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Exergy Analysis of Gas-Steam Combined Cycle Power Plant

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ABSTRACT
The present work deals with Exergy analysis of 415 MW natural gas based Gas-steam combined cycle power plant at NTPC, Anta. A parametric study has been carried out and the effect of thermodynamic operating variables on the performance of combined cycle plant was studied. The operating variables considered in the present work are: inlet air temperature to compressor, turbine inlet temperature TIT, and pressure ratio of compressor. The results obtained show that rate of Exergy destruction in combustor is much more than the rate of Exergy destruction in other components of the plant and the condenser has minimum Exergy destruction in the plant. For an increase in inlet air temperature from 273 K to 313 K, second law efficiency decreases by 1.5% and the Exergy destruction rate increases by 1.25%. Similarly for an increase in TIT from 1273 K to 1473 K second law efficiency increases by 10.7% and the Exergy destruction rate decreases by 8.5%. The Exergy destruction rate in compressor, combustor, HRSG, and steam turbine increases with increasing TIT and in gas turbine and exhaust it decreases with increasing TIT. With an increases in pressure ratio from 8 to 12 the second law Efficiency of the combined cycle increases by 1.12% and increases by 0.32% for an increase in pressure ratio from 8 to 16 at TIT 1100^{0}C and T_0 = 27^{0}C. Exergy destruction rate of combined cycle decreases by 0.99% for an increase in pressure ratio from 8 to12 and it decreases by 0.3% for an increase in pressure ratio from 8 to 16 for same TIT and T_0. The rate of Exergy destruction in compressor, gas turbine, and exhaust gases increases with increasing pressure ratio and in combustor, HRSG, and in steam turbine it decreases with increasing pressure ratio.

Keywords:- gas turbine, steam turbine, combined cycle, ambient air temperature, turbine inlet temperature.

INTRODUCTION

The increased awareness that the world's Exergy resources are limited has caused many countries to reexamine their Exergy policies and take drastic measures in eliminating waste. It has also sparked interest in the scientific community to take a closer look on the Exergy conversion devices and to develop new techniques to better utilize the existing limited resources. It is generally said in everyday conversation and also in scientific discussion that the Exergy is consumed but this claim however conflicts with the first law of thermodynamic which states that the total amount of Exergy conserved even though form of Exergy may change from one to another.

Expressions as “Exergy consumption”, “Exergy saving” and even “Exergy conservation”, when used refer to “Exergy” as intense Exergy available from fossil fuels or from condensed uranium. But, it is confusing to use one of the most well established scientific terms, Exergy, to mean “to be conserved” and to be consumed simultaneously. That is why the thermodynamic concept, Exergy, is used to articulate what is consumed. There are two essential tools for thermodynamic design and optimization of thermal systems such Exergy analysis and Exergy analysis (referring to the first and second law of thermodynamic analysis)

The first law of thermodynamics deals with the quantity of Exergy and asserts that Exergy cannot be created or destroyed. The law acts merely as a necessary tool for the book keeping of Exergy during a process and offers no challenges for the engineer. The second law, however, deals with the quality of Exergy. More specifically, it is concerned with the degradation of Exergy during a process, the entropy generation and it offers plenty of room for improvement.

Recently, Exergy analysis has become a key aspect in providing a better understanding of the process, to quantify sources of inefficiency, to distinguish quality of Exergy (of heat). Exergy is defined as the maximum theoretical useful work obtained as a result of interaction of system with a reference environment. Exergy is only conserved for a reversible process, but it is partly consumed in an irreversible process. Thus, Exergy is never in balance for real processes. For a real process, the Exergy input always exceeds the Exergy output; this imbalance is due to the Exergy destruction. Exergy destruction is a measure of irreversibility that is the source of performance. Therefore, an Exergy analysis assessing the magnitude of Exergy destruction identifies the location, magnitude and the sources of thermodynamic inefficiencies in a thermal system. This provides useful information for the improving the overall efficiency and cost effectiveness of a system and comparing the performance of two systems.

Exergy is generally not conserved as Exergy but some quantity of Exergy is destroyed in the system. Exergy destruction is the measure of irreversibility that is source of performance loss. Therefore, an Exergy analysis assessing the magnitude of Exergy destruction identifies the location, magnitude and source of thermodynamic inefficiencies in a thermal system. Exergy analysis usually predicts the thermodynamic performance of an Exergy system and the efficiency of the system components by accurately quantifying the entropy generation within components.

II THERMODYNAMIC MODELING

The schematic diagram of dual pressure combined cycle under consideration is shown in figure.1. In this arrangement a simple gas turbine cycle is used as topping cycle and a steam cycle is used as bottoming cycle. The waste heat of gas turbine exhaust which is at high temperature is recovered in a dual pressure heat recovery steam generator.

Air from atmosphere is compressed in a compressor to a higher pressure. The fuel is injected into combustor and is burnt with compressed air at constant pressure. The hot combustion products from the combustor are allowed to expand in a gas turbine to produce useful work. The heat rejected by gas turbine is highly appreciable as the exhaust temperature is very high. This waste heat is utilized in a dual pressure heat recovery steam generator to produce steam which is expanded in steam turbine working on Rankin cycle to produce additional power.

The steam expanded up to condenser pressure in LP steam turbine is condensed to saturated water state in condenser and pumped to deareator through condensate extraction pump. In deareator dissolved gases are removed by using the steam bled from low pressure turbine resulting in saturated feed water at deareator pressure. The hot saturated water from deareator is pumped to the economizer section of HRSG through LP and HP boiler feed pumps respectively and is heated to the saturation temperature corresponding to their pressures. The saturated water is converted into steam in the LP and HP evaporators and further the steam is superheated in the corresponding superheaters. The superheated steam from HP super heater is expanded in a high pressure steam turbine and is mixed with superheated steam supplied from LP super heater. The enthalpy of mixed steam is obtained by applying the adiabatic mixing process. The mixed steam is then expanded in low pressure steam turbine and appropriate part of steam is bled for deareation process at required pressure.

