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THERMODYNAMIC PERFORMANCE INVESTIGATION OF GAS/STEAM COMBINED CYCLE BASED ON EXERGY ANALYSIS

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ABSTRACT

In the era of high energy demand gas turbine based combined cycle power plant (CCPP) has a critical role in power generation. Present paper deals with the thermodynamic performance investigation of gas/steam combined cycle on the basis of exergy analysis. The gas turbine cycle was taken as higher-temperature topping cycle and steam power cycle was embraced as lower temperature bottoming cycle. In this paper, the second law examination of combined cycle power plant with single pressure HRSG has been accounted for. This work plans to determine component-wise exergetic proficiency, the reasons for exergy demolition in the significant cycle component and to recognize the explanations behind the component-wise inefficiencies. The component wise exergy analysis of a conceptual power plant cycle operating under a pressure ratio of 23 and turbine inlet temperature (1700K) is carried out with a prime objective to highlight the major sites of exergy destruction in the cycle. The work generated by gas turbine is 521.36kJ/kg while for steam turbine it is 184.79kJ/kg. The performance of cycle on the basis of second law efficiency was assessed to be 52.45%.

***Keywords:* Combined Cycle, Exergy Analysis, Exergy Destruction, HRSG, Irreversibility, Gas Turbine, Steam Turbine.**

I. INTRODUCTION

Optimization of power generation system is one of the major concerns in the energy engineering field. The increase in the prices of energy is making it important to utilize the energy up to its maximum extent [1]. Gas turbine power plant (GTPP) is advantageous over steam cycle power plants with respect to relatively low capital investment, lesser installation time, high reliability without complexity, quick starting as well as eco-friendly. Besides all the advantages, gas turbine based power plants have the potential of being integrated into the steam turbine cycle by utilizing the thermal energy of the exhaust of the gas turbine. Gas/steam based combined cycle power plants (CCPP) offers high thermal efficiencies and improved power output. In the proposed combined cycle there is a topping cycle, a heat recovery steam generator and a bottoming cycle. The topping cycle is the Brayton cycle and bottoming cycle is Rankine cycle. The purpose of HRSG is to integrate topping and bottoming cycle for their synergetic operation. The heat leaving gas turbine exhaust is utilized in HRSG. The energy analysis of any energy system does not lead to the analysis of qualitative losses which occur in individual components of the system. The exergy-based analysis is a promising tool to analyze any energy based system. It also highlights the inefficiency of any energy based system. On the other hand, the second law analysis deals with the qualitative study of energy and determines the maximum amount of theoretical useful work obtained till the system comes to a dead state with reference to a specified environment. Exergy analysis focuses on the destruction of available energy in contrast to energy analysis which focuses on conservation of energy. Second law analysis determines the component level irreversibility and also the magnitude of irreversibility. The second law efficiency of CCPP is generally less than first law efficiency [2].

