

Assessment of thermodynamic irreversibility in different zones of a heavy fuel oil fired high pressure boiler

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Abstract This paper presents a comprehensive second law-based thermodynamic analysis of a heavy oil-fired high-pressure boiler used in a steam power plant. The entire boiler region is divided into four zones to determine the magnitudes of energy loss and irreversibility taking place in these zones so that measures may be taken to reduce them. The energy analysis reveals that the maximum loss of combustion energy of the fuel takes place in the last zone where the exhaust gasses leave the chimney and taken away by these gases, whereas the exergy analysis reveals that the irreversibility rate in the first zone, i.e., furnace region, is the highest as compared to that in other zones. In this zone, 29.26 % of the fuel exergy is destroyed, while the energy loss is zero. In the chimney zone, where the energy loss is the maximum, the exergy destruction is just 4.64 %. Therefore, the real loss is actually taking place in the boiler furnace due to the generation of entropy. The chemical reactions between the fuel and air are the main cause of irreversibility. Decreasing either the percentage of excess combustion air or the exhaust gas temperature or both will cause the irreversibility rate in the furnace to decrease.

Keywords Net calorific value · Standard chemical exergy · Heavy fuel oil · Boiler · Irreversibility

List of symbols

\bar{C}_p^h	Mean molar isobaric heat capacity for enthalpy of a gas/kJ kmol ⁻¹ K ⁻¹
\bar{C}_p^ψ	Mean molar isobaric exergy capacity of a gas/kJ kmol ⁻¹ K ⁻¹
FD	Forced draft
$g_1, g_2 \dots g_6$	Nodal points in air and gas passage
h	Specific enthalpy/kJ kg ⁻¹
h_{ph}	Specific physical enthalpy/kJ kg ⁻¹
h_d	Specific enthalpy of devaluation/kJ kg ⁻¹
\bar{h}_{ph}	Molar specific physical enthalpy/kJ kmol ⁻¹
H	Enthalpy/kJ
ID	Induced draft
i	Irreversibility rate/kW
\bar{h}_d^0	Standard molar specific enthalpy of devaluation/kJ kmol ⁻¹
m	Mass of a component in 100 kg mass of fuel/kg
M	Molecular mass
\dot{m}	Mass flow rate/kg s ⁻¹
n	Number of kmols of a component in 100 kg mass of fuel
\dot{n}	Molar flow rate/kmol s ⁻¹
$(NCV)^0$	Net calorific value of fuel/kJ kg ⁻¹
p	Absolute pressure/Pa
\dot{Q}	Heat-transfer rate/kW
\bar{R}	Universal gas constant (=8.31451 kJ kmol ⁻¹ K ⁻¹)
s	Specific entropy/kJ kg ⁻¹ K ⁻¹
T	Absolute temperature/K
w_1, w_2	Nodal points in the water and steam passage
\dot{W}_s	Shaft work done rate/kW
x	Mole fraction
\dot{X}	Exergy flow rate/kW

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Greek letters

ψ	Specific exergy/kJ kg ⁻¹
ψ_{Ph}	Specific physical exergy/kJ kg ⁻¹
ψ_{Ch}	Specific chemical exergy/kJ kg ⁻¹
$\bar{\psi}$	Molar specific exergy/kJ kmol ⁻¹
$\bar{\psi}_{Ph}$	Molar specific physical exergy/kJ kmol ⁻¹
$\bar{\psi}_{Ch}$	Molar specific chemical exergy/kJ kmol ⁻¹
ψ_{Ch}^0	Standard specific chemical exergy/kJ kg ⁻¹
Δ	Used for a change in any parameter

Subscripts

a	Air
CV	Control volume
d	Devaluation
e	Exit
f	Fuel
g	Gas
g_1, g_2, \dots, g_6	For the nodal points in air and gas passage
i	Inlet
j	A component
P	Product
R	Reactant
w_1, w_2	For the nodal points in the water and steam passage
0	For environmental state
I, II, III, and IV	Zones in the boiler

Superscripts

0	For standard state
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Introduction

Boiler is a major component of a steam power plant. In the boiler, steam is generated at the desired rate and at the desired pressure and temperature by burning fuel in its furnace. The performance of the power plant depends a great deal on the performance of the boiler. It is a complex integration of furnace, evaporator, superheater, economizer, and air preheater along with various auxiliaries such as burners, fans, pumps, and chimney. The demand for the higher efficiency of the boiler and associated plant is continuously increasing due to diminishing fossil fuel resources and growing concerns for the environment. This requires improvement in the boiler design. Before any effort to improve the boiler design is undertaken, a thorough thermodynamic analysis of the boiler under consideration is required to locate the areas of maximum thermodynamic losses. The areas of maximum thermodynamic losses are the areas which have the maximum potential for improvement.

The traditional method of thermodynamic analysis is based on the first law of thermodynamics in which an

energy balance is applied on the system under consideration, e.g., [1]. Energy balances provide no information about internal losses and changes in energy quality during a process. The second law analysis, also called exergy analysis, overcomes this short coming of the first law and is based on the concept of exergy defined as the maximum useful work that can be obtained in a process in which the system comes to equilibrium with the surroundings or attains the dead state [2, 3]. In this analysis, an exergy balance is applied to a process or a system to know how much of the exergy input to the system under consideration has been utilized and how much has been lost. The loss of exergy or irreversibility provides a quantitative measure of process inefficiency.

A number of researchers have made contributions to the theory and practice of exergy analysis [2, 4–12]. Many researchers have applied this analysis to study and improve the thermodynamic performance of steam power plants. Nag and De [13] applied a thermodynamic analysis based on the minimization of entropy generation to the optimal design and operation of a heat recovery steam generator (HRSG) generating saturated steam for a combined gas and steam power cycle. Dincer and Al-Muslim [14] conducted a thermodynamic analysis of a Rankine cycle reheat steam power plant in terms of the first law of thermodynamic analysis (i.e., energy analysis) and the second law analysis (i.e., exergy analysis), using a spreadsheet calculation technique. Sengupta et al. [15] conducted exergy analysis of a coal-based 210-MW thermal power plant. Kopac and Hilalci [16] studied the effect of ambient temperature on the efficiency of the regenerative and reheat Catalagzı steam power plant in Turkey. Aljundi [17] conducted energy and exergy analysis of a 66-MW steam power plant unit in Jordan that uses heavy oil as fuel. Teerawat and Adisorn [18] determined best design and operating conditions for the maximum power plant efficiency of both subcritical and supercritical pulverized coal-fired power plants. Guo and Yang [19] established models for energy analysis, entropy analysis, and exergy analysis for a boiler operating under unsteady conditions. Singh and Kaushik [20] carried out the exergy analysis of a 67.5-MW unit of a pulverized coal-fired steam power plant situated in India. This analysis has also been successfully applied to improve and innovate other power generation systems such as gas turbine cycles [21], gas–steam combined cycle [22], Kalina cycles [23–25], absorption power/cooling combined cycles [26], and cogeneration [27]; and various other systems such as solar-assisted ground-source heat pump greenhouse heating system [28], low-temperature solar thermal electric generation using regenerative organic Rankine cycle [29], solar cookers [30], waste heat recovery system with Canopus heat exchanger in an industrial refrigeration and air-conditioning unit [31], domestic refrigerator using eco-

friendly R290/R600a refrigerant mixture [32], and solar air heater [33].

