Exergetic Analysis of Combustion Chamber of a Combined Heat and Power System

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Abstract
In the present analysis mathematical modeling for a 30MW cogeneration cycle is done and effect of cycle pressure ratio, inlet air temperature and turbine inlet temperature (TIT) is studied for the combustion chamber. Cogeneration is the production of electrical energy and useful thermal energy from the same energy source that is why it is called combined heat and power (CHP) system. From the results it is being found that there is an increase of exergy destruction by 31.30% when the inlet air temperature is increased from 5°C to 50°C. Increased exergy destruction shows that performance of combustion chamber deteriorates with the increase in inlet air temperature. A different pattern for the exergy destruction is observed when compressor pressure ratio is increased. From a compression ratio of 5 to 15 there is a decrease of exergy destruction in combustion chamber and after that it increases. After a compression ratio of 26, performance of system starts deteriorate and regenerator is no longer useful. That is why in the present analysis exergy destruction in the combustion chamber is studied only upto a pressure ratio of 26.

Keywords: cogeneration; compressor; inlet air temperature; pressure ratio; TIT.

1. Introduction
A combustor converts the chemical energy in the fuel to heat energy which is transferred to the working fluid [1-4]. There are several different approaches for modeling of reactors. These include stoichiometric, equilibrium, kinetic, etc. The design of gas turbine combustion system is a complex process involving fluid dynamics, combustion and mechanical design. The most common fuel for gas turbines are liquid petroleum distillates and natural gas. Combustion in the normal, open-cycle, gas turbine is a continuous process in which fuel is burned in the air supplied by the compressor and electric spark is required for initiating the combustion process and thereafter the flame must be self-sustained [5-10]. Gas turbine combustion is a steady flow process in which a hydrocarbon fuel is burned with a large amount of excess air, to keep the turbine inlet temperature at an appropriate value. Now a day’s control of emissions has become the most important factor in the design of industrial gas turbine. Combustion equations express conservation of mass in molecular terms following the rearrangement of molecules during the combustion process. The oxygen required for stoichiometric combustion can be found from the general equation:

\[
C_xH_y + nO_2 \rightarrow aCO_2 + bH_2O
\]

Where
\[
a = x, \quad b = (y/2) \quad \text{and} \quad n = x + (y/4)
\]

Each kilogram of oxygen will be accompanied by \((76.7/23.3)\) kg of nitrogen, which is normally considered to be inert and to appear unchanged in the exhaust; at the temperatures in the primary zone, however, small amount of oxides of nitrogen are formed. The combustion equation assumes complete combustion of carbon to \(CO_2\) but incomplete combustion can result in to small amounts of carbon monoxide and unburned hydrocarbons (UHC) being present in the exhaust [11]. The gas turbine uses a large quantity of excess air, resulting in considerable oxygen in the exhaust; the amount can be deducted from the total oxygen in the incoming air less that required for combustion. Thus the exhaust of any gas turbine consist primary of \(N_2, O_2, CO_2\) and \(H_2O\) and composition can be expressed in terms of either gravimetric (by mass) or molar (by volume) composition. The main factors of importance in assessing combustion chamber performance are (a) pressure loss, (b) combustion efficiency, (c) outlet temperature distribution, (d) stability limits and (e) combustion intensity. Till now not much attention is being paid on the concentration of the different constituents of combustion products [11-16]. In the present analysis a computer program is being executed in the software Engineering Equation Solver (EES) for a 30 MW gas turbine. Results are analyzed for different concentration of combustion products.

2. Mathematical Modeling
The scheme outlined below has been numerically studied using a code developed in EES. The code is based on fundamental thermodynamic relations, including real gas behavior and pressure losses.
In the present system ambient air is coming to air compressor and after compression its temperature and pressure is increased. This compressed air is passed through regenerator. In regenerator compressed air is entering from one side and combustion gases coming out of gas turbine from the other. High temperature combustion gases transfer their heat to the compressed air. After gaining heat this compressed air comes to combustion chamber and fuel is added in it. After burning in air chemical energy of fuel is converted into thermal energy. Temperature of combustion products coming out of combustion chamber depends upon turbine inlet temperature. Combustion product temperature is controlled by making A/F mixture a lean mixture. Gasses coming out from gas turbine are having a large amount of thermal energy. Some part of this thermal energy is transferred to compressed air in regenerator and remaining part is absorbed by high pressure water in steam generator. Flue gas temperature coming out of steam generator is dependent upon the dew point temperature of flue gases. This dew point temperature decides the temperature at which flue gases must enter the stack. Each component is modeled by mass and energy balances. If the system operates in a steady-state, steady flow condition and all the nonreacting gases are arbitrarily assigned as zero thermo mechanical enthalpy, entropy, and exergy at the condition of ambient pressure and temperature regardless of their chemical composition, then the entropy of mixing different gaseous components can be neglected, and the general exergy-balance equation is given by