Several researchers have studied various aspects of combined cycle power plant and second law analysis of different energy based systems. A literature review of CCPP and exergy analysis is discussed below. M. J. Javadi et al. [3] carried out the first law analysis of the 116 MW gas turbine power plant in Iran, and also investigated the component-wise exergy analysis. E. Bilgen [4] carried out the exergy and energy analysis of the cogeneration power plant. He also developed an algorithm and the results shows good agreement with the previously reported data. A Vidal et al [5] investigated the exergy analysis of a new proposed combined power cycle integrated with refrigeration cycle. To investigate the effect of various parameters on the new cycle in various operating conditions, second irreversible case was analyzed. Sanjay et al [6] performed a parametric first and second law analysis of gas-steam reheat combined cycle using closed-loop-steam-cooling. The author suggests that steam cooling has the better cooling effects w.r.t air film cooling technique. M. Ameri et al [7] carried out the exergy efficiency of a 420MW NEKA CCPP in Tehran, Iran. The author suggests that combustion chamber is the main source of irreversibility followed by a gas turbine, duct burner, and HRSG. NEKA CCPP showed exergy efficiency of 47.5% and 44% with and without duct burner. Y. Hesli et al [8] studied the exergetic performance of a conventional recuperated GTPP integrated with the high temperature solid oxide fuel cell.. The result highlights that increase in TIT has the adverse effect of the exergetic efficiency as it declines with increase in TIT. Abdul Khaliq [9] investigated the second law analysis of a proposed trigeneration system based on conventional GT for high temperature heat addition while HRSG and vapour refrigeration systems were adopted for process heating and cold production respectively. Isam H. Aljundi [10] carried out the exergy and energy analysis of a steam power plant in Jordan. The basic objective of that study was to highlight the component wise quantitative exergy destruction. The percentage of exergy destruction was the maximum in the boiler, followed by the condenser and others. Young Sik Kim et al [11] carried out a trial to modify the compressor and gas turbine units to avoid the surge margin and overheating of blading in GT of an IGCC as its real operating condition is changed when integrated with gasifier and air separation unit (ASU) both. The author quoted the result that there is an increase in terms of both power output and efficiency when both turbine and compressor modifications are enabled. O. Singh and Kaushik [2] calculated the second law analysis of combined triple power cycles. The simulations of natural gas fired CCPP situated in India and KALINA cycle of Iceland were done in MATLAB and then results were compared and a possibility of KALINA cycle in Indian operating condition were investigated and effect of various parameters were also analysed . Sanjay and B. N. Prasad [12] carried out first and second law analysis of inter-cooled combustion turbine based CCPP and an effort has been taken for the maximization of efficiency by selecting an appropriate pressure ratio. The author states that intercooling increase the output of around 20% . M. Ghazikhani et al [13] carried out the second law analysis of GT with air bottoming cycle (ABC) and results were compared with conventional GT. The prime focus of the study was the investigation of work output, specific fuel consumption and exergy destruction. A. K. Mohapatra and Sanjay [14] carried out the parametric study of the effect of different operating parameters such as TIT, combustor inlet temperature (CIT), pressure ratio on a cooled gas turbine power plant with two inlet air cooling techniques. The author describes the effect of integration of two techniques positively on plant performance. M. A. Rosen [15] deal with Second law analysis approach and its implications on different power plant units. Nitul and Sanjay [16] have performed the exergy analysis to investigate the effect of compression ratio and air fuel ratio on rational efficiency for a combined cycle. The result shows that higher compression ratio and lower air/fuel ratio leads to lower exergy destruction.

II. CYCLE DESCRIPTION

Fig. 1 illustrates the proposed cycle configuration under investigation. The diagram outlines the configuration of basic gas turbine cycle integrated with single pressure HRSG steam turbine cycle. This configuration as a whole is known as combined cycle with single pressure HRSG. Air is taken from the atmosphere at an ambient condition which enters to air compressor (AC), which in turn compresses air to a higher pressure level. The compressed air which exits the air compressor enters the combustion chamber, where the burning of natural gas in the presence of air takes place, which in succession increase its temperature and energy level. Thereafter the expansion of pressurized high temperature flue gases takes place in a gas turbine. The HRSG is installed for the recovery of heat content in gas turbine exhaust for steam generation purpose. Here single pressure HRSG is being used. HRSG is generally equipped with an economiser, evaporator and super heater. Water from the deaerator when passes through the HRSG unit, the steam of required pressure level and quality is obtained. In HRSG unit deaerated water enters the economiser by passing through boiler feed pump, where sensible heating of the feed water takes place as heat is added in the economiser at constant pressure. Then this sensibly heated water is passed through the evaporator where heat added at constant evaporator pressure is just used for the conversion of water into steam. After the phase change of the water, steam produced is then passed through the superheater for the further temperature rise and then superheated steam is expanded in a steam turbine for the power generation purpose. The expanded steam is then passed through the condenser where by phase change the steam is converted into liquid phase i.e. water. The condensate extraction pump (CEP) then supply this water to the deaerator unit where de oxidation of feed water takes place using bled steam from a steam turbine.

III. MATHEMATICAL MODELLING

Mathematical modeling of various components of the proposed cycle to evaluate its thermodynamic performance has been detailed in the sub-sections as below.