In the exergy analysis of the coal-fired steam power plant carried out by Singh and Kaushik [20], it was found that the maximum destruction of fuel exergy occurs in the boiler system where about 63 % of fuel exergy is destroyed. Therefore, in this work, a heavy oil-fired high-pressure boiler of a steam power plant is considered and divided into four subregions. Energy and exergy analysis of each sub-region has been carried out separately to determine the magnitude of energy and exergy losses taking place in each subregion. The aim of this study is to identify the locations of thermodynamic inefficiencies in different zones of the considered boiler and to assess their magnitude and causes.

Boiler description

The schematic diagram of the boiler under consideration is shown in Fig. 1. It is a heavy fuel oil-fired, natural circulation, artificial draft, water-tube boiler. Water at 200–250 °C (depending on the operating pressure of the boiler) from the high-pressure feed-water heater enters the economizer and leaves it as a two-phase mixture of low dryness fraction. It then enters the steam drum. From the steam drum, it flows

through insulated downcomers into the headers situated outside the furnace. The whole furnace region is surrounded by tubes backed by refractory walls. The headers are connected to these tubes so that water flows through them and these tubes act as risers. The water in the riser tubes receives heat from the combustion gases mainly by radiation. As the burned gases move upward and secondary air is added, the effect of radiation is reduced and the convection becomes predominant. The heat transfer in this zone takes place by convection as well as radiation. So, radiant and convective evaporators are located in this zone. As the gases travel further, the heat transfer takes place purely by convection and the temperature is the highest here. Therefore, super-heater is located in this zone. As the gases transfer most of their heat to the evaporating water and superheating steam, their temperature reduces. Therefore, in this zone, heat recovery units like economizer and air preheater are located. The circulation of water in the tubes of the boiler is caused by natural convection due to density difference caused by the temperature gradient. The arrangement of parallel tubing is used to reduce the pressure loss and to have better control over the quality of steam. A combination of forced and induced draft is provided by means of two fans known as FD Fan and ID Fan. The specifications and the operating data of the boiler are presented in Table 1.

Fig. 1 Schematic diagram of the boiler plant

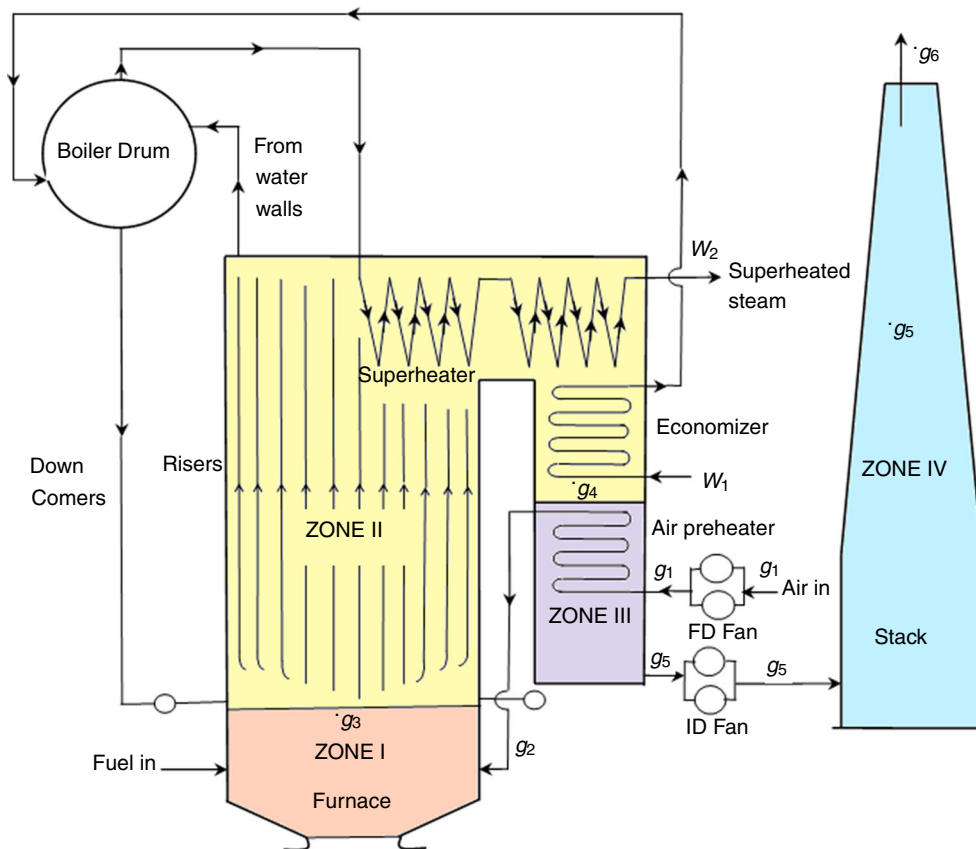


Table 1 Specifications of the boiler, its mountings and accessories, and operating data

Operating data	Value
Boiler type	Heavy oil-fired water-tube boiler
Steam-generation capacity	275 t h ⁻¹
Design pressure	110 kg cm ⁻² (g)
Superheater outlet pressure	95 kg cm ⁻² (g)
Superheater type	Two-stage superheater
Temperature at superheater outlet	520 °C
Economizer type	Plain tube
Fuel used	Heavy fuel oil
Density of fuel at 15 °C	0.9705 g mL ⁻¹
Ultimate analysis of fuel	Carbon 83.7 %, hydrogen 10.15 %, sulfur 3.76 %, oxygen 1.2 %, nitrogen 1.15 %, ash content 0.04 %
Mass flow rate of fuel	5 kg s ⁻¹
F.D. Fan motor power	400 kW
I.D. Fan motor power	400 kW
Feed pump motor power	2000 kW
Temperature of water entering the economizer	221 °C
Pressure of water entering the economizer	105 kg cm ⁻² (g)
Temperature of air entering the air preheater	25 °C
Pressure of air entering the air preheater	1.01326 bar
Ambient pressure	1.01325 bar
Ambient temperature	25 °C

Net calorific value and standard chemical exergy of fuel

Number of kmols of various components in 100 kg mass of fuel

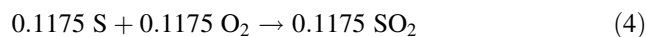
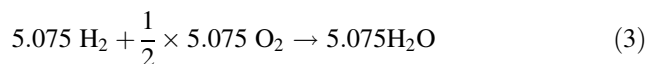
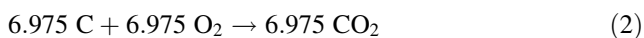
Number of kmols of a component in 100 kg mass of fuel is given by

$$n = \frac{m}{M} \quad (1)$$

Using the equation above, the number of kmols of C, H₂, S, O₂, and N₂ in 100 kg mass of fuel is calculated as 6.975, 5.075, 0.1175, 0.0375, and 0.041, respectively.

