Research Article

Arzu Keven*

Exergetic performance analyses of three different cogeneration plants

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Abstract: Energy cost and consumption are increasing, so that efficiency becomes more important, from the production to the end users. Energy reserves are limited; increasing demand and environmental concerns make efficiency important. To provide cost-effective and clean power generation, gas turbines and cogeneration are becoming the key technology that must be researched and developed. Three different cogeneration cycles are analyzed by using first and second laws of thermodynamics and exergy analysis method. The three cycles are basic, air heated, and fuel air heated cycles. The performance analysis of the devices such as turbine, recuperator, combustion chamber, compressor, and heat exchanger for the whole cycle is obtained and discussed. For different excess air rates, compression ratios (\( \gamma \)), and ambient (inlet air) temperatures, the Z factor (ratio of lost exergy to useful exergy), specific fuel consumption, and specific work and other performance parameters of the devices were obtained and discussed. It is found that excess air rates have the most effect on the performance of the three cycles.

Keywords: cogeneration, power, air-fuel heating, exergy

1 Introduction

Energy cost and consumption are increasing, so that efficiency becomes more important, from the production to the end users. Energy reserves are limited; increasing demand and environmental concerns make efficiency important. To provide cost-effective and clean power generation, gas turbine and cogeneration are becoming the key technology that must be researched and developed.

Cogeneration is a concept that is used to define the production of electricity and heat in a single process [1–3]. Cogeneration has many advantages over the traditional systems such as fast start-up time, compact size, more economical, higher efficiency, safe and reliable operation, dual fuel capacity, lower weight per unit power, fuel flexibility, and less environmental emissions. Different kinds of fuels like biogas, natural gas, alcohols, biomass, naphtha, mixed fuels, refinery residues are used in gas turbine systems. Improving the performance of gas turbine cogeneration plants is currently an important goal. Publications and studies on development and research studies in this field are about reheating, intercooling, optimization, increasing ignition temperature, better component design, increasing pressure ratio, improving cooling and combustion technologies, using advanced materials, recovery, and integration [4–6].

Cogeneration plants are applied in industry, buildings, and other places. A cogeneration plant suitable for a purpose is preferred according to many criteria such as heat power ratio, efficiency, and temperature. To increase the efficiency, decreasing the auxiliary power consumption, increasing the gas turbine inlet temperature, preheating the fuel and air, cooling the gas turbine, intercooling, using hydrogen-cooled generators, humidified and high inlet air pressure, low compressor inlet air temperature, steam injection, multiple pressure cycles with reheat, and increased excess air ratios are required. The exergy efficiency of gas turbine cogeneration cycles can be increased by reducing exergy destruction and energy losses, some of which can be avoided and others not. Unnecessary heat transfers to the environment and to other components and flows should be avoided. Chemical reactions, preheating the reactants, and using more air than necessary are the main sources of inefficiency. The irreversibility of combustion and heat transfer is greater than the irreversibility of friction, mixing, and unlimited expansion. The use of throttling should be minimized or replaced with turbines for power recovery [7–9].

The heat energy in the exhaust gas of turbines can be used for heating, absorption cooling, or other thermal purposes. One of the most efficient ways to use energy
is cogeneration, which enables the simultaneous production of heat and electricity from the same energy source. Cogeneration systems generally consist of a compressor, a gas turbine, a combustion chamber, and a heat recovery steam generator (HRSG). The energy usage factors of real cogeneration systems are around 85–90%, of which approximately 40% is electricity and 50% is thermal energy [10,11]. Gas turbine cogeneration systems on the market are defined by factors such as pressure ratio, efficiency, exhaust temperature, power output, and ignition temperature. Many research studies are made on finding good evaluation criteria. These studies are often unsatisfactory as there are fewer criteria, parameters, and loops. For a better design and optimization process of cogeneration plants, detailed knowledge of the factors affecting their performance and operating conditions is needed [12–14].

For metallurgical reasons, the maximum temperature must be below a certain value. This can be achieved by using about two to four times the air theoretically required for the complete combustion of the loops. Air properties have a major influence on exhaust gas properties and therefore temperatures drop. Accordingly, when the compression ratio \( (P_2/P_1) \) increases, the compressor outlet temperature \( (T_2) \) increases, and thus, the efficiency increases. High temperature is limited for metallurgical reasons. Therefore, adding a recuperator increases the outlet temperature of the compressor, which increases the efficiency of the cycle. Some of the work produced by the turbine is spent on the compressor. So the compressor work has a very important impact on cycle efficiency [15–17].