The temperature entropy diagram for the combined cycle is shown in figure 2.

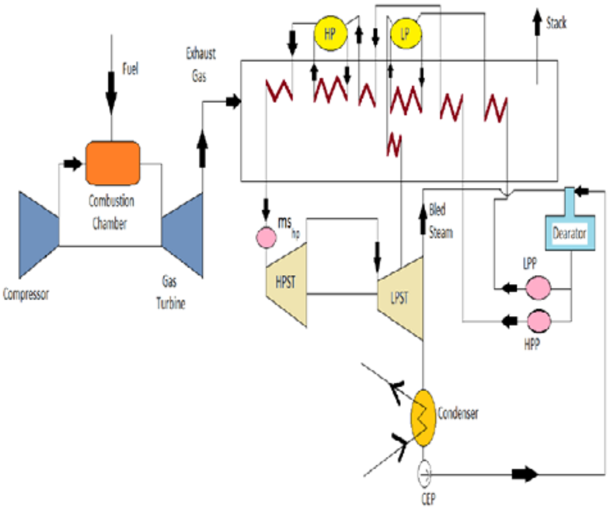

Figure 1: Schematic diagram of combined cycle power plant.

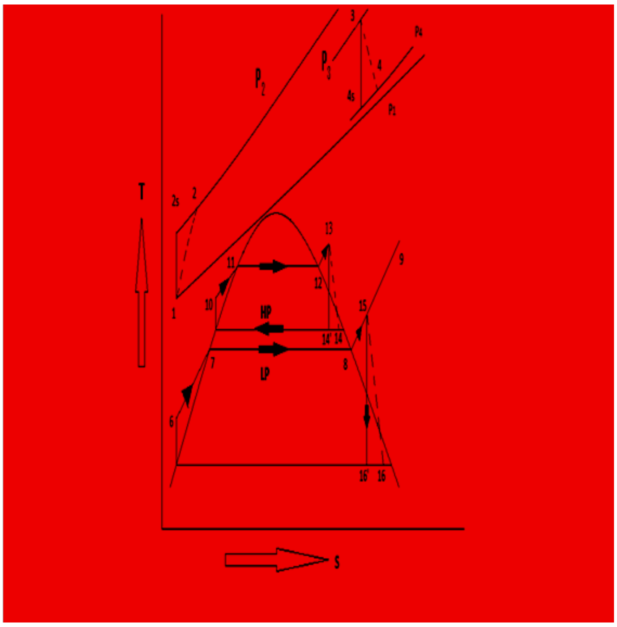

Figure 2: Temperature – Entropy diagram of combined cycle power plant.

2.1GAS TURBINE CYCLE MODEL

It is assumed that the compressor efficiency and the turbine efficiency are represented by η_c and η_t respectively. The ideal and actual processes are represented by full and dashed lines on the temperature entropy diagram as shown in figure 2.

Air compressor model:

The compression ratio R_P of the compressor can be defined as

$$r_p = \frac{P_2}{P_1} \quad \text{--------------------(1)}$$

Where P_1 and P_2 are compressor inlet and outlet pressures respectively

The isentropic efficiency of compressor is expressed as

$$\eta_c = \frac{T_{2s} - T_1}{T_2 - T_1} \quad \text{-------------- (2)}$$

Where

T_1 = compressor inlet temperature

T_{2S} = compressor isentropic outlet temperature

T_2 = compressor actual discharge temperature

The actual discharge temperature of the compressor is calculated from eq 3

$$T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_C} \quad \text{-------------- (3)}$$

The work of the compressor (W_C) can be calculated as

$$W_C = \dot{m}_a \times c_{pa} \times (T_2 - T_1) \quad \text{------- (4)}$$

Where C_{pa} is the specific heat of air which can be fitted by eq 5 for the range of 200K< T<800K.

$$C_{pa} = 1.0189 \times 10^3 - 0.13784T_a + 1.9843\times 10^{-4}T_a^2 + 4.2399\times 10^{-7}T_a^3 - 3.7632 \times 10^{-10}T_a^4 \quad (5)$$

Irreversibility of compressor can be obtained by

$$I_{comp} = m_a T_0 (s_2 - s_1)$$

$$I_{Comp} = m_a\ T_0\ [C_{pa} \log_e (T_2 / T_1) - R \log_e (p_2 / p_1)] \quad \text{....... (6)}$$

Combustion Chamber Model: From the Exergy balance in the combustion chamber

$$\dot{m}_a\ C_{pa}\ T_2 + \dot{m}_f \times LHV + \dot{m}_f\ C_{pf}\ T_f = (\dot{m}_a + \dot{m}_f)\ C_{pg} \times TIT \quad \text{............... .... (7)}$$

Where $\dot{m}_f$ is the fuel mass flow rate (kg/s), $\dot{m}_a$ is the air mass flow rate(kg/s), LHV is the lower heat value , TIT = T_3 is turbine inlet temperature, c_{pf} is the specific heat of fuel and T_f is the temperature of the fuel, and c_{pg} is the specific heat of flue gases.

The specific heat of flue gases c_{pg} is given by eq 7

$$C_{pg} = 1.8083 - 2.3127 \times 10^{-3}\ T + 4.045 \times 10^{-6}\ T^2 - 1.7363 \times 10^{-9}\ T^3 \quad \text{................ (8)}$$

The fuel air ratio (f) is expressed as Eq 8

$$f = \frac{\dot{m}_f}{\dot{m}_a} = \frac{C_{pg} \times TIT - C_{pa} \times T_2}{LHV - C_{pg} \times TIT} \quad \ldots\ldots. (9)$$

Irreversibility in the combustion chamber is calculated as

$I_{comb} = T_0\,[(s_p)_3 - (s_r)_2]$ KJ/kg

Where $(s_r)_2 = (s_a)_2 + (s_f)_2$

Here the subscripts p, r a, and f stand for products, reactants, air, and fuel respectively.