Chemical equations for complete combustion of fuel

The chemical equations for the combustion of combustible components of fuel can be written as



Stoichiometric air–fuel ratio

Theoretical oxygen to be supplied by air
= Oxygen required for complete combustion of fuel
– Oxygen already present in the fuel

Air contains 21.0 % oxygen and 79 % nitrogen by volume. Therefore, for each kmol of oxygen, 3.76 kmol of nitrogen is involved.

Therefore, the number of kmols of air required for complete combustion of 100 kg of fuel can be given as,

$$n_{\text{air}} = (1 + 3.76) \times \text{Number of kmols of oxygen to be supplied by air}$$

Therefore, the theoretical mass of air required for complete combustion of 100 kg of fuel can be given as,

$$m_{\text{air}} = n_{\text{air}} \cdot M_{\text{air}} \quad (5)$$

Theoretical air–fuel ratio on a mass basis can be given as,

$$\text{AF}_{\text{theoretical}} = \frac{m_{\text{air}}}{100} \quad (6)$$

Actual air–fuel ratio

The boiler uses 30 % excess air to ensure complete combustion of fuel.

Therefore,

$$\text{Actual air – fuel ratio} = 1.3 \times \text{AF}_{\text{theoretical}}$$

Mass flow rate of air

Mass flow rate of air,

$$\dot{m}_a = \text{AF}_{\text{actual}} \times \dot{m}_f \quad (7)$$

Number of kmols of products for the combustion of 100 kg of fuel

From the combustion equations, for 100 kg of fuel, $n_{\text{CO}_2} = 6.975$, $n_{\text{H}_2\text{O}} = 5.075$, and $n_{\text{SO}_2} = 0.1175$ kmol.

The boiler uses 30 % excess air. Therefore, O₂ in the products of combustion is given by,

$$n_{\text{O}_2} = 0.3 \times \text{No. of kmol of oxygen to be supplied by air theoretically} \\ = 2.878 \text{ kmol}/100 \text{ kg of fuel}$$

Nitrogen does not take part in the combustion. Therefore, N_2 in the products of combustion is same as that present in the fuel and incoming air, i.e.,

Net calorific value of fuel

$$\begin{aligned}
 (\text{NCV})^0 &= - \text{Enthalpy of combustion} \\
 &= \Delta H^0 \\
 &= - \left(H_{d,P}^0 - H_{d,R}^0 \right) \\
 &= - \left[\sum_P n_P \cdot \bar{h}_{d,P}^0 - \sum_R n_R \cdot \bar{h}_{d,R}^0 \right] \\
 &= - \left[\left(n_{\text{CO}_2} \cdot \bar{h}_{d,\text{CO}_2}^0 + n_{\text{H}_2\text{O}} \cdot \bar{h}_{d,\text{H}_2\text{O}}^0 + n_{\text{N}_2} \cdot \bar{h}_{d,\text{N}_2}^0 + n_{\text{O}_2} \cdot \bar{h}_{d,\text{O}_2}^0 + n_{\text{SO}_2} \cdot \bar{h}_{d,\text{SO}_2}^0 \right) \right. \\
 &\quad \left. - \left(n_C \cdot \bar{h}_{d,C}^0 + n_{\text{H}_2} \cdot \bar{h}_{d,\text{H}_2}^0 + n_{\text{N}_2} \cdot \bar{h}_{d,\text{N}_2}^0 + n_S \cdot \bar{h}_{d,S}^0 + n_{\text{O}_2} \cdot \bar{h}_{d,\text{O}_2}^0 \right) \right]
 \end{aligned} \tag{11}$$

$$\begin{aligned}
 n_{\text{N}_2} &= 0.041 + 1.3 \times 3.76 \\
 &\times \left(6.975 + \frac{5.075}{2} + 0.1175 - 0.0375 \right) \\
 &= 46.929 \text{ kmol/100 kg of fuel}
 \end{aligned}$$

Mass flow rate of gases

Mass of a gaseous component of the products per 100 kg of fuel is given by,

$$m = n \cdot M \tag{8}$$

Using the equation above, the masses of CO_2 , H_2O , N_2 , O_2 , and SO_2 can be calculated. The total mass of combustion gases formed per 100 kg of fuel is the sum of the masses of all the gaseous components.

Therefore, the mass flow rate of combustion gases is given by,

$$\begin{aligned}
 \dot{m}_g &= \text{Mass of combustion gases formed per kg of fuel} \\
 &\times \dot{m}_f
 \end{aligned} \tag{9}$$

where,

$$\dot{m}_f = \text{Mass flow rate of fuel in kg s}^{-1}$$

Mass of ash/soot

$$\begin{aligned}
 &\text{Rate of ash/soot formed per second} \\
 &= \frac{\text{Ash content of fuel (\% by mass)}}{100} \times \dot{m}_f
 \end{aligned} \tag{10}$$

The net calorific value of fuel is found to be 40.1 MJ kg^{-1} .

Chemical exergy of fuel

For finding the chemical exergy of the fuel, the method explained by Kaushik and Singh [34] has been followed. The ratio of standard specific chemical exergy to the net calorific value for liquid fuels without moisture content is given by [2]

$$\begin{aligned}
 \varphi &= 1.0401 + 0.1728 \frac{h}{c} + 0.0432 \frac{o}{c} \\
 &+ 0.2169 \frac{s}{c} \left(1 - 2.0628 \frac{h}{c} \right)
 \end{aligned} \tag{12}$$

Therefore, the standard specific chemical exergy of the fuel is given by

$$\psi_{\text{Ch},f}^0 = \varphi \cdot (\text{NCV})^0 \tag{13}$$

Using the above relations, the standard specific chemical exergy of fuel is found to be 42.9 MJ kg^{-1} .