In this study, for different excess air rates, compression ratio \( (r) \), and inlet air temperature for bsc, ah and fah cogeneration systems are analyzed. The Z factor (ratio of lost exergy to useful exergy), the SFC (specific fuel consumption), the specific work of all cycles and the turbines, the HRSG, the combustion chambers, and the recuperators efficiencies are obtained, compared, and discussed.

### 2 Materials and methods

Cogeneration plants include some components in which the main component is the turbine. In those components, chemical compositions, pressures, and temperatures are changed. The assumptions made in the analysis of the three cycles are as follows: the air and exhaust are taken as an ideal gas mixture. Works are in a continuous regime. There is no NOx at combustion, and combustion is complete. Natural gas is taken as methane that accepted ideal gas. As heat loss is taken zero in devices except for 2% of the UHV of the fuel during combustion in the combustion chamber. The kinetic-potential energies are ignored [8,18,19].

It is seen from Figure 1 that the air is taken into the compressor and after compressing is sent to the combustion chamber for combustion with natural gas. A part of the exhaust energy is transformed into a generator as mechanical energy in a gas turbine to produce electricity. The remaining energy of the exhaust gases is used in an HRSG component to produce hot water or steam.

In Figure 2, the schema of the air heating (ah) cogeneration plant is given. It is seen that the compressed air is heated by transferring the exhaust energy with a recuperator at the inlet of the combustion chamber. By adding a recuperator at the outlet of the compressor, the combustion chamber’s outlet temperature and the work obtained from the turbine increase.

In Figure 3, the schema of an air fuel heating (fah) cogeneration plant is given. It is shown that the fuel and the air are heated with recuperators. By adding a recuperator at the inlet of the combustion chamber, the combustion chamber’s outlet temperature, the efficiency of the cycle, and the work obtained from the turbine increase.

![Figure 1: Schema of a basic (bsc) cogeneration plant.](image1.png)

![Figure 2: Schema of an air heating (ah) cogeneration plant.](image2.png)

![Figure 3: Schema of an air fuel heating (fah) cogeneration plant.](image3.png)
For steady state and open system, the energy equation:

\[
Q_{CV} - W_{CV} + \sum_{\text{in}} \dot{m}_{\text{in}} \left( h_{\text{in}} + \frac{V_{\text{in}}^2}{2} + g z_{\text{in}} \right) - \sum_{\text{out}} \dot{m}_{\text{out}} \left( h_{\text{out}} + \frac{V_{\text{out}}^2}{2} + g z_{\text{out}} \right) = 0.
\]

The law for steady state conservation mass:

\[
\sum \dot{m}_{\text{in}} = \sum \dot{m}_{\text{out}}.
\]

Efficiency or the overall efficiency of the system:

\[
\eta = \frac{W + Q_{\text{steam}}}{Q_{\text{Fuel,inlet}}}. \tag{3}
\]

Electrical efficiency of the system:

\[
\eta_{\text{el}} = \frac{W}{Q_{\text{Fuel,inlet}}}. \tag{4}
\]

Heat efficiency of the system:

\[
\eta_{\text{heat}} = \frac{Q_{\text{steam}}}{Q_{\text{Fuel,inlet}}}. \tag{5}
\]

In the combustion chamber, the chemical energy of the fuel is converted to thermal energy by chemical reaction. In the calculations, it is taken as the combustion ideally and completely. The chemical reaction in the combustion chamber is:

\[
\hat{\Delta} \text{CH}_4 + (0.7748 \text{N}_2 + 0.2059 \text{O}_2 + 0.0003 \text{CO}_2 + 0.019 \text{H}_2\text{O}) \\
\rightarrow (1 + \hat{\Delta})(\text{X}_N \text{N}_2 + \text{X}_O \text{O}_2 + \text{X}_\text{CO}_2 + \text{X}_\text{H}_2\text{O}_2).
\]

The stoichiometric value of the air is the minimum value of air required to complete the theoretical combustion. However, to complete the combustion, more air is always used than the theoretical amount. The excess air ratio is the rate of the real quantity of air to the theoretical air [20–22]. Availability or exergy is the theoretical maximum quantity of useful work. That can be obtained at the end of a reversible process if equilibrium with the environment is reached. Exergy has two components: physical–chemical [23,24]. For mixed substances, the physical exergy of ideal gas mixtures is as follows:

\[
e_{\text{phy}} = \int_{T_i}^{T_o} \left( \frac{\tilde{e}_{\text{pol}}(T_o) dT}{T_o} - T_{\text{R}} \tilde{R} \frac{P_i}{P_o} \right). \tag{6}
\]