Thus $I_{comb} = T_0\,[(s_p)_3 - (s_a)_2 - (s_f)_2]$

$= T_0\,\{[(s_p)_3 - (s_p)_0] + (s_p)_0 - [(s_a)_2 - (s_a)_0] - (s_a)_0 - [(s_f)_2 - (s_f)_0] - (s_f)_0\}$

But $(s_p)_0 - [(s_f)_0 + (s_a)_0] = (\Delta s)_0$

$I_{comb} = T_0\,\{[(s_p)_3 - (s_p)_0] - [(s_a)_2 - (s_a)_0] + (\Delta s)_0\}$

$I_{comb} = T_0\,\{[C_{pg} \log_e (T_3/T_0) - R_g \log_e (p_3 / p_0)] - [C_{pa} \log_e (T_2 / T_0) - R_a \log_e (p_2 / p_0)] + (\Delta s)_0\}$ ………(10)

Where $T_0 (\Delta S)_0 = (\Delta G)_0 - (\Delta H)_0$

$= m_f \times (LCV)_0\,(\Phi - 1)$ KW

Gas Turbine Model: the isentropic efficiency of gas turbine is given by Eq (11)

$$\eta_t = \frac{T_3 - T_4}{T_3 - T_{4s}} \quad \ldots\ldots..(11)$$

Where

T_3 = Turbine inlet temperature

T_{4s} = isentropic discharge temperature of gas turbine

T_4 = actual discharge temperature of gas turbine

The actual discharge temperature of gas turbine can be given by Eq (12)

$$T_4 = T_3 - \eta_t (T_3 - T_{4s}) \quad \ldots\ldots\ldots.. (12)$$

The shaft work of the gas turbine is given by Eq (13)

$$W_{GT} = \dot{m}_g \times C_{pg} (T_3 - T_4) \quad \ldots\ldots (13)$$

The net work of the gas turbine is determined by Eq (14)

$$(W_{GT})_{net} = W_{GT} - W_C \quad \ldots\ldots.. (14)$$

Irreversibility of gas turbine is given by

$I_{GT} = m_g T_0 (s_4 - s_3)$

$= m_g T_0 [C_{pg} \log_e (T_4 / T_3) - R_g \log_e (P4 / P_3)]$ ………… (15)

The specific fuel consumption is determined by Eq (16)

$$sfc = \frac{3600 \times \dot{m}_f}{(W_{net})} \quad \ldots\ldots\ldots.(16)$$

The heat supplied is given by Eq (17)

$$Qadd = \dot{m}_f \times LHV \quad \ldots\ldots\ldots\ldots (17)$$

The thermal efficiency of gas turbine is determined by Eq (18)

$$(\eta_{th})_{GT} = \frac{(W_{GT})net}{\dot{m}_f \times LHV} \quad \ldots\ldots\ldots. (18)$$

2.2 STEAM TURBINE CYCLE MODEL: It is assumed that the steam turbine efficiency and pump efficiency are represented by η_{st} and η_p respectively. The ideal and actual processes on temperature – Entropy diagram are represented by full and dashed lines as shown in figure.

Heat Recovery Steam Generator Model: A dual pressure HRSG is considered here for combined cycle gas turbine plant. By applying Exergy balance for gas and water in each part of the HRSG the gas temperature and water properties are calculated by solving the following equations,

Heat available with the exhaust gases from gas turbine is given as

$$Q_{av} = \dot{m}_g \times C_{pg} \times (T_4 - T_5) \times h_{f1} \quad \ldots\ldots. (19)$$

Where T_5 is the exhaust temperature of the flue gases from HRSG, and h_{f1} is the heat loss factor whose value ranges from 0.98 to 0.99.

The HP superheater duty is expressed as

$$(Q_{sh})_{HP} = ms_{HP}\,(h_{13} - h_{12}) = \dot{m}_g \times c_{pg} \times (T_4 - T_g) \quad \ldots\ldots..(20)$$

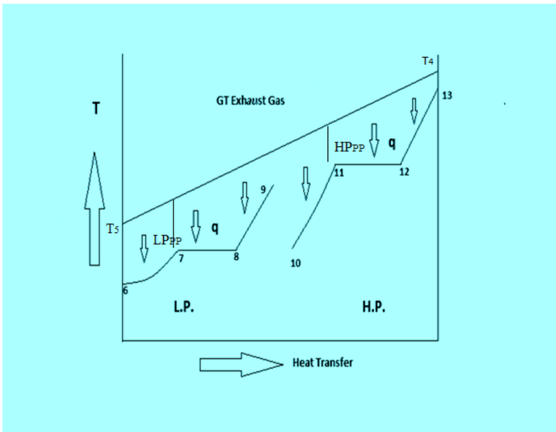

Figure 3: temperatures – heat transfer diagram of HRSG.

The thermal analysis of HRSG depends on the designed pinch point $(\Delta T)_{PP}$ and approach point$(\Delta T)_{AP}$. The temperature of the gas leaving the HP evaporator is given by

$T_{g1} = T_{11} + (\Delta T_{PP})_{HP}$ ....(21)

Where T_{11} is the saturation steam temperature corresponding to HP superheater pressure and $(\Delta T_{PP})_{HP}$ is the pinch point temperature difference in the HP side of HRSG.

