THEORETICAL ANALYSIS OF AIR CONDITIONING BY EVAPORATIVE COOLING INFLUENCE ON GAS TURBINE CYCLES PERFORMANCE

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Abstract. This paper studies the influence of evaporative cooling air conditioning for a gas turbine. A mathematical model of semi-empiric nature for both the evaporative panel and the gas turbine cycle was developed and implemented for computational simulation. The model used for the evaporative panel is based on mass and energy balances as well as in available empirical correlations for commercial evaporative panels. The modeling of the gas turbine cycle is based in non-dimensional numbers obtained from performance maps reproducing actual performance of the compressor and turbine. Simulation results are shown for a wole Test Reference Year in Brasilia allowing to quantify the advantages of evaporative cooling in power output increasing and NOx emission reduction.

Keywords: modeling, simulation, evaporative cooling, gas turbine, energy

1. Introduction

With the corroboration of the expectations about Brazilian economy growth an inevitable increase in the country energy demand will occur inexorably. Such a scenario, associated with the difficulties of generation from hydroelectric plants, in periods low rain precipitations, has motivated the production of energy by alternative ways, as for example, by means of gas turbine power plants using natural gas.

Gas turbines are worldwide used in the generation of electricity in thermal plants, in different geographic places with variable climatic conditions of temperature and humidity. The conditioning of the air that enters the compressor in a gas turbine cycle is a common technique applied which causes the inlet air density increase allowing some augmentation in net power for generation (Alhazmy and Najjar, 2004). Gas turbines are very sensible to ambient air wet bulb temperature. For instance, if the gas turbine operates with an ambient temperature of 35°C a reduction about 20% of its capacity of generation may occur when compared to ISO standard condition of $15°$C. This is due to the direct influence o air density on the amount of air introduced in the combustion chamber (at higher temperatures, air presents low density and, therefore the air mass supplied to the turbine is reduced). Previous studies (Guimarães 2000, Bassily 2001) have demonstrated the advantages of the use of evaporative panels in the cooling of the air of turbines the gas. Different techniques can be used for inlet air conditioning, among them evaporative cooling by means of contact panels represents a low cost and effective alternative. The evaporative cooling of the air at gas turbine inlet can increase power production by 2 to 4% in hot and dry climates (De Lucia, 1995 and 1997, Mohanty and Paloso Jr, 1995 and Wang and Chiou, 2002).

This paper presents a theoretical analysis of direct evaporative cooling systems influence on a gas turbine cycle performance. A mathematical modeling is carried out considering two different approaches: (i) for the gas turbine the model is developed through characteristics non-dimensional numbers and performance maps available for both the compressor and the turbine, and (ii) application of a mathematical model, proposed by Castro and Pimenta (2003) for a direct evaporative systems based in commercial cellulose panels.

2. Description of the system

The standard Brayton cycle is the ideal cycle for the thermodynamics analysis of gas turbine generation systems. Figure 1 presents the scheme adopted in the present work which is based in an open Brayton cycle using a direct evaporative panel for conditioning of the air at the compressor inlet.

During the gas turbine cycle the ambient air is initially cooled and humid in the direct evaporative cooler (RED) due to simultaneous mass and heat exchange between the air stream and the wet surface of the panel. After that air, enters the compressor (C) being compressed to state 2. In the a combustion chamber (CC) air is mixture with the fuel occurring the burning of this mixture which generates combustion products that in state 3 are supplied to the gas turbine (T) where it expands generating mechanical shaft work used to move a generator (G).
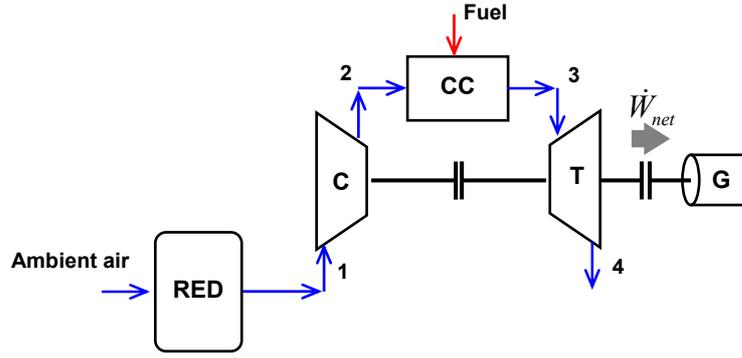
Figure 1. Gas turbine open cycle with compressor inlet air conditioned by a direct evaporative pannel.

