Application of exergy balances for the optimization of regenerative gas turbine cycle

Saria Abed *, Tahar Khir

Applied Thermodynamic Research Unit (99/UR/11-21), National School of Engineers of Gabes (ENIG), University of Gabes, Gabes 6029 Gabes, TUNISIA

Abstract: An exergetic and second law efficiency analysis was performed for a 100 MW regenerative gas turbine. The effect of compressor pressure ratio and turbine inlet temperature on exergy destroyed and second law efficiency was investigated using quantitative exergy balance for each component and for the whole system. The results are given graphically with the appropriate discussion and conclusion.

Key words: exergy analysis, optimization, regenerator, gas turbine.

Regenerator model

<table>
<thead>
<tr>
<th>parameter</th>
<th>H</th>
<th>Lₐ</th>
<th>Lₑ</th>
<th>X</th>
<th>Y</th>
<th>E</th>
</tr>
</thead>
<tbody>
<tr>
<td>Value (m)</td>
<td>1.2376</td>
<td>1.004</td>
<td>1.506</td>
<td>0.05</td>
<td>1.76*X</td>
<td>0.0002</td>
</tr>
</tbody>
</table>

* Corresponding author: Saria Abed
E-mail: syria495@gmail.com.
1. Introduction

In gas turbine units, the temperature of the exhaust gases leaving the turbine is often considerably higher than the temperature of compressed air leaves compressor [1,2,3]. Therefore, the high pressure air leaving the compressor can be heated up by transferring heat to it from the hot gases leaving the turbine by a heat exchanger called regenerator before expelling to atmosphere. The thermal efficiency of gas turbine cycle increases as a result of regeneration since the portion of energy of exhaust gases that is normally rejected to the surrounding is now used to preheat the compressed air entering the combustion chamber this, in turn, decrease the mass of fuel required for the same turbine inlet temperature $T_3$ [1,3].

In thermodynamics, the concern is not only for the quantity of energy but also for quality of energy [4]. The first law efficiency does not take into account the quality of energy.

The current work is aimed to analyze the gas turbine efficiency with alternative regenerator from second law. Exergy analysis has been widely applied to gas turbine and gas turbine based power plants. Ebadi and Gorji-Bandpy [5] have performed an exergetic analysis of a gas turbine power plant and showed that the plant’s efficiency and exergy destruction are greatly affected by the variation in turbine inlet gas temperature.

2. Simple gas turbine

In the simple cycle, from the compressor inlet point 1, the ambient air is compressed to reach the high pressure at point 2. No heat is added, however, the compression increases the air temperature, so that at the compressor outlet the air is at high temperature and pressure. Leaving the compressor, the air enters into the combustion chamber where the fuel is injected and the combustion occurs practically at constant pressure. In fact, the temperature of exhaust gas exiting the turbine is usually much higher than the temperature of the air leaving the compressor.

Therefore, the high pressure air leaving the compressor can be heated by heat transfer from the exhaust gas in a heat exchanger acting as a recuperator regenerator. [6].

The GT with regenerator is shown in Figure 1.

The thermal efficiency is

$$\eta_{reg} = \frac{m_p c_p T_3 \left( 1 - \frac{1 - T_f}{1 - T_i} \right) - \Delta h_{fg} \left( T_f - T_i \right)}{m_p c_l e_{cc} - \varepsilon_{min} (T_2 - T_4)}$$

(1)

3. Exergy study of the regenerative cycle

3.1 Exergy in the compressor

For all equipment the exergy efficiency is:

$$\eta_{ex,i} = \frac{E_{i,p}}{E_{i,f}}$$

(2)

We denotes by $p$: product and $f$: fuel.