The chemical exergy of gas mixtures is as follows [25,26]:

\[
e_{\text{chem,mix}} = \sum_i x_i e_{\text{chem,i}} + R T_{\text{R}} \sum_i x_i \ln x_i. \tag{7}
\]

For a flow or control mass, the total exergy is as follows:

\[
\tilde{E} = \tilde{E}_{\text{phy}} + \tilde{E}_{\text{chem}}. \tag{8}
\]

For open systems exergy equation:

\[
\sum_i \dot{m}_i h_i - \sum_j T_o S_j - \sum_i \dot{m}_i h_i + \sum_j T_o S_j + \sum_k Q_k - \sum_k T_o \frac{T_j}{T_k}
\]

\[
- \dot{W} = \dot{E}_{\text{loss}}. \tag{9}
\]

In Table 1, the energy, mass, and entropy equations of the devices for basic (bsc) plants are shown. In Table 2,

<table>
<thead>
<tr>
<th>Component</th>
<th>Mass equation</th>
<th>Energy equation</th>
<th>Entropy equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compressor</td>
<td>(\dot{m}_1 = \dot{m}_2)</td>
<td>(\dot{m}_3 h_1 + W_C = \dot{m}_2 h_3)</td>
<td>(\dot{m}_3 s_1 - \dot{m}<em>3 s_2 + S</em>{\text{gen,CC}} = 0)</td>
</tr>
<tr>
<td>Turbine</td>
<td>(\dot{m}_3 = \dot{m}_4)</td>
<td>(\dot{m}_5 h_4 + W_T + W_C + \dot{m}_6 h_4)</td>
<td>(\dot{m}_5 s_3 - \dot{m}<em>5 s_4 + S</em>{\text{gen,T}} = 0)</td>
</tr>
<tr>
<td>HRSG</td>
<td>(\dot{m}_4 = \dot{m}_5)</td>
<td>(\dot{m}_6 h_5 + \dot{m}_7 h_7 = \dot{m}_5 h_5 + \dot{m}_7 h_8)</td>
<td>(\dot{m}_5 s_4 + \dot{m}_5 s_7 - \dot{m}_5 s_5 - \dot{m}<em>5 s_8 + S</em>{\text{gen,HRSG}} = 0)</td>
</tr>
<tr>
<td>Combustion chamber</td>
<td>(\dot{m}_2 + \dot{m}_6 = \dot{m}_3)</td>
<td>(\dot{m}_2 h_2 + \dot{m}_6 h_6 = \dot{m}_3 h_3 + 0.02 \dot{m}_7 \text{LHV})</td>
<td>(\dot{m}_2 s_2 + \dot{m}_6 s_6 - \dot{m}<em>2 s_3 + S</em>{\text{gen,CC}} = 0)</td>
</tr>
<tr>
<td>Overall cycle</td>
<td>(\dot{h}_0 = f(\dot{T}_i))</td>
<td>(\dot{h}_0 = f(T_i, P_i))</td>
<td>(\dot{Q}<em>{\text{loss,CC}} = \dot{Q}</em>{\text{eg,CC}} - \dot{h}<em>{\text{eg,out}} - \dot{h}</em>{\text{eg,in}} - W_f - \dot{m}<em>{\text{steam}}(h</em>{\text{water,in}} - h_{\text{steam,out}}) = 0)</td>
</tr>
<tr>
<td></td>
<td>(\dot{Q}<em>{\text{loss,CC}} = 0.02 \dot{m}</em>{\text{fuel}} \text{LHV}_{\text{CH}_4})</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 3: Schema of an air fuel heating (fah) cogeneration plant.
3 Results and discussion

In this article, the normal conditions are taken as $P_0 = 101.3$ kPa and $T_0 = 25^\circ$C. Compressor’s inlet air mass flow $m_{\text{air}} = 91.3$ kg/s, fuel mass flow is $m_{\text{fuel}} = 1.64$ kg/s, isentropic efficiencies for turbine and compressor are taken as $\eta_{\text{izC}} = \eta_{\text{izT}} = 0.86$, compressor compression rate $r = 10$, the outlet temperature of the recuperator is $T_{\text{recout}} = 850$ K, the produced steam or hot water temperature is $T_{\text{steam}} = 485.57$ K, and the outlet temperature of the HRSG is $T_{\text{exh}} = 426$ K [8,12,17].