3. Mathematical Model

The mathematical model implemented for the system under study involves the following main considerations: (i) adiabatic evaporative panel; (ii) gas turbine operating according an open Brayton cycle in steady state, neglecting changes in kinetic and potential energy; (iii) pressure drops trough compressor, adiabatic combustion chamber and turbine were neglected; (iv) no changes in absolute humidity of the air; (v) turbine exhaust gases at local ambient atmospheric pressure.

3.1. Evaporative Panel

The direct evaporative cooling process works essentially with the conversion of sensible heat in latent heat. The surrounding ambient air is cooled by evaporation of the water from the wet surface of the panel to the air. The addition of water vapor to the air increases its latent heat and relative humidity. If it the process is adiabatic, this increase of the latent heat is compensated by a reduction of the sensible heat and consequent reduction of the dry bulb temperature of air (Fig. 2).
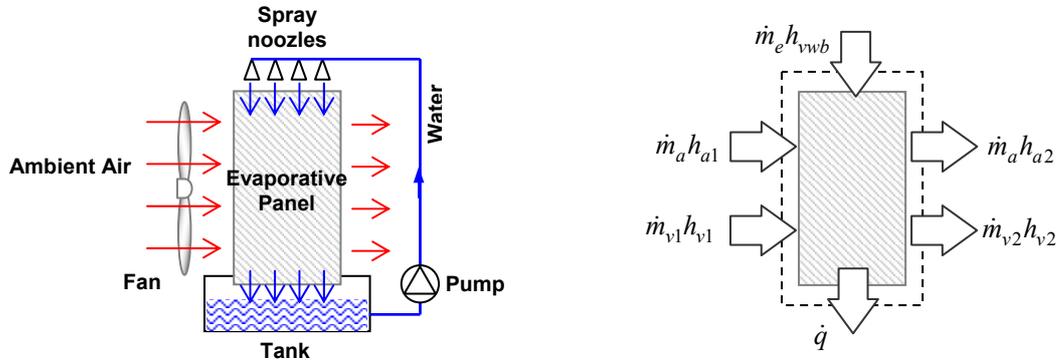
Figure 2. Direct evaporative panel: (a) schematic view, (b) panel control volume.

The direct evaporative panel can basically be considered as a heat exchanger in cross flow (Castro and Pimenta, 2003). Then, applying an analysis based in classical heat transfer theory, the LMTD method (Log Mean Difference Temperature) can be used. So, the rate of heat transfer from air to water in the wet panel surface, $\dot{q}$, is given by

$$\dot{q} = h_H \cdot A_S \cdot \Delta T_{LM} \tag{1}$$

where h_H represents the heat transfer coefficient, A_S is the wet surface area of the panel and ΔT_{LM} the log mean temperature difference, given by,

$$\Delta T_{LM} = \frac{(T_2 - T_1)}{\ln[(T_2 - T_{wb})/(T_1 - T_{wb})]} \tag{2}$$

where T_1, T_2 and T_{wb} are the dry bulb temperatures at the evaporative panel inlet, outlet and the wet bulb temperature of surroundings ambient air.

Substituting Eq. (2) into Eq. (1) and regrouping terms, follows that,

$$1 - \frac{T_1 - T_2}{T_1 - T_{wb}} = \exp\left(-\frac{h_H \cdot A_S \cdot (T_1 - T_2)}{\dot{q}}\right)$$ (3)

where the terms $(T_1-T_2)/(T_1-T_{wb})$ is defined as the panel effectiveness ε, i.e.,

$$\varepsilon = \frac{T_1 - T_2}{T_1 - T_{wb}}$$ (4)

From Eq.(3) the effectiveness of the evaporative panel can also be expressed as,

$$\varepsilon = 1 - \exp\left(-\frac{h_H \cdot A_S \cdot (T_1 - T_2)}{q}\right)$$ (5)

Analyzing Eq. (4) one notices that a 100% effectiveness corresponds to an air outlet temperature equal to the ambient air wet bulb temperature. Besides, from Eq. (5) it is possible to observe that such limit effectiveness requires a combination of a great area of surface with a high heat transfer coefficient.

