PERFORMANCE ANALYSIS OF REHEATED GAS TURBINE BASED POWER PLANT CYCLE

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ABSTRACT

The present work deals with the thermodynamic analysis of reheat gas turbine based power plant cycle. The work is carried out by performing energy and exergy analysis of each components of the cycle. Energy, exergy and mass balance approach had been taken for mathematical modeling of entire cycle. Energy analysis provides thermal efficiency of cycle while exergy analysis gives the actual value of cycle efficiency or rational efficiency of the cycle. Performing exergy analysis gives information about the irreversibility present in cycle in terms of its location and magnitude. Exergy analysis provide the true performance of the cycle as it inform that how much potential is utilized and how much is wasted in terms of net-work and exergy destruction respectively. Aforesaid cycle is investigated for different operating parameters to see the effect of this on performance parameters. The cycle results for turbine inlet temperature 1800K and compressor pressure ratio 30 is being net-work is 1.026 MJ/kg and exergy efficiency is 41.38 %.

Keywords: Reheat cycle, exergy analysis, irreversibility, gas turbine, Turbine inlet temperature, compressor pressure ratio.

I. INTRODUCTION

The power plants in view of gas turbines are most encouraging and proficient wellspring of power generation. Gas turbine is a promising energy conversion system which utilizes natural gas as its fuel. The lower emission rate and clean energy makes it advantageous over other such energy conversion systems. Today's energy scenario requires such energy conversion system which satisfies the need of power and come-up with energy security alongside lower emission rate [1]. Exergy examination has been utilized as a valuable tool to decide component-wise thermodynamic inefficiencies of the presented cycles. In this field H. Chandra et al. [2] have reported thermodynamic analysis of simple closed baryon cycle. In order to achieve higher performance from the gas turbine cycle the turbine inlet temperature must be high enough and that required cooling of turbine blades. Sanjay et al. [3] have reported aftereffects of parametric examination taking into account energy and exergy investigation of reheat-gas-steam combined cycle and have reported that by utilizing closed loop steam cooling, the plant thermal efficiency can be enhanced up to 62%. J. B. Young et al. [4] and R. C. Wilcock et al.

[5] have examined the impact of the blade cooling on the cycle performance of gas turbine. J.H. Horlock et al. [6] have detailed a methodology for calculating the amount of coolant requirement. Sanjay et al. [7] have examined the impact of seven distinct methods of blade cooling on thermodynamic performance of power plant cycle. With reference to above literature exergy analysis of complex gas turbine cycle is as of now area of interest for researchers. The present paper is about energy and exergy examination of one such complex cycle, reheat gas turbine based power plant cycle.

II. SYSTEM CONFIGURATION

The schematic diagram of reheat gas turbine based power plant is shown in Fig.1. Air is taken from the environment at surrounding condition which goes through the compressor, which compresses air to a required pressure level. From that point these air goes through combustion chamber (CC1). In CC1 natural gas is burnt in the presence of these airs to increase its temperature and energy level. Thereafter it expands in expansion stages of higher pressure turbine (HPT). As this is a reheat cycle the exit of turbine is again send to the another combustion chamber (CC2) where its temperature and energy level is increased by burning the fuel. Again these hot gases are expanded in lower pressure turbine (LPT).Table 3shows the stream report of cycle including mass flow rate, pressure, temperature, enthalpy and exergy values.