In Figure 4, variations of the $Z$ factor (ratio of lost exergy to useful exergy) with the ambient temperature for bsc, ah and fah cogeneration systems are given. It is shown that increasing ambient temperature increases

![Figure 4: Variation of the $Z$ factor (ratio of lost exergy to useful exergy) with the ambient temperature for bsc, ah and fah cogeneration systems.](image)
the Z factor for the three cycles. Decreasing the inlet air ambient temperature of the three cycles from 308 to 275°C, the Z factors decrease by about 0.3% for the bsc cycle and about 2% for the ah and fah cycles, which is important. Increasing inlet air temperatures decrease the cycles’ lost exergy.

In Figure 5, variations of the Z factor with the compression ratio \( (r) \) for bsc, ah and fah cogeneration systems are given. It is shown that increasing the compression ratio \( (r) \) decreases the Z factor of the three plants. Decreasing the compression ratio \( (r) \) of the three cycles from 16 to 5, the Z factors increase by about 8% for bsc cycle and about 11% for ah and fah cycles.

In Figure 6, variations of the Z factor (ratio of lost exergy to useful exergy) with excess air rates for bsc, ah and fah cycles are given. It is shown that increasing

![Graph showing variations of Z factor with the compression ratio](image)

**Figure 5:** Variation of the Z factor with the compression ratio \( (r) \) for bsc, ah and fah cogeneration systems.

![Graph showing variations of Z factor with excess air rates](image)

**Figure 6:** Variation of the Z factor with excess air rates for bsc, ah and fah cogeneration systems.
excess air rates decreases the Z factor of the three cycles. Increasing excess air rates of the systems from 1.3 to 3.5, the Z factors of the cycles increase by about 4.5% for bsc cycle and about 23% for ah and fah cycles. For basic cycle, there is an optimum point at 2.2 excess air rates that shows the minimum lost exergy.

In Figure 7, variations of the SFC-specific fuel consumption with the compression ratio (r) for bsc, ah and fah cogeneration systems are given. That is shown as increasing the compression ratio (r) decreases the SFC of the three cycles. Decreasing the compression ratio (r) of the three cycles from 16 to 5, the Z factors of the systems increase by about 20% for bsc cycle and about 28% for ah and fah cycles.

In Figure 8, variations of the SFC-specific fuel consumption with the ambient temperature for bsc, ah and fah cogeneration plants are given. That is shown as increasing the ambient temperature increases the SFC of the three plants. Decreasing the ambient temperature of the three cycles from 308 to 275°C, the SFC-specific fuel consumption of the cycles decreases by about 6.5% for bsc cycle and about 9% for ah and fah cycles.
In Figure 9, variations of the SFC with excess air rates for bsc, ah and fah cogeneration plants are given. It is shown that increasing the excess air rates decreases the SFC (specific fuel consumption) of the three plants. Decreasing excess air rates of the three cycles from 3.5 to 1.3, the SFC-specific fuel consumptions of the cycles increase by about 52% for bsc cycle and about 135% for ah and fah cycles. For the basic cycle, there is an optimum point at 2.9 excess air rates that shows the minimum specific fuel consumption.

In Figure 10, variations of the specific work with ambient temperature for bsc, ah and fah cogeneration plants are given. That is shown as increasing the ambient temperature decreases the specific work of the three cycles. Decreasing the ambient temperature of the three cycles from 308 to 275°C, the specific work increases by about 7% for bsc cycle and about 11% for ah and fah plants.

In Figure 11, variations of the specific work with the compression ratio ($r$) for bsc, ah and fah cogeneration plants are given. That is shown as increasing the compression ratio ($r$) decreases the specific work of the three plants. Decreasing the compression ratio ($r$) of the three cycles from 16 to 5, the specific work increases by about 7% for bsc cycle and about 13% for ah and fah cycles.

In Figure 12, variations of the specific work with excess air rates for bsc, ah and fah cogeneration plants...
are given. It is shown that increasing excess air rates decreases the specific work of the three cycles. Decreasing excess air rates of the three cycles from 3.5 to 1.3, the specific work increases by about 76% for bsc cycle and about 14% for ah and fah cycles. For excess air rates, 1.5 for bsc cycle and 1.8 for ah and fah cycles, there is a maximum value that should be taken into account.

In Figure 13, variations of the turbine efficiency with excess air rates for bsc, ah and fah cogeneration plants are given. It is shown that increasing excess air rates increases the turbine efficiency of the three cycles. Increasing excess air rates of the three cycles from 1.3 to 3.5, the turbine efficiency increases by about 15% for bsc cycle and about 20% for ah and fah plants.

In Figure 14, variations of the turbine efficiency with ambient temperature for bsc, ah and fah cogeneration plants are given. That is shown as increasing ambient temperature decreases the turbine efficiency of the three cycles.