Then, the air temperature leaving the evaporative panel (T_2) towards the compressor suction can be calculated from Eq. (4) as,

$$T_2 = T_{db} - \varepsilon \cdot (T_{dw} - T_{wb})$$ (6)

On the other hand, the mass transfer rate in the water vapor state, can be expressed by an equation analogous to Eq.(1), i.e.,

$$\dot{m}_e = h_M \cdot A_S \cdot \Delta\rho_{LM}$$ (7)

where h_M is the mass transfer coefficient and $\Delta\rho_{LM}$ is the log mean density difference, given by,

$$\Delta\rho_{LM} = \frac{(\rho_{v2} - \rho_{v1})}{\ln[(\rho_{v2} - \rho_{vwb})/(\rho_{v1} - \rho_{vwb})]}$$ (8)

where ρ_{v1} and ρ_{v2} are, respectively, the densities of the water vapor at evaporative panel inlet and outlet with ρ_{vwb} being the water vapor density at wet bulb temperature of air.

For the determination of the heat and mass transfer coefficients the following correlations developed by Dowdy e Karabash (1987, apud, Castro and Pimenta, 2003), were considered,

$$Nu = 0,10 \cdot \left(\frac{L_e}{L}\right)^{0,12} \cdot Re^{0,8} \cdot Pr^{1/3}$$ (9)

$$Sh = 0,08 \cdot \left(\frac{L_e}{L}\right)^{0,12} \cdot Re^{0,8} \cdot Sc^{1/3}$$ (10)

where *Nu, Re, Pr, Sh* and *Sc* are the Nusselt, Reynolds, Prandtl, Sherwood and Schimdt numbers, respect., L is the panel thickness and Le is a characteristic panel thickness.

3.2. Brayton cycle

The analysis of the gas turbine cycle is carried out with the help of performance maps found in literature (Saravanamuttoo at al, 1996) for the compressor and the turbine. Such kind of information could be get from gas turbine manufacturers, however they are not easily found, instead this information is, generally, confidential. The performance maps allow to estimate the isentropic efficiency for both the compressor and the turbine as a function of their operational condition which is expressed in terms of non-dimensional numbers (Tab. 1).

For the simulation, initially, a design condition for the gas turbine was proposed as described by the curves represented in Fig's. 3a to d. A reference condition for Brasilia was considered corresponding to a wet bulb temperature of 297 K, a dry bulb temperature of 305 K and an atmospheric pressure of 89,859 bar. Together with these temperatures and pressure one also fixed a maximum efficiency of the compressor in 0,8 a compression ratio of 3 and the non-dimensional rotation equal to 1.

Table 1. Non-dimensional numbers (index "0" represents stagnation conditions).

Rotation speed	Mass air flow	Compression ratio	Expansion ratio	Gas mass flow
N	$m\dfrac{\sqrt{T_{02}}}{P_{02}}$	P_{03}/P_{02}	P_{04}/P_{05}	$m\dfrac{\sqrt{T_{04}}}{P_{04}}$

With these data in hands a "sub-routine" was developed, through the EES software (F-Chart, 2001), for calculation of the maximum air flow admitted for the compressor in the reference condition. The stagnation conditions had been also taken into account for the calculations of the compression power, the power generated in the turbine and for the generated net power in cycle.

(a)

(b)

(c)

