Performances Analysis of a Gas Turbine Plant with Coated Blades

Ion V. Ion[a], Jorge Martins[b], Krisztina Uzuneanu[a] and Anibal Portinha[c]

[a] "Dunărea de Jos" University, Thermodynamics and Heat Engines Dept., 47 Domnească St., 800008 Galati, Romania

[b] University of Minho, Mechanical Engineering Dept., Campus de Azurém, 4800 Guimarães, Portugal

[c] University of Minho, Physics Dept., Campus de Azurém, 4800 Guimarães, Portugal

Abstract

Nowadays with the purposes of further increase the efficiency of gas turbines plants a higher gas turbine inlet temperature is required. This makes the use of new materials essential. Super alloy developments (with directional and single crystal solidification) allow its operation above 1000ºC under higher stresses. The highly thermal loaded, parts of gas turbines are usually protected with a MCrAlY (M-Ni, Co) bond coat, coating which provides oxidation protection and better thermo-mechanical compatibility with a ceramic thermal barrier coating (TBC). Thermal barrier coatings allow higher inlet temperatures for the same cooling rates or even reducing and simplifying the cooling systems. In order to show the effect of thermal barrier coatings application on turbine blades, numerical models were developed that calculate the gain in thermal efficiency, net power and the pollutant emissions of the turbine plants

1. Introduction

There are two possible ways to increase the thermal efficiency of a gas-turbine cycle (assuming the basic efficiency equation $\eta = 1 - \frac{T_{min}}{T_{max}}$ (Carnot efficiency)): decreasing T_{min} and increasing T_{max}. The first is limited by the ambient temperature (T_0) and by the size of the heat exchanger surface. The increase of T_{max} creates strength problem on the blades and vanes material. It is possible to avoid these problems by keeping the blades and vanes material temperature below T_{max}, using coated blades and vanes or/and cooling these elements by air from the air compressor [1]. When the air cooling technique is used, a higher combustion temperature may be used, being compensated by the cooling effect of the cooling air.

Zirconia (ZrO_2) stabilized with 8 wt% Y_2O_3 is the most common material to be applied in thermal barrier coatings (TBC's) owing to its excellent properties: low thermal conductivity; high toughness and thermal expansion coefficient similar to the substrate and intermediate metallic coatings; good wear and erosion resistance and a good resistance to thermal shocks. TBC's improve performance at high temperatures, allowing higher inlet temperatures for the same cooling rates or even reducing and simplifying the cooling systems [2-3]. Also, with the increase of hot gas temperature some benefits concerning reduction of emissions like the unburned hydrocarbons and CO are expected. However, the nitrogen oxide (NO_x) emissions would tend to increase. Nowadays the requests for emissions reduction (nitrogen oxides, carbon oxide and unburned hydrocarbons) are very important. The increase of the operative temperatures can improve the specific fuel consumption but also may increase the NO_x formation. The abatement of NO_x emissions, can be achieved by the control of flame temperature (staged combustion, injection of water and steam into the flame), by improving the combustors design or by catalytic reactions.

In order to show the effect of thermal barrier coatings application on turbine blades, numerical models were developed that calculate the gain in thermal efficiency and net power of the turbine plants. The application of thermal barrier coatings allows a reduction of the temperature on metal parts, increasing their life time and improving the gas turbines performances.

2. The physical model

To perform the thermodynamic analysis of a gas turbine plant it was considered the gas turbine plant with regeneration shown in Figure 1, which works under the Joule-Brayton cycle (also shown in Fig. 1) and has the combustion chamber and the first turbine stages cooled by air drawn from the air compressor. The plant produces electricity using methane (natural gas) as fuel.