Then for the compressor:

$$E_{p,c} = \dot{m}_a \left( (c_{p,a} * (T_2 - T_1)) - T_0 \right) * (c_{p,a} * \ln \frac{T_2}{T_3}) - R_a * \ln \left( \frac{P_2}{P_1} \right)$$

(3)

$$E_{f,c} = W_c = \dot{m}_a \left( c_{p,a} * (T_2 - T_1) \right)$$

(4)

3.2 Exergy in the combustion chamber

$$E_{f,cc} = E_{fuel}$$

$$E_{f,cc} = \dot{m}_f * PCI$$

$$E_{p,cc} = E_3 - E_5$$

$$E_5 = \dot{m}_a \left( c_{p,a} * (T_5 - T_0) - T_0 \right) * (c_{p,a} * \ln \frac{T_2}{T_0}) - R \ln \left( \frac{P_5}{P_0} \right)$$

(8)

$$ex_3 = ex_3^{ph} + ex_3^{chim}$$

$$ex_3^{ph} = \dot{m}_g \left( c_{p,g} * (T_3 - T_0) - T_0 \right) * (c_{p,g} * \ln \frac{T_2}{T_0}) - R \ln \left( \frac{P_3}{P_0} \right)$$

(9)

$$ex_3^{chim} = \dot{m}_g \left( c_{p,g} * (T_3 - T_0) - T_0 \right) * (c_{p,g} * \ln \frac{T_2}{T_0}) - R \ln \left( \frac{P_3}{P_0} \right)$$

(10)
4. Numerical simulation

To analyze the influence operating variables on the cycle performances, a simulation code is established using the software EES for a net power of 100 MW. The operating conditions and the air properties are defined according to those usually considered for the STEG (Tunisian Society of Electricity and Gas) power plant. The operating variable ranges are given in Table 1. The figures bellow shows some variation of exergy efficiencies with some operating parameters:

\[ \text{Fig 1. GT with regenerator operating mode} \]

\[
\begin{align*}
\text{ex}_{3}^{\text{chim}} &= \sum_{i=1}^{n} X_i \cdot \text{ex}_{i}^{\text{ch}} + R \cdot T_0 \sum_{i=1}^{n} X_i \cdot \ln(X_i) \\
\text{ex}_{i}^{\text{ch}} &= \mu_{i,0} - \mu_{i}^{e} \quad \text{(11)}
\end{align*}
\]

The reaction of combustion is:
\[ a\text{CH}_4 + b\text{C}_2\text{H}_6 + d\text{C}_3\text{H}_8 + e\text{N}_2 + k\text{CO}_2 + j\text{O}_2 \rightarrow l\text{CO}_2 + m\text{N}_2 + n\text{O}_2 + o\text{H}_2\text{O} + p\text{NO} \quad \text{(13)} \]

4.3 exergy in the regenerator

\[
\begin{align*}
\dot{E}_{p,\text{reg}} &= \dot{E}_5 - \dot{E}_2 \\
\dot{E}_{f,\text{reg}} &= \dot{E}_4 - \dot{E}_6 \quad \text{(14, 15)}
\end{align*}
\]

3.4 exergy in the turbine

\[
\begin{align*}
\dot{E}_{p,t} &= \dot{W}_t \quad \text{(16)}
\end{align*}
\]

\[
\begin{align*}
\dot{E}_{f,t} &= \dot{E}_3 - \dot{E}_4 \quad \text{(17)}
\end{align*}
\]

3.5 exergy of the cycle

\[
\eta_{\text{ex,cycle}} = \frac{\dot{W}_{\text{net}}}{\dot{E}_{\text{fuel}}} \quad \text{(18)}
\]


### Table 1. range of variable of operating parameters

<table>
<thead>
<tr>
<th>Variables</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Input temperature</td>
<td>10-45 °C</td>
</tr>
<tr>
<td>Air humidity ratio</td>
<td>0.45-0.5</td>
</tr>
<tr>
<td>Pressure rate</td>
<td>10 – 16</td>
</tr>
<tr>
<td>Hot gas temperature</td>
<td>800-1600</td>
</tr>
</tbody>
</table>

Fig4. variation of exergy efficiency with pressure ratio.

### 5. Conclusions

A numerical simulation is performed for a gas turbine of 100 MW of net power with steam injection device. According to obtained results, following remarks can be pointed out:

- The exergy efficiency of the cycle decreases with the increase of ambient temperature
- The exergy efficiency of the cycle increases as the pressure ratio increases and the combustion temperature.
- The combustion chamber has the lower exergy efficiency.

### References