Various zones in the boiler

A high-pressure boiler is a very complex assembly of certain components like burners, evaporators, superheaters, economizers, air heaters, and others. The functions of these components are interrelated. Energy and exergy losses in these components are also different. Therefore, for the purpose of accurate analysis and to assess the magnitude and exact location of maximum thermodynamic losses, the entire boiler region has been divided into four zones:

Zone I	This zone includes the furnace region in which the combustion of fuel is assumed to be adiabatic and at constant pressure
Zone II	It is the region of medium to high temperature in which evaporator, superheater, and economizer are located. Here, the heat transfer takes place between the hot products of combustion and water
Zone III	It is the region of comparatively lower temperature in which air preheater is located. Here, the heat transfer takes place between hot gases and air. Pressure remains constant at p^0
Zone IV	This is the region of mixing of the products of combustion with ambient air

Thermodynamic modeling and analysis

Assumptions

1. A steady-state operation of the boiler is considered.
2. The boiler furnace, the evaporator section, and the air preheater section are perfectly insulated from the surroundings so that no heat loss takes place in zones I, II, and III.
3. In zone I, where the furnace is located, the dissociation of the combustion products at high temperature is neglected.
4. The work required by ID and FD fans is neglected.
5. The only loss of heat energy is taking place through the exhaust gases.
6. Pressure drops in pipe lines are neglected.

Zone I

Mass balance

Molar flow rates of the reactants for the given fuel consumption rate and with combustion with 30 % excess air are determined and presented in Table 2.

The mass balance for the control volume (zone I) is given by

$$\sum \dot{m}_R = \sum \dot{m}_P \quad (14)$$

or

$$\sum \dot{n}_R \cdot M_R = \sum \dot{n}_P \cdot M_P \quad (15)$$

By applying the mass balance, the molar flow rate of the components of the products of combustion is determined and presented in Table 3.

Energy balance

The energy balance for any combustion process taking place in a control volume is given by

$$\dot{Q}_{CV} + \dot{m}_f(\text{NCV})^0 = \left(\sum \dot{n}_P \bar{h}_{Ph,P} - \sum \dot{n}_R \bar{h}_{Ph,R} \right) + \dot{W}_s \quad (16)$$

As the reactants are fuel and air, Eq. (16) can be rewritten as

$$\dot{Q}_{CV} + \dot{m}_f(\text{NCV})^0 = \left(\sum \dot{n}_P \cdot \bar{h}_{Ph,P} - \dot{n}_f \cdot \bar{h}_{Ph,f} - \dot{n}_a \cdot \bar{h}_{Ph,a} \right) + \dot{W}_s \quad (17)$$

Table 2 Molar flow rates, mole fractions, and mass flow rates of the components of reactants with 30 % excess air

Reactants	Components of fuel					Components of air	
	C	H ₂	S	O ₂	N ₂	O ₂	N ₂
$\dot{n}_R/\text{kmol s}^{-1}$	0.3487	0.2537	0.0059	0.001875	0.00205	0.62351	2.3444
x_R	0.56956	0.41439	0.009636	0.003062	0.003348	0.2100838	0.78991
$\dot{m}_R = \dot{n}_R \cdot M_R / \text{kg s}^{-1}$	4.185	0.5075	0.188	0.06	0.0574	19.9524	65.6433

Table 3 Molar flow rates, mole fractions, and mass flow rates of the components of the products of combustion with 30 % excess air

Product	CO ₂	H ₂ O	SO ₂	O ₂	N ₂	Total
$\dot{n}_P/\text{kmol s}^{-1}$	0.3487	0.2537	0.00587	0.1438	2.34645	3.0987
x_P	0.1125	0.08187	0.00189	0.0464	0.7572	1.00
$\dot{m}_P = \dot{n}_P \cdot M_P / \text{kg s}^{-1}$	15.345	4.5675	0.376	4.6044	65.7006	90.5935

For zone I, there is no heat transfer across the boundary of the control volume and no shaft work is done, i.e., $\dot{Q}_{CV} = 0, \dot{W}_s = 0$.

As fuel is supplied at standard temperature ($T^0 = 298.15$ K) and standard pressure ($p^0 = 0.101325$ MPa), $\bar{h}_{Ph,f} = 0$

Therefore, the energy equation becomes

$$\dot{m}_f(\text{NCV})^0 = \sum \dot{n}_P \cdot \bar{h}_{Ph,P} - \dot{n}_a \cdot \bar{h}_{Ph,a} \tag{18}$$

or

$$\dot{m}_f(\text{NCV})^0 = (T_{g_3} - T_0) \sum \dot{n}_P \cdot \bar{C}_{p,P}^h - (\dot{n}_{O_2,\text{air}} + \dot{n}_{N_2,\text{air}}) \cdot (T_{g_2} - T_0) \cdot \bar{C}_{p,a}^h \tag{19}$$

Therefore,

$$T_{g_3} = T_0 + \frac{\dot{m}_f(\text{NCV})^0 + (\dot{n}_{O_2,\text{air}} + \dot{n}_{N_2,\text{air}}) \cdot (T_{g_2} - T_0) \cdot \bar{C}_{p,a}^h}{\sum \dot{n}_P \cdot \bar{C}_{p,P}^h} \tag{20}$$

As isobaric heat capacity of a gas is a function of temperature, its value will vary as the temperature varies. Therefore, to get accurate result, the above equation is to be solved by iteration. In any iteration, the value of isobaric heat capacity for a particular gas is to be taken corresponding to the value of temperature. We have to keep on doing iterations till the difference between two successive values of temperature is negligibly small.

By iteration procedure, we get

$$T_{g_3} = 2144.63 \text{ K}$$

Values of ‘molar isobaric heat capacity for enthalpy’ for different gases at different temperatures are obtained from [2].

Exergy balance

The exergy balance for any control volume is given by

$$\sum_i \dot{m}_i \psi_i + \sum_j \left(1 - \frac{T_0}{T_j}\right) \dot{Q}_{CV,j} = \sum_e \dot{m}_e \psi_e + \dot{W}_s + \dot{I} \tag{21}$$

For the zone I, $\dot{Q}_{CV} = 0, \dot{W}_s = 0$. The reactants (fuel and air) enter the control volume at inlet, and the products of combustion leave at exit.