(d)
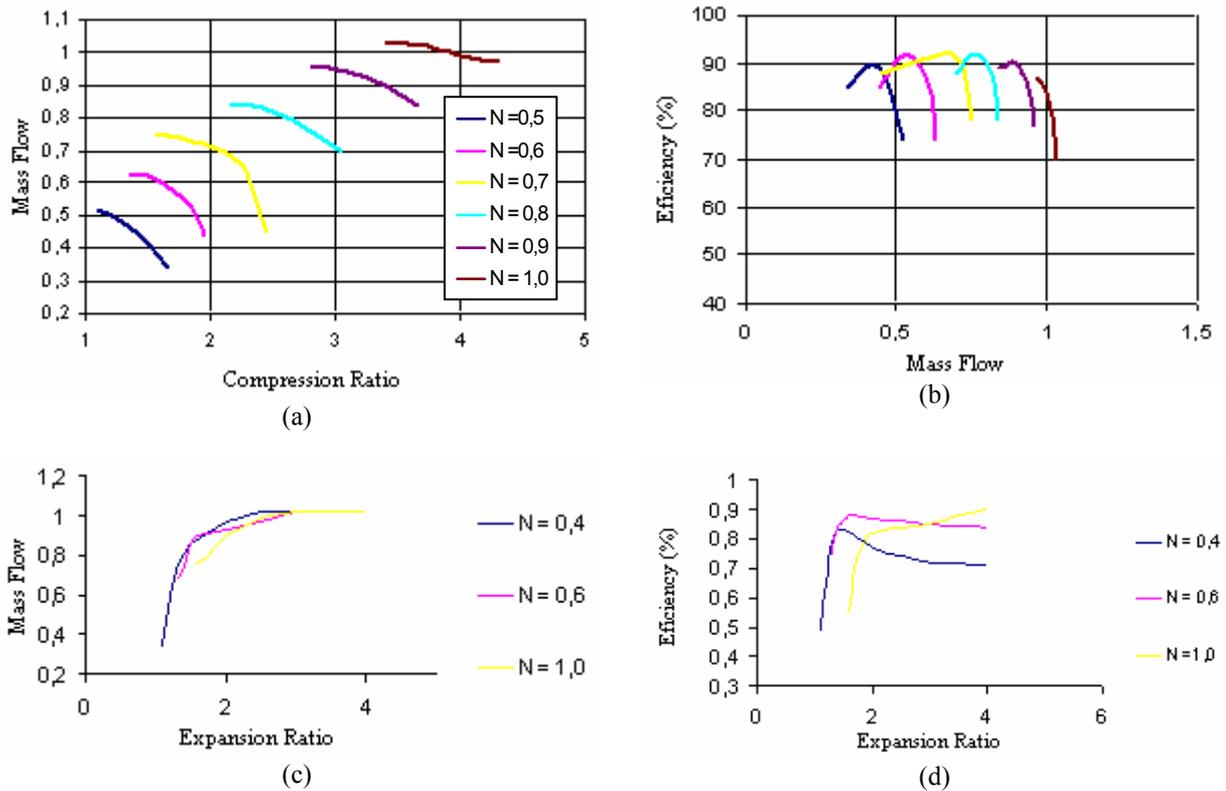
Figure 3. Gas turbine performance maps: (a) air mass flow rate *vs*. compression ratio, (b) compressor efficiency *vs*. air mass flow rate, (c) turbine efficiency *vs*. expansion ratio, (d) air mass flow rate *vs*. expansion ratio.

3.2.1. Compressor

With the non-dimensional performance data, a least squares fit technique was applied in order to get functions adjusted to each map. The air mass flow, $\dot{m}_{ar}$, rate was then calculated by means of,

$$\dot{m}_{ar} = \left(\dot{m}_{ar,adm} \cdot P_{02}\right)\big/ \sqrt{T_{02}} \tag{11}$$

where P_{02} and T_{02} are respectively the stagnations pressure and temperature at the compressor inlet.

The stagnation temperature difference between compressor inlet and outlet is given by,

$$\Delta T_{032} = \frac{T_{02}}{\eta_c} * \left(\left(\mathrm{Pr}_{c;adm} \right)^{\left(\frac{\gamma-1}{\gamma} \right)} - 1 \right) \tag{12}$$

where h_c the compressor efficiency, $Pr_{c;adm}$ is the non-dimensional compression ratio and γ is the specific heats ratio. Thus the stagnation temperature at the compressor outlet, T_{03}, is calculated by,

$$T_{03} = \Delta T_{032} + T_{02} \tag{13}$$

With the non-dimensional compression ratio and the stagnation pressure at the compressor inlet, the stagnation pressure at compressor outlet, P_{03}, is given from,

$$\mathrm{Pr}_{c;adm} = \frac{P_{03}}{P_{02}} \tag{14}$$

Finally, with the stagnation temperatures and pressures the stagnation enthalpies at compressor inlet (h_{02}) and outlet h_{03}), can be found by means of EES built in thermo-physical properties functions, allowing to compute the required compression power ($\dot{W}_c$) by,

$$\dot{W}_c = \dot{m}_{ar} \cdot \left(h_{03} - h_{02} \right) \tag{15}$$

3.2.2. Combustion chamber

For the combustion chamber simulation a chemical equilibrium program called GASEQ (Morley, 2005) was used. The program simulated the burning of methane (CH_4) with humid air ($N_2 + O_2 + H_2O$) according to the following chemical reaction,

$$\left(N_2 + O_2 \right) + CH_4 + H_2O \rightarrow N_2 + H_2O + CO_2 + CO + O_2 + H_2 + NO \tag{16}$$
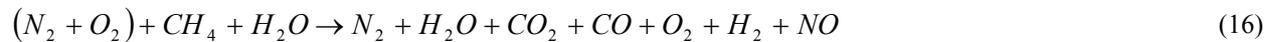
For the simulation a burning with an air excess was considered, although only 24 % of air used in the combustion while 76 % was bypassed. The amount of water vapor used in the process is gotten through a average of the amount of present water vapor in air during all the year.

