Gas Turbine System with Interstage Cooling and Steam Injection Using Oxy-combustion

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Abstract: - The intake air temperature influences the power output of a gas turbine. In order to control it, the paper considers the interstage air cooling by adiabatic humidification. Steam injection after the combustor is considered to control the flue gases temperature in the inlet section of the turbine. Both techniques are intended to increase the shaft work rate of the gas turbine. The first section of the paper studies the influences exerted by the intake air temperature and by the injected steam/air ratio on the flue gas temperature and composition. The second section tackles the analysis of the compression and expansion processes respectively, allowing the estimation of the shaft work of the turbine. The third section of the paper considers the opportunity of using oxy-combustion to increase the power of the turbine or to counterbalance the effect of the compressor power consumption increase due to high intake air temperatures.

Key-Words: - gas turbine, interstage cooling, adiabatic humidification, steam injection, oxy-combustion

1. Introduction

In the field of gas turbine systems, a great deal of research work has been dedicated to gas turbine repowering techniques [1, 2], new working fluids [12, 13, 14], renewable and recovered energy [3÷11], second law analysis, modeling, and simulation [15÷21], new solutions to improve the components performance (compressor, combustor, turbine) [22, 23, 24].

The intake air temperature has an important influence on the system's performance, since a higher air temperature means an increased power consumption of the compressor, resulting in a corresponding decrease of the useful shaft work supplied by the turbine. It is known that adiabatic humidification of unsaturated atmospheric air results in a temperature decrease. The paper considers a power control technique consisting of interstage cooling of the combustion air by means of adiabatic humidification (injection of pulverized water in the air stream).

Steam injection before the turbine reduces the temperature and increases the mass flow rate that enters the turbine. The second objective of the paper is to study the regulating effect of steam injection in the flue gases that leave the combustor, in order to reduce the turbine intake temperature. The third objective is the analysis of an increased percentage of oxygen in the combustion air in terms of improved performance of the gas turbine system.

2. The Gas Turbine System

Figure 1 shows the schematic of the gas turbine system. It consists of a two-stage compressor (C1 and C2), three adiabatic humidifiers (AH1 in the intake section of the compressor, AH2 between the two compression stages, and AH3 after the second stage), a regenerative heat exchanger (IHE), the combustor (CC), the steam injection unit (SI), and the gas turbine (T). Two fictitious dissociation cells (DC and FDC respectively) have been inserted in order to perform dissociation calculi necessary to obtain the composition of the flue gases after the combustor and after the steam injection unit.

The sequence of processes is as follows:

0 – 1: oxygen-air mixing; 1 – 2: first adiabatic humidification; 2 – 3: first compression (adiabatic); 3 – 4: interstage adiabatic humidification; 4 – 5: second compression (adiabatic); 5 – 6: final adiabatic humidification; 6 – 7: regenerative heating; 7 – 8; combustion; 8 – 9: dissociation; 9 – i steam injection; i – 10: dissociation; 10 – 11: adiabatic expansion; 11 – 12: regenerative cooling of the flue gases.

3. Mathematical Approach

3.1. Main Equations

– saturation pressure of the water vapor:

$$ln\, p_{w_s} = A_1 T^{-1} + A_2 + A_3 T + A_4 T^2 + A_5 T^3 + A_6\, ln\, T \quad (1)$$

The coefficients $A_1 \ldots A_6$ have been taken from [26].