Therefore, Eq. (21) can be written as

$$\dot{m}_f \cdot \psi_f + \dot{m}_a \cdot \psi_{g_2} = \sum_P \dot{m}_P \psi_P + \dot{I}_I \tag{22}$$

Therefore, irreversibility rate in zone I is given by

$$\begin{aligned} \dot{I}_I &= \dot{m}_f \cdot \psi_f + \dot{m}_a \cdot \psi_{g_2} - \sum_P \dot{m}_P \psi_P \\ &= \dot{m}_f \cdot \psi_f + \dot{n}_a \cdot \bar{\psi}_{g_2} - \sum_P \dot{n}_P \cdot \bar{\psi}_P \end{aligned} \tag{23}$$

Specific exergy of fuel Specific exergy of the fuel is the sum of specific physical and chemical exergies,

$$\psi_f = \psi_{Ph,f} + \psi_{Ch,f} \tag{24}$$

As fuel is supplied at environmental state (p_0, T_0), its physical exergy is zero, i.e., $\psi_{Ph,f} = 0$ As $T_f = T_0 = T^0 = 298.15$ K and $p_f = p_0 = p^0 = 0.101325$ MPa, the specific chemical exergy of fuel will be its standard specific chemical exergy, i.e., $\psi_{Ch,f} = \psi_{Ch,f}^0$

Therefore,

$$\psi_f = \psi_{Ch,f} = \psi_{Ch,f}^0 \tag{25}$$

Specific exergy of air Molar specific exergy of air at inlet to zone I is the sum of specific physical and chemical exergies

$$\bar{\psi}_{g_2} = \bar{\psi}_{Ph,g_2} + \bar{\psi}_{Ch,g_2} \tag{26}$$

As air consists of reference gases such as $N_2, O_2,$ and CO_2 which are at their partial pressures, its molar specific chemical exergy is zero, i.e., $\bar{\psi}_{Ch,g_2} = 0$

The molar specific physical exergy of air at inlet to zone I is given by [2]

$$\bar{\psi}_{Ph,g_2} = \bar{C}_{p,g_2}^\psi (T_{g_2} - T_0) + \bar{R}T_0 \ln\left(\frac{p_{g_2}}{p_0}\right) \tag{27}$$

Since, pressure of air at inlet to zone I, i.e., $p_{g_2} = p_0 = p^0 = 0.101325$ MPa, the second term on the right-hand side of Eq. (27) will become zero.

Specific exergy of products of combustion Molar specific exergy of products of combustion is the sum of specific physical and chemical exergies

$$\bar{\psi}_P = \bar{\psi}_{Ph,P} + \bar{\psi}_{Ch,P} \tag{28}$$

The molar specific physical exergy of products of combustion at exit from zone I is given by [2]

$$\begin{aligned} \bar{\psi}_{Ph,g_3} = \bar{\psi}_{Ph,P} &= (T_{g_3} - T_0) \sum_P x_P \cdot \bar{C}_{p,P}^\psi + \bar{R}T_0 \ln\left(\frac{p_{g_3}}{p_0}\right) \\ &= (T_{g_3} - T_0) \left[x_{CO_2} \cdot \bar{C}_{p,CO_2}^\psi + x_{H_2O} \cdot \bar{C}_{p,H_2O}^\psi \right. \\ &\quad \left. + x_{SO_2} \cdot \bar{C}_{p,SO_2}^\psi + x_{O_2} \cdot \bar{C}_{p,O_2}^\psi + x_{N_2} \cdot \bar{C}_{p,N_2}^\psi \right] \\ &\quad + \bar{R}T_0 \ln\left(\frac{p_{g_3}}{p_0}\right) \end{aligned} \tag{29}$$

Since combustion is taking place at constant pressure $p_2 = p_0 = 0.101325$ MPa, in Eq. (29), $\bar{R}T_0 \ln\left(\frac{p_{g_3}}{p_0}\right) = 0$.

Values of 'molar isobaric exergy capacity' for different gases at different temperatures are obtained from [2].

The molar specific chemical exergy of products is given by [2]

$$\begin{aligned} \bar{\psi}_{Ch,P} &= \sum_P x_P \cdot \bar{\psi}_{Ch,P} + \bar{R}T_0 \sum_P x_P \cdot \ln x_P \\ &= [x_{CO_2} \cdot \bar{\psi}_{Ch,CO_2} + x_{H_2O} \cdot \bar{\psi}_{Ch,H_2O} + x_{SO_2} \cdot \bar{\psi}_{Ch,SO_2} \\ &\quad + x_{O_2} \cdot \bar{\psi}_{Ch,O_2} + x_{N_2} \cdot \bar{\psi}_{Ch,N_2}] \\ &\quad + \bar{R} \cdot T_0 [x_{CO_2} \cdot \ln x_{CO_2} + x_{H_2O} \cdot \ln x_{H_2O} \\ &\quad + x_{SO_2} \cdot \ln x_{SO_2} + x_{O_2} \cdot \ln x_{O_2} + x_{N_2} \cdot \ln x_{N_2}] \end{aligned} \quad (30)$$

Values of standard molar specific chemical exergy of components of the products of combustion are obtained from [2].

Zone II

Mass balance

The mass balance or equation of continuity for the zone II is given by For the gas passage,

$$\dot{m}_{g_3} = \dot{m}_{g_4} = \dot{m}_P \quad (31)$$

For the water passage,

$$\dot{m}_{w_1} = \dot{m}_{w_2} = \dot{m}_W \quad (32)$$

Energy balance

The energy balance for the control volume with no chemical reaction is given by

$$\dot{Q}_{CV} + \sum \dot{m}_i h_i = \sum \dot{m}_e h_e + \dot{W}_s \quad (33)$$

or

$$\dot{Q}_{CV} + \sum \dot{m}_i (h_{Ph} + h_d)_i = \sum \dot{m}_e (h_{Ph} + h_d)_e + \dot{W}_s \quad (34)$$

or

$$\begin{aligned} \dot{Q}_{CV} + \sum \dot{m}_i h_{Ph,i} + \sum \dot{m}_i h_{d,i} \\ = \sum \dot{m}_e h_{Ph,e} + \sum \dot{m}_e h_{d,e} + \dot{W}_s \end{aligned} \quad (35)$$

Since it is purely a physical process and there is no change in chemical composition of the two fluid streams, there will be no change in the enthalpy of devaluation of fluids between inlet and exit, i.e., $h_{d,i} = h_{d,e}$

Therefore, Eq. (35) becomes

$$\dot{Q}_{CV} + \sum \dot{m}_i h_{Ph,i} = \sum \dot{m}_e h_{Ph,e} + \dot{W}_s \quad (36)$$

or

$$\begin{aligned} \dot{Q}_{CV} + \sum \dot{m}_{g_3} h_{Ph,g_3} + \dot{m}_{w_1} h_{Ph,w_1} \\ = \sum \dot{m}_{g_4} h_{Ph,g_4} + \dot{m}_{w_2} h_{Ph,w_2} + \dot{W}_s \end{aligned} \quad (37)$$

For the zone II, $\dot{Q}_{CV} = 0$, $\dot{W}_s = 0$

$$\therefore \sum \dot{m}_{g_3} h_{Ph,g_3} + \dot{m}_{w_1} h_{Ph,w_1} = \sum \dot{m}_{g_4} h_{Ph,g_4} + \dot{m}_{w_2} h_{Ph,w_2} \quad (38)$$