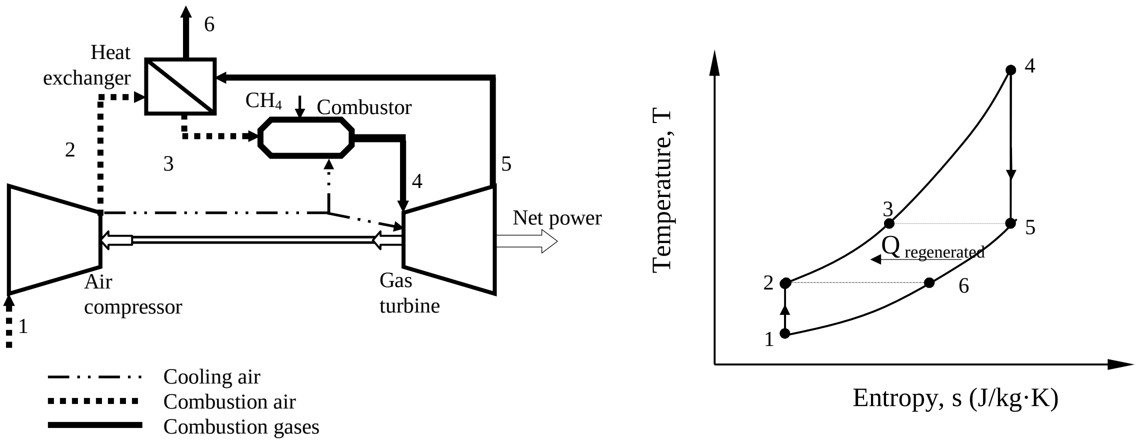

Fig. 1. Gas turbine plant with heat regeneration.

3. The thermodynamic model

In this model the following assumptions are made: the working fluids (air and combustion gases) are considered ideal gases with constant specific heats; all the components are adiabatic; different mass flow rates through the compressor (just air) and the gas turbine (combustion gases) are considered, the difference being equal to the mass flow rate of the fuel.

Air compressor power:

$$\dot{W}_{AC} = \dot{m}_a \cdot (1+\beta) \cdot c_{p,a}\left(T_2 - T_1\right), \quad \text{kW} \tag{1}$$

where: $T_2 = T_1\left\{1+\frac{1}{\eta_{AC}}\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma_a-1}{\gamma_a}}-1\right]\right\}$; $\dot{m}_a$ -combustion air mass flow rate, kg/s; β=(13-15%)-air fraction used for the cooling of combustion chamber (β_{CC} =1.6%), rotor blades (β_{RB}=2·1.9%), stator blades (β_{SB}=2·2.5%) and turbine disc (β_{TD}=0.5%); $c_{p,a}$=1.004 kJ/(kg·K), air specific heat; γ_a=1.4- air specific heat ratio; R_a=0.287kJ/(kg·K) - gas constant for air, $\eta_{AC} = 0.8468$, air compressor isentropic efficiency [4].

Combustion chamber mass and heat balances:

$$\dot{m}_g = \dot{m}_a(1+\beta_{CC}) + \dot{m}_f \,;\; \dot{m}_a \cdot c_{p,a}(T_3 + \beta_{CC}T_2) + \eta_{CC} \cdot \dot{m}_f \cdot LHV = \dot{m}_g \cdot c_{p,g}T_4 \tag{2}$$

where: $p_4 = p_3(1-\Delta p)$, $\Delta p = 0.05$; $\eta_{CC} = 0.98$ combustor efficiency [4]; $\dot{m}_g$ - flue gas mass flow rate, kg/s; $\dot{m}_f$ - fuel mass flow rate, kg/s; *LHV*=50 000 kJ/kg (lower heating value of methane); *AFR*=(m_a/m_f)-air-fuel ratio; $c_{p,g}$=1.17 kJ/(kg·K)-flue gas specific heat;R_g=0.29 kJ/(kg·K)-gas constant for flue gas.