\[
E_w = \sum_{i=1}^{n} (E_{qi}) + \sum_{i}^{m} ne - \sum_{i}^{m} me - E_D
\]

For single stream flow

\[
E_w = (E_q) + me_{in} - me_{out} - me_{p}
\]

Specific exergy is given by

\[
e = (C_{pa} + wC_{pv})T_a \left( \frac{T}{T_a} - 1 - \ln \frac{T}{T_a} \right) + (1 + \tilde{w})R_a T_a \ln \frac{P}{P_a} + R_a T_a \left[ (1 + \tilde{w}) \ln \left( \frac{1 + \tilde{w}}{1 + \tilde{w}} \right)^{\frac{T_a}{T_a}} \right] + \tilde{w} \ln \left( \frac{\tilde{w}}{T_a} \right)
\]

Where \( \tilde{w} = 1.608w \)

The mass, energy, and exergy balances of the component of the plant are given below.

### 2.1. Air Compressors

The inlet and outlet humidity ratios will be the same. The energy balance yields the compressor work \( W_c \) and compressor outlet temperature.

\[
W_{ct} = W_{t0}
\]

\[
\frac{T_{t0}}{T_{t1}} = \left[ \frac{T_{t1}}{T_{t0}} \right]^{\gamma - 1} \gamma
\]

\[
W_c = h_{t0} - h_{t1}
\]

The exergy balance for the compressor gives the exergy destruction \( e_{DC} \) as following

\[
e_{DC} = W_c + (e_{w} - e_{m})
\]
2.2. Gas Turbines

For the expansion ratio, \( r_e \) temperature at the exit of the turbine isentropic process can be calculated by

\[
T_{go} = T_{go}^{(\frac{\gamma - 1}{\gamma})} r_e
\]

The actual temperature \( T_{go} \) at the exit of the turbine can be calculated by

\[
\eta_t = \frac{T_{go} - T_{go}}{T_{go} - T_{go}}
\]

And the inlet and outlet humidity ratios will be the same

\[
w_{gi} = w_{go}
\]

The energy balance yields the turbine work \( W_T \) given by the following relation

\[
W_T = h_{gi} - h_{go}
\]

The exergy balance for the turbine gives the exergy destruction \( e_{DT} \) as following

\[
e_{DT} = (e_{gi} - e_{go}) - W_T
\]

2.3. Combustor

Mass of fuel supplied can be calculated by energy balance between the energy supplied by the fuel and change in enthalpy of combustion products due to this heat addition. Change in enthalpy of air after adding fuel in combustion chamber is

\[
\eta_c m_f CV = h_{go} - h_{gi} = h_3 - h_2
\]

The exergy balance of the combustion chamber yields exergy destruction

\[
e_{DCC} = m_{jCC} e_{jCC} + e_{ai} - e_{go}
\]

Where

\[
e_{jCC} = \Delta g_f + R_f T_a \ln{\frac{p_f}{p_a}}
\]

and

\[
\Delta g_f = \Delta H_f - T_{av} (s_p - s_R)
\]

\( (s_p - s_R) \) is the entropy change during the combustion process and is given as

\[
s_p - s_R = x_a \left[ C_{pa} \ln{\frac{T_{go}}{T_a}} - R \ln{\frac{p_{go}}{p_a}} \right] + x_c \left[ C_{pa} \ln{\frac{T_{go}}{T_a}} - R \ln{\frac{p_{go}}{p_a}} \right]
\]

where

\[
x_a = \frac{1}{1 + w} \quad \text{and} \quad x_c = \frac{w}{1 + w}
\]

The effect of water vapor present in fuel is neglected and humidity ratio at combustion chamber outlet will be higher than inlet.