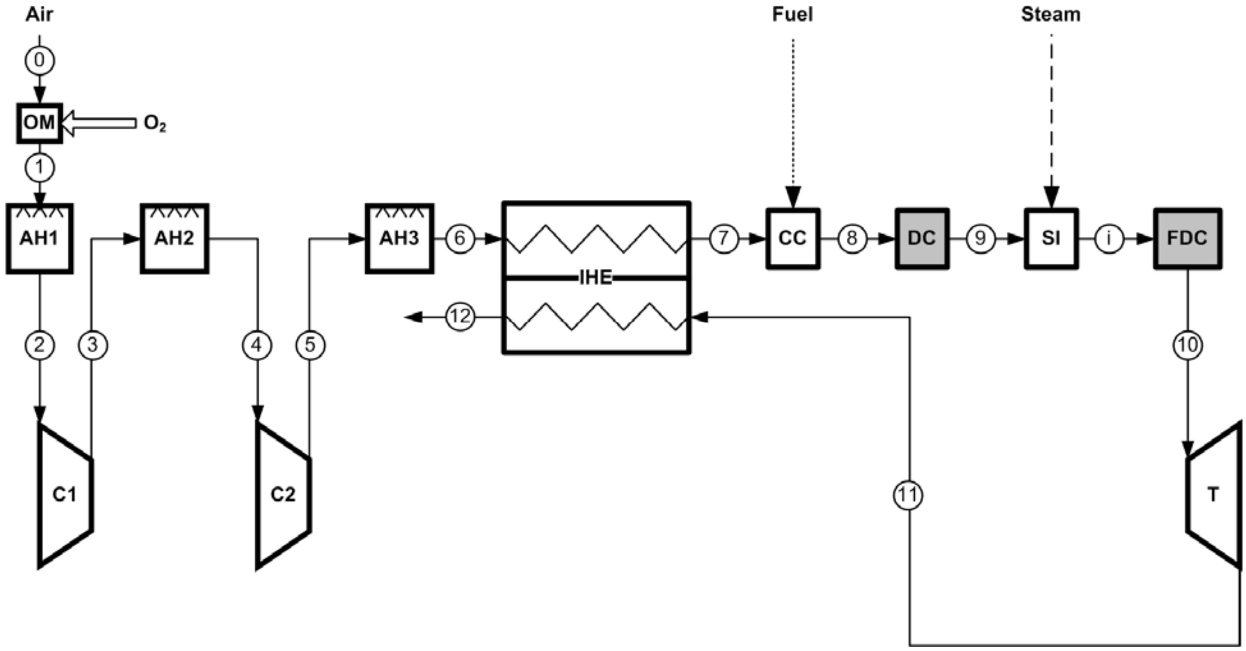

Fig. 1.

– water vapor mole fraction in dry air or gases:

$$x = \frac{\phi p_{ws}}{p - \phi p_{ws}} \quad \left[\frac{\text{kmoles } H_2O}{\text{kmoles dry gases}}\right] \qquad (2)$$

– enthalpy of moist air or gases:
- for processes without dissociation:

$$h_n = \sum_i \left(r_i \int_{T_0}^{T} c_{p,i} dT \right)_{dg} + x \left(L_0 + \int_{T_0}^{T} c_{p,w} dT \right) \qquad (3)$$

– enthalpy of moist air or gases:
- for processes with dissociation:

$$h_n^d = \sum_j \left[n_j \left(h_{f,j}^0 + \int_{T_0}^{T} c_{p,j} dT \right) \right]_{gas}^{d} \qquad (4)$$

– entropy of moist air or gases:

$$s = \sum_i \left(r_i \int_{T_0}^{T} c_{p,i} dT \right)_{dg} - R \, ln \frac{p(1 - r_w)}{p_0} + x \left(\frac{L_0}{T_0} + \int_{T_0}^{T} c_{p,w} \frac{dT}{T} - R \frac{r_w p}{p_{ws}(T_0)} \right) \qquad (5)$$

In the above equations: $p_0 = 1$bar, $T_0 = 273.15$K for dry gases, $p_{ws}(T_0) = 610.8$ Pa and $T_0 = 273.15$K for water vapor.

The combustion calculus model has been developed by us in our previous work referring to the possibilities to implement oxy-combustion in supercritical steam boilers [27].

3.2. Processes not Involving Dissociation

– moist air-oxygen mixing ($0 \rightarrow 1$) – isobaric and isothermal:

$$\left(0.21 + n_{O_2}^{inj}\right) M_{O_2} + 0.79 M_{N_2} = r_{O_2}^{dg} M_{O_2} + \left(1 - r_{O_2}^{dg}\right) M_{N_2} \qquad (6)$$

where $n_{O_2}^{inj}$ represents the moles of injected oxygen.

– adiabatic humidification ($1 \rightarrow 2$, $3 \rightarrow 4$, $5 \rightarrow 6$):

$$\left[h_n\left(T, r_i, x\right)\right]_{in} = \left(h_n\left(T, r_i, x\right)\right)_{out} \qquad (7)$$

– irreversible adiabatic compressions ($2 \rightarrow 3$, $4 \rightarrow 5$) and expansion ($10 \rightarrow 11$):

$$\left(h_n^{out}\right)_{irrev} = \left(h_n^{in}\right)_{irrev} + \frac{\left(h_n^{out} - h_n^{in}\right)_{rev}}{\eta_s} \qquad (8)$$

