A thermodynamic optimization of a gas turbine cycle with a supplementary regenerator

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Abstract. International concern regarding pollution related to maritime transport sector and energy efficiency desiderate have determined specialists to search for more suitable alternatives to internal combustion engines. Due to their advantages, gas turbine systems are met in different marine propulsion applications, such as passenger ships or military vessels. Gas turbines will be even more intensively used in the future, an outcome of this reality being the need of their performance increment. A direction to be followed, considered in this study, is the combination between the improvement of the gas turbine efficiency by including a regenerator in the layout of the plant and, also, the rise of the temperature at the inlet of the turbine (TIT). Will be considered an open cycle gas turbine. The thermodynamic analysis is about the assessment of different influences such as: regenerative effectiveness versus the heat added in the combustor, regenerative effectiveness versus the work consumed by the compressor, regenerative effectiveness versus the thermal efficiency, regenerative effectiveness versus the specific fuel consumption, temperature at inlet of the turbine (TIT) versus the thermal efficiency. The results obtained will indicate the fact that both actions will lead to a gain in the performance of this technology.

1. Introduction
In present times, gas turbines, which are a type of internal combustion engines, are commonly spread technologies; they use the energy from burnt gases and air mixture that spin a turbine to produce power.

In order to obtain a working fluid with a high pressure, requested for the expansion, it is needed the presence of a compressor. Most of the times, are in use axial compressors or centrifugal compressors, since the amount of working fluid and speed are more.

In the original Brayton cycle, air is compressed by the help of a reciprocating piston, the process being considered to be isentropic (theoretically). This compressed air is led to a mixing chamber, where it is introduced the fuel.

The heated and pressurized air-fuel mixture is burned in an expansion cylinder, resulting released energy. Heated air and burning products will expand within a piston/cylinder (theoretically considered at constant entropy); part of the work extracted by the piston/cylinder will drive the compressor by the help of a crankshaft system.

The international concern regarding pollutant emissions decrease and energy efficiency increase made specialist to search for new technologies able to substitute or, at least, to diminish the use of conventional piston engines; one of these technologies being gas turbines [1].
Gas turbines are environment friendly, and cheap; they also show the following advantages: high flexibility, low complexity and small sizes, multi fuel capability, an acceptable volume to power ratio than piston engines [2].

These are some factors affecting the economics of power plants based on gas turbines: low cost of fuel, efficiency in operation, low maintenance cost and first cost [3].

The efficiency of the gas turbine is affected by the atmospheric air temperature, meaning that an increase of this temperature has a negative impact on the performance of the system [4-6].

Open and closed gas turbine cycles might be met in use. In the first case, air is mixed with combustion products, resulted from the combustion chamber – which is also the hot source, the resulted working fluid is expanded in the turbine; after this, the thermodynamic system is released in the atmosphere – which is the cold source.

For the closed cycle gas turbines, the hot source (combustion chamber) and cold source (atmospheric air) are featured by the presence of heat transfer surfaces. Thus, the working fluid has no direct contact with burning products or with the cooling agent.

Because the amount of working fluid is kept constant within the cycle, might be used more expensive gases, having superior properties compared with the air.

Marine application of gas turbines refers to some merchant ships (such as oil tankers or passenger ships) and aircraft carriers.

In the actual framework, various studies led to different manners of improving the efficiency of these systems [7].

Reheating – enables the increase of the thermal efficiency and regeneration has the same goal as reheating; also efficiency increase is reached through out combined cycle-meaning a gas turbine together with a steam turbine.

This paper deals with regeneration – which is involving a regenerator (in fact a heat exchanger) able to extract the heat from the burning gases evacuated by the turbine. The second working fluid in the regenerator is the compressed atmospheric air.

The position of the regenerator in the layout of the plant is between the compressor and the combustion chamber [8, 9].

In order to achieve supplementary performance improvement, in this study will be also discussed the increment of the temperature at the turbine inlet (TIT).

Will be found that the increase of both regenerator efficiency and temperature at the turbine inlet will be reflected in the improvement of the gas turbine system efficiency.

2. Methods and materials

Our thermodynamic analysis is developed on the basis of the schematic layout of the gas turbine with a supplementary regenerator plant, provided in figure 1.

In this study the regenerator is a counter flow heat exchanger, meaning that the two fluids move in opposite directions. The operation of this plant is based on the cycle given in figure 2.

The ideal cycle is modelled as an adiabatic reversible compression (1-2) and expansion (3-4), at constant entropy (as sein in figure 2). The real cycle results in an increase in the entropy, due to irreversibilities, as sein for the processes 1-2’ and 3-4’.

In order to perform the mentioned analysis, it is introduced the following thermodynamic model-provided through out equations (1) - (13), according to [10, 11]:

- specific heat of air at constant pressure, depending on the environment temperature ($T_0$) is given by equation (1):

$$c_{pair} = \left(1.0189 \cdot 10^3\right) - \left(0.13784 T_0\right) + \left(1.9843 \cdot 10^{-4} T_0^2\right) + \left(4.2399 \cdot 10^{-7} T_0^3\right) - \left(3.7632 \cdot 10^{-10} T_0^4\right)$$

- specific heat of burning gases at constant pressure is given by equation (2):

$$c_{p_{bg}} = 0.0086 c_{p_{N_2}} + 0.7154 c_{p_{O_2}} + 0.0648 c_{p_{Ar}} + 0.1346 c_{p_{O_2}} + 0.0666 c_{p_{CO_2}}$$

(2)
• efficiencies of compressor, turbine and regenerator are given by equations (3), (4) and (5):

\[ \eta_{comp} = \frac{T_2 - T'_1}{T'_2 - T'_1} \]  

(3)

\[ \eta_{turb} = \frac{T_3 - T'_4}{T_3 - T'_4} \]  