Chemical equation for methane combustion with excess air (the dissociation process is not considered):

$$CH_4 + 2(O_2 + 3.76N_2) \rightarrow H_2O + 2CO_2 + 7.52N_2 + 2O_2 \tag{3}$$

Heat exchanger balance:

$$\dot{m}_a c_{p,a}\left(T_3 - T_2\right) = \dot{m}_{g1} c_{p,g}\left(T_5 - T_6\right);\; T_3 - T_2 = \varepsilon_{HE}\left(T_5 - T_2\right) \tag{4}$$

where: γ_g=1.33 - flue gas specific heat ratio; $\eta_{GT} = 0.8468$ - gas turbine isentropic efficiency [4]; ε_{HE} =0.8-heat exchanger effectiveness [5]; $\dot{m}_{g1} = \dot{m}_g + \beta \cdot \dot{m}_a$; $T_5 = T_4\left\{1-\eta_{GT}\left[1-\left(\frac{p_4}{p_5}\right)^{\frac{1-\gamma_g}{\gamma_g}}\right]\right\}$.

Gas turbine power:

$$\dot{W}_{GT} = \dot{m}_g c_{p,g}(T_4 - T_5), \quad \text{kW} \tag{5}$$

(to simplify the model it is assumed that the cooling air for blades and vanes does not expand in turbine)

Net power:

$$\dot{W}_{net} = \dot{W}_{GT} - \dot{W}_{AC}, \quad \text{kW} \tag{6}$$

First-law efficiency of the gas turbine plant:

$$\eta_I = \frac{\dot{W}_{net,out}}{\dot{Q}_{in}} = \frac{\dot{W}_{net}}{\dot{m}_f LHV} = \frac{\dot{W}_{GT} - \dot{W}_{AC}}{\dot{m}_f LHV} \tag{7}$$

For a complete thermodynamic analysis it is necessary to apply the second law of thermodynamic, too. In this way we can identify the mechanisms and system components that are responsible for thermodynamic losses and to see the extent of these losses (generated entropy or destroyed exergy). To calculate the total entropy generation rate in the entire system we can precede to summing up the entropy generated due to each particular irreversibility mechanism. This approach enables observing the direct interaction between various design concepts and the irreversibilities. For the total system considered in Figure 2, the entropy generation occurs in air compression, combustion gases expansion, heat transfer, combustion and in combustion gases mixing with the atmosphere (external irreversibility).

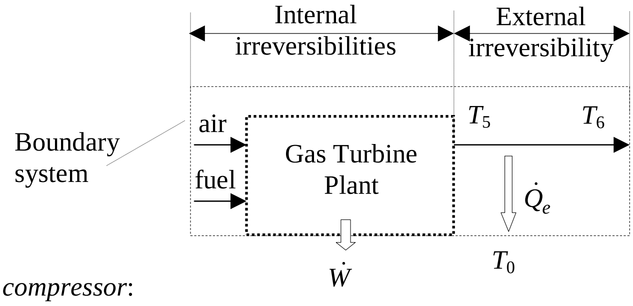

Fig. 2. Schematic diagram of the total system.

The entropy generation rate associated with the air compressor:

$$\dot{S}_{gen,AC} = \dot{m}_a (1+\beta)(s_2 - s_1) = \dot{m}_a (1+\beta) c_{p,a} \ln\left\{1 + \frac{1}{\eta_{AC}}\left[\left(\frac{p_2}{p_1}\right)^{\frac{\gamma_a - 1}{\gamma_a}} - 1\right]\right\}, \text{ kJ/K} \tag{8}$$

The entropy generation rate in combustion chamber (due to friction, mixing and heat transfer processes that accompany the combustion process):

$$\dot{S}_{gen,CC} = \dot{m}_g s_{g4} - \dot{m}_a s_{a3} - \dot{m}_f s_f - \dot{m}_a \beta_{CC} s_{a2}, \text{ kJ/K} \tag{9}$$

where: s_f - fuel absolute entropy, kJ/(kg·K). It is calculated using the third law of thermodynamics [6]:

$$s_f(T,p) = s_f^0(T,p_0) - R_f \ln\frac{p}{p_0} =$$

$$= 0.51967\left(1.971324\ln T + 7.874586\cdot 10^{-3} T - 0.524296\cdot 10^{-6} T^2 + 8.873728\right) - 0.51967\ln\frac{p}{p_0}$$

s_{aj} (j=2, 3) – air absolute entropy, kJ/(kg·K):