From Eqs. (31), (32), and (38), we get

$$\sum \dot{m}_P h_{Ph,g_3} + \dot{m}_W h_{Ph,w_1} = \sum \dot{m}_P h_{Ph,g_4} + \dot{m}_W h_{Ph,w_2} \quad (39)$$

or

$$\sum \dot{n}_P \bar{h}_{Ph,g_3} + \dot{m}_W h_{Ph,w_1} = \sum \dot{n}_P \bar{h}_{Ph,g_4} + \dot{m}_W h_{Ph,w_2} \quad (40)$$

or

$$\begin{aligned} (T_{g_3} - T_0) \sum \dot{n}_P \cdot \bar{C}_{p,g_3}^h + \dot{m}_W h_{Ph,w_1} \\ = (T_{g_4} - T_0) \sum \dot{n}_P \cdot \bar{C}_{p,g_4}^h + \dot{m}_W h_{Ph,w_2} \end{aligned} \quad (41)$$

$$\therefore T_{g_4} = T_0 + \frac{(T_{g_3} - T_0) \sum \dot{n}_P \cdot \bar{C}_{p,g_3}^h + \dot{m}_W (h_{Ph,w_1} - h_{Ph,w_2})}{\sum \dot{n}_P \cdot \bar{C}_{p,g_4}^h} \quad (42)$$

By iteration procedure, we get

$$T_{g_4} = 490.416 \text{ K}$$

Exergy balance

The exergy balance for any control volume is given by Eq. (21)

For the zone II, $\dot{Q}_{CV} = 0$, $\dot{W}_s = 0$. Therefore, the exergy balance equation for this zone can be written as

$$\sum_i \dot{m}_i \psi_i = \sum_e \dot{m}_e \psi_e + \dot{I}_{II} \quad (43)$$

or

$$\dot{m}_{g_3} \psi_{g_3} + \dot{m}_{w_1} \psi_{w_1} = \dot{m}_{g_4} \psi_{g_4} + \dot{m}_{w_2} \psi_{w_2} + \dot{I}_{II} \quad (44)$$

From Eqs. (31), (32), and (44), we get

$$\dot{m}_P \psi_{g_3} + \dot{m}_W \psi_{w_1} = \dot{m}_P \psi_{g_4} + \dot{m}_W \psi_{w_2} + \dot{I}_{II} \quad (45)$$

Therefore, irreversibility rate in zone II is given by,

$$\begin{aligned} \dot{I}_{II} &= \dot{m}_P \psi_{g_3} - \dot{m}_P \psi_{g_4} + \dot{m}_W \psi_{w_1} - \dot{m}_W \psi_{w_2} \\ &= \dot{n}_P (\bar{\psi}_{g_3} - \bar{\psi}_{g_4}) + \dot{m}_W (\psi_{w_1} - \psi_{w_2}) \\ &= \dot{n}_P \left[(\bar{\psi}_{Ph,g_3} + \bar{\psi}_{Ch,g_3}) - (\bar{\psi}_{Ph,g_4} + \bar{\psi}_{Ch,g_4}) \right] \\ &\quad + \dot{m}_W [(\psi_{Ph,w_1} + \psi_{Ch,w_1}) - (\psi_{Ph,w_2} + \psi_{Ch,w_2})] \end{aligned} \quad (46)$$

Since the process taking place in zone II is purely a physical process and there is no change in chemical

composition of the two fluid streams, $\bar{\psi}_{Ch,P_2} = \bar{\psi}_{Ch,P_3}$ and $\psi_{Ch,w_2} = \psi_{Ch,s_3}$.

Therefore,

$$\dot{I}_{II} = \dot{n}_P (\bar{\psi}_{Ph,g_3} - \bar{\psi}_{Ph,g_4}) + \dot{m}_w (\psi_{Ph,w_1} - \psi_{Ph,w_2}) \quad (47)$$

Equation (29) has been used for finding the molar specific physical exergy of products of combustion at inlet to zone II. The same equation can be used for finding the molar specific physical exergy of products of combustion at exit from zone with a slight change that the temperature in this case is T_{g_4} .

$$\sum \dot{m}_P h_{Ph,g_4} + \dot{m}_a h_{Ph,g_1} = \sum \dot{m}_P h_{Ph,g_5} + \dot{m}_a h_{Ph,g_2} \quad (53)$$

or

$$\sum \dot{n}_P \bar{h}_{Ph,g_4} + \dot{n}_a \bar{h}_{Ph,g_1} = \sum \dot{n}_P \bar{h}_{Ph,g_5} + \dot{n}_a \bar{h}_{Ph,g_2} \quad (54)$$

or

$$\begin{aligned} (T_{g_4} - T_0) \sum \dot{n}_P \cdot \bar{C}_{p,g_4}^h + (\dot{n}_{O_2} + \dot{n}_{N_2})(T_{g_1} - T_0) \cdot \bar{C}_{p,g_1}^h \\ = (T_{g_5} - T_0) \sum \dot{n}_P \cdot \bar{C}_{p,g_5}^h + (\dot{n}_{O_2} + \dot{n}_{N_2})(T_{g_2} - T_0) \\ \cdot \bar{C}_{p,g_2}^h \end{aligned} \quad (55)$$

Therefore,

$$T_{g_5} = T_0 + \frac{(T_{g_4} - T_0) \sum \dot{n}_P \cdot \bar{C}_{p,g_4}^h + (\dot{n}_{O_2} + \dot{n}_{N_2}) [(T_{g_1} - T_0) \cdot \bar{C}_{p,g_1}^h - (T_{g_2} - T_0) \cdot \bar{C}_{p,g_2}^h]}{\sum \dot{n}_P \cdot \bar{C}_{p,g_5}^h} \quad (56)$$

Now,

$$\begin{aligned} \psi_{Ph,w_1} - \psi_{Ph,w_2} = & [(h_{Ph,w_1} - T_0 s_{w_1}) - (h_{Ph,w_0} - T_0 s_{w_0})] \\ & - [(h_{Ph,w_2} - T_0 s_{w_2}) - (h_{Ph,w_0} - T_0 s_{w_0})] \\ = & (h_{Ph,w_1} - h_{Ph,w_2}) - T_0 (s_{w_1} - s_{w_2}) \end{aligned} \quad (48)$$

Zone III

Mass balance

The mass balance or equation of continuity for zone III is given by For the gas passage,

$$\dot{m}_{g_4} = \dot{m}_{g_5} = \dot{m}_P \quad (49)$$

For the air passage,

$$\dot{m}_{g_1} = \dot{m}_{g_2} = \dot{m}_a \quad (50)$$

Energy balance

The energy balance is given by Eq. (36) For the zone III, $\dot{Q}_{CV} = 0$ and $\dot{W}_s = 0$.