**Figure 11:** Variation of the specific work with the compression ratio ($r$) for bsc, ah and fah cogeneration plants.

**Figure 12:** Variation of the specific work with excess air rates for bsc, ah and fah cogeneration plants.

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*Arzu Keven*
Decreasing the ambient temperature of the three cycles from 308 to 275°C, the turbine efficiency of the cycles increases by about 1.9% for bsc cycle and about 1% for ah and fah cycles. Increasing ambient temperature decreases the turbine efficiency.

In Figure 15, variations of the turbine efficiency with the compression ratio ($r$) for bsc, ah and fah cogeneration plants are given. It is concluded that increasing the compression ratio ($r$) decreases the turbine efficiency of the three cycles. Decreasing the compression ratio ($r$) of the three cycles from 16 to 5, the turbine efficiency of the cycles decreases by about 9.5% for bsc cycle and about 3.7% for ah and fah cycles.

In Figure 16, variations of the recuperator efficiency with excess air rates for ah and fah cogeneration plants are given. That is shown as increasing excess air rates increases the recuperator efficiency of the ah and fah cycles. Increasing the combustion ratio from 6 to 16 increases the recuperator efficiency by about 10–14% of the ah and fah cycles. Increasing excess air rates of the ah and fah cycles from 1.3 to 3.5, the recuperator efficiency of the cycles for the compressing ratio 16 ($r = 16$) increases

**Figure 13:** Variation of the turbine efficiency with excess air rates for bsc, ah and fah cogeneration plants.

**Figure 14:** Variation of the turbine efficiency with ambient temperature for bsc, ah and fah cogeneration plants.
about 25%, for the compressing ratio 10 \((r = 10)\) increases by about 32.3%, and for the compressing ratio 6 \((r = 6)\) increases about 31.8%.

In Figure 17, variations of the combustion chamber efficiency with excess air rates for bsc, ah and fah cogeneration plants are given. That is shown as increasing the excess air rates decreases the combustion chamber efficiency of the bsc cycle. However, it increases the combustion chamber efficiency of the ah and fah cycles. Increasing the excess air rates of the three cycles from 1.3 to 3.5, the combustion chamber efficiency of the bsc cycle decreases by about 3%, and for ah and fah cycles, the increases are about 1.6% and 0.7%, respectively.

In Figure 18, variations of the HRSG efficiency with excess air rates for bsc, ah and fah cogeneration plants are given. That is shown as increasing the excess air rates
increases the HRSG efficiency cycles. Increasing the excess air rates of the three cycles from 1.3 to 3.5, the HRSG efficiency of the cycles increases by about 34% for bsc cycle and about 44% for ah and 45% for fah cycles.

4 Conclusions

In this study, for different excess air rates, compression ratio ($r$) and inlet air temperature for bsc, ah and fah cogeneration systems are analyzed. The $Z$ factor (ratio...
of lost exergy to useful exergy), the SFC, the specific work of all cycles and the turbines, the HRSG, the combustion chambers, and the recuperators efficiencies are obtained, compared, and discussed.

Excess air rates have the most effect on the Z factor ratio of lost exergy to useful exergy, and compression ratio (r) is more effective than the ambient temperature for bsc, ah and fah cogeneration systems. Decreasing the ambient temperature decreases the Z factor. Decreasing excess air rates increases the SFC-specific fuel consumption by about 50–130%, decreasing compression ratio (r) increases the SFC by about 20–28%, and decreasing the ambient temperature decreases the SFC (specific fuel consumption) by about 7–9%, of the three cycles. Decreasing excess air rates increases the specific work by about 14–76%, decreasing compression ratio (r) increases the specific work by about 7–13%, and decreasing the ambient temperature increases the specific work by about 7–11%, of the three cycles. In addition, for excess air rates 1.5 for bsc cycle and 1.8 for ah and fah cycles, there is a maximum value that should be taken into account. Increasing excess air rates increases the turbine efficiency by about 15–20%, the recuperator efficiency by about 25–32%, the HRSG efficiency by about 34–45% for the three cycles and increases by about 1% for of the ah and fah and decreases about 1% for the bsc cycles combustion chamber efficiency. Decreasing the compression ratio (r) of the three cycles from 16 to 5, the turbine efficiency and the recuperator efficiency decrease by about 4–14%. Decreasing the ambient temperature of the three cycles from 308 to 275°C, the turbine efficiency of the cycles increases by about 1.9% for bsc cycle and about 1% for ah and fah cycles.

In conclusion, decreasing inlet air temperature by absorption cooling or if possible by other methods should be taken into consideration. Also, for optimum working conditions, an optimization process on excess air rates and compression ratio should be carried out.

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References