$$s_{aj}(T,p) = -4.7306\cdot 10^{-14} T^4 + 3.9579\cdot 10^{-10} T^3 - 1.3797\cdot 10^{-6} T^2 + 2.9099\cdot 10^{-3} T + 1.089 - 0.287\ln\frac{p}{p_0}$$

s_{gk} (k=4, 5, 6) – combustion gases absolute entropy, kJ/(kg·K):

$$s_{gk} = \sum_l \mu_l s_l(T, x_l p) = \sum_l \mu_l \left[s_l(T,p_0) - R_l \ln\left(\frac{x_l p}{p_0}\right)\right];$$

$$s_l(T,p_0) = R_l\left(c_1 \ln T + c_2 T + \frac{c_3}{2}T^2 + \frac{c_4}{3}T^3 + \frac{c_5}{4}T^4 + c_6\right) \quad \text{(see Table I),}$$

μ_l, x_l – mass fraction and molar fraction of the *l*-th element (l=CO_2, N_2, H_2O, O_2).

Table I – Constants used for the calculation of entropy

	c_1	c_2	c_3	c_4	c_5	c_6	R
CO_2	2.4007797	$8.7350957\cdot 10^{-6}$	$-6.6070878\cdot 10^{-6}$	$2.0021861\cdot 10^{-9}$	$6.3274039\cdot 10^{-16}$	9.6951457	0.189
H_2O	4.0701275	$-1.1084499\cdot 10^{-3}$	$4.1521180\cdot 10^{-6}$	$-2.9637404\cdot 10^{-9}$	$8.0702103\cdot 10^{-13}$	-0.32270046	0.46149
N_2	3.6748261	$-1.2081500\cdot 10^{-3}$	$2.3240102\cdot 10^{-6}$	$-6.3217559\cdot 10^{-10}$	$-2.2577253\cdot 10^{-13}$	2.3580424	0.297
O_2	3.6255985	$-1.8782184\cdot 10^{-3}$	$7.0554544\cdot 10^{-6}$	$-6.7635137\cdot 10^{-9}$	$2.1555993\cdot 10^{-12}$	4.3052778	0.26

The entropy generation rate in the heat exchanger (due to the heat transfer across the temperature gap between the working fluids and due to pressure drop):

$$\dot{S}_{gen,HE} = \dot{m}_a(s_{a3} - s_{a2}) + \dot{m}_{g1}(s_{g6} - s_{g5}) = \dot{m}_a\left(c_{p,a}\ln\frac{T_3}{T_2} - R_a\ln\frac{p_3}{p_2}\right) + \dot{m}_{g1}\left(c_{p,g}\ln\frac{T_5}{T_6} - R_g\ln\frac{p_5}{p_6}\right) \tag{10}$$

The entropy generation rate in gas turbine:

$$\dot{S}_{gen,GT} = \dot{m}_{g1}s_{g5} - \dot{m}_g s_{g4} - \dot{m}_a(\beta_{TD} + \beta_{SB})s_{a2}\text{, kJ/K} \tag{11}$$

The external entropy generation rate:

$$\dot{S}_{gen,ex} = \dot{m}_{g1}\left(c_{p,g}\ln\frac{T_0}{T_6} - R_g\ln\frac{p_0}{p_6} + c_{p,g}\frac{T_6 - T_0}{T_0}\right)\text{, kJ/K} \tag{12}$$

Summing up the equations (8-12) we obtain the entropy generation rate associated with the total system:

$$\dot{S}_{gen,tot} = \dot{S}_{gen,AC} + \dot{S}_{gen,HE} + \dot{S}_{gen,CC} + \dot{S}_{gen,GT} + \dot{S}_{gen,ex}\text{, kJ/K} \tag{13}$$

The available work destroyed through thermodynamic irreversibilites is:

$$\dot{W}_{lost} = T_0\dot{S}_{gen,tot}\text{, kJ/K} \tag{14}$$

The second-law efficiency of the gas turbine plant [1]:

$$\eta_{II} = \frac{\dot{W}_{net}}{(\dot{W}_{net})_{rev}} = \frac{\dot{W}_{net}}{\dot{W}_{net} + \dot{W}_{lost}} \tag{15}$$