3.1 Air Model

Air is taken as a working fluid of the cycle in the compressor. The thermodynamic properties of air enthalpy, entropy and exergy have been mathematically modeled as under:

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

$$h = c_p(T - T_0) \quad .. \quad (1)$$

$$s = c_p ln\left(\frac{T}{T_0}\right) - Rln\left(\frac{p}{p_0}\right) \quad \text{...... (2)}$$

$$e = h - T_0 . s = h - T_0 \varphi + RT_0 ln\,(p - p_0) \quad \text{....... (3)}$$

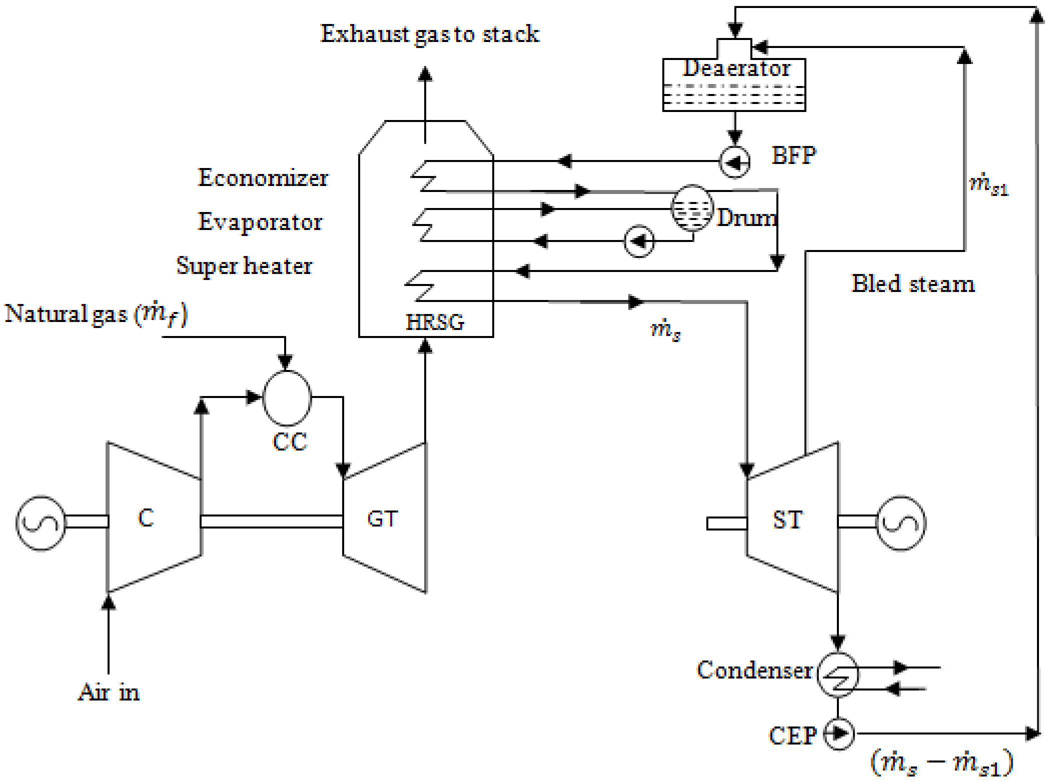

Figure 1 Schematic diagram of combined cycle power plant with single pressure HRSG

3.2 Air Compressor (AC)

Working fluid air enters to the axial flow compressor at ambient conditions. The compressor compresses the air to a higher pressure level simultaneously raising its temperature and energy level. The pressure and temperature of the air at the exit of air compressor are determined by the following equations.

$$p_2 = p_1 * r_{pc} \quad \text{... (4)}$$

$$T_2{}' = T_1\{1 + \frac{1}{\eta_{AC}}[(\frac{p_2}{p_1})^{\frac{1-\gamma_a}{\gamma_a}} - 1]\} \quad \text{..... (5)}$$

Exergy loss in the compressor for 1 kg of air is

$$\Omega_{comp} = T_0(s_2' - s_1) = T_0(c_{pa} ln\frac{T_2{}'}{T_1} - R_a ln\frac{p_2}{p_1}) \quad \text{.. (6)}$$

$$W_C = c_{pa}(T_2{}' - T_1) \quad \text{.... (7)}$$

3.2Combustion Chamber (CC)

The combustion chamber experiences various types of losses which are taken care by adopting a pressure loss of around 2% of the entry pressure at its exit and combustion chamber inefficiency of 0.5 %. Natural gas has been taken as a fuel and is burnt in the presence of compressed air to raise its temperature. The combustion products at high temperature lead into the gas turbine where expansion of gases takes place. Mass and energy balance equations for combustion chamber are as mentioned below:-

$$\dot{m}_g = \dot{m}_a + \dot{m}_f \qquad \text{...... (8)}$$

$$\dot{m}_f \cdot LHV \cdot \eta_{comb} = \dot{m}_g \cdot h_3 - \dot{m}_a \cdot h_2 \qquad \text{.... (9)}$$

