows an increase trend while the exergy efficiency of the cycle decreases at the same parameters. The graph further describes the behaviour of increasing r_{pc} values over fixed TIT values which further shows an increase in both coolant mass fraction as well as exergy efficiency due to the reasons explained earlier.

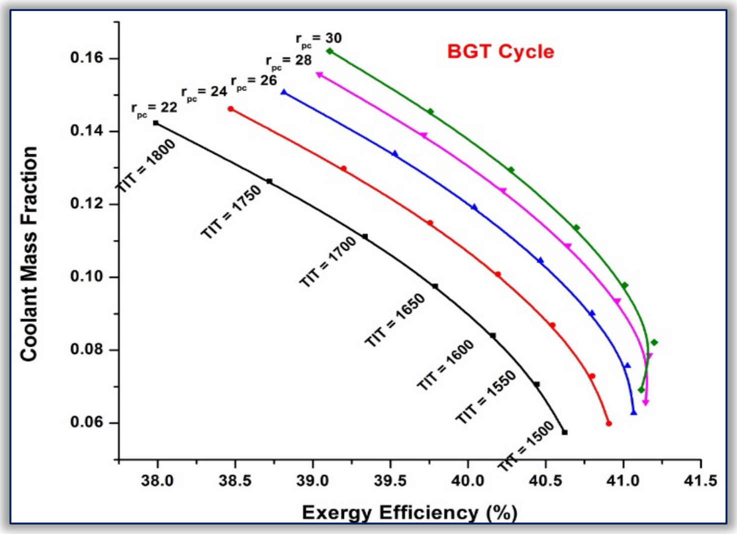

Figure 6. Effect of operating parameters on coolant mass fraction and exergy efficiency for BGT cycle

Figure 7 describes the performance behaviour of coolant mass fraction of reheated cooled gas turbine cycle with second law efficiency (exergy efficiency). It shows that the coolant mass fraction and exergy efficiency both shows increasing trend with increase in r_{pc} while keeping a fixed value of TIT. It is due to the fact that higher r_{pc} results increase in coolant air temperature as well as it also results saving in mass flow rate of fuel required for combustion. An increase in TIT value at a fixed r_{pc} shows an increase in coolant mass fraction (higher temperature as well as more number of stages required cooling) while the value of exergy efficiency decreases on increasing the TIT (increase in fuel mass flow rate).

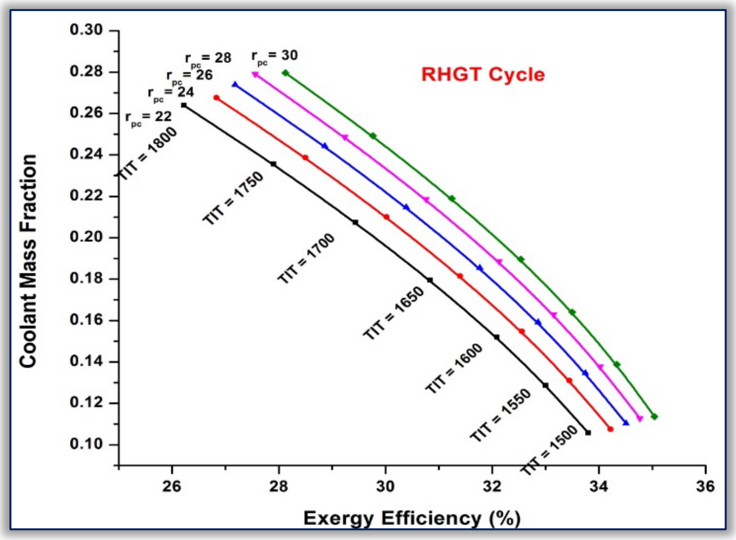

Figure 7. Effect of operating parameters on coolant mass fraction and exergy efficiency for RHGT cycle

In Figure 8, variation of coolant mass fraction with the TIT values for both basic and reheat cooled gas turbine cycle at a fixed value of r_{pc}=30 has been illustrated. The values of coolant mass fraction is increased along with the increase in TIT in both basic and reheat cooled gas turbine cycle because higher TIT causes an increase in coolant mass fraction for more number of stages to get cooled. The coolant requirement for the both basic and reheat gas turbine cycle increases with increase in TIT but the trend followed in basic cooled gas turbine is little gradual while for cooled reheated gas turbine it follows a little sharp trend.