4. Pollutant emissions

To predict pollutant emissions it was considered a dry low NO_x combustor. In this combustor the emissions are reduced only by flame temperature control using a staged combustion.
The emissions are estimated using the following semi-analytical correlations [5]:

$$NO_x = \frac{15\cdot 10^{13.75}\cdot\tau^{0.5}\cdot e^{\frac{-71100}{T_{pz}}}}{p_3^{0.05}\left(\frac{\Delta p_{a,APH}}{p_3}\right)^{0.5}},\quad \text{g/kg fuel} \tag{16}$$

$$CO = \frac{0.018\cdot e^{\frac{7800}{T_{pz}}}}{p_3^2\cdot\tau\left(\frac{\Delta p_{a,HE}}{p_3}\right)^{0.5}},\quad \text{g/kg fuel} \tag{17}$$

where: T_{pz} – the adiabatic temperature in the primary zone of the combustor:

$$T_{pz} = A\left(\frac{1}{AFR}\right)^{\alpha}\left(\frac{p_3}{p_0}\right)^{x}\left(\frac{T_3}{T_0}\right)^{y}\Psi^{z}e^{\beta(\sigma+\lambda)^2} \tag{18}$$

p_3 – combustion pressure, kPa; p_0=101.300 kPa; ψ – H/C atomic ratio (for methane ψ=4); x, y, z – quadratic functions of $\frac{1}{AF}$ [7]; A, α, β, λ – constants [7]; τ – residence time in the combustor (τ=0.002s) [7].

5. Results and discussion

As equation (3) shows, we can increase the inlet temperature in the gas turbine (T_4) either by increasing the mass fuel flow rate ($\dot{m}_f$) or by reducing the mass air flow rate ($\dot{m}_a$). In this analysis it was chosen the increasing of gas turbine inlet temperature by variation of mass rate of fuel flow from 1.627 to 8 kg/s. The maximum possible inlet temperature occurs for stoichiometric combustion and depends on the lower heating value (*LHV*) of the fuel. The net output power of the gas turbine plant increases with the increasing of gas turbine inlet temperature (see figure 3), this augmentation being almost linear and for each increase of 100 K it increases nearly 120 MW. The first-law efficiency (η_I) and the second-law efficiency (η_{II}) also increase with the inlet temperature (figure 3). Compared with the first-law efficiency the second law efficiency increases more slowly with the rise of the inlet temperature and it shows higher values.
In Figure 4 we can see the change of first and second-law efficiencies as a function of air-fuel ratio. As predicted, there is an initial raise with a subsequent reduction of efficiencies, as the air-fuel ratio increases.

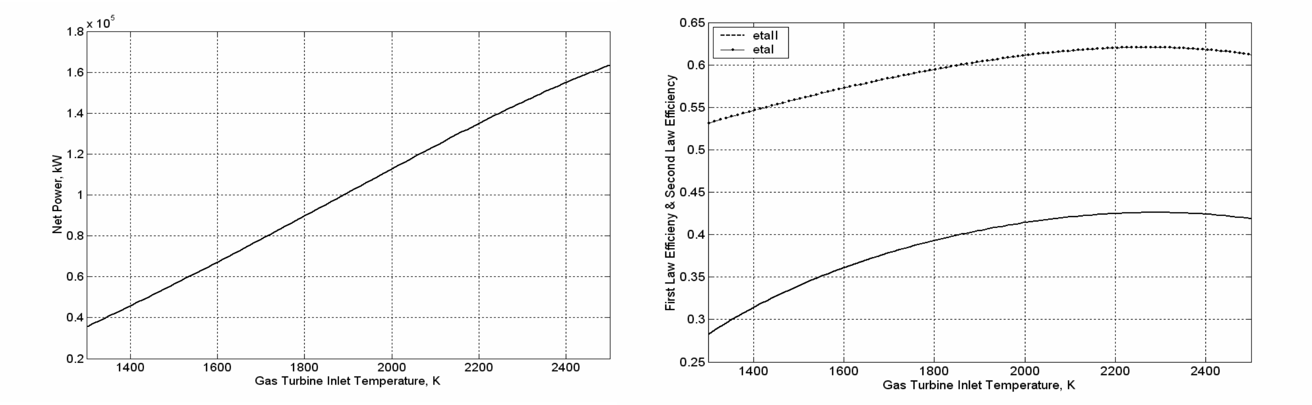

Fig. 3. Variation of the net power $(\dot{W}_{net})$, first-law efficiency (η_I) and second-law efficiency (η_{II}) with gas turbine inlet temperature.