The LP superheater duty is expressed as

$(Q_{sh})_{LP} = ms_{LP}\ (h_9 - h_8) = \dot{m}_g \times c_{pg} \times (T_{g1} - T_{g2})$ .... (22)

The temperature of the hot exhaust gases leaving the HRSG is given as

$ms_{LP}\ (h_9 - h_6) + ms_{HP}\ (h_{11} - h_{10}) = \dot{m}_g \times c_{pg} \times (T_{g1} - T_5)$ .... (23)

Loss of Exergy of flue gases in HRSG is given by

$(B_4 - B_5) = m_g\ [C_{pg}\ (T_4 - T_5) - T_0\{C_{pg}\ \log_e (T_4/T_5) - R\ \log_e (P_4/P_5)\}]$ .... (24)

Total gain of Exergy in LP side of HRSG can be obtained by multiplying by mass flow rate of steam

$B_9 - B_6 = m_{SLP} \times [C_{pw}\ (T_7 - T_6) + h_{fg} + C_{ps}(T_9 - T_8)] - T_0[C_{pw}\ \log_e (T_7/T_6) + (h_{fg}\ /T_8) + C_{ps}\ \log_e (T_9/T_8)]$ KW .... (25)

Total gain of Exergy of water in HP side of HRSG can be obtained by multiplying the mass flow rate of steam in HP side

$B_{13} - B_{10} = m_{SHP}\ [C_{pw}\ (T_{11} - T_{10}) + (h_{fg}) + C_{ps}\ (T_{13} - T_{12})] - T_0\ [C_{pw}\ \log_e (T_{11}/T_{10}) + (h_{fg}\ /\ T_{12}) + C_{ps}\ \log_e (T_{13}/T_{12})]$ KW .... (26)

The irreversibility in HRSG can be calculated by subtracting eq 25 and 26 from 24

Steam turbine model: The steam at high pressure and high temperature obtained from HP superheater is expand in the HP steam turbine and then mixed with LP steam obtained from LP superheater and expand upto the condenser pressure in LP steam turbine. The Exergy balance gives

$W_{HPST} = ms_{HP}\ (h_{13} - h_{14})$ .... (27)

And

$W_{LPST} = ms_{HP} + ms_{LP}\ (h_{15} - h_{16})$ .... (28)

The irreversibility in HP and LP steam turbines are given by

$I_{HPST} = ms_{HP} \times T_0\ (s_{13} - s_{14})$ .... (29)

$I_{LPST} = ms_{LP} \times T_0\ (s_{15} - s_{16})$ .... (30)

Condenser model: The heat rejected from the condenser is expressed as

$Q_{rej} = (ms_{HP} + ms_{LP} - ms_{bled})\ (h_{16} - h_{f17})$. (31)

The Exergy destruction in the condenser is given by considering two streams – steam and cooling water.

$I_{Cond} = (ms_{HP} + ms_{LP} - ms_{bled})\ (s_{out} - s_{in})_{steam} + m_w T_0\ (s_{f,out} - s_{f,in})_{cw}$ .... (32)

Pump model: The condensate from the condenser is extracted by the HP and LP pump and raised to LP and HP economizer pressure. The corresponding work is given by

$W_{hpp} = m_w \times v_{17}\ (p_{10} - p_c)$ .... (33)

$W_{lpp} = m_w \times v_{17}\ (p_6 - p_c)$ .... (34)

Therefore the net work of ST power plant is

$(W_{net})_{ST} = W_{ST} - W_P$ .... (35)

The efficiency of steam turbine power plant is

$\eta_{st} = \frac{(W_{net})_{st}}{Q_{av}}$ .... (36)

The overall thermal efficiency of the combined cycle gas turbine plant is given by

$\eta_{cc} = \frac{(W_{net})_{GT} + (W_{net})_{ST}}{m_f \times LHV}$ .... (37)

Exhaust or stack: rate of Exergy loss due to exhaust flue gases or stack is given by

$I_{exhaust} = m_g\ c_{pg}\ [(T_5 - T_0) - T_0 \ln T_5/T_0]$ (38)

Where T_5 = Temperature of flue gases leaving HRSG

T_0 = Ambient temperature

Exergetic or second law efficiency of the combined cycle is defined as

$(\eta_{cc})_{II} = \frac{(W_{net})_{GT} + (W_{net})_{ST}}{m_f \times LHV \times \emptyset}$ .... (39)

Φ - Exergy flux = $\Delta G/\Delta H$

ΔG - Change in standard Gibbs function of the reaction kJ/kg

ΔH - Change in the enthalpy of reaction kJ/kg

The value of Φ is taken as 1.04 to 1.045 for natural gas and depends upon the ratio of hydrogen to carbon in the fuel.

III RESULT AND DISCUSSION

In present work Exergy and Exergy analysis of natural gas based gas-steam combined cycle plant have been carried out and the effect of different thermodynamic operating variables on performance of combined cycle have been investigated and the results are presented in the tables 3.1-3.2 and figures 4 to 14. First and second law efficiencies of the combined cycle are calculated at 300K ambient temperature and TIT = 1005^0C and R_P =12.2 by applying Exergy and Exergy balance equations. Similarly Exergy destruction rate at different components of the plant are calculate and listed in table 3.1. The total Exergy distribution in the plant is listed in table 3.2 The results show that the combustor is the most significant Exergy destructor in the combined cycle plant and 39% of the total Exergy destroy in it.gas turbine and steam turbine are the other chief sources of irreversibility and Exergy destruction in these components is 5% and 3%. The Exergy destruction rate is minimum in the condenser and is 1% of the total Exergy input.