2.4. Air Preheater (regenerator)

Temperature of air \( (T_{ao}) \) at the exit of a heat exchanger can be calculated by

\[
\text{Heat Exchanger Effectiveness} = \frac{T_{ao} - T_{in}}{T_{go} - T_{in}}
\]

Applying the energy balance equation on the heat exchanger yields
Here $h_{ra}$ and $h_{rao}$ represents the enthalpy of air at the regenerator inlet and outlet respectively and $h_{rgi}$ and $h_{rgo}$ represents the enthalpy of combustion products at the regenerator inlet and outlet respectively. The exergy balance of the heat exchanger (HE) yields exergy destruction.

Exergy Destruction in HE = $(e_{ai} - e_{ao}) + (e_{gi} - e_{go})$

2.5. Heat Recovery Steam Generator (HRSG)

For regenerator and steam generator simple mass and enthalpy balance equations are used. The amount of heat transferred to water may be calculated as

$$Q_p = h_{gi} - h_{go}$$

Temperature of gas at regenerator outlet may be given as

$$m_p (h_{gi} - h_{go}) = m_u (h_{uo} - h_{wu})$$

Exergy destroyed in HRSG can be calculated by

Exergy Destruction in HRSG = $(e_{gi} - e_{go}) - m_w (e_{wo} - e_{wi})$

Inlet and outlet humidity ratio will remain same.

Mathematical modeling described above is used for the cogeneration cycle. A program is executed in software EES to study cogeneration cycle performance for different parameters. Real gas and water properties are inbuilt in the software. Results obtained are discussed in the following section.

3. Results and Discussion

The conventional definition of efficiency of a combustor indicates how much thermal energy is available for use from the stored chemical energy of the fuel. The losses in a combustor that accounts for the decrease in the efficiency are due to unburnt fuel, incomplete combustion and heat loss to the surrounding across the combustor wall. For typical atmospheric combustion systems, about 1/3rd of the fuel exergy becomes unavailable due to the inherent irreversibilities in the combustor. Most of this irreversibility is associated with the internal heat transfer within the combustor between the products and reactants. Such heat transfer becomes inevitable in both premixed and diffusion flames, where highly energetic product molecules are free to exchange energy with unreacted fuel and air molecules. Lior [15] and Lior et al. [16] outlined the necessity of second-law-based analysis of combustion processes with the following objectives:

(1) Identification of the specific phenomena/processes that have large exergy losses or irreversibilities,

(2) Understanding of why these losses occur,

(3) Evaluation of how they change with the changes in the process parameters and configuration, and

(4) As a consequence of all the above, suggestions on how the process could be improved.

A combustion system, in general, is a multicomponent and multiphase system. The physical processes occurring in the system can be classified broadly in two groups, namely: (i) transport processes and (ii) chemical reactions. The transport processes pertain to the transport of mass momentum and energy, which involve the processes of diffusion and convection of those quantities. The turbulence plays an additional key role by transporting mass momentum and energy through turbulent eddies along with the transport of the quantities through molecular diffusion and flow-aided convection. The transport processes are inherently irreversible due to thermodynamic dissipation in the processes occurring under a finite potential gradient. The oxidation between fuel and oxidizer in a combustion system takes place through a number of reaction steps involving the production of intermediate species in the form of compounds, elements, radicals, molecules and atoms. The rate of any chemical reaction is guided either by the kinetics of the reaction or by the rate of diffusive transport of the reacting molecules to come in contact for possible collision for reaction. One way of analyzing the performance of a combustor is by the exergy balance across the combustor. Considering the fuel and air entering the combustor either separately, or in the form of a mixture, it is possible to calculate the exergy flow rate at the inlet to the combustor. It will comprise the chemical exergy of the fuel and the thermomechanical (or physical) exergy of the fuel and air. With the increase in inlet air temperature exergy destruction in the combustion chamber is increased. In the combustion chamber chemical energy of fuel is converted into thermal energy and some part of Gibbs free energy is lost in the process due to which exergy destruction takes place. As the inlet air temperature (IAT) increases exergy...
destruction in the combustion chamber (CC) increases. There is an increase of exergy destruction by 31.30% when the inlet air temperature is increased from 5°C to 50°C. Increased exergy destruction shows that performance of combustion chamber deteriorates with the increase in inlet air temperature. A different pattern for the exergy destruction is observed when cycle pressure ratio (CR) is increased.