Therefore,

$$\sum \dot{m}_i h_{Ph,i} = \sum \dot{m}_e h_{Ph,e} \quad (51)$$

or

$$\sum \dot{m}_{g_4} h_{Ph,g_4} + \dot{m}_{g_1} h_{Ph,g_1} = \sum \dot{m}_{g_5} h_{Ph,g_5} + \dot{m}_{g_2} h_{Ph,g_2} \quad (52)$$

From Eqs. (49), (50), and (52),

By iteration procedure, we get

$$T_{g_3} = 429.412 \text{ K}$$

Exergy balance

For zone III, $\dot{Q}_{CV} = 0$ and $\dot{W}_s = 0$. Therefore, Eq. (43) is applicable for this zone also which can be further written as

$$\dot{m}_{g_4} \psi_{g_4} + \dot{m}_{g_1} \psi_{g_1} = \dot{m}_{g_5} \psi_{g_5} + \dot{m}_{g_2} \psi_{g_2} + \dot{I}_{III} \quad (57)$$

From Eqs. (49), (50), and (57),

$$\dot{m}_P \psi_{g_4} + \dot{m}_a \psi_{g_1} = \dot{m}_P \psi_{g_5} + \dot{m}_a \psi_{g_2} + \dot{I}_{III} \quad (58)$$

Therefore, irreversibility rate in zone III is given by,

$$\begin{aligned} \dot{I}_{III} = & \dot{m}_P (\psi_{g_4} - \psi_{g_5}) + \dot{m}_a (\psi_{g_1} - \psi_{g_2}) \\ = & \dot{n}_P (\bar{\psi}_{g_4} - \bar{\psi}_{g_5}) + \dot{n}_a (\bar{\psi}_{g_1} - \bar{\psi}_{g_2}) \\ = & \dot{n}_P [(\bar{\psi}_{Ph,g_4} + \bar{\psi}_{Ch,g_4}) - (\bar{\psi}_{Ph,g_5} + \bar{\psi}_{Ch,g_5})] \\ & + \dot{n}_a [(\bar{\psi}_{Ph,g_1} + \bar{\psi}_{Ch,g_1}) - (\bar{\psi}_{Ph,g_2} + \bar{\psi}_{Ch,g_2})] \end{aligned} \quad (59)$$

Since the process taking place in zone III is purely a physical process and there is no change in chemical composition of the two fluid streams, $\bar{\psi}_{Ch,P_3} = \bar{\psi}_{Ch,P_4}$ and $\bar{\psi}_{Ch,a_1} = \bar{\psi}_{Ch,a_2}$.

Therefore,

$$\dot{I}_{III} = \dot{n}_P (\bar{\psi}_{Ph,g_4} - \bar{\psi}_{Ph,g_5}) + \dot{n}_a (\bar{\psi}_{Ph,g_1} - \bar{\psi}_{Ph,g_2}) \quad (60)$$

The molar specific physical exergy of products of combustion at inlet and exit from zone III can be calculated by Eq. (29) by using temperatures T_{g_4} and T_{g_5} , respectively.

As air enters zone III at environmental state (p_0, T_0), its physical exergy is zero, i.e., $\bar{\psi}_{Ph,g_1} = 0$. The molar specific physical exergy of air at exit from zone III can be calculated by Eq. (27).

Zone IV

In this zone, all the exergy of gases given by \dot{X}_{g_5} is lost through dissipation (mixing, cooling, etc.).

Hence, irreversibility rate in zone IV is given by,

$$\dot{I}_{IV} = \dot{X}_{g_5} = \dot{m}_p \cdot \psi_{g_5} \tag{61}$$

Therefore, total irreversibility rate in the boiler is given by,

$$\dot{I}_{Boiler} = \dot{I}_I + \dot{I}_{II} + \dot{I}_{III} + \dot{I}_{IV} \tag{62}$$

Energy and exergy efficiencies of the boiler

Energy efficiency of the boiler is given by

$$\begin{aligned} \eta_{th} &= \frac{\text{Heat added to water in the boiler}}{\text{Lower heating value of fuel}} \\ &= \frac{\dot{m}_w (h_{Ph,w_2} - h_{Ph,w_1})}{\dot{m}_{fuel} (NCV)^0} \end{aligned} \tag{63}$$

Since the function of the boiler is to increase the exergy of the stream of water by taking exergy of the fuel as input, its exergy efficiency is given by

$$\begin{aligned} \eta_{ex} &= \frac{\dot{X}_{steam} - \dot{X}_{water}}{\dot{X}_{fuel}} \\ &= \frac{\dot{m}_w (\psi_{steam} - \psi_{water})}{\dot{m}_{fuel} \cdot \psi_{fuel}} \end{aligned} \tag{64}$$

Results and discussion

The considered boiler was analyzed considering the environmental temperature and pressure of 298.15 K and 100.8 kPa, respectively, as reference conditions. The results of the exergy analyses are summarized in Table 4.

The only loss of heat energy has been assumed to be taking place through the exhaust gases. The energy efficiency of the boiler under consideration is (94.81 %). This efficiency is based on the lower heating value (LHV) of the fuel which assumes that the water vapor formed during combustion leaves as vapor itself so that it takes into account the losses occurring in the boiler furnace due to moisture present in the fuel. The energy analysis reveals

