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Energy and Exergy Analysis of a Coal Fired Boiler of 250MW Thermal Power Plant

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Abstract: The present study deals with application of useful concept of energy and exergy utilization to a coal fired boiler of a thermal power plant. In this research paper a simplified method is used in which the energy and the exergy efficiencies are calculated using either input and output energy/exergy or energy and exergy changes of the fluid streams during the processes. Objective of this research is to find out locations and magnitudes of major energy losses and exergy destruction in the boiler under different loading conditions. In the study, boiler is divided in three zones viz. combustion chamber, heat exchanger and heat recovery subsystem and analyzed component wise as well as section wise. Energy and exergy efficiencies of different components and subsystems are determined and exergy destruction is also calculated under different loading conditions. Above parameters are also determined for whole boiler as a single unit.

In component wise analysis final superheater shows minimum energy efficiency but quantitatively maximum energy and exergy losses occurs in evaporator. In the subsystem wise analysis maximum energy and exergy losses were found in heat exchangers, thus the loss in this zone needs to be controlled. Boiler as a whole has maximum 86.93% energy efficiency at 80% TMCR while 46.78% maximum exergy efficiency is found at 100% BMCR.

Keywords: Energy loss, Exergy destruction, Exergy efficiency, Irreversibility.

I. INTRODUCTION

Worldwide increasing per capita energy demand along with increasing population has asked the governments to establish more and more power generation industries. The viable options are thermal power plants with coal as the primary energy source. Continuously increasing use of coal, besides rapidly depleting reserve and many serious environmental issues may cause the growth in worldwide coal demand up to 94.7% in between 2003 to 2030 [1]. In India's total power generation capacity share of thermal power plants were 67% and that of coal based thermal power plants were 59% in April 2017 [2]. The above figure told that any effort succeeding to small improvement in energy efficiency will contribute a lot in reducing the coal consumption.

Old conventional practices of thermodynamics first law based energy analyses are not sufficient as it does not provide any information regarding qualitative utilization of energy. Hence the concept of exergy analysis which is based on the thermodynamics first and second law together is gaining more importance and being effectively used as more reliable scientific tool.

Main objective of this research work is to carry out the energy and exergy analysis of the boiler of a 250MW coal fired steam power plant to quantify the energy and exergy losses and to identify the components causing major losses. Useful concept of energy and exergy utilization is used by R. Saidur et al. [3]. In their study they have shown the energy and exergy flows in an industrial boiler and found that energy and exergy efficiencies of the boiler were 72.46% and 24.89% respectively. Also they found that the combustor was the component causing highest exergy destruction followed by the heat exchanger of the boiler system. They also suggested some energy saving measures. L. Pattanayak and S. K. Ayyagari [4] presented the study dealing with the energy and exergy analysis of a 500MW coal fired boiler in design and off-design conditions at constant pressure mode of operation. They used the component wise modeling and simulation of the boiler and found that the exergy destruction was maximum in the combustor then next in heat exchanger. Krishan Kumar et al. [5] carried out the exergy analysis of boiler and found that maximum exergy destruction occurs due to combustion process. V. D. Shimpi and M. Yadav [6] in an energy and exergy analysis of boiler of 210MW coal based thermal power plant found the energy and exergy efficieny as 73.96% and 39.85% and concluded that exergy destruction increases with increasing reference temperature. S. J. Gulhane and A. K.Thakur [7] in an exergy analysis of boiler of 6MW captive power plant determined the amount and sources of irreversibilities in boiler to locate the place of largest exergy destruction and to identify the system component needed to be redesigned. S. S. Fard and A. A. Davoodi [8] use oxygen enrichment of combustion air as one of the technique for improvement of combustion systems. In their study of a typical coal based steam generator the optimum enrichment level observed was 35% by volume at which the exergy and energy efficiencies were 44.9% and

93.3% respectively. A. H. Rana and J. R. Mehta [9] carried out the energy and exergy analysis of circulating fluidized bed boiler and worked out the energy and exergy efficiencies as 85.6% and 41% and identified the combustion chamber as the source of major exergy destruction. P. S. Hada and I. H. Shah [10] in a first law and second law analysis of a lignite fired boiler used in a 30MW thermal power plant found the two efficiencies as 85.52% and 33.73%. S. Arefdehgani and O. K. Sadaghiyani [11] in a boiler parametric study of a gas fired steam power plant found the boiler energy efficiency as 91.54% and 86.17% based on lower and higher heating values respectively which were in logical agreement with experimental data. They studied the effect of several measures and concluded that by decreasing excess air fraction from 0.4 to 0.15 energy and exergy efficiency of the plant increases by 0.497% and 0.46% respectively. Also two efficiencies increase nearly 2.196% when stack gas temperature decreases from 159 to 97^{o}C. L. Pattanayak [12] in an exergy analysis of a 500MW pulverised coal fired boiler evaluated the performance with varying excess air and conclude that with 0.5% increase in excess O_2% caused 0.65% decrease in combustion chamber efficiency and 0.47% boiler exergy efficiency. A. Babu et al. [13] in a research project identified the sulphuric acid corrrosion as the root cause of failure of utility boiler economiser tubes. They also determined the exergy efficiency of economiser, superheater, combustor and entire boiler as well. Ankit Patel [14] in an energy and exergy of a boiler with different grades of coal found that major inefficiencies are due to the heat transfer and combustion processes. Mukesh Gupta and Raj Kumar [15] analyze the effect of hot air temperature on the thermoeconomic performance of a boiler used in a coal fired thermal power plant and optimize the boiler as a trade off between the unit product cost of hot air and unit product cost of the boiler system. M. Bakhshes and A. Vosough [16] in a study found that the decrease in stack gas temperature and excess air quantity improves the performance of a power plant with a gas fired steam generator. L. Pattanayak [17] in a steady state energy and exergy analysis of pulverized coal fired boiler using Ebsilon software for component wise modeling and simulation found that maximum exergy destruction occurs at combustor followed by heat exchanger. T. Ashok Kumar et al. [18] carried out the exergy analysis of a coal based 63MW circulating fluidized bed boiler and found that the boiler system utilises 88.41% of the total energy supplied to the plant and nearly 6.7% of heat supplied is carried away by the exhaust gas. Ameet P. Koringa and N. S. Mehta [19] in a mathematical analysis of boiler by using first and second law of thermodynamics determined the boiler energy and exergy efficiencies as 71.6% and 25% respectively. M. Siddharth Bhatt and S. Jothibasu [20] in a syudy for performance enhancement in coal fired thermal power plants with specific reference to the boilers evaluated the efficiencies and performance indices of the equipment. They investigated the generic problems associated with design, operation and fuel inputs and suggested some appropriate solutions. Jun-Ta WU [21] in a research work on the optimization of boiler efficiency based on Artificial Bee Colony (ABC) algorithm found that the optimal value of parameters for optimization of boiler efficiency can quickly be obtained by the algorithm. S. S. L. Patel and G. K. Agrawal [22] in an assessment of boiler losses and investigation of influencing parameters found the dry flue gases as the biggest source followed by the loss due to hydrogen in fuel.