From a compression ratio of 5 to 15 there is a decrease of exergy destruction in combustion chamber and after that it increases. After a compression ratio of 26, performance of system starts deteriorate and regenerator is no longer useful. That is why in the present analysis exergy destruction in the combustion chamber is studied only upto a pressure ratio of 26.

As the inlet air temperature increases fuel consumption also increases. As it may be seen in the figure.3, there is an increase of 11.65% in fuel consumption with an increase in IAT from 5°C to 50°C. As it may be seen from the equation of combustion chamber exergy destruction that as the mass of fuel injected will increase, exergy destruction in combustion chamber will also increase.

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Figure 1: Exergy destruction in combustion chamber with change in IAT

Figure 2: Exergy destruction in combustion chamber with change in CR

Figure 3: Fuel consumption in combustion chamber with change in IAT
Compression ratio of compressor also affects the mass of fuel to be injected in combustion chamber but in an irregular fashion. Upto a pressure ratio of 15 there is decrease in fuel consumption and after that it starts increasing. Similar pattern is observed for the exergy destruction in combustion chamber.

![Figure 4. Fuel consumption in combustion chamber with change in CR](image)

As the turbine inlet temperature increases, exergy destruction in combustion chamber decreases (figure 5). It is due to the reason that, increase in temperature inside the combustion chamber increases the fuel utilization efficiency. Due to increased fuel utilization efficiency, exergy destruction in combustion chamber is decreased. With increase in TIT from 900°C to 1300°C, exergy destruction is decreased by 6.22%.

![Figure 5. Exergy destruction in combustion chamber with change in turbine inlet temperature (°C)](image)

If heat is transferred from a high temperature to lower temperature then its quality goes down and exergy destruction takes place. Exergy (or available energy, or availability) is the maximum useful work that can be extracted from a quantity of energy and refers to the quality of energy. Thus, though the energy is conserved in the process of conversion, its quality deteriorates and less work can be obtained with each conversion. The various irreversible processes encountered within the combustor leads to certain degree of exergy loss. It is observed that the second-law efficiency is the maximum for the stoichiometric supply of air. The lower product gas temperature at the exit for a lean mixture as a result of the excess air supply reduces the second-law efficiency of the combustor. This is despite the fact that all the energy stored in the fuel is contained in the product gas with the complete combustion of the fuel in case of a lean fuel–air mixture in the adiabatic combustor. Therefore, it is clear that neither the completeness of combustion nor the energy content of the product gas determines the exergy-based performance of the combustor. As the product gas temperature at the combustor exit decreases with the increase in the excess supply of air, the maximum ability of it to perform useful work decreases. Therefore, the second-law efficiency decreases. The same decrease in exergetic efficiency is also observed with the insufficient supply of air, when the exit temperature decreases because of the incomplete release of fuel’s stored energy. The maximum efficiency, which is obtained with the stoichiometric
supply of air, may be considered as the ideal situation for every inlet air temperature. This ideal efficiency again increases with the increase in the inlet air temperature which increases the temperature of the product gas.

Adebiyi [17] showed from a similar analysis in an adiabatic combustor, assuming complete combustion without dissociation and constant specific heats of air and product gas, that the maximum second-law efficiency attainable for a combustion engine is 70% with CH₄ as the fuel and keeping the product temperature within the acceptable limit guided by the metallurgical constraint. The second-law efficiency obtained from the exergy approach gives a direct measurement of the performance of the combustor. The larger the irreversibilities, lower will be the value of the second-law efficiency. However, this approach does not throw any light on the causes of the irreversibilities. In a combustor, several transport and chemical processes take place simultaneously, which are irreversible. The contribution of the processes towards entropy generation at different locations within the combustor is required to be known to investigate the primary causes of exergy loss. It is also not practically possible to improve the performance of a combustor without such information.