It is well known that the second law analysis offers a better understanding of what is happening during a process and indicates the causes of the inefficiencies with this purpose. A second law analysis was performed for each plant component. Figure 5 shows the variation of entropy generation rate associated with each plant component against the inlet temperature. The entropy generation rate due to heat transfer ($\dot{S}_{gen,HE}$), air compression ($\dot{S}_{gen,AC}$), combustion gas expansion ($\dot{S}_{gen,GT}$) and combustion gases mixing with the atmosphere ($\dot{S}_{gen,ex}$) are negligible compared to the combustion process ($\dot{S}_{gen,CC}$). The entropy generation rate due to combustion gases mixing with the atmosphere increases with the increase of inlet temperature.

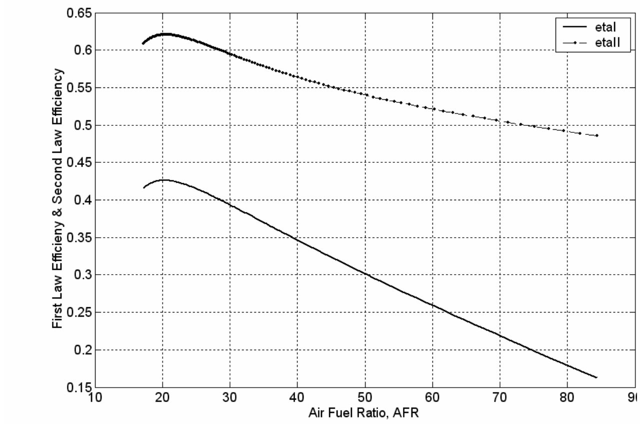

Fig. 4. First-law efficiency (η_I) and second-law efficiency (η_{II}) as a function of air-fuel ratio (AFR).

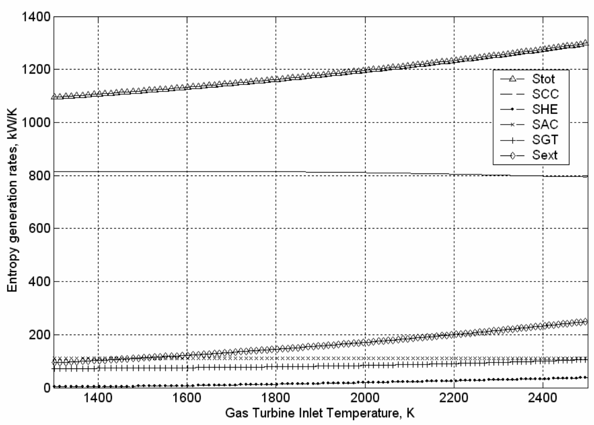

Fig. 5. Total entropy generation rate and entropy generation rate of each component.

We can see in figure 6 that for an increase of gas turbine inlet temperature of 173 K (from 1600 K to 1773 K), the thermal efficiency decreases about 3%. This figure also shows that the η_I decreases from 38.6% to 34.6% at 1600K and from 41.5% to 38,·2% at 1773K, as the fraction of cooling air flow rate changes from 7% to 15%.

The variation of $(\dot{W}_{net})$ with the mass of cooling air can be seen in figure 6 and this power decreases as the fraction of cooling air flow rate raises.

The NO_x and CO emissions, function of the adiabatic flame temperature in the primary zone of the combustor, are plotted in figure 7. As expected the NO_x emissions increase with the combustion temperature. The NO_x increases exponentially with the temperature above about 1750 K. Our results prove this effect, as can be seen in the NO_x curve on figure 7.