– regenerative heating ($6 \rightarrow 7$) and cooling ($11 \rightarrow 12$):

$$h_7\left(T, r_i, x\right) - h_6\left(T, r_i, x,\right) = \frac{n_{dg} \text{ kmoles}}{\text{kmole dry air}} \left[h_{11}\left(T, r_i, x\right) - h_{12}\left(T, r_i, x\right)\right] \qquad (9)$$

3.3. Combustion calculus

First we calculate the mole fractions of products in flue gases per kg of burned fuel and finally the total amount of kmoles, by addition. The adiabatic flame temperature T_8 results as a solution of the heat balance of the combustor:

ISSN: 1790-5095

ISBN: 978-960-6766-97-8

$$
\begin{aligned}
&H_i + h_{fuel}(T_8) + \sum_i \left(n_i \int_{T_0}^{T_7} c_{p,i} dT \right)_{dry\ air} = \\
&= \sum_j \left(n_j \int_{T_0}^{T_8} c_{p,j} dT \right)_{fg}
\end{aligned}
\tag{10}
$$

We have considered dissociation and we have computed the flue gases composition and temperature after dissociation by using the fictitious dissociation cell DC.

3.4. Calculus of steam injection

The purpose of steam injection is mainly to regulate the temperature of flue gases in the inlet section of the gas turbine (T_{10}). To make the steam injection calculus possible, we have divided the process into three stages:

a) adiabatic – isobaric mixing of flue gases and steam in the steam injection chamber (SI). The only component that changes in terms of mole fraction is water vapor, due to the injection of n_{steam} kmoles. As a result, the temperature of the flue gases-steam mix equals T_i, which has to be determined.

b) isothermal recombination of the dissociated products of the combustion process. This stage is a fictitious one, necessary to determine the primary quantities dubbed y_1^i, y_2^i, y_3^i, and y_4^i, which represent the initial values used to start the dissociation calculus algorithm.

c) dissociation at T_8, imposed as the turbine inlet temperature. The process occurs in the fictitious final dissociation cell FDC and uses as input data the primary quantities determined in the previous stage.

The governing equations of the three stages are:

– for the first stage (adiabatic mixing):

$$
\begin{aligned}
&\sum_k \left(n_k(T_9) \int_{T_0}^{T_9} c_{p,k} dT \right)_{fg}^{d} + n_{steam} h_{n,steam} = \\
&= \sum_m \left(n_m \int_{T_0}^{T_i} c_{p,m} dT \right)_{fg}^{i}
\end{aligned}
\tag{11}
$$

– for the second stage (isothermal recombination):

$$
\begin{aligned}
y_1^i &= n_{CO_2}^d + n_{CO_2}^d \\
y_2^i &= n_{H_2O}^d + n_{steam} + n_{H_2}^d + n_{OH}^d + n_H^d \\
y_3^i &= n_{N_2}^d + 0.5 n_{NO}^d + n_N^d \\
y_4^i &= n_{O_2}^d + 0.5 n_{NO}^d + n_O^d
\end{aligned}
\tag{12}
$$

– for the third stage (dissociation at T_{10}):

$$
\begin{aligned}
&\sum_{j=1}^{4} \left[y_j^i \left(h_{f,j}^0 + \int_{T_0}^{T_i} c_{p,j} dT \right) \right]_{fg}^{i} = \\
&= \sum_k \left[n_k(T_{10}) \left(h_{f,k}^0 + \int_{T_0}^{T_{10}} c_{p,k} dT \right) \right]_{g}^{d}
\end{aligned}
\tag{13}
$$

Since the mathematical model of the processes has two degrees of freedom (n_{steam} and T_i) and, what's more, T_i depends on the amount of injection steam, the solution of the problem resides in applying a trial-and-error technique (see Fig. 2):