The present approach fails to provide this data and calls for a methodology, which considers the irreversibility as a field variable to find its distribution within the entire combustor. In the flow field of a combustor, the nonequilibrium conditions are due to the exchange of momentum, energy and mass of different species (multicomponent) within the fluid and at the solid boundaries. These nonequilibrium phenomena cause a continuous generation of entropy in the flow field. The entropy generation is due to the irreversible nature of heat transfer, mass diffusion, viscous effects within the fluid and at the solid boundaries, chemical reaction, coupling effects between heat and mass transfers and body force effects. Every irreversible process can be viewed as the relevant flux driven by the corresponding potential, e.g. the flux of heat is driven by the temperature gradient. The highest exergy exists in the fuel, which represents the maximum potential of the fuel to perform work. When this chemical energy is transformed into thermal energy, some portion (which depends on the final temperature) of the initial availability is destroyed. An amount of this exergy that is destroyed increases for lower final temperatures of the product, i.e. for lower flame temperature. The higher initial temperature of the reactant decreases the destruction in exergy in the combustion process due to the higher temperature of the product, which retains more exergy contained in it.

**Nomenclature**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$E$</td>
<td>Exergy rate (kJ/s)</td>
</tr>
<tr>
<td>$H_f$</td>
<td>Heat supplied by fuel [kJ/kg (dry air)]</td>
</tr>
<tr>
<td>$\Delta H_r$</td>
<td>Heat of reaction of fuel (kJ/kg of fuel)</td>
</tr>
<tr>
<td>$R$</td>
<td>Gas constant (kJ/kg K)</td>
</tr>
<tr>
<td>$T$</td>
<td>Absolute temperature (K)</td>
</tr>
<tr>
<td>$T_P$</td>
<td>Process heat temperature (°C)</td>
</tr>
<tr>
<td>$W$</td>
<td>Work (kJ/kg (dry air))</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat at constant pressure (kJ/kg K)</td>
</tr>
<tr>
<td>$c_v$</td>
<td>Specific heat at constant pressure (kJ/kg K)</td>
</tr>
<tr>
<td>$e$</td>
<td>Specific exergy (kJ/kg (dry air))</td>
</tr>
<tr>
<td>$e_P$</td>
<td>Specific exergy associated with process heat (kJ/kg (dry air))</td>
</tr>
<tr>
<td>$g_r$</td>
<td>Gibbs function of fuel (kJ/kg)</td>
</tr>
<tr>
<td>$h$</td>
<td>Enthalpy (kJ/kg (dry air))</td>
</tr>
<tr>
<td>$h_f$</td>
<td>Enthalpy of saturated water at process steam pressure (kJ/kg)</td>
</tr>
<tr>
<td>$h_g$</td>
<td>Enthalpy of saturated vapor at process steam pressure (kJ/kg)</td>
</tr>
<tr>
<td>$m$</td>
<td>Mass (kg)</td>
</tr>
<tr>
<td>$n$</td>
<td>Number of moles</td>
</tr>
<tr>
<td>$p$</td>
<td>Pressure (bar)</td>
</tr>
<tr>
<td>$Q_P$</td>
<td>Process heat (kJ/kg (dry air))</td>
</tr>
<tr>
<td>$re$</td>
<td>Expansion ratio</td>
</tr>
<tr>
<td>$rp$</td>
<td>Pressure ratio</td>
</tr>
<tr>
<td>$s$</td>
<td>Entropy (kJ/kg K)</td>
</tr>
<tr>
<td>$t$</td>
<td>Temperature (°C)</td>
</tr>
<tr>
<td>$v$</td>
<td>Specific volume (m³/kg)</td>
</tr>
</tbody>
</table>

**Greek Symbols**

- $\omega$ = Humidity ratio (kilogram of water vapor per kilogram of dry air)
- $\phi$ = Relative humidity (%)
- $\varepsilon$ = Effectiveness (%)
- $\eta$ = Efficiency (%)
\( \gamma = \text{Specific heat ratio} \)

\( \lambda = \text{Fuel- air ratio on molar basis} \)

**Subscripts**

- \( C \) = Compressor
- \( CC \) = Combustion chamber
- \( D \) = Destruction
- \( HRSG \) = Heat recovery steam generator
- \( P \) = Product
- \( Q \) = Heat
- \( R \) = Reactant
- \( T \) = Turbine
- \( W \) = Work
- \( a \) = Ambient air
- \( av \) = Average
- \( f \) = Fuel
- \( g \) = Gas
- \( i \) = Inlet
- \( l \) = Liquid
- \( o \) = Outlet
- \( r \) = Regenerator
- \( s \) = Isentropic
- \( v \) = Water vapor
- \( w \) = Water