II. BOILER DESCRIPTION

Schematic diagram of the boiler under study is shown in Fig.1. Boiler is equipped with the accessories viz. three superheaters namely Low Temperature Superheater (LTSH), Platen Superheater (SHPL) and Final Superheater (SHF), one Reheater (RH) located between platen and final superheaters, Economiser (ECO) and air preheater(APH). Important operating parameters of the boiler are given in Table 1 to 4.

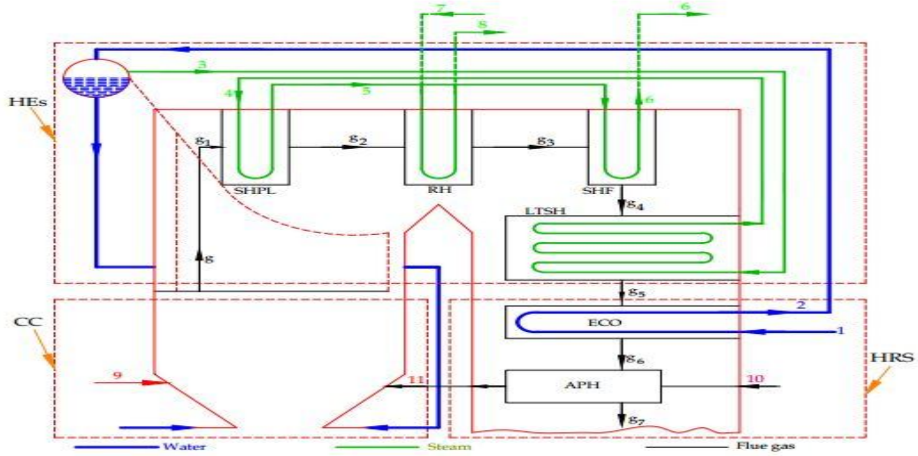

Fig.1 Schematic diagram of boiler

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TABLE 1: Process parameters of boiler at 100% BMCR*

Point	Physical State	T (°C)	p (bar)	m (kg/s)	h (kJ/kg)	s (kJ/kgK)
1	Water	247	171.32	230.36	1071.28	2.766
2	Water	289	169.16	230.36	1284.64	3.161
3	Steam	351	167.19	230.36	2640	5.2
4	Steam	402	162.77	230.36	2935	5.821
5	Steam	509	157.36	230.36	3322	6.305
6	Steam	540	152.44	225	3440	6.45
7	Steam	344	38.75	191.3	3094	6.562
8	Steam	540	36.68	191.3	3550	7.243
9	Coal	85	1.038	51.41	13800	0.7
10	Air	33	1.035	244.36	39	6.9
11	Air	295	1.024	274.55	309	7.623
g	Flue gas	1560	1.013	303.33	2604	7.9
g_1	Flue gas	1139	1.0125	303.33	1503	7.595
g_2	Flue gas	925	1.0121	303.33	1173	7.47
g_3	Flue gas	730	1.0118	303.33	876	7.321
g_4	Flue gas	629	1.0111	303.33	754	7.245
g_5	Flue gas	463	1.0097	303.33	514	7.156
g_6	Flue gas	325	1.006	303.33	351	7.0437
g_7	Flue gas	138	0.9909	327.27	147	6.554

*BMCR- Boiler Maximum Continuous Rating

TABLE 2: Process parameters of boiler at 100% TMCR*

Point	Physical State	T (°C)	p (bar)	m (kg/s)	h (kJ/kg)	s (kJ/kgK)
1	Water	246	167.78	205.14	1077.2	2.7761
2	Water	290	165.72	205.14	1290	3.161
3	Steam	349	162.96	205.14	2631	5.314
4	Steam	403	160.21	210	2947	5.815
5	Steam	508	155.68	210	3325	6.3
6	Steam	540	151.45	205.14	3426.24	6.46
7	Steam	344	37.17	183.55	3088.43	6.58
8	Steam	540	35.2	183.55	3534.66	7.26
9	Coal	85	1.0337	47.05	13800	0.7
10	Air	33	1.0316	216.11	39	6.9
11	Air	295	1.0231	251.22	309	7.623
g	Flue gas	1560	1.013	277.5	2604	7.9
g_1	Flue gas	1155	1.0125	277.5	1524.6	7.6
g_2	Flue gas	860	1.012	277.5	1171	7.428
g_3	Flue gas	722	1.0118	277.5	859	7.32
g_4	Flue gas	621	1.0115	277.5	747	7.24
g_5	Flue gas	457	1.0102	277.5	510	7.155
g_6	Flue gas	323	1.0075	277.5	351	7.0437
g_7	Flue gas	125	0.9938	300.94	139	6.5187