With the combustion products discriminated we have them determined its enthalpy for the calculation of the power generated in the turbine.

3.2.3. Turbine

For the turbine the temperature difference between inlet and outlet is given by,

$$\Delta T_{054} = \frac{T_{04}}{\eta_t} * \left(\left(\mathrm{Pr}_{e;adm} \right)^{\left(\frac{\gamma-1}{\gamma} \right)} - 1 \right) \tag{17}$$

where T_{04} is the temperature at the turbine inlet, η_t is the turbine efficiency and $Pr_{e;adm}$ is the non-dimensional pressure expansion, while T_{04} is calculated through,

$$\sqrt{\frac{T_{04}}{T_{02}}} = \frac{\dot{m}_{gás;adm} \left(P_{04}/P_{02} \right)}{\dot{m}_{ar;adm}} \tag{18}$$

with $m_{gás;adm}$ being the non-dimensional gas mass flow and P_{04} the stagnation pressure at the turbine inlet. The temperature at turbine outlet T_{05} can be calculated from,

$$T_{04} = \Delta T_{054} + T_{05} \tag{19}$$

With the stagnation temperatures and pressures and the enthalpies of the combustion products the power generated by the turbine can be calculated as,.

$$\dot{W}_t = \dot{m}_{gás} \cdot \left(h_{04} - h_{05} \right) \tag{20}$$

where h_{04} and h_{05} are respectively the enthalpies of the combustion products at turbine inlet and outlet.

4. Results and discussion

4.1. Compressor

The compression power required by the compressor for a whole Test Reference Year (TRY) of operation in Brasilia is represented in Fig. 4. The linee in red shows the power required without the use of the evaporative panel while the curve in blue shows the compression power when the evaporative panel is used. As expected compression power required by the compressor when the evaporative panel is used is always smaller than when none air cooling is used. The compression power decrease with the use of the evaporative panel (RED) was about 1,52 % in the year, with a total economy of approximately 93,200 kWh.
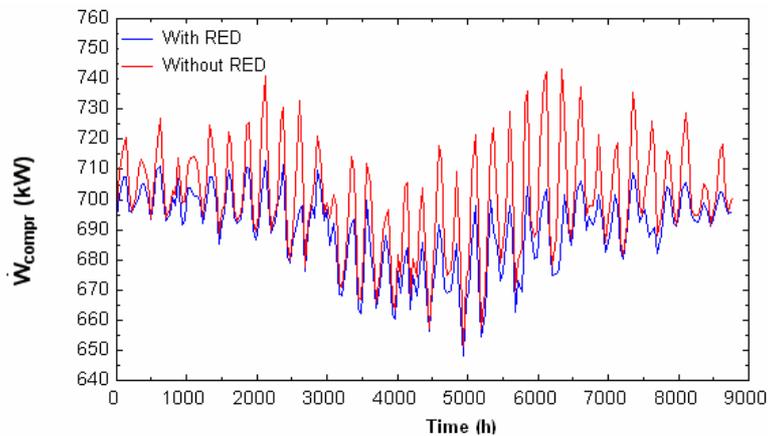
Figure 4. Power required for the compressor during Brasilia TRY

4.2. Combustion chamber

The results for combustion chamber are shown in Figs. 5 and 6 below, for this were used GASEQ -Chemical Equilibrium Program (Morley, 1998). The amount of water steam used in the process is get through an average of the amount of present water steam in air during all the year, this amount react with CH_4 and $N_2 + O_2$ (air). The products of this combustion are used to calculated the enthropy of the flow inlet the turbine.