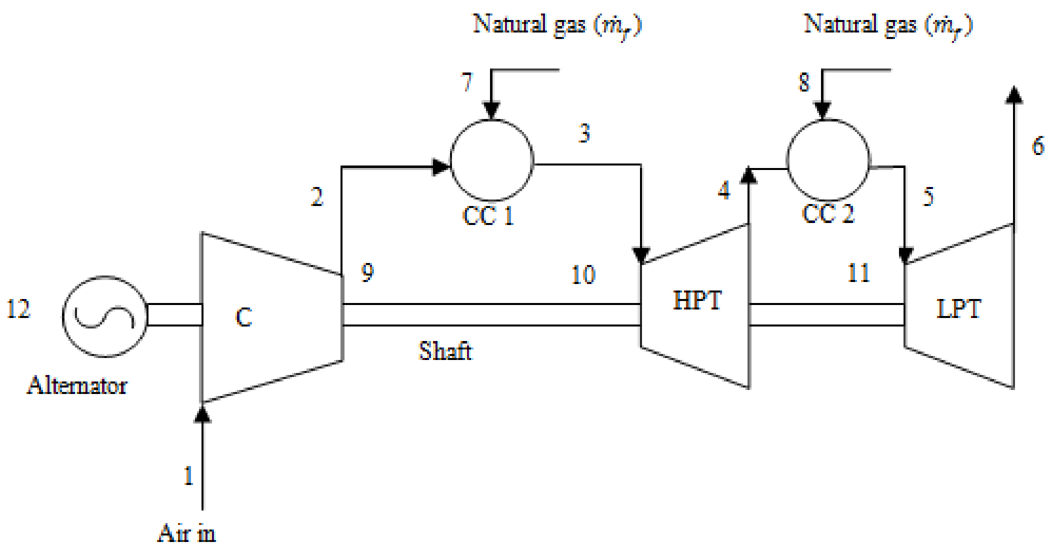

Figure 1 Schematic diagram of reheat gas turbine based power plant cycle.

III. Thermodynamic Model

Air has been adopted as a working fluid for the cycle. The thermodynamic properties of air take account of enthalpy, entropy, exergy and c_p and these all are modeled as under:

The specific heat of air is based on polynomial given by Touloukian and Tadash [22]

$$c_{pa} = 1.023204 - 1.76021 * 10^{-4}T + 4.0205 * 10^{-7}T^2 - 4.87272 * 10^{-11}T^3 \quad (1)$$

The specific heat of combustion gases has been modelled based on polynomial [22]

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

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

$$h = \int_{T_0}^{T} c_p(T)\, dT \quad (3)$$

$$\theta = \int_{T_0}^{T} c_p(T) \frac{dT}{T} \quad (4)$$

$$s = \theta - R\ln(\frac{p}{p_0}) \quad (5)$$

$$E = h - T_0.s = h - T_0.\theta + RT_0 \ln(p - p_0) \quad (6)$$

IV. RESULT AND DISCUSSION

The results have been simulated using MATLAB program which is based on modeling presented and using operating parameters detailed in table 1.

Table 1 Operating performance parameters of cycle.

Parameters	Symbol	Units
Gas properties	$c_p = f(T)$	kJ kg^{-1} k^{-1}
	$Enthalpy = \int c_p(T)dT$	kJ kg^{-1}
Compressor	Isentropic efficiency = 88.0	%
	Mechanical efficiency = 98.5	%
Combustor	Combustor efficiency = 99.5	%
Gas turbine	Isentropic efficiency = 90.0	%
	Exhaust pressure = 1.08	Bar
Mass flow rate of air	1	kg/s

Table 2 Stream report of the cycle

Stream	Mass flow rate (kg/s)	Temperature(K)	Pressure(bar)	Enthalpy(kJ/kg)	Exergy(MW)
1	1	288	1.0132	0	0
2	1	866.4169	30.396	644.4394	0.602843
3	1.035517	1800	29.78808	2055.68	1.736528
4	1.035517	1246.137	5.67196	1233.199	0.867553
5	1.056952	1800	5.55852	2055.68	1.604463
6	1.056952	1251.69	1.08	1241.329	0.726313
7	0.035517	288	44.68212	42000	1.546904
8	0.021435	288	8.337781	42000	0.933567
9	-	-	-	-	0.644439
10	-	-	-	-	0.838918
11	-	-	-	-	0.847819
12	-	-	-	-	1.026663

Table 3 Exergy analysis of cycle

Component	Exergy destruction (MW)	Exergy loss (MW)	Exergy Efficiency (%)
C	0.041596	0	93.54533
CC 1	0.382281	0.030938	80.77825
HPT	0.030056	0	96.54117
CC 2	0.177987	0.018671	89.08135
LPT	0.030331	0.726313	96.54606
System	0.662251	0.775923	41.38983

Table 2 is the representation of streams to and from direction with essential thermodynamic properties of same.

Table 3 is the exergy analysis summary of the cycle having operating parameters as TIT = 1800K and $r_{pc} = 30$.

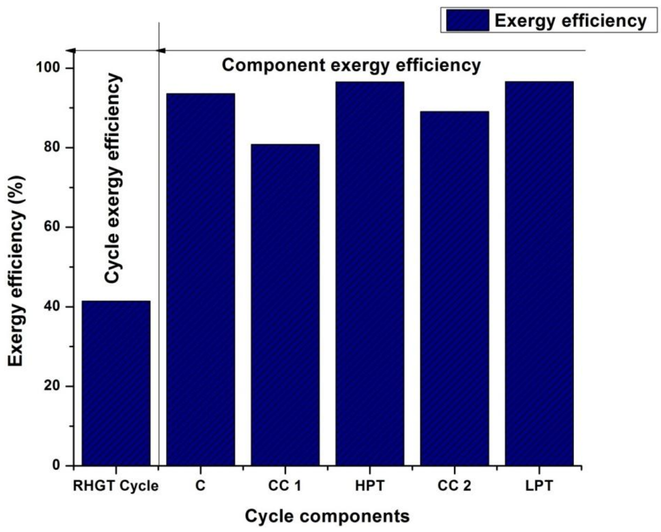

Figure 2 Exergy efficiency of cycle and component-wise exergy efficiency.