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TABLE 3: Process parameters of boiler at 80% TMCR

Point	Physical State	T (^oC)	p (bar)	m (kg/s)	h (kJ/kg)	s (kJ/kgK)
1	Water	236	161.5	165.11	1042	2.6652
2	Water	282	160.1	165.11	1247.36	3.0868
3	Steam	346	158.1	165.11	2660	5.35
4	Steam	406	155.6	174.13	2950	5.86
5	Steam	511	152.64	174.13	3327	6.32
6	Steam	540	149.8	165.11	3432	6.475
7	Steam	338	30.1	148.91	3089.27	6.67
8	Steam	540	28.52	148.91	3549.73	7.37
9	Coal	85	1.029	38.61	13800	0.7
10	Air	33	1.099	175.83	39	6.9
11	Air	285	1.092	206	296	7.61
g	Flue gas	1560	1.013	227.5	2604	7.9
g_1	Flue gas	1136	1.0125	227.5	1499.5	7.595
g_2	Flue gas	856	1.0123	227.5	1134	7.428
g_3	Flue gas	688	1.0121	227.5	825.6	7.31
g_4	Flue gas	594	1.0118	227.5	718.84	7.225
g_5	Flue gas	442	1.011	227.5	504.8	7.153
g_6	Flue gas	307	1.0091	227.5	336	7.035
g_7	Flue gas	118	0.9991	249.8	126	6.495

TABLE 4: Process parameters of boiler at 50% TMCR

Point	Physical State	T (^oC)	p (bar)	m (kg/s)	h (kJ/kg)	s (kJ/kgK)
1	Water	214	154.6	108.39	945	2.463
2	Water	269	153.42	108.39	1173	2.969
3	Steam	343	151.75	108.39	2681	5.407
4	Steam	411	150.67	118.78	2976	5.9
5	Steam	519	149.3	118.78	3350	6.37
6	Steam	540	147.92	108.39	3417.45	6.5
7	Steam	323	18.78	94.3	3076.71	6.87
8	Steam	540	17.8	94.3	3551	7.57
9	Coal	85	1.0234	26.16	13800	0.7
10	Air	33	1.09	117.55	39	6.9
11	Air	260	1.086	145.16	269	7.59
g	Flue gas	1470	1.013	159.72	2480	7.8
g_1	Flue gas	1079	1.0125	159.72	1409.17	7.5
g_2	Flue gas	770	1.0124	159.72	1059.6	7.385
g_3	Flue gas	627	1.0123	159.72	751.6	7.214
g_4	Flue gas	551	1.0122	159.72	678	7.155
g_5	Flue gas	420	1.0118	159.72	473	7.13
g_6	Flue gas	278	1.0108	159.72	290	7.01
g_7	Flue gas	115	0.9954	180.72	122.9	6.53

III. ENERGY AND EXERGY ANALYSIS

For the purpose of energy and exergy analysis of the boiler it is divided into three subsystems as stated below and shown in Fig.1
Combustion Chamber or Combustor (CC),
Heat Exchangers (HEs) – this includes evaporator (EVA), superheaters (SHs) and reheater (RH), and
Heat Recovery System (HRS) – this includes economiser (ECO) and air preheater (APH).

A. Energy analysis

Boiler, either component wise, subsystem wise or as a whole can be considered as a steady flow process in an open system as there is no parametric change with time at any particular load. In a steady flow process of an open system three types of energy transfer across the control surface are namely work transfer, heat transfer and energy associated with mass transfer and/or flow [23].
Thus, for a steady flow process in an open system, the first law of thermodynamics can be written as [23-25]:

$$\Sigma Q + m\,(h_i + C_i^2/2 + gZ_i) = W + m\,(h_0 + C_0^2/2 + gZ_0) \tag{1}$$

Where Q is the heat transfer to the system from source at temperature T, W is the net work developed by the system, m and C are mass flow rate and bulk velocity of the working fluid respectively, Z is the altitude of the stream above datum and g is the acceleration due to gravity.
The energy or first law efficiency η_I of a system and/or system component is defined as the ratio of energy output to the energy input to system/component [23] i.e.

$$\eta_I = \text{Desired output energy/ Input energy supplied} = E_{out}/E_{in} \tag{2}$$

And Energy loss, $E_{loss} = E_{in} - E_{out}$ (3)

B. Exergy analysis

Exergy in most common term is the maximum obtainable work from a system, a stream of matter and/or heat interaction with environment as the datum state. For any thermodynamic system the maximum obtainable work is equal to that of work delivered by the reversible cyclic process or Carnot work, which is given by:

$$W_{max} = \eta_{carnot} \times Q \tag{4}$$

Or exergy, $\psi = Q\,(1 - T_0/T)$ as $\eta_{max} = \eta_{carnot} = 1 - T_0/T$ (5)

Where T_0 is ambient temperature and T is the temperature of heat source/sink at which the heat is transferred or rejected.
Second law analysis uses both the mass balance and energy quality degradation with the entropy generation. Exergy analysis is a useful method to complement but not to replace energy analysis.
The exergy balance equation for a steady flow process of an open system neglecting kinetic and potential energy changes is given by:

$$\Sigma Q\,(1 - T_0/T) + \Sigma m\,\varepsilon_{in} = \psi_w + \Sigma m\,\varepsilon_{out} + I_{destroyed} \tag{6}$$