Table 3.1 Exergy analysis of combined cycle plant at T_0, =27^0C, TIT= 1005^0C, and R_P =12.2

Component	Rate of Exergy destruction (KW)
Compressor	20382
Combustor	345683.19
Gas turbine	42141
HRSG	40339.5
Steam turbine	25105.95
Condenser	7314
Stack or exhaust	11752.8
Unaccounted	2202.17

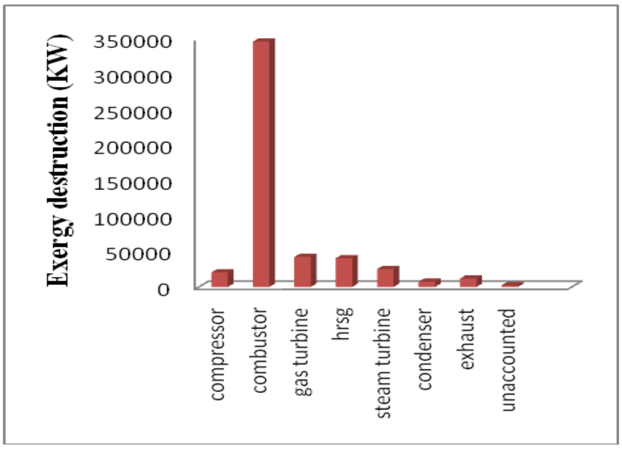

Figure 4: Exergy destruction rate at different components of the plant at TIT = 1005^0C and T_0 =27^0C and R_P = 12.2.

Table 3.2 Exergy balance for combined cycle plant atT_0 = 27^0C, TIT =1005^0C, and R_P =12.2

Exergy input (KW)	Power output (KW)	Exergy destruction (KW)
$-\Delta G_0$ = 894777.38	$(W_{net})_{GT}$ = 251298.57	Compressor: 20382
	$(W_{net})_{ST}$ = 148558.2	Combustor : 345683.19
		Gas turbine : 42141
		HRSG : 40339.5
		Steam turbine : 25105.95
		Exhaust gases : 11752.8
		Condenser : 7314
		Unaccounted : 2202.17
Input = 894777.38 KW	Total output = 399856.77 KW	Total losses = 494920.61 KW

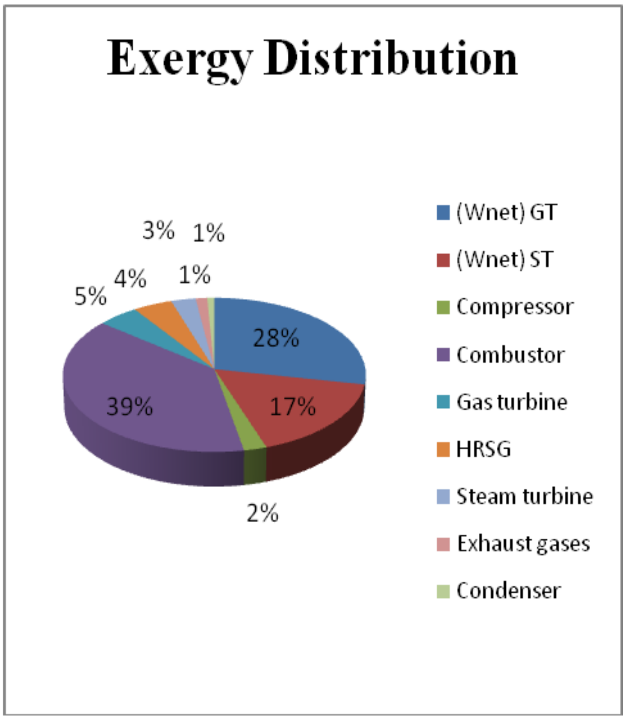

Figure 5: Exergy balance for the combined cycle plant.

Figures 6 and 7 show the effect of inlet air temperature of compressor on the Exergy efficiency and Exergy destruction rate of the combined cycle. It is clear from the figure that the Exergy efficiency of combined cycle decreases by 1.52% and the Exergy destruction rate increases by 1.25% as the inlet air temperature increases from 273K to 313K. The reduction in combined cycle Exergy efficiency is small as compared to the gas turbine cycle because of insignificant change in steam cycle efficiency with inlet air temperature.

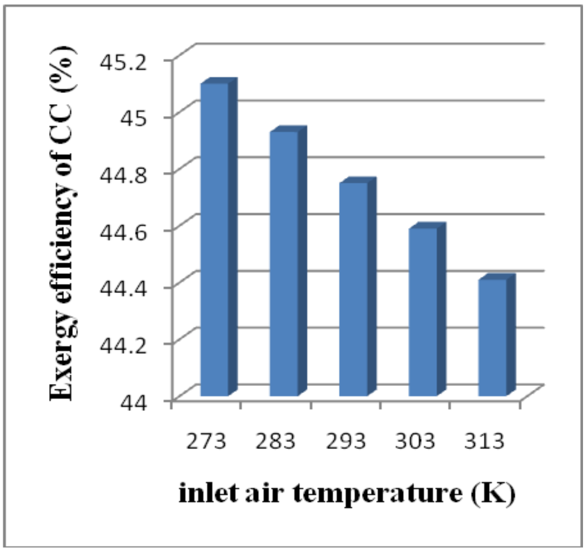

Figure 6: Effect of inlet air temperature on Exergy efficiency of CC.

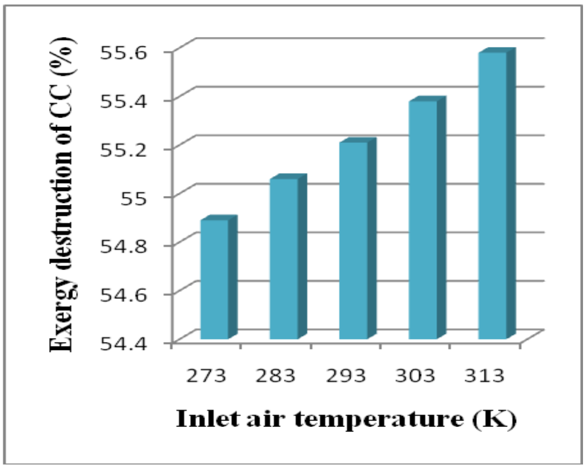

Figure 7: Effect of inlet air temperature on Exergy destruction of combined cycle.