Figure 2 shows the cycle exergy efficiency and component-wise exergy efficiency of the cycle with TIT = 1800K and r_{pc} = 30. In figure it is showing that cycle exhibits exergy efficiency around 40% and in the component combustion chamber shows comparatively lower exergy efficiency from other cycle components.

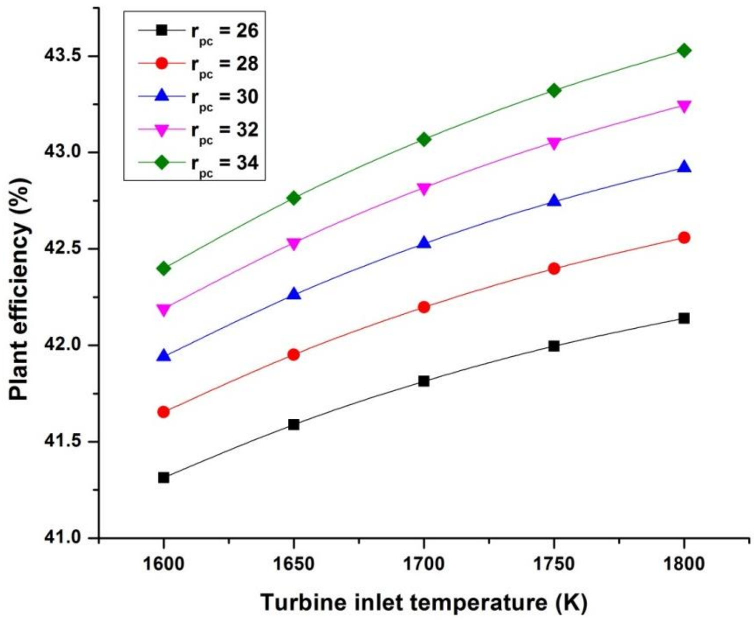

Figure 3 Variation of plant efficiency with TIT

Figure 3 is the graph between turbine inlet temperature and plant efficiency. It can be concluded from figure as TIT increases the plant efficiency also gets increased for constant compressor pressure ratio as TIT provides higher net-work comparatively. From figure it is also clear that with increase in compressor pressure ratio plant efficiency shows increasing trend.

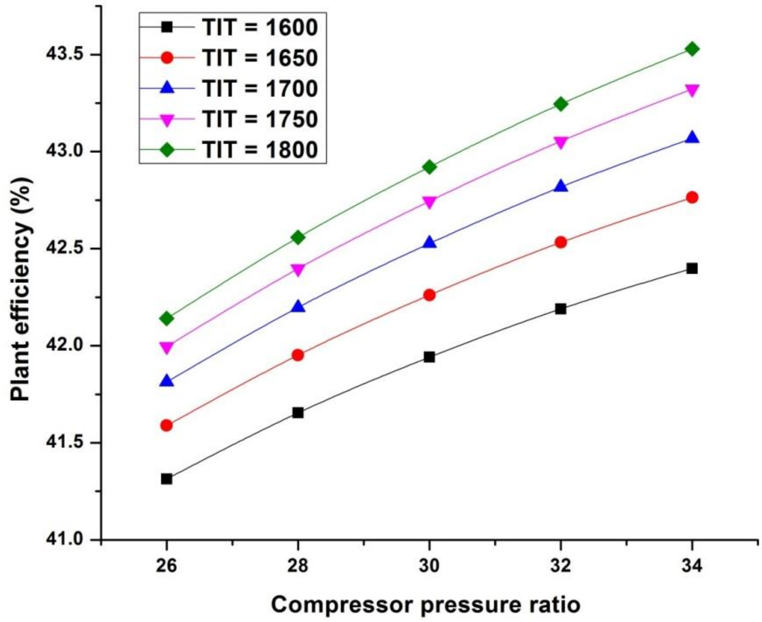

Figure 4 Compressor pressure ratio Vs plant efficiency

Figure 4 is the graph between plant efficiency and compressor pressure ratio. Figure shows that as r_{pc} increases in magnitude the curve of plant efficiency shows increasing trend for fixed TIT. As r_{pc} increases two conclusions can be drawn first it results in increased compressor work and second it results in decrease of mass flow rate of fuel. The increased plant efficiency shows dominating effect of lower fuel requirement. By analyzing the cycle it can be state that increasing TIT will also help to increases the plant efficiency for fixed compressor pressure ratio.