Where $I_{destroyed} = T_0\,\Delta S$ or $T_0\,(S_{gen})$

Also total exergy, $\psi = m\,\varepsilon = m\,[(h - h_0) - T_0\,(s - s_0)]$ (7)

In the above equations ε_{in} and ε_{out} are specific exergy associated with mass inflow and outflows respectively, ψ_w is useful workdone on/by system, $I_{destroyed}$ is irreversibility or exergy destroyed or exergy loss. Possible causes for irreversibility are heat transfer through finite temperature difference, mixing of fluids at different temperatures and mechanical friction. Exergy analysis is an effective means to pinpoint the losses due to irreversibility.
The exergy or second law efficiency is defined as,

η_{II} = Actual thermal efficiency/Maximum possible thermal efficiency
= Exergy output /Exergy input = Product exergy/ Input exergy

i.e. $\eta_{II} = \psi_p/\psi_{in}$ or ψ_{out}/ψ_{in} (8)

And Exergy destruction, $I_{destroyed} = \psi_{in} - \psi_{out}$ (9)

C. *Component/ Subsystem wise energy and exergy analysis*

1)Combustion chamber :Energy analysis- Heat transfer across the combustion chamber control surface can be considered as zero as it is well insulated i.e. $\Sigma Q = 0$. Also there is no work transfer, therefore W = 0. Again change in kinetic and potential energies of the fluid streams are usually negligible. With these considerations the energy quantities remained for analysis are only the total energy of the incoming streams i.e. of coal and air and outgoing mixture i.e. of products as a flue gas. Thus, the total energy of incoming streams [25].

$E_{in} = m_f h_f + m_a h_a$

Where m_f, h_f = mass flow rate and specific enthalpy (HHV) of coal and m_a, h_a = mass flow rate and specific enthalpy of air at the inlet to combustion chamber.

Now with reference to boiler schematic diagram shown in Fig.1

$$E_{in} = m_9 h_9 + m_{11} h_{11} \tag{10}$$

And total energy of outgoing mixture,

$$E_{out} = m_g h_g \tag{11}$$

Where m_g and h_g are the mass flow rate and specific enthalpy of hot combustion product i.e. flue gas respectively.

Exergy analysis- In combustion chamber quantity Q $(1–T_0/T) = 0$, and $\psi_w = 0$, therefore, equation (6) will reduced to:

$$\Sigma m\, \varepsilon_{in} = \Sigma m\, \varepsilon_{out} + T_0\, \Delta S \text{ or } \psi_{in} = \psi_{out} + T_0\, \Delta S \tag{12}$$

$$\text{or} \quad T_0\, \Delta S = \psi_{in} - \psi_{out} \tag{13}$$

With reference to the coal fired boiler under study,

$$T_0 \Delta S = (m\varepsilon)_f + (m\varepsilon)_a - (m\varepsilon)_g = m_9 \varepsilon_9 + m_{11} \varepsilon_{11} - m_g\, \varepsilon_g \tag{14}$$

Where ‘f’ stands for fuel i.e. coal, ‘a’ for air and ‘g’ for gas.

$$\text{Also } T_0\, \Delta S = m_9(h_9 - T_0 s_9) + m_{11}(h_{11} - T_0\, s_{11}) - m_g(h_g - T_0 s_g) \tag{15}$$

Entropy of flue gas and hot products are obtained from Cengel and Michel Boles [26]

$$\psi_{in} = m_9 \varepsilon_9 + m_{11}\, \varepsilon_{11} = m_9(h_9 - T_0 s_9) + m_{11}(h_{11} - T_0 s_{11})$$

$$\text{or} \quad \psi_{in} = E_{in} - m_9 T_0 s_9 - m_{11} T_0 s_{11} \tag{16}$$

$$\text{and} \quad \psi_{out} = m_g\, \varepsilon_g = m_g(h_g - T_0 s_g)$$

$$\text{or} \quad \psi_{out} = E_{out} - m_g T_0 s_g \tag{17}$$

2) Heat exchanger: Heat exchangers are merely used to transfer heat from hot stream to cold stream. Normally the performance of heat exchangers are represented by their effectiveness which is the ratio of increase in heat content of the cold stream to the decrease in heat content of hot stream. In case of boiler heat exchangers, their performance is evaluated in terms of first and second law efficiencies which are similar to energy effectiveness and exergy effectiveness respectively. Thus, for heat exchangers relations used are [26, 27]:

$$\eta_I = (\Delta E)_c / (\Delta E)_h \tag{18}$$

$$\text{and} \quad \eta_{II} = (\Delta\psi)_c / (\Delta\psi)_h \tag{19}$$

$$\text{Energy loss, } E_{loss} = (\Delta E)_h - (\Delta E)_c \tag{20}$$

$$\text{and exergy loss, } \psi_{loss} = (\Delta\psi)_h - (\Delta\psi)_c \tag{21}$$

Where $(\Delta E)_c$ and $(\Delta\psi)_c$ are the increase in energy and exergy of cold fluid while ΔE_h and $(\Delta\psi)_h$ are decrease in energy and exergy of hot fluid respectively.

The energy and exergy efficiencies of each components of the heat exchanger subsystem are analyzed as above.

3) Heat recovery subsystem: Heat recovery subsystem comprises economiser and air preheater. Economiser exchanges heat from hot flue gas to feed water and air preheater transfers heat from flue gas to air. Therefore, both the components are type of heat exchangers and they are analyzed so as discussed under heat exchanger section.