Mass flow rate of fuel required has been calculated using mass and energy balance of the combustion chamber for 1 kg of air:

$$\frac{\dot{m}_f}{\dot{m}_a} = \frac{[c_{pg} \cdot T_3 - c_{pa} \cdot T_2{}']}{[\eta_{cc} \cdot LHV - c_{pg} \cdot T_3]} \qquad \text{..... (10)}$$

The exergy of fuel

$$e_{xf} = \dot{m}_f (LHV)_f \cdot \Phi \qquad \text{...... (11)}$$

Exergy loss in combustion chamber is

$$\Omega_{cc} = T_0[\left\{(s_p)_3 - (s_p)_0\right\} - \{(s_a)_2 - (s_a)_0\} + (\Delta s_{react})_0]$$

$$= T_0[\left\{c_{pg} ln\frac{T_3}{T_0} - R_g ln\frac{p_3}{p_0}\right\} - c_{pa} ln\frac{T_2{}'}{T_0} + R_a ln\frac{p_2}{p_1} + (\Delta s_{react})_0) \qquad \text{.... (12)}$$

Where

$$T_0(\Delta s_{react})_0 = \dot{m}_f \cdot LHV \cdot (\Phi - 1) \qquad \text{...... (13)}$$

3.4 Gas Turbine (GT)

This high pressure and temperature flue gases sequentially expand in the stages of axial flow gas turbine, thus power is developed by the continuous removal of energy content from the flue gases.

$$p_3 = p_2(1 - \Delta p_{cc}) \qquad \text{.... (14)}$$

Where $\Delta p_{cc} = 0.02$

$$T_4{}' = T_3\{1 - \eta_{GT}[1 - (\frac{p_3}{p_4})^{\frac{1-\gamma_g}{\gamma_g}}]\} \qquad \text{..... (15)}$$

Where $p_4 = 1.08\ bar$

$$\Omega_{GT} = (1 + \dot{m}_f/\dot{m}_a) \cdot T_0(s_4' - s_3) = (1 + \dot{m}_f/\dot{m}_a) \cdot T_0[c_{pg} ln\frac{T_4{}'}{T_3} - R_g ln\frac{p_4}{p_3}]$$

..... (16)

$$W_{GT} = (1+\dot{m}_f/\dot{m}_a)c_{pg}(T_3 - T_4') \quad \text{.... (17)}$$

$$W_{gt,net} = W_{GT} - W_C = \left(1+\frac{\dot{m}_f}{\dot{m}_a}\right)c_{pg}(T_3 - T_4) - c_{pa}(T_2 - T_1) \quad \text{.... (18)}$$

HRSG

The heat recovery steam generator (HRSG) is generally used for the recovery of energy from the high temperature gas turbine exhaust. The unit generally contains an economiser, an evaporator and super heater. The deaerated water enters the HRSG unit and exits as a superheated steam.

The energy and exergy balance equations are given as under:-

Here, h_5= 3440, h_{f10} (Deaerator outlet) = 504.8 kJ/kg

Energy balance yields

$$(1+\dot{m}_f/\dot{m}_a)(h_4' - h_{stack})\ \varepsilon_{hrsg} = \dot{m}_s(h_5 - h_{f10}) \quad \text{.... (19)}$$

The exergy loss in HRSG is given by

$$\Omega_{hrsg} = T_0[\dot{m}_3 \cdot (s_{hrsg,out} - s_{hrsg,in})_{s/w} + (1+\dot{m}_f/\dot{m}_a)\cdot(s_{stack} - s_{hrsg,in})_{gas}] \quad \text{..... (20)}$$

$$= T_0[\dot{m}_3 \cdot (s_5 - s_{f10}) + \left(1+\frac{\dot{m}_f}{\dot{m}_a}\right)\cdot\left(c_{pg}\,ln\frac{T_{stack}}{T_{hrsg,in}} - R_g\,ln\frac{p_{stack}}{p_{hrsg,in}}\right)] \quad \text{.... (21)}$$

3.5 Exhaust

The exhaust gas exergy loss is given by

$$\Omega_{Exh} = \left(1+\frac{\dot{m}_f}{\dot{m}_a}\right)c_{pg}[(T_{stack} - T_0) - T_0 \cdot ln\frac{T_{stack}}{T_0}] \quad \text{.. (22)}$$

3.6 Steam turbine

The superheated steam from the HRSG enters the steam turbine where the continuous expansion of steam takes place for work extraction.