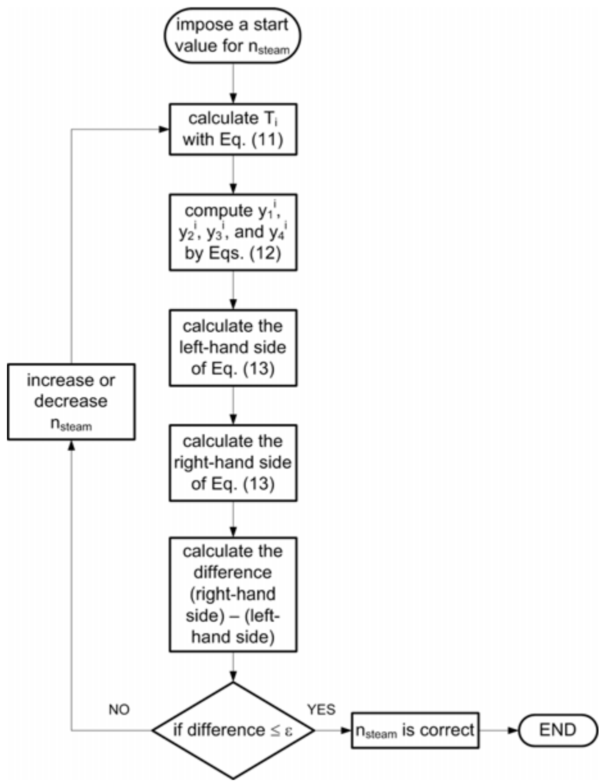

Fig. 2

4. Numerical Results

In order to apply the model, we have assumed the following start data:

$\phi_2 = \phi_4 = \phi_6 = 0.98$;
$p_2 = p_1 = p_0$;
$p_4 = p_3 = \sqrt{p_2 p_5}$;
$T_{10} = 1373.15$ K;
$p_5 = p_6 = 15 p_0 = 1.1 p_7$;
$p_7 = p_8 = p_9 = p_i = p_{10}$;
$p_{11} = 1.1 p_{12} = 1.1 p_0$;

The isentropic efficiencies of the adiabatic compression and expansion equal 0.75 and 0.85 respectively. The plots resulted from the numerical calculus are shown in Figs. 3 through 6. Fig. 3 shows the enthalpy values in the main state points on the flow path for an excess air equal to 1.5 considering two values of the oxygen contents in the combustion air (atmospheric: 21%, respectively 35%), in the hypothesis of an inlet air relative humidity equal to 0.5. The enthalpy values have been computed for two inlet air temperatures: 273 K, respectively 313 K. As shown in Fig. 3, an increased oxygen mole fraction results in a higher enthalpy in the state points situated after the steam injection unit. The influence of the inlet air temperature on the enthalpy is noticeable in the compression stages (states *1*, *2*, and 3), whereas the final state *4* (the compressor outlet) shows practically no discrepancy due to the final humidifying which levels all differences.

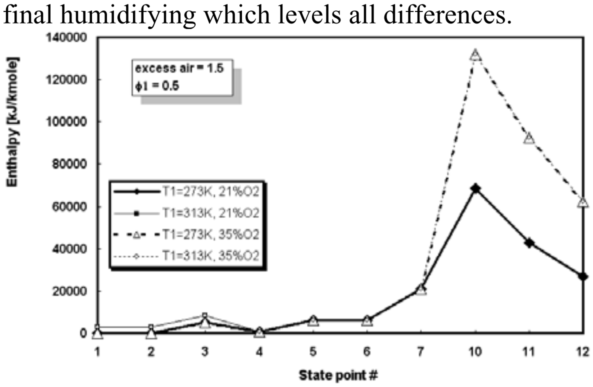

Fig. 3.

Figure 4 presents the shaft work consumption of the compressor in terms of enthalpy difference in both oxygen mole fraction scenarios, for the two inlet air temperatures. As expected, the oxygen contents exerts practically no influence on the shaft work transfer necessary to power the compressor, whereas the inlet air temperature increase results in an increased shaft work transfer (500 kJ/kmole for $\Delta T = 40$ K, representing a 0.5% increase). It is important to emphasize that this shaft work increase results under conditions of interstage adiabatic humidification. In the absence of its cooling effect, for the same temperature increase, the shaft work rise lies in the range (7 … 9%), which is 14 to 18 times higher. The plot in Fig. 5 presents the shaft work transfer of the turbine under the above conditions. The fact that the intake air temperature has no effect can be easily explained when one notices that the temperature of the flue gases entering the turbine is regulated by the amount of injected steam so as to have a fixed value (1373 K). On the other hand, the shaft work output of the turbine strongly depends on the oxygen mole fraction in the combustion air. Fig. 6 presents the net shaft work transfer of the gas turbine system. Due to the cooling effect of the adiabatic humidification, the impact of an increased inlet temperature is practically nil: it determines a very slight decrease of the net shaft work. Without adiabatic humidification, the loss in net shaft work is much higher. An increased oxygen mole fraction in the combustion air determines a significant increase of the net shaft work, which is very beneficial to the gas turbine performance.