4) Overall boiler analysis: Considering all three subsystems of the boiler together and analysing them as a single system as below:

Energy analysis: $E_{in} = m_9 h_9$

$$E_{out} = (m_6\, h_6 - m_1 h_1) + m_7\, (h_8 - h_7) \tag{22}$$

Exergy analysis: $(\Delta\psi)_h$ or $\psi_{in} = m_f\, \varepsilon_f = m_9 \times \gamma \times \text{LHV}$ (23)

Where ε_f is the specific exergy of fuel based on lower heating value (LHV) and is given by $\varepsilon_f = \gamma \times \text{LHV}$ [23] and γ is the exergy factor.

Value of γ can be determined for any fuel C_xH_y by [24-26]

$$\gamma = 1.033 + 0.0169\, (y/x) - 0.0698/x \tag{24}$$

Also the value of γ for different grades of coal varies between 1.06 to 1.10 Table C1 Appendix C [23]

$$\begin{aligned}(\Delta\psi)_c \text{ or } \psi_{out} &= (m_6\, \varepsilon_6 - m_1\, \varepsilon_1) + m_7\, (\varepsilon_8 - \varepsilon_7)\\ &= m_6\, (h_6 - T_0\, s_6) - m_1\, (h_1 - T_0\, s_1) + m_7\, [(h_8 - h_7) - T_0\, (s_8 - s_7)]\\ &= (\Delta E)_c - T_0\, [m_6\, s_6 - m_1 s_1 + m_7\, (s_8 - s_7)]\end{aligned} \tag{25}$$

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IV. RESULTS AND DISCUSSION

A detailed energy and exergy analysis has been carried out in order to determine the location and magnitude of energy and exergy losses in boiler components and boiler subsystems. First and second law analysis is performed across the each component and subsystems of boiler for different loading conditions by using parameters as given in Table 1 to 4. Results are shown in Fig. 2 to14.

Exergetic analysis has been carried out with dead state condition of T_0=27°C and p_0=1.013bar. Combustion is assumed/considered a steady flow process and maximum temperature of combustion product in all loading condition is taken as 1560°C. It is also assumed that the kinetic and potential energies are negligible. Fig. 2 to 5 shows that even energy efficiency being satisfactory for most of the boiler components (except SHPL and SHF), exergy efficiency is low. Lower energy efficiency of SHPL and SHF highlights the need to decide their better location and orientation as well as the possibility of optimizing relative velocity of two fluid streams passing through these devices.

Subsystem wise analysis results shown in Fig. 6 to 9 depicts that even energy efficiency being highest the exergy efficiency of the combustion chamber is least among three zones, though from exergy loss by quantity it stands next to heat exchanger system. As Fig. 10 to 13 shows, highest exergy destruction takes place in heat exchanger system amounting to near 70% in all the four loading conditions. Finally, from the Fig. 10 to 13 the maximum energy efficiency of the boiler is found at 80%TMCR as 86.93% and exergy efficiency is maximum 46.78% at 100%BMCR. Exergy efficiency at 100%TMCR condition is more by 1.38% than at 50%TMCR and 0.5% than at 80% TMCR. Further analysis shows that boiler energy efficiency varies by fraction of percentage from 100% TMCR to 50%TMCR but, exergy efficiency reduces by more than 1% with the same variation in load .Thus, operation under part load has more intense and negative impact on exergetic performance than on energetic performance. Therefore, study suggests that operation of the plant at or around 100%TMCR condition will be economical both from energetic and exergetic point of view.