Figure 8 shows the effect of inlet air temperature on the Exergy destruction in different components of the combined cycle. It is clear from the figure that the combustor dominates in the picture of Exergy destruction as compared to the other components of the plant. It is also seen from the figure that the Exergy destruction rate in combustor and in exhaust flue gases decreases with increasing inlet air temperature. The reason behind this is that the mass of fuel added in the combustor for the fixed TIT decreases as the inlet air temperature increases due to increase in discharge temperature of compressor. The another reason behind this is that the temperature difference between the flame temperature and working fluid decreases with increasing inlet air temperature which results in reduction in external irreversibility. The Exergy destruction rate in exhaust flue gases decreases due to reduction in mass flow rate of exhaust gas due to increasing inlet air temperature. The inlet air temperature has insignificant effect on Exergy destruction in compressor, gas turbine, HRSG, and in steam turbine.

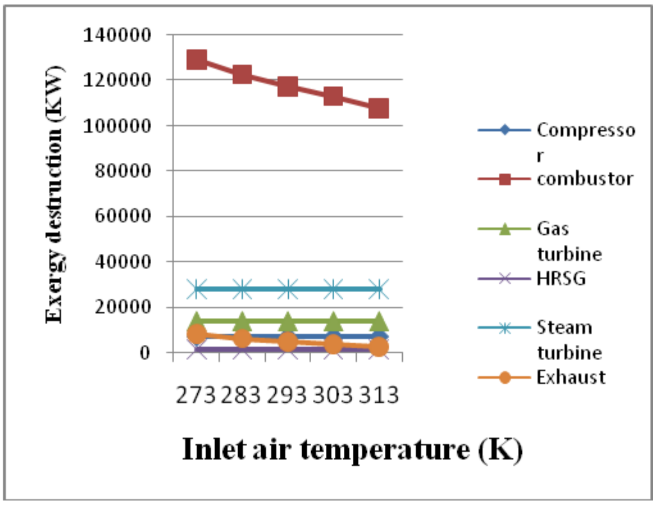

Figure 8: Effect of inlet air temperature on Exergy destruction rate of different components of combined cycle at TIT 1005^{0}C and R_P =12.2.

Figure 9 and10 shows the effect of TIT on Exergy efficiency and Exergy destruction rate of combined cycle. It is clear from the figures that Exergy efficiency of combined cycle increases with increasing TIT due to increase in net power output of both topping cycle and bottoming cycle with increasing TIT. For an increase of TIT from 1273 K to 1373 K the Exergy efficiency of combined cycle increases by 6.05% and it increases by 4.4% for an increase of TIT from 1373 K to 1473 K. The improvement in Exergy efficiency of combined cycle is slower in higher range of TIT from 1373 K to 1473 K than that of the TIT range of 1273 K to 1373 K. The Exergy destruction rate of combined cycle decreases with increasing TIT. For an increase of TIT from 1273 K to 1373 K the Exergy destruction rate of combined cycle decreases by 4.84% and it decreases by 3.93% for an increase of TIT from 1373K to 1473K.

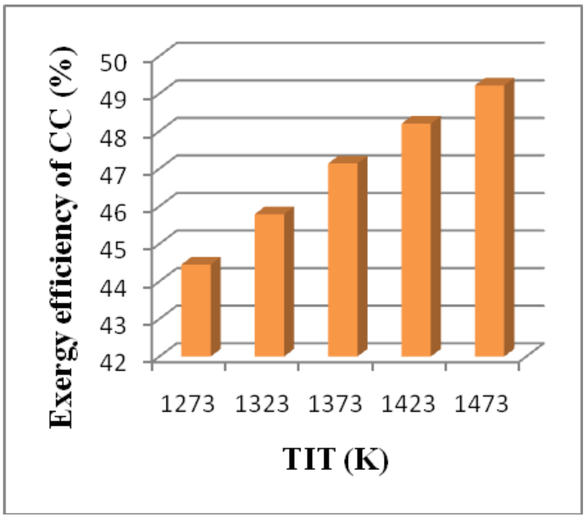

Figure 9: Effect of TIT on Exergy efficiency of combined cycle at R_P = 12.2.

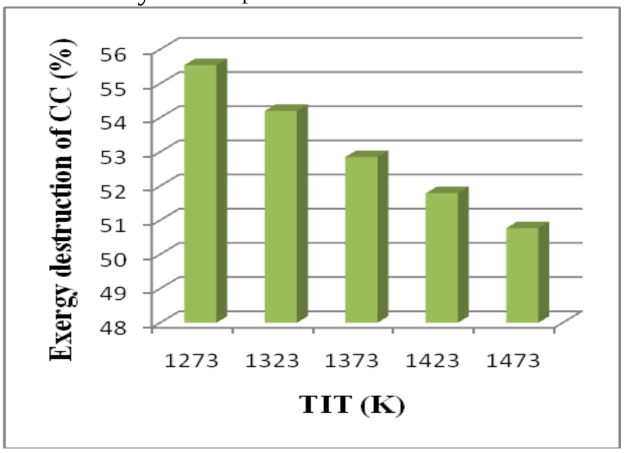

Figure 10: Effect of TIT on Exergy destruction of combined cycle at R_P = 12.2

Figure 11 shows the effect of TIT on Exergy destruction rate in different components of the combined cycle plant. It is clear from the figure that the combustor dominates in the picture of Exergy destruction as compared to the other components of the plant. It is also seen from the figure that the Exergy destruction rate in the compressor, combustor, HRSG, and in the steam turbine increases with increasing TIT and the Exergy destruction rate in the gas turbine and exhaust flue gases decreases with increasing TIT.

The reason behind this is that the mass of fuel added in the combustor increases with increasing TIT for a fixed pressure ratio of compressor. Any increase in mass flow rate of fuel results in increase in Exergy destruction in combustor. The reason behind this is that the external irreversibility increases with increasing TIT due to increase in temperature difference between the flame and working fluid. The reason of increase in Exergy destruction rate in HRSG is that with increasing TIT the exhaust temperature of gas turbine increases and temperature difference between the gas turbine exhaust and fixed maximum temperature of steam also increases which results in increase in external irreversibility.