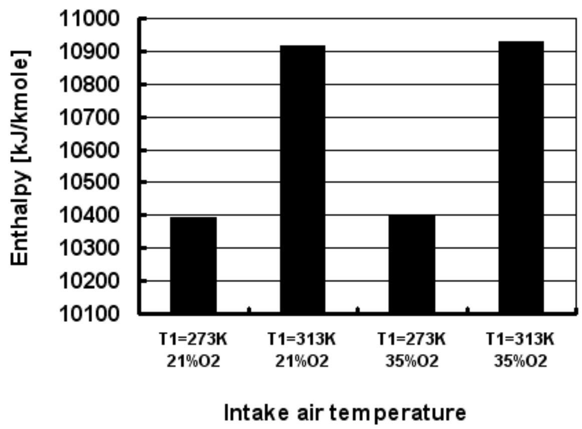

Fig. 4

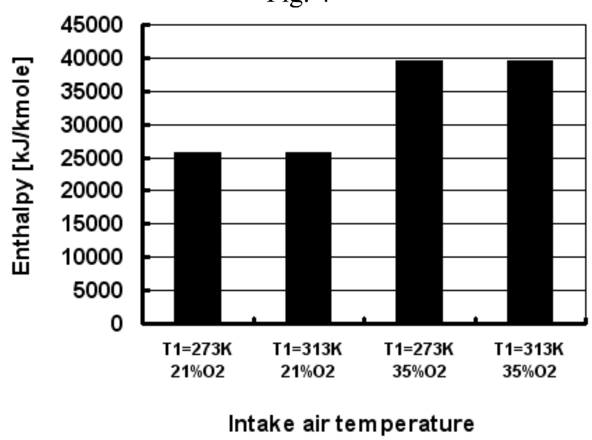

Fig. 5

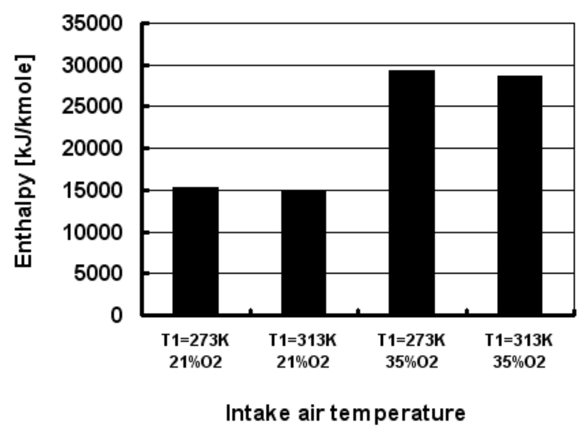

Fig. 6

The presence of the regenerative heat exchanger has an impact only on the First Law efficiency, as shown in Table 1.

O_2 Mole Fraction in Combustion Air	First Law Efficiency	
	T_1 = 273 K	T_1=313 K
With Regenerative Heat Exchanger		
21% O_2	0.396	0.383
35% O_2	0.323	0.317
Without Regenerative Heat Exchanger		
21% O_2	0.246	0.241
35% O_2	0.232	0.223

Table 1

5. Conclusions

The paper deals with the impact of interstage cooling by means of adiabatic humidification during the compression process of the combustion air in a gas turbine system, using atmospheric and oxygen-enriched air as well. In order to control the flue gases temperature in the inlet section of the turbine so as to limit it to a certain value (in our case, 1373 K), steam injection is used. The paper puts forward a mathematical model for actual thermodynamic processes (irreversible, with temperature-dependent heat capacities) and combustion with dissociation. The analysis of the numerical results obtained allows to reach the following conclusions:

– adiabatic humidification of the compressed air proved to be effective, resulting in a cooling of about 100°C. This cooling reduces the mechanical power consumption of the compressor and thus increases the net shaft work rate.

– steam injection is effective as a control and "fine tuning" technique of the temperature in the inlet section of the turbine. By maintaining a constant value of the turbine inlet temperature (T_{10} = 1100°C) by means of steam injection, the flue gases temperature leaving the turbine is in a very slight measure influenced by the relative humidity and temperature of the air sucked by the compressor.

– the oxygen mole fraction in the combustion air practically has no influence on the temperatures along the flow path of the system, except the outlet section of the turbine, but even here the difference is very weak. On the other hand, the oxygen mole fraction strongly influences the enthalpy of the flue gases entering the turbine due to the fact that combustion with higher oxygen contents in air results in higher temperature values requiring an increased steam injection ratio. Thus, the steam-flue gases mixture's enthalpy is much higher and consequently, the shaft work rate output of the turbine increases, which has a beneficial effect on the net shaft work rate, as the compression shaft work consumption is much less dependent on the O_2 mole fraction.