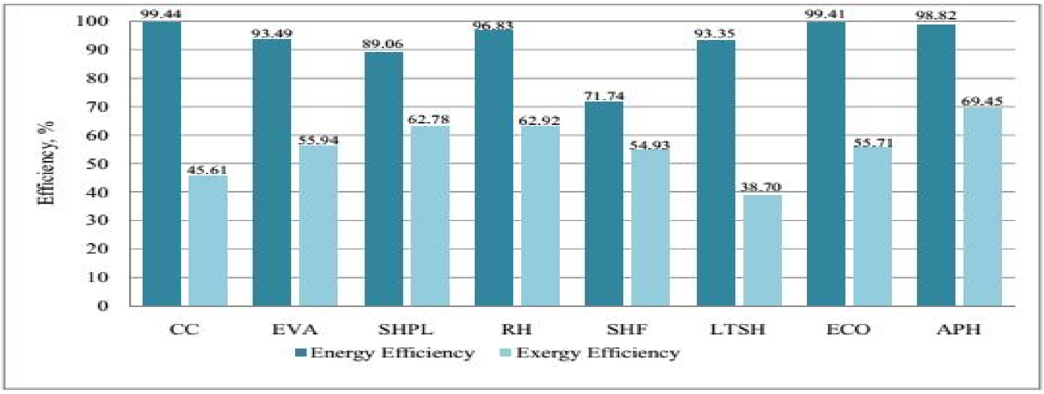

Fig.2 Component wise energy and exergy efficiency at 100% BMCR

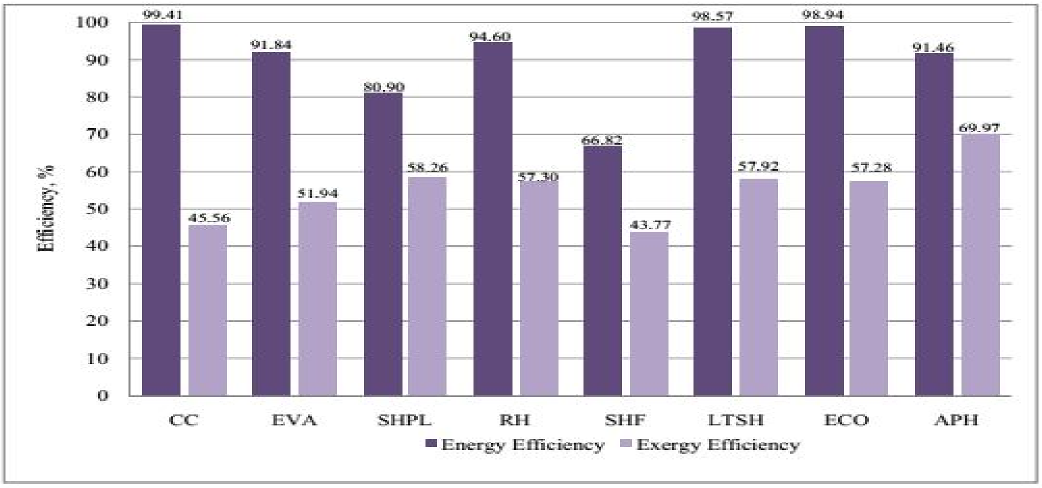

Fig.3 Component wise energy and exergy efficiency at 100% TMCR

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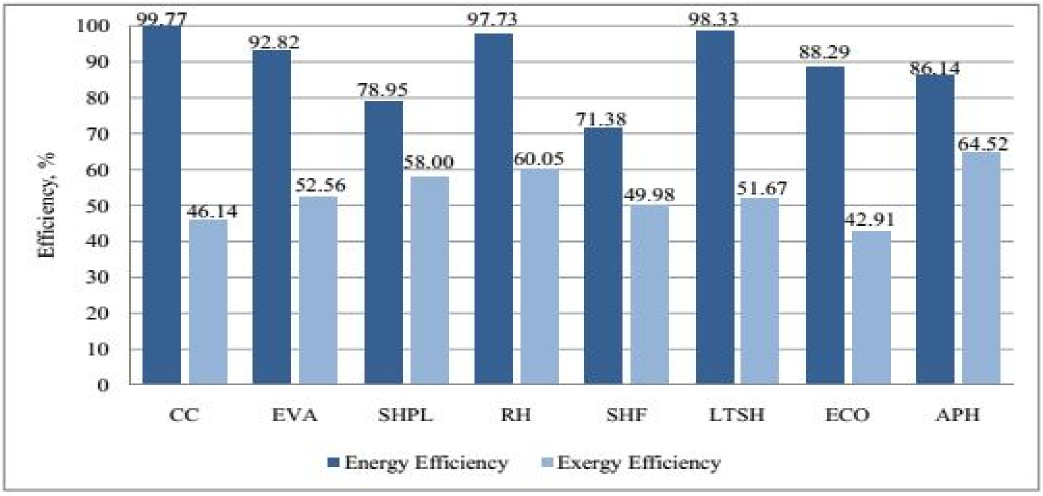


Fig.4 Component wise energy and exergy efficiency at 80% TMCR

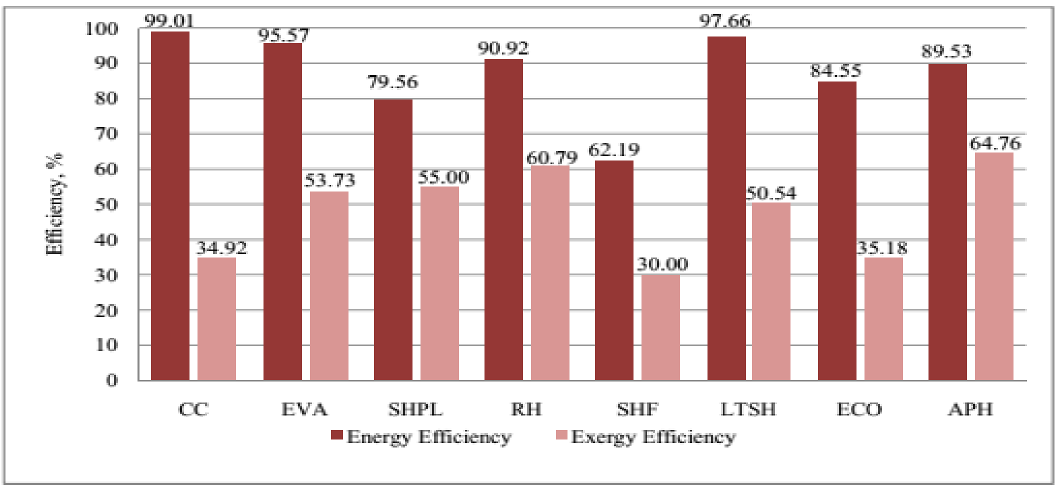
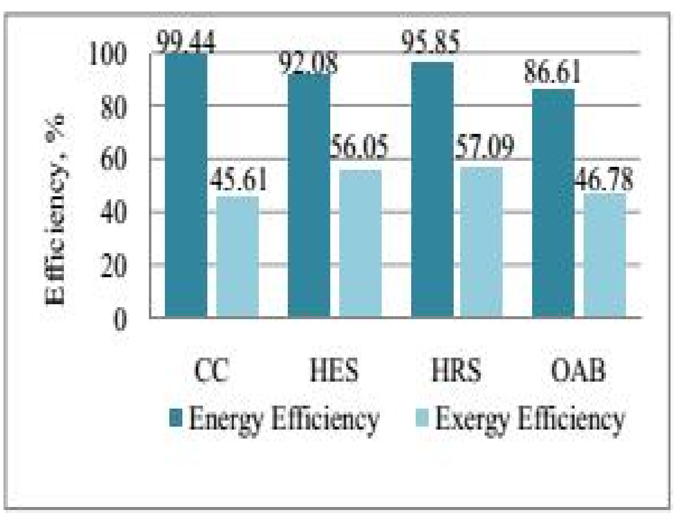