It may be remarked that, thanks to the use of the evaporative panel, the occurrence of a greater amount of steam water on air and a lesser temperature, leads to a decrease in the NOx emission from 0,04928 kg/mol against 0,74347 kg/mol for the process without RED. This is related to the reduction of the flame temperature, as discussed in the literature (Najjar, 1996, Wang and Chiou , 2004).
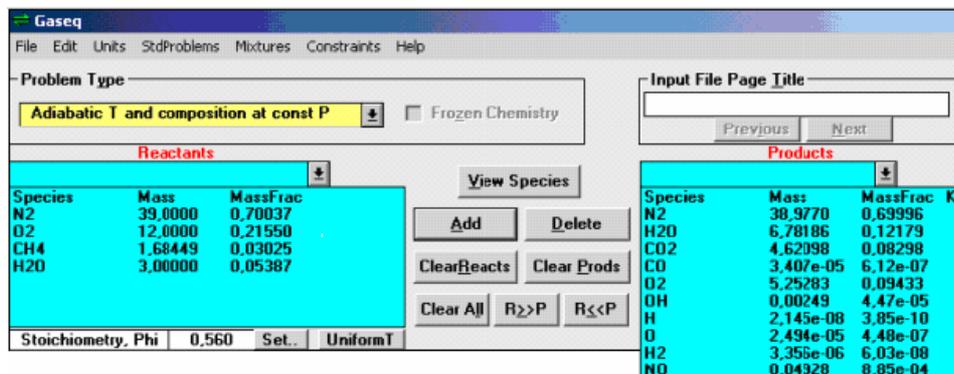
Figure 5. Simulation of the combustion with RED
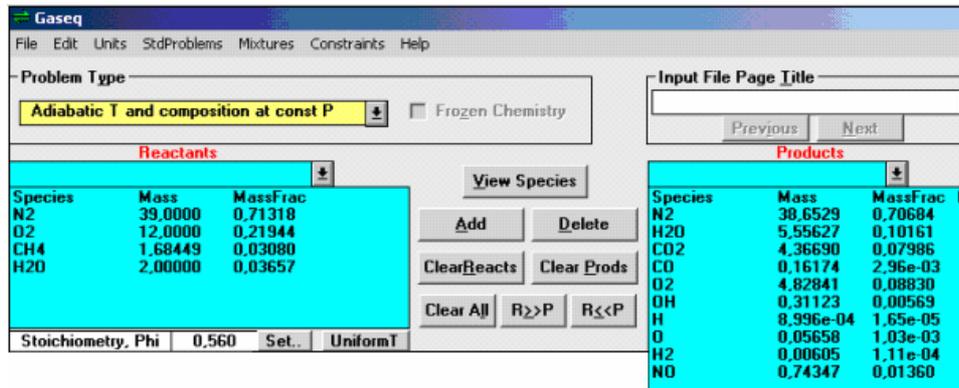
Figure 6. Simulation of the combustion without RED

4.3. Turbine

Figure 7 presents the simulation results for the overall (Fig. 7a) and net (Fig. 7b) turbine power output for a whole year of operation in Brasília. For the overall turbine power production an increase of 1,26 % is obtained corresponding to about 520.833 kWh more energy generated. With relation to the net power an increase of 1,75 % is observed when using the RED, generating a economy of about 614.042 kWh.
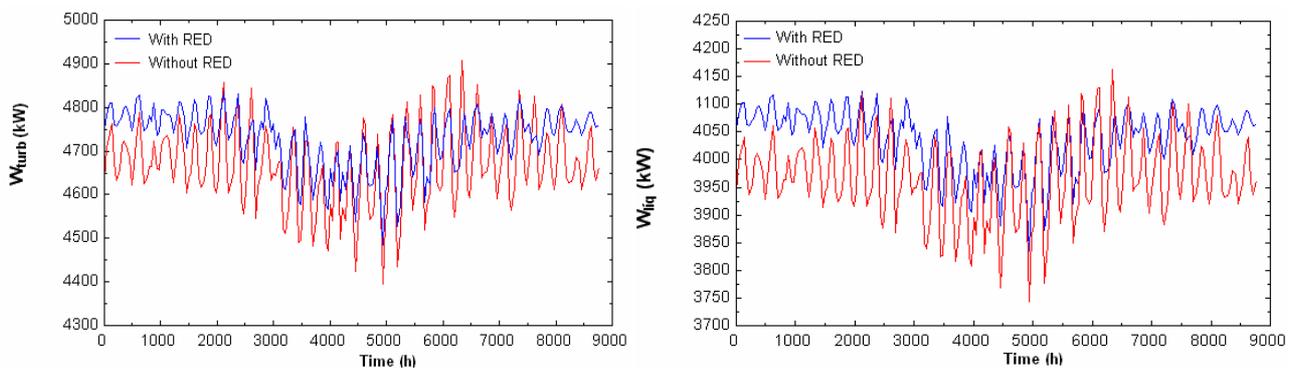
Figure 7. Turbine power output by the turbine during Brasilia TRY: (a) overall power generation, (b) net power generation

5. Conclusions

The simulation results from the analysis of the influence of evaporative conditioning of air supplied to a gas turbine power plant allow both confirming and quantifying the expected gains in power production. Besides the performance improvement an important aspect is related to the NOx reduction in the emissions form the combustion chambers.

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