Figures 9 explain the behaviour of exergetic performance and specific work output of basic and reheat cooled gas turbine cycle. The column chart clearly indicates that exergy efficiency of basic gas turbine is better as compared to reheat cycle for same operating parameters. The combustion chamber is the main source of exergy destruction and as in reheat cycle fuel required is also higher; these are the reasons for lesser exergy efficiency in case of reheat cycle. The column chart also

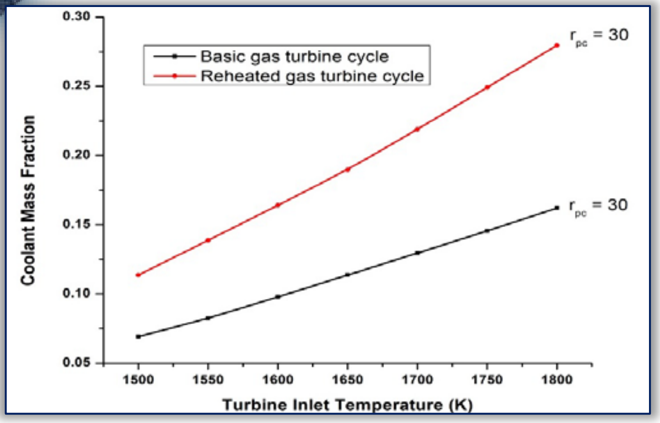

Figure 8. Comparative analysis of BGT and RHGT for coolant mass fraction with TIT variation (fixed r_{pc})

5. CONCLUSIONS

A systematic component modelling followed by exergy analysis of proposed basic and reheat gas turbine based power plants incorporating air film blade cooling has been carried out. Based on the analysis of the results discussed in previous section following conclusions have been drawn:

— The coolant mass flow requirement for both the BGT and RHGT increases with increase in TIT by keeping the compressor pressure ratio constant.

— The specific work output of the BGT increases on increasing the TIT at a fixed r_{pc} while for the RHGT specific plant work initially increases sharply on increase in TIT and again it starts decreasing after a certain TIT.

— The exergy efficiency curve of both the cycle shows falling nature with an increase in TIT at a fixed compressor pressure ratio.

— Exergy efficiency of the basic cooled gas turbine cycle is better as compared to the reheat cycle for the case TIT = 1800K & r_{pc} = 30.

— Plant specific output is find to be more in case of reheat cycle as compared to basic cooled gas turbine for TIT = 1800K & r_{pc} = 30, as in reheat cycle the expansion takes place in two steps i.e. first in high pressure turbine and again in low pressure turbine.

compares the plant specific work output of both the cycles which is greater in case of RHGT cycle. In reheat cycle expansion takes in two stages of turbine i.e. in high pressure turbine and again in low pressure turbine which results higher plant specific work output compare to BGT cycle for same operating parameters.

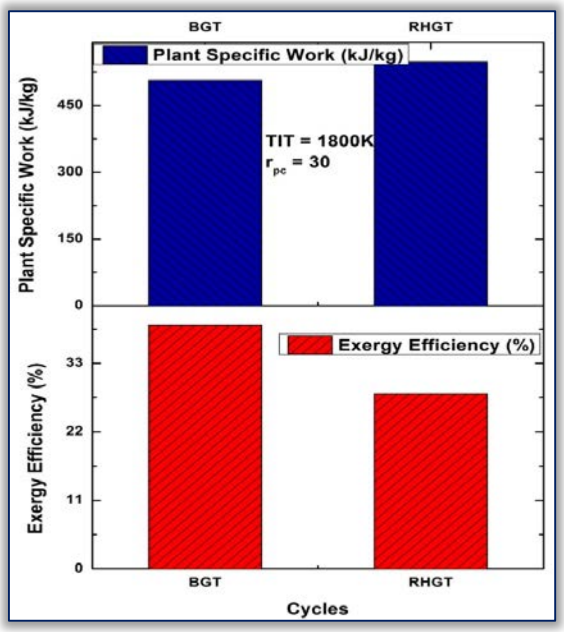

Figure 9. Comparative analysis of BGT and RHGT cycle on plant specific work and exergy efficiency

As thermodynamic analysis of air film blade cooled gas turbine is closer to the actual gas turbine based power plant in comparison to thermodynamic analysis of similar cycles reported earlier as blade cooling had not been considered. Power plant designers adopting proposed model are more likely to come up with results closer to the real life power plant cycles than otherwise. Hence, this work is a step in that direction and is likely to lead to further research using proposed modelling of cooled gas turbines.

Nomenclature

c_p	Specific heat at constant pressure (kJ/kgK)	s	Specific entropy (kJ/kgK)
ex	Specific exergy of the stream (kJ/kg)	S	Entropy (kJ/K)
h	Specific enthalpy of the stream (kJ/kg)	S_{gen}	Entropy generation (kJ/K)
$\dot{m}$	Mass flow rate (kg/s)	T	Temperature (K)
p	Pressure (bar)	T_0	Reference or ambient temperature (K)
p_0	Reference or ambient pressure (kpa)	W	Work (kW)

Subscripts

a	Air	f	Fuel
c	Compressor	Gen	Generation
comb	Combustion chamber	G	Gas
in	Inlet	a	Air
Out	Outlet	c	Compressor

Acronyms

BGT	Brayton gas turbine cycle	GT	Gas turbine
C	Compressor	RHGT	Reheated gas turbine cycle
CC	Combustion chamber	TIT	Turbine inlet temperature

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