Fig.5 Component wise energy and exergy efficiency at 50% TMCR



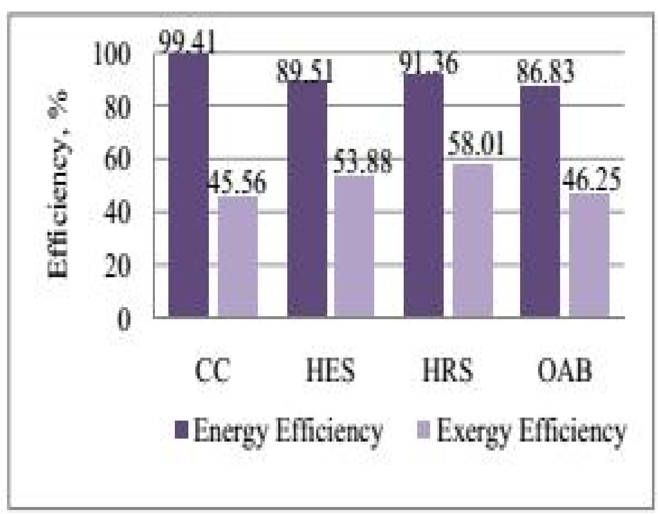


Fig.6 Subsystem wise energy and exergy efficiency at 100% BMCR Fig.7 Subsystem wise energy and exergy efficiency at 100% TMCR