(4)

\[ \varepsilon_{reg} = \frac{T_3 - T'_2}{T_4 - T'_2} \]  

(5)

• specific work consumed by the compressor is: given by equation (6)

\[ l_{comp} = c_{pair} \frac{k_a - 1}{\beta k_a - 1} \frac{\beta}{\eta_{comp}} \]  

(6)

where:
\( \beta \) - compression ratio
\( k_a \) - adiabatic coefficient of air

• specific work produced by the turbine is given by equation (7):

\[ l_{turb} = c_{pbg} \frac{T_3}{\eta_{turb}} \left( 1 - \frac{1}{\beta} \right) \]  

(7)

where:
\( k_g \) - adiabatic coefficient of burning gases

• net specific work is assessed with equation (8):

\[ l_{net} = l_{turb} - l_{comp} \]  

(8)

• power output is estimated with equation (9):

\[ P = m_{air} \cdot l_{net} \]  

(9)

where:
\( m_{air} \) - mass flow of air

• air-fuel ratio is calculated with equation (10):

\[ AIRFR = \frac{c_{pbg} (T_3 - c_{pair} T_2)}{\eta_b \cdot LCV - c_{pbg} T_3} \]  

(10)

where:
\( \eta_b \) - efficiency of burning
\( \eta_b \in (0.98 - 0.99) \)

LCV - lower calorific value of fuel

• specific fuel consumption is given by equation (11):
\[ S_{FC} = \frac{3600}{AIRFR \cdot l_{net}} \]  

(11)

- thermal efficiency is given by equation (12):

\[ \eta = \frac{l_{net}}{q_{intr}} \]  

(12)

where:

- \( q_{intr} \) – the heat introduced in the cycle, it is given by equation (13):

\[
q_{intr} = c_{pg} \left[ T_4 - T_1 \left( 1 - \varepsilon_{reg} \right) \left( 1 + \frac{\beta (k_a - 1) / k_a - 1}{\eta_{comp}} \right) \right. \\
- \varepsilon_{reg} T_4 \left[ 1 - \eta_{turb} \left( 1 - \frac{1}{\beta (k_g - 1) / k_g} \right) \right] \right]
\]  

(13)

\[ \begin{align*}
\text{Figure 1.} & \quad \text{Schematic representation of a gas turbine with a supplementary regenerator power plant.} \\
\text{Figure 2.} & \quad \text{T-s diagram of the cycle: (1253461) \quad \text{ideal cycle; (12'534'61) \quad \text{real cycle.}}}
\end{align*} \]

3. Results and discussions

The following results have been obtained for a compression ratio of 30, an efficiency of the turbine of 90\%, an efficiency of the compressor first of 80\% and second of 84\%, temperature values at turbine inlet are in the range (133-1500) K; the fuel considered is diesel.

The following figures are given to depict the dependence between heat introduced and efficiency of regenerator (figure 3), thermal efficiency and efficiency of regenerator (figure 4), specific fuel consumption and efficiency of regenerator (figure 5), temperature at the turbine inlet and thermal efficiency (figure 6), specific fuel consumption and temperature at turbine inlet (figure 7).
Figure 3. Dependence between $q_{intr} - \varepsilon_{reg}$.

Figure 4. Dependence between $\eta_t - \varepsilon_{reg}$.

Figure 5. Dependence between $S_pFC - \varepsilon_{reg}$. 
The efficiency of the regenerator variation affects the amount of heat introduced in the cycle as well the thermal efficiency and the specific fuel consumption.

More specifically, the increase of the efficiency of the regenerator will decrease the amount of heat; also, when increasing the efficiency of the compressor, values of the heat are also lower. The highest amount of heat (1390 kJ) is introduced on the cycle when the efficiency of the regenerator is smallest (55%) and the efficiency of the compressor is 84%; decreasing the efficiency of the compressor to 80%, the amount of heat will be diminished with 1%.

The increase of the efficiency of regenerator will be, on the other hand, reflected in the increase of the thermal efficiency and in the decrease of the specific fuel consumption. Meanwhile the increase of the efficiency of the compressor will lead to better values of the thermal efficiency and lower fuel consumptions. For the highest value considered for the efficiency of the regenerator (85%) are obtained the best thermal efficiency (35.1%) and the smallest specific fuel consumption (0.64kg/kgWh), for 84% the efficiency of the compressor; the decrease of this value to 80% will be
reflected in a 4.56% decrease of the thermal efficiency and a 5.03% increase in the specific fuel consumption.

When increasing the temperature at the inlet of the turbine, will be noticed that improved values of the thermal efficiencies are obtained, together with a lower fuel combustion. Rising of the efficiency of the compressor has the same result: better values of the thermal efficiency and lower fuel consumption. For the highest value considered for the temperature at the inlet of the turbine (1500 K) are obtained the best thermal efficiency (36%) and the most convenient specific fuel consumption (0.25 kg/ kgWh) - when the efficiency of the compressor is 84%; the decrease of this efficiency to 80% leads to 8.33% decrease on the thermal efficiency and 13.79% increase in the specific fuel consumption.

4. Conclusions
At international level it is registered a high interest in the increase of power plants efficiency since this increment is reflected in a lower operation cost, due to the diminish of the fuel consumption; but this decrease has as result the diminishment of noxes in the atmosphere.

Results shown in this paper indicate that the inclusion of a regenerator in the gas turbine layout has a benefic output: efficiency improvement. It was observed that the increase of the regenerator efficiency is found in the thermal efficiency increase and specific fuel consumption decrease.

The second improvement direction followed in the paper focuses on the rise of the temperature at the turbine inlet. Thus, was found that a better thermal efficiency is obtained when this temperature reaches high values.

5. References
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