The Exergy destruction rate in steam turbine increases due to increasing mass flow rate of total steam with increasing TIT. The reduction in Exergy destruction rate in exhaust flue gases is due to the reduction in temperature of flue gases leaving the HRSG with increasing TIT. The Exergy destruction rate in gas turbine decreases due to increasing temperature ratio of combustion products with increasing TIT. For an increase in TIT from 1273K to 1473K the Exergy destruction rate in combustor increases by 23.6MW and in steam turbine and HRSG it increases by 11.9 MW and 6.8 MW respectively. The Exergy destruction rate in exhaust flue gases and gas turbine decreases by 1.7 MW and 3.1 MW respectively for the same increase in TIT from 1273K to 1473K.

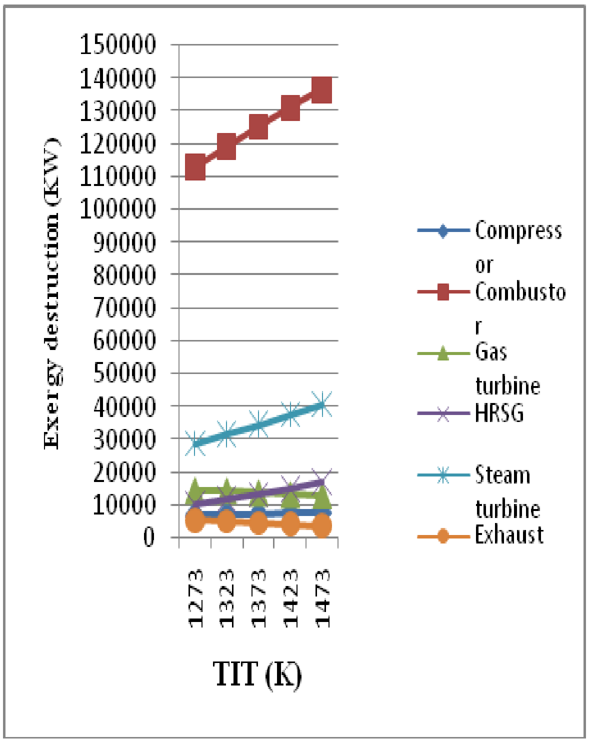

Figure 11: Effect of TIT on Exergy destruction rate in different components of combined cycle at $R_P = 12.2$.

Figure 12 and 13 shows the effect of pressure ratio on Exergy efficiency and Exergy destruction rate of combined cycle. It is clear from the figure that the Exergy efficiency of combined cycle first increases with increasing pressure ratio and then starts decreasing with increasing pressure ratio. The Exergy efficiency of combined cycle increases by 1.1% for an increase in pressure ratio from 8 to 12 and it increases by 0.34% for an increase in pressure ratio from 8 to 16.

The Exergy destruction rate of combined cycle first decreases with increasing pressure ratio and then starts increasing. The Exergy destruction rate of combined cycle decreases by 1% for an increase in pressure ratio from 8 to 12 and it decreases by 0.3% for an increase in pressure ratio from 8 to 16.

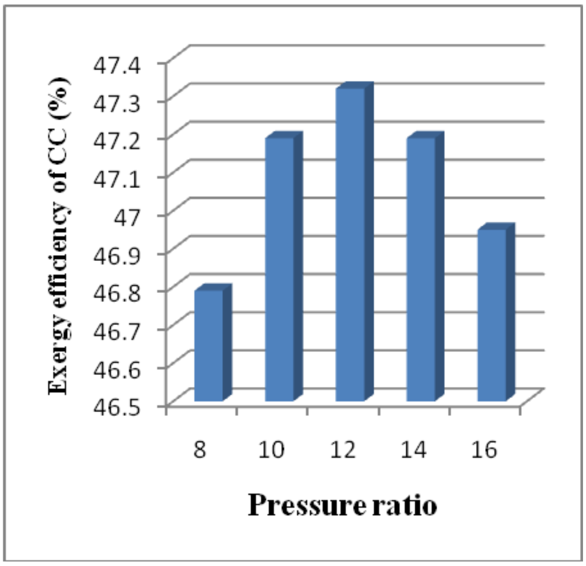

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Figure 12: Effect of pressure ratio on Exergy efficiency of combined cycle.

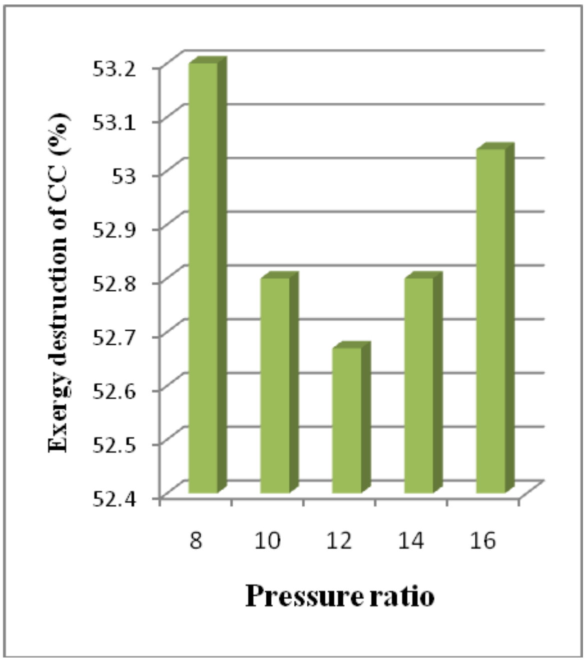

Figure 13: Effect of pressure ratio on Exergy destruction of combined cycle.