The exergy loss in steam turbine is given by

$$\Omega_{ST} = \dot{m}_s \cdot T_0(s_7' - s_5) \quad \text{..... (23)}$$

$$W_{st,net} = \dot{m}_s(h_5 - h_6') + (\dot{m}_s - \dot{m}_{s1})(h_6' - h_7') \quad \text{.... (24)}$$

Here $\dot{m}_{s1}$ can be calculated from energy balance of the deaerator

$$\dot{m}_{s1} = \frac{\dot{m}_s(h_{f6} - h_{f1})}{(h_6' - h_{f7})} \quad \text{...... (25)}$$

3.7 Condenser

The condenser is generally a cross flow heat exchanger which is employed to remove the heat content of the exhaust steam from the steam turbine so as to condense it and achieve vacuum pressure for enhanced expansion of steam in the steam turbine.

Energy balance at condenser is given by

$$\dot{m}_w = \frac{\dot{m}_s(h_7' - h_{f8})}{c_{pg}\cdot\Delta T_w} \quad \text{...... (26)}$$

$$\Omega_{cond} = \dot{m}_s \cdot T_0\left(s_{f8} - s_7^{'}\right) + \dot{m}_w \cdot T_0\left(s_{f,out} - s_{f,in}\right)_{water} \quad \text{... (27)}$$

Feed Pump

The purpose of installing feed pump is to pressurise the deaerated water at deaerator pressure to the boiler pressure. It consumes a negligible amount of work compared to the net work output.

The exergy loss in feed pump is given by

$$\Omega_{cond} = \dot{m}_s \cdot T_0\left(s_{f9}^{'} - s_{f8}\right) \quad \text{..... (28)}$$

IV. RESULTS AND DISCUSSION

Based on the modelling, governing equation and different input parameters, cycle performance has been plotted in the form of different curves. The input parameters and exergy balance for discussed cycle have been represented in the form of tables 1 and 2.

All the calculations have been made using air/gas as the working fluid under operating conditions as detailed in Table 1.

Table 1 Input data for analysis [17-18]

Parameter	Symbol	Unit
Air/Gas Properties	T_0 =288	K
	p_0 = 1	bar
Compressor	Isentropic efficiency = 86	%
	Mechanical efficiency = 99	%
	Compressor Pressure ratio = 23	
Combustion Chamber	Combustion chamber efficiency (η_{cc})=98	%
	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
Gas Turbine	Isentropic efficiency =86	%
	Exhaust pressure = 1.08	bar
	TIT = 1700	K
Mass flow rate of air	$\dot{m}_a = 1$	kg/s
HRSG	Effectiveness = 0.92	
Steam Turbine	Isentropic efficiency =86	%
	Exhaust/condenser pressure = .05	bar
	TIT = 773	K

Table 2 shows the exergy distribution percentage within the cycle, depicting the quantum of exergy that is consumed/lost in different components of the cycle. As we all know that the presence of irreversibility's within different components causes' exergy destruction. The percent of exergy balance of the discussed cycle has been presented in table 2. It highlights that exergy input to the cycle is 1316.7kJ/kg which is further distributed in the form of power output and exergy destruction among different components of the cycle. The total power output of cycle is 690.63 kJ/kg, while combustion chamber shows the maximum amount of exergy destruction (27.32%) followed by steam turbine (2.34%) and air compressor (2.04%)

Table 2 Exergy balance in combined cycle

Exergy input (%)	Power output %	Component	Exergy loss (%)
100	39.59	Air Compressor	2.04
	14.03	Combustion Chamber	27.32
		Gas Turbine	1.68
		HRSG	0.67
		Exhaust	1.91
		Steam turbine	2.34
		Condenser	1.4
		Feed pump	0.09
		Unaccounted	10.11
Input = 1316.7 kJ/kg	Output = 690.63 kJ/kg		Total = 626.05 kJ/kg

$$\text{Second law efficiency} = \frac{w_{out}}{e_{x,f}} = \frac{690.63}{1316.7} = 52.45\%$$