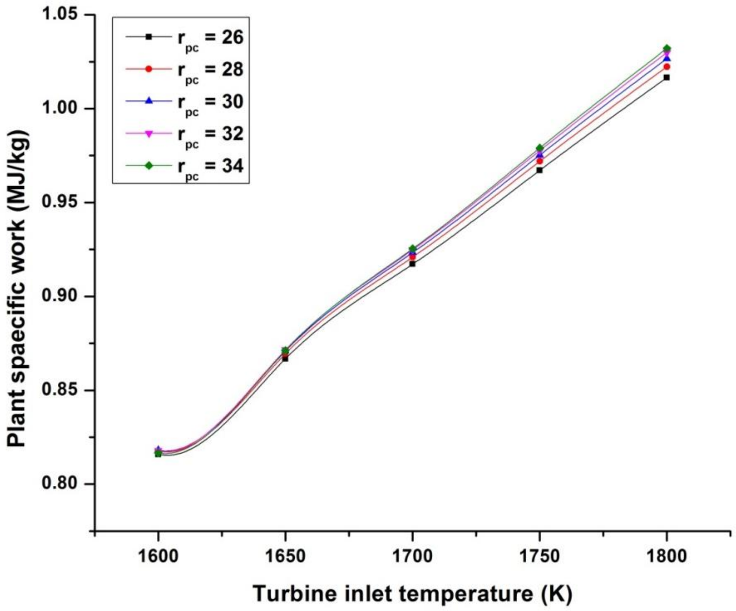

Figure 5 Variation of plant specific work with TIT

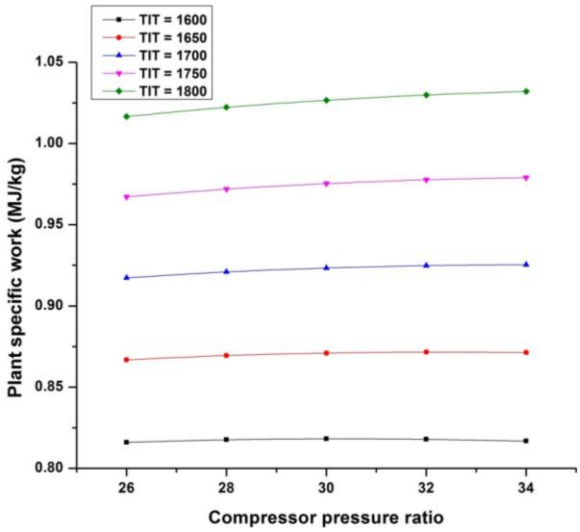

Figure 6 Graph between plant specific work and compressor pressure ratio.

Fig. 5 is the graphical representation of the plant specific work variation with the turbine inlet temperature. Figure shows that for constant r_{pc} the plant specific work considerably increases with increased value of TIT. The reason behind this behavior of curve is higher TIT provides more expansion work comparatively and due to reheat cycle we observe rapid change in plan specific work. From fig it can also be seen that change in r_{pc} for constant TIT doesn't make rapid change in graph.

Fig. 6 is variation of plant specific work with the compressors pressure ratio. Figure shows that for varying compressor pressure ratio at lower TIT the slope of plant specific work is almost zero but for higher TIT the curve shows increasing trend for increasing values of r_{pc}. This happens because as we go for higher r_{pc} the net-work decreases but as this is a reheat cycle and cycle obtained its work from two expansion turbines so the net result shows slight increase in plant specific work. For higher TIT the increasing trend of plant specific work can be easily seen.

V. CONCLUSION

Based on the comprehensive thermodynamic analysis of reheat gas turbine based power plant cycle, the following conclusions have been drawn:

- The component level thermodynamic analysis suggests that losses arise due to irreversibility's within the components of the cycle.
- The energy efficiency for reheat gas turbine cycle has been observed as (42.92%).
- The exergy efficiency for reheat gas turbine cycle was found to be (41.38%).
- The reheat gas turbine cycle plant specific work was observed as 1.026MJ/kg while the trend of plant specific work is of increasing nature for increasing TIT.
- Plant efficiency graphs suggest its increasing behaviour with increasing value of TIT and r_{pc}.

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