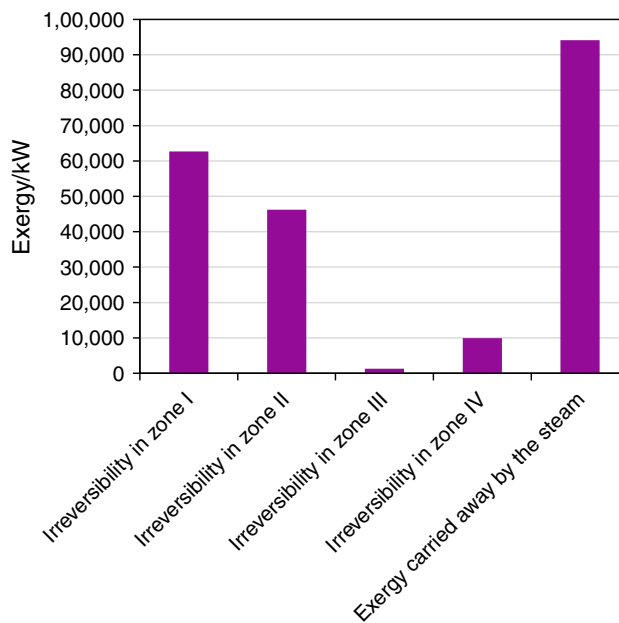
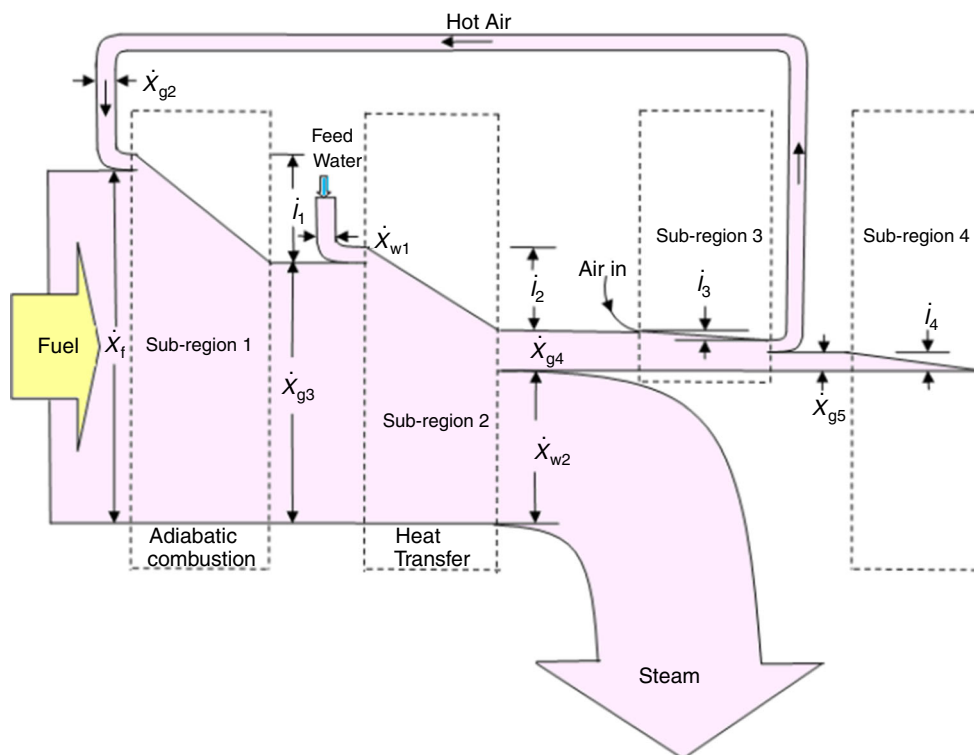


Fig. 2 Irreversibility rates in various zones of the boiler and exergy carried away by the steam

Table 4 Exergy balance of the boiler and the percentage ratio of exergy to fuel exergy input

	kW	Percent ratio
Fuel exergy input	214,313.205	100
Irreversibility rate in the zone I	62,707.91	29.26
Irreversibility rate in the zone II	46,248.462	21.58
Irreversibility rate in the zone III	1,258.4253	0.59
Irreversibility rate in the zone IV	9,940.1437	4.64
Exergy carried away by the stream of boiling water	94,158.256	43.93
Exergy efficiency of the boiler = 43.93 %		

Fig. 3 Grassmann diagram showing exergy flow and irreversibilities in various subregions of the boiler plant



that maximum combustion energy of the fuel is lost in zone IV of the boiler and taken away by the exhaust gases. So, from the point of view of energy analysis, the major loss of energy occurs in zone IV.

Energy efficiency and the energy losses are based on the first law of thermodynamics and do not provide a true picture of the processes taking place in the boiler. This analysis also does not tell anything about the quality of energy going waste along with the exhaust gases. Actually, the loss of energy through exhaust gases is thermodynamically insignificant due to its low quality. Moreover, this loss cannot be avoided completely as the flue gases are to be discharged into the atmosphere at higher temperature to remain above dew point temperature to prevent acid corrosion problems and for proper dispersion of exhaust gases into the atmosphere [35].

The exergy analyses of different zones of the boiler have been carried out at reference environmental condition of $T_o = 298.15$ K and $p_o = 100.8$ kPa. The exergy efficiency of the boiler under consideration is found to be 43.93 %. The irreversibility rates in various zones and exergy efficiency of the boiler expressed as percent of fuel exergy input are presented in Table 4 and are illustrated in the bar diagram in Fig. 2. The exergy flow and irreversibility rates in different zones are shown in the Grassmann diagram in Fig. 3. It is found that the irreversibility rate in the zone I is the highest as compared to that in other zones. In this zone, 29.26 % of fuel exergy is destroyed, while the energy loss is zero. In zone II, the exergy destruction is 21.58 %. In the

chimney zone, where the energy loss is the maximum, the exergy destruction is just 4.64 %. Therefore, the real loss is actually taking place in the boiler furnace due to the generation of entropy. The chemical reactions between the fuel and the air are the main cause of irreversibility. The performance of the boiler can, therefore, be improved by reducing the thermodynamic irreversibility in the furnace and in the vaporization zone.

Conclusions

Fossil fuel resources are being depleted quickly due to increasing energy demand. Therefore, there is a need to use these resources judiciously to extend their life. This is possible by improving the efficiency of fuel-burning systems. Moreover, the pollution hazard arising out of fossil fuel burning has also necessitated the improvement in the efficiency of these systems. In view of this, the designers of the boilers and energy experts should consider the second law or exergy analysis in designing this equipment. This analysis enables one to evaluate quantitatively the causes of thermodynamic irreversibilities of thermal and chemical processes. Poor thermodynamic performance is the results of exergy destructions in combustion and heat-transfer processes. In the present work, the method for the calculation of chemical exergy of heavy fuel oil and the method of carrying out exergy analysis of a boiler have been presented.

In the boiler under consideration, the maximum energy loss was found in the zone IV, while maximum destruction of fuel exergy was found in the zone I (29.26 %) followed by zone II (21.58 %) where the heat energy loss has been assumed to be zero. So, the real loss is taking place in zone I. Therefore, based on the exergy analysis, it can be concluded that the components that should be considered for improvement in the plant are the boiler furnace and the evaporator section. Furnace, being the major source of irreversibility, has the largest improvement potential as compared to other components of the boiler. The evaporator section being the second largest destroyer of exergy also needs careful attention. The chemical reaction between air and fuel in the combustion process is the main source of irreversibility in the boiler furnace. Decreasing the temperature difference between the reactants and the products will cause the irreversibility rate in the boiler furnace to decrease. This can be achieved by decreasing the percentage of excess combustion air and increasing the inlet air temperature further by utilizing the exhaust heat further.

It would be worthwhile to note that the methods and the work presented herein will prove to be useful to the analysts and researchers not only in conducting the exergy analysis of any other boiler but also helping in analyzing other combustion systems as well.

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