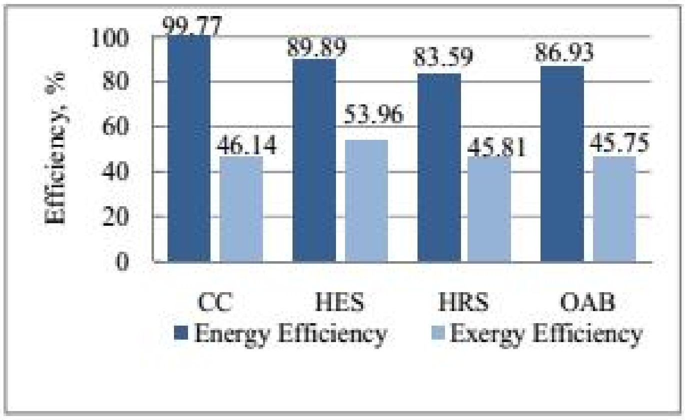


Fig.8 Subsystem wise energy and exergy efficiency at 80% TMCR

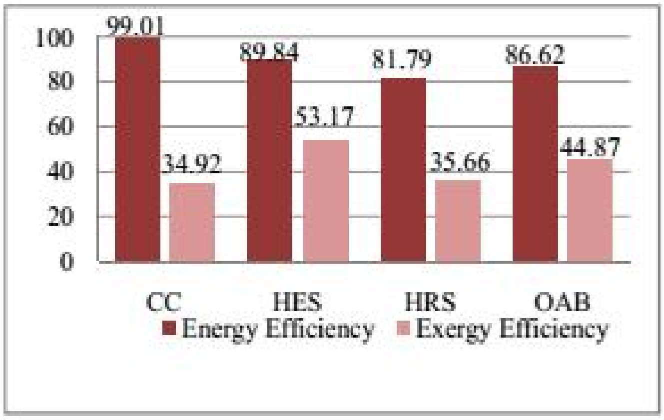


Fig.9 Subsystem wise energy and exergy efficiency at 50% TMCR

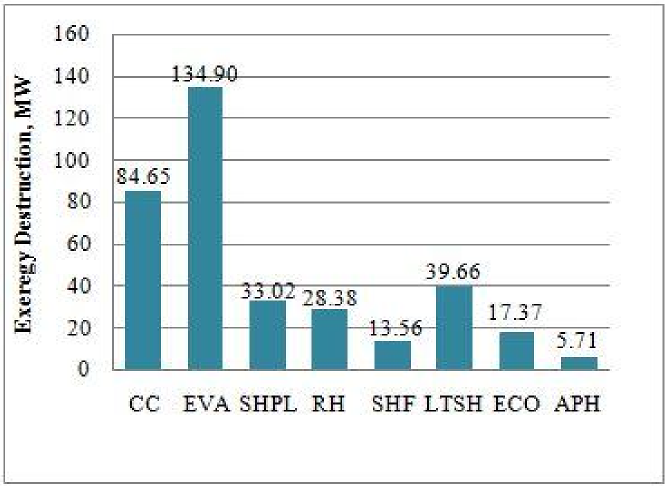


Fig.10 Component wise exergy destruction at 100% BMCR

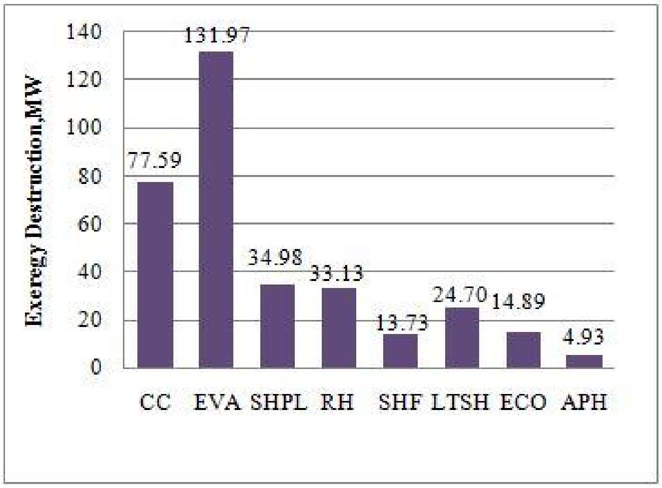


Fig.11 Component wise exergy destruction at 100% TMCR

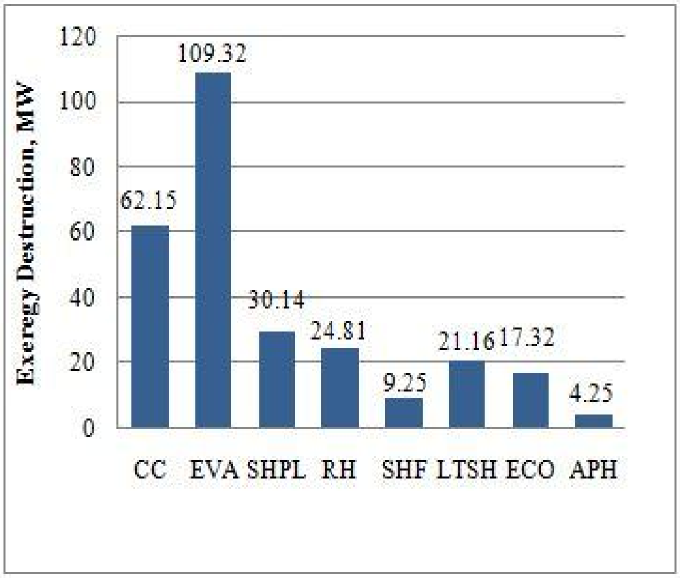


Fig.12 Component wise exergy destruction at 80% TMCR

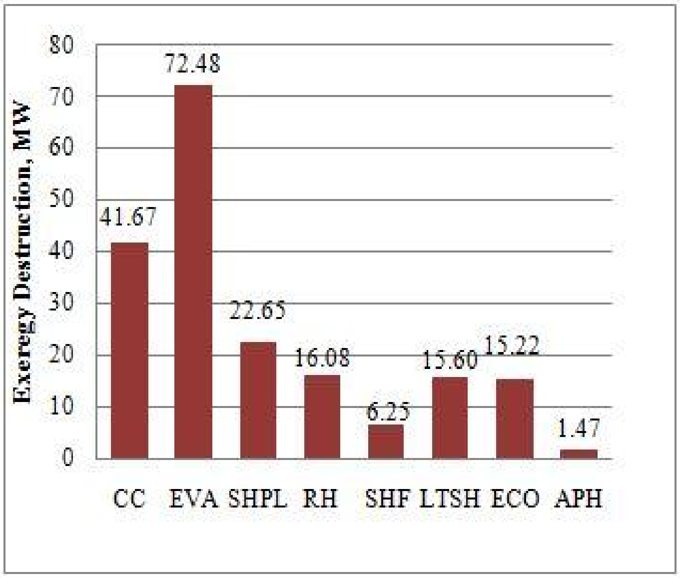


Fig.13 Component wise exergy destruction at 50% TMCR

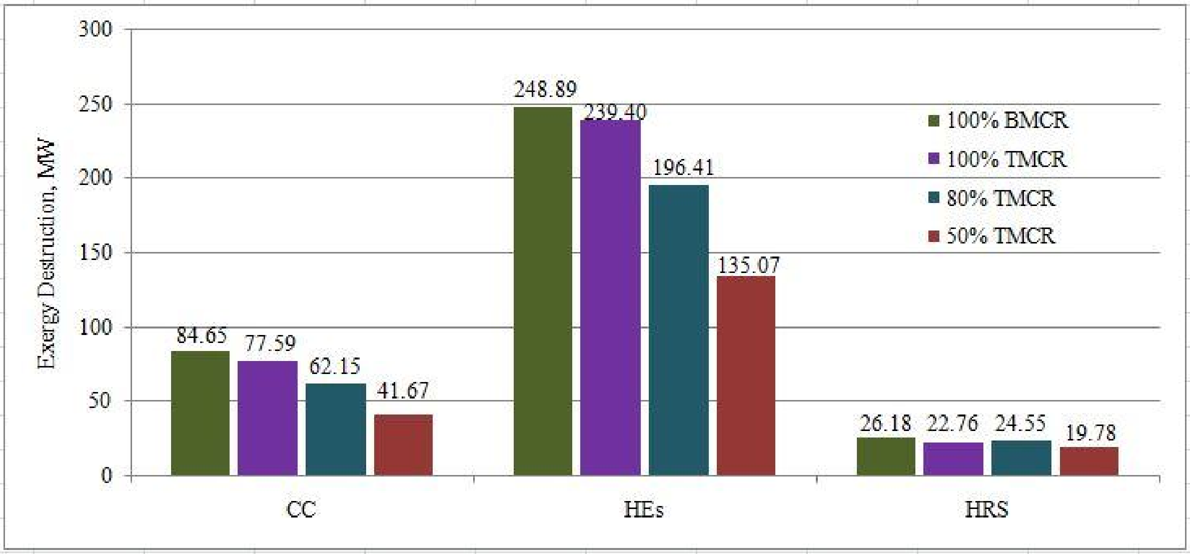


Fig.14 Subsystem wise exergy destruction at different loads

V. CONCLUSION

In this paper thermodynamic analysis of a reheat high pressure boiler is conducted. Exergy efficiency is found less than the energy efficiency for all the components in all the loading conditions and this difference is higher in the components of the region where temperature gradient of two fluid streams is more. Accordingly, the maximum exergy loss is in evaporator having highest temperature difference between hot gas stream and feed water. Thus, an attempt should be made to further increase the feed water temperature. Next to the evaporator is the combustion chamber where maximum exergy loss occurs and possible reasons could be the irreversibility in chemical reaction, incomplete combustion, evaporation of H_2 and moisture in coal etc. Therefore, reducing the chemical irreversibility and optimizing the excess air level will be helpful in improving combustion chamber exergy efficiency.

VI. FUTURE SCOPE OF WORK

This work was focussed to energy and exergy analysis of steam generating unit. A similar analysis for turbine cycle of plant (i.e. remaining components of the plant) should be conducted so that a thorough picture about saving potential and optimization of plant performance can be obtained.

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