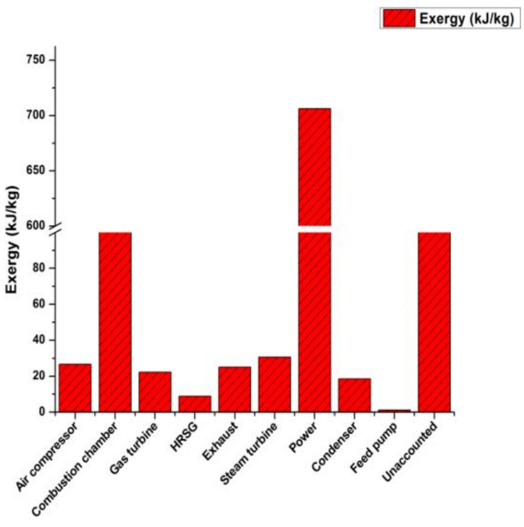

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Figure 2 Component-Wise Exergy Distribution (Gain/Loss) Of Combined Cycle Power Plant

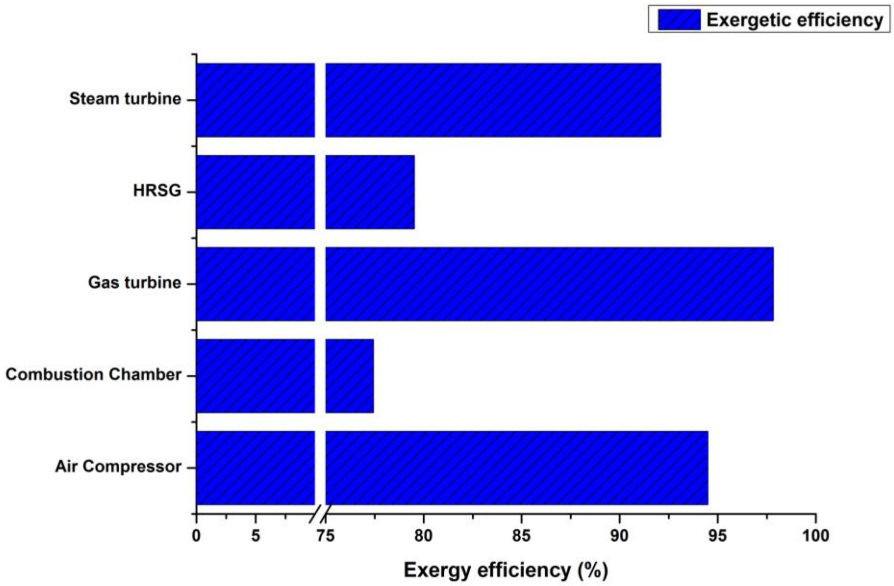

Figure 3 Component-wise exergy efficiency of combined cycle power plant

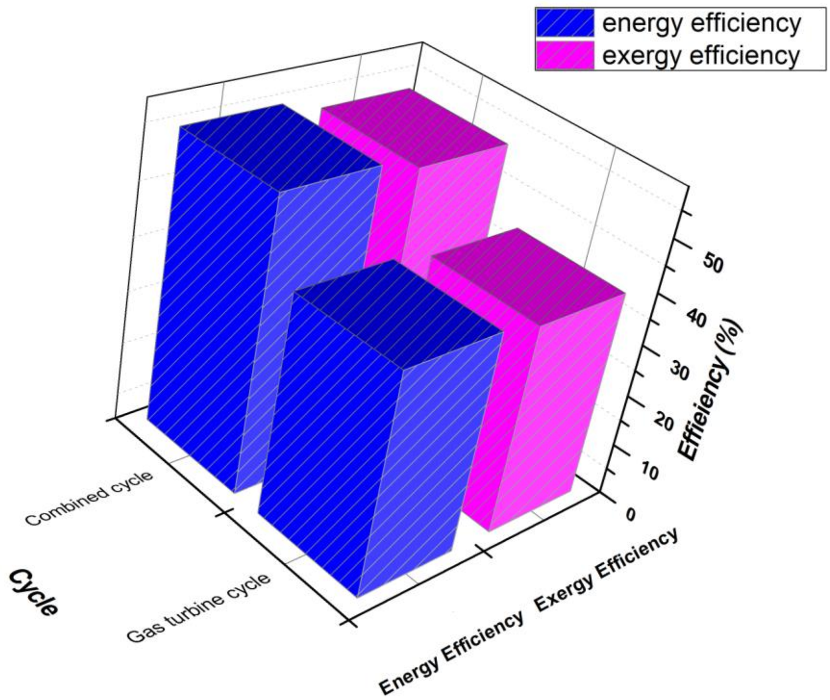

Figure 4 Energy And Exergy Efficiency of Gas Turbine Cycle and Combined Cycle

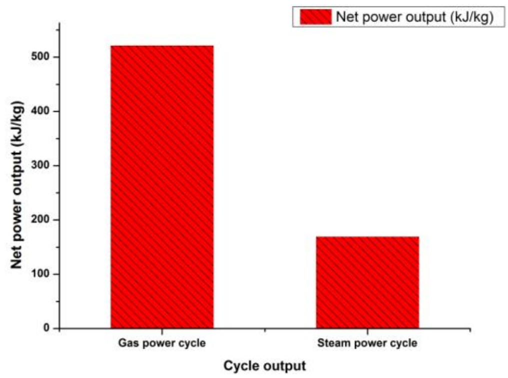

Figure 5 Net power output Vs cycle output

Figure 2 diagrams the component-wise rate exergy destruction in the combined cycle. It is clear from the graph that the exergy destruction is found the most extreme in the combustion unit which demonstrates great concurrence with the past works, additionally it is supported from the thermodynamic perspective that exergy destruction in the combustion chamber is observed to be 25%-30%, which for our situation it is around 359.5kJ/kg (27.32%). The following most astounding exergy annihilation is in a steam turbine (2.34%), trailed by an air compressor (2.04%).The unaccounted exergy destruction is around one-tenth (10.11%) of the aggregate cycle exergy destruction.

Figure 3 expresses the component-wise exergetic efficiency of combined cycle power plant. The exergetic efficiency is characterized as the proportion of exergy recovered to the exergy supplied, which further clarifies the greatest measure of work which can be acquired when the system comes in equilibrium with environment. The outcome demonstrates that exergy efficiency of the gas turbine is around 97.84% which portrays a magnificent work change rate, trailed via air compressor (94.5%), steam turbine (92.1%), HRSG (79.52%) and combustion chamber (77.3%) separately.

Figure 4 explains the comparative study of variation of exergy and energy efficiency of both combined cycle power plant and gas turbine cycle. It can be obviously concluded from the above graph that energy and exergy efficiency of the gas turbine cycle is 41.37% and 39.59% respectively, while for combined cycle it is 54.81% and 52.45% respectively. This shows the good agreement for the better performance of combined cycle over basic gas turbine cycle.