Figure 14 shows the effect of pressure ratio on Exergy destruction rate in different components of combined cycle plant. It is clear from the figure that the combustor dominates in the picture of Exergy destruction as compared to the other components of the plant. It is also seen from the figure that the Exergy destruction rate in the combustor, steam turbine, and in HRSG decreases with increasing pressure ratio whereas in compressor, gas turbine and in exhaust flue gases it increases with increasing pressure ratio. The reason behind this is that with increasing pressure ratio the mass of fuel added in combustor decreases for fixed TIT and the discharged temperature of compressor also increases with increasing pressure ratio resulting in reduction in external irreversibility due to decrease in temperature difference of flame temperature and of working fluid. Similarly the Exergy destruction in HRSG decreases with increasing pressure ratio because of reducing temperature difference between exhaust temperature of gas turbine and fixed maximum steam temperature in HRSG. The Exergy destruction rate in steam turbine decreases due to reducing mass flow rate of total steam in HRSG. The Exergy destruction rate in compressor and gas turbine increases with increasing pressure ratio because of increasing external irreversibility due to increase in temperature difference at inlet and outlet of these components. The increase in Exergy destruction in exhaust flue gas is due to the increase in exhaust temperature of flue gases leaving the HRSG with increasing pressure ratio due to reduction in total heat recovery in HRSG. For an increase in pressure ratio from 8 to 12 the Exergy destruction rate in combustor decreases by about 14 MW and it decreases by 27.8 MW for an increase in pressure ratio from 8 to 16. Similarly for an increase in pressure ratio from 8 to 12 the Exergy destruction rate in HRSG decreases by 4.2 MW and it decreases by 6.5 MW for an increase in pressure ratio from 8 to 16. The Exergy destruction rate in steam turbine decreases by 6.6 MW for an increase in pressure ratio from 8 to 12 and it decreases by 10.7 MW for an increase in pressure ratio from 8 to 16. The Exergy destruction rate in compressor increases by 0.98 MW for an increase in pressure ratio from 8 to 12 and it increases by 1.5 MW for an increase from 8 to 16. The Exergy destruction rate in the gas turbine increases by 4.1 MW for an increase in pressure ratio from 8 to 12 and it increases by 7.3 MW for an increase in pressure ratio from 8 to 16. The Exergy destruction rate in exhaust flue gases increases by 0.93MW for an increase in pressure ratio from 8 to 12 and it increases by 1.5MW for an increase in pressure ratio from 8 to 16. The overall Exergy destruction rate of the plant decreases with increasing pressure ratio.

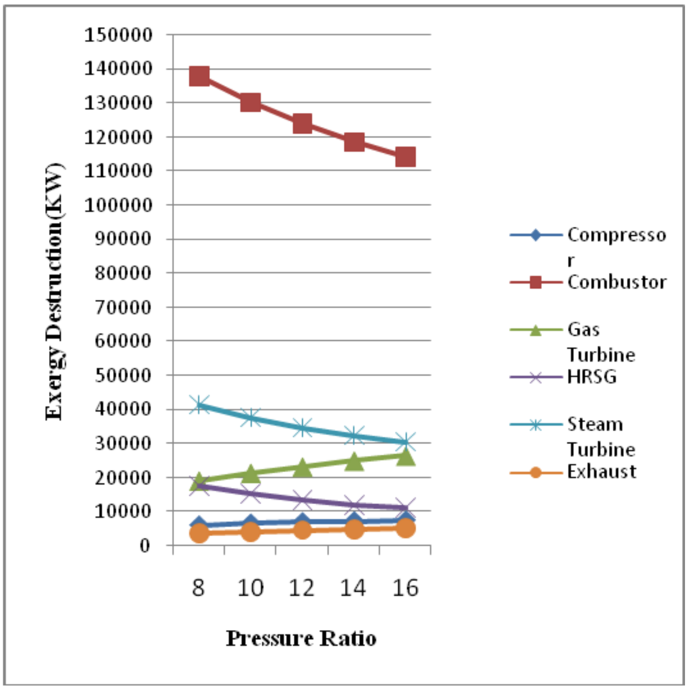

Figure 14: Effect of pressure ratio on Exergy destruction in different components of combined cycle at TIT 1100^{0}C.

IV CONCLUSION

On the basis of Exergy and Exergy analysis carried out on natural gas based 415MW gas-steam combined cycle power plant, the following important conclusions are made:

1. The combustion chamber and turbines are found to be chief means of irreversibility in the combined cycle plant due to the chemical reaction and large temperature difference between the burners and working fluid and thus have largest improvement potential.
2. The rate of Exergy destruction in combustor is about 39% of the total Exergy in and in gas and steam turbine is 5% and 3%.
3. The net power output, first and second law efficiency decreases with increasing inlet temperature to compressor and overall Exergy

destruction rate increases with increasing inlet air temperature.

4. The net power output, first and second law efficiency increases with increasing TIT and overall Exergy destruction rate decreases with increasing TIT.

5. The net power output decreases with increasing pressure ratio and first and second law efficiency and overall Exergy destruction rate decreases with increasing pressure ratio.

6. The change in first and second law efficiency with varying inlet air temperature, turbine inlet temperature, and pressure ratio is identical. The maximum first law efficiency corresponds to maximum second law efficiency.

7. At higher TIT and lower pressure ratio the Exergy destruction rate in the components combustor, HRSG, and steam turbine are higher.

8. At lower TIT and higher pressure ratio the Exergy destruction rate in the components gas turbine, compressor and exhaust are higher.

V FUTURE SCOPE OF WORK

Despite this detailed study, there is further scope of work as follows:

- The optimum turbine inlet temperature and pressure ratio should be the next focus of study for minimizing the total Exergy destruction rate in all the components.
- The effect of number of pressure levels of HRSG and the maximum steam pressure can be investigated for optimization of the combined cycle plant.
- More work is required on chemical Exergy analysis in combustor.

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