Increasing the mass flow rate of blade cooling air promotes an increase in the efficiency and in the net power because it allows higher inlet temperatures by promoting the intensification of heat removal. This raise in inlet temperature is accompanied by an increase in the insulating effect allowing, lower temperatures in the metal parts even for higher gas inlet temperatures.

Other parameters like the pressure ratio in gas turbines and compressors may prove to be more important than the increase in the inlet temperature for the increase of power plants performance.

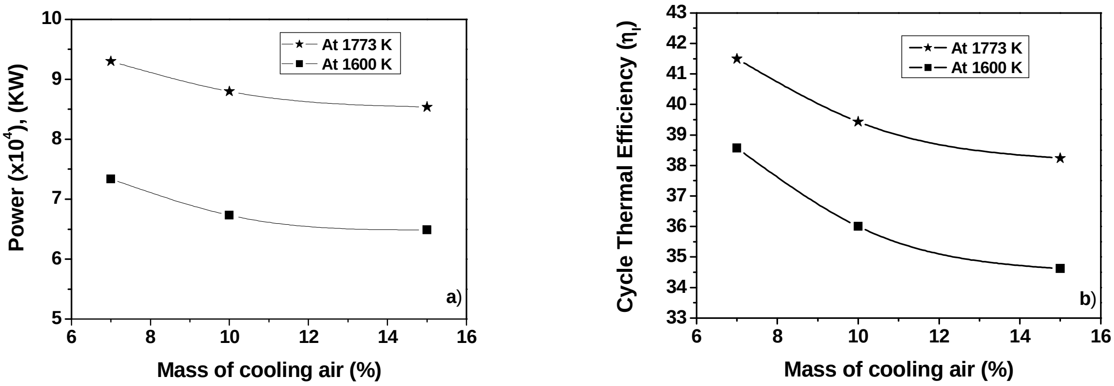

Fig. 6. Variation of the net power ($\dot{W}_{net}$) and cycle thermal efficiency (η_I) at the minimum (1600 K) and at the maximum (1773 K) inlet temperature, as a function of the fraction of cooling air (β).

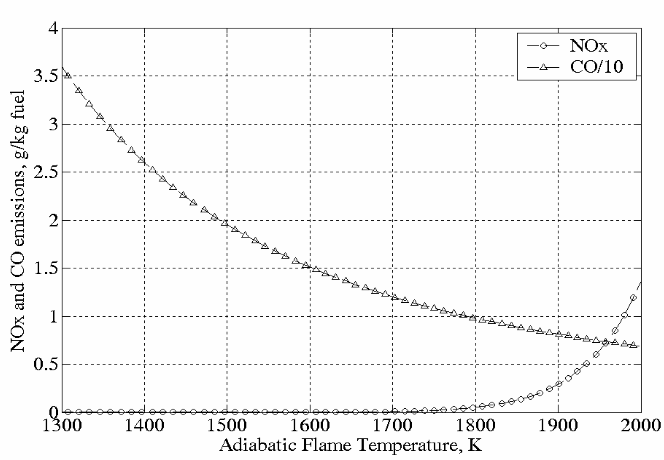

Fig. 7. NO_x and CO emissions function of the adiabatic temperature in the primary zone of the combustor.

6. Conclusions

For the thermodynamic analysis, the model is an important step forward among other studies, as it uses the second law of thermodynamic in order to identify and quantify the thermodynamic losses in the plant operation. Also, it considers a different mass flow rate though the compressor (just air) and the gas turbine (burned gases), the difference being equal to the mass flow rate of the fuel. This analysis shows that the augmentation of turbine inlet temperature is possible either using coated blades or cooling the elements of first two turbine stages. Increasing the mass flow rate of blade cooling air increases the temperature reduction and the thermodynamic performance. However the gain in the temperature reduction is more important than the loss in the performance (about 1.4% in thermal efficiency). As expected, increasing the inlet temperature, the NO_x emission increases (mainly above 1750K) while the CO emission decreases.

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