Figure 5 represents the comparison of the net power output of gas turbine and steam turbine cycles individually. It is clear from the plot that net power output of gas turbine cycle is 521.36 kJ/kg while for steam cycle it is 169.27 kJ/kg, which explains that the power output of the gas turbine is almost three times the steam cycle and represents the sizing ratio of these machines.

V. CONCLUSION

Based on the comprehensive thermodynamic analysis of combined cycle with single pressure HRSG, 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 gas turbine and combined cycle has been observed as (41.37%) and (54.81%) respectively.
- The second law efficiency for gas turbine and combined cycle was found to be (39.59%) and (52.45%) respectively.
- The gas turbine cycle power output was observed as 521.36kJ/kg while for steam turbine cycle it was 184.79kJ/kg. The total power output for combined cycle was 690.63kJ/kg.
- Exergy destruction arising due to component irreversibility's is maximum in the combustion chamber at 27.32% followed by air compressor and exhaust gas stream at 2.04% and 1.91% respectively.

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Nomenclature

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

Subscripts

a	Air
c	Compressor
comb	Combustion chamber

d	Destruction
ex	Exergy
f	Fuel
Gen	Generation
G	Gas
in	Inlet
Out	Outlet
sat	Saturated
T	Turbine
w	Water
I	First law
II	Second law
1, 2, 3	State points

Greek symbols

φ	Thermodynamic property function
ε	Effectiveness
η	Efficiency
ω	Availability per unit mass of gas
Ω_d	Exergy destruction rate

Acronyms

A	Alternator
ABC	Air bottoming cycle
AC	Air compressor
ATR	Auto thermal reformer
ASU	Air separation unit
BGT	Brayton gas turbine cycle
BFP	Boiler feed pump
C	Compressor
CEP	Condensate extraction pump
CC	Combustion chamber
CCPP	Combined cycle power plant
CIT	Combustor inlet temperature
GTPP	Gas turbine power plant
GT	Gas turbine
HRSG	Heat recovery steam generator
I	Irreversibility
IGCC	Integrated gas combined cycle
TIT	Turbine inlet temperature