– the density of the flue gases is very slightly influenced by the intake air parameters (temperature, relative humidity, and O_2 mole fraction).

– the First Law efficiency is influenced by the intake air temperature and oxygen mole fraction, but the main influence is due to the heat regeneration, which justifies the integration of the regenerative heat exchanger in the system configuration.

6. Nomenclature

c_p - mole heat capacity (kJ/kmole K)
h_f^0 - enthalpy of formation (kJ/kmole)
h_n - mole enthalpy (kJ/kmole)
H_i - lower heating value (kJ/kg fuel)
L_0 - water mole latent heat at 0°C (kJ/kmole)
M - molar mass (kg/kmole)
n - number of kmoles
p - pressure (N/m^2)
p_0 - standard state pressure (N/m^2)
p - pressure
r - mole fraction
R - universal constant of ideal gases
s_n - mole entropy (kJ/kmole K)
T - temperature (K)
T_0 - standard state temperature (K)
x - mole humidity (kmole/kmole dry gases)
y_i - primary product kmoles (kmole/kg fuel)

Greek letter symbols

ϕ - relative humidity

Subscripts

da - dry air
dg - dry gases
fg - flue gases
g - gases (after the steam injection)
w - water vapor in air

Superscripts

d - dissociated
da - dry air
dg - dry gases
fg - flue gases
i - intermediary

References:

[1] *Gas Turbine World, Handbook*, Pequot Publishing Inc., 1996

[2] Andersen, S., Houbak, N., Petersen, S., Hansen, S., *Improved Repowering Techniques For Power Plants*, ECOS'98, volume II, Nancy, France, 1998, pp. 583-590

[3] Consonni, S., Larson, E.D., *Biomass – Gasifier/Aeroderivative Gas Turbine Cycles, Part A+B, Technologies And Performance Modeling, and Performance Calculation And Economic Assessment*, J. of Engineering for Gas Turbines and Power, Volume 118, 1996, pp. 507-515, and 516-525

[4] Paisley, M.A., Litt, R.D., Overend, R.P., Bain, R.L., *Gas Turbine Power Generation From Biomass Gasification*, ASME COGEN – TURBO, volume 9, 1994, pp. 625-629

[5] Elmegaard, B., Korving, A., *Analysis Of An Integrated Biomass Gasification/Combined Cycle Plant*, ECOS'98, volume II, Nancy, France, 1998, pp. 591-580

[6] Bammert, K., *A General Review Of Closed – Cycle Gas Turbines Using Fossil, Nuclear And Solar Energy*, Verlag Karl Thiemig, Munich, Germany, 1975

[7] Anheden, M., Ahlroth, M., *System Studies On A Biomass Fired CHP Closed Gas Turbine With A CFB Furnace*, ECOS'98, volume II, Nancy, France, 1998, pp. 651-658

[8] Ishida, M., Jin, H., *A New Advanced Power – Generation System Using Chemical Looping Combustion*, Energy, 19, 1994, pp. 415-422

[9] Call, F.W., *Work And Syngas From Partial Oxidation Of Carbon Fuels By Chemical Looping*, ECOS'98, volume II, Nancy, France, 1998, pp. 659-665

[10] Leveika, R., Sauer, H.J., Jr., Anderson, M.D., *Cogeneration Alternatives For A University Campus*, ECOS'98, volume II, Nancy, France, 1998, pp. 789-798

[11] ASHRAE, *Systems And Equipment Handbook*, Chapter 7, Cogeneration Systems, 1992

[12] Bolland, O., Saether, S., *New Concepts For Natural Gas Fired Power Plants Which Simplify The Recovery Of Carbon Dioxide*, First International Conference On Carbon Dioxide Removal, Amsterdam, Netherlands, Energy Convers. Mgmt., volume 33, no. 5-8, 1992, pp. 467-475

[13] Mathieu, Ph., Nihart, R., *Sensitivity Analysis Of The MATIANT Cycle*, ECOS'98, volume II, Nancy, France 1998, pp. 775-787

[14] Moran, M.J., Sciubba, E., *Exergy Analysis: Principles And Practice*, Journal of Engineering for Gas Turbines and Power, no. 4, 1994, pp. 116-122

[15] Falcetta, M.F., Sciubba, E., *Exergy – Based Analysis Of The Start – Up Transient Of A Combined Cycle Plant*, ECOS'98, volume II, Nancy, France, 1998, pp. 775-787

[16] Laine, J., *Exergy Optimisation In Steam Power Plants, I.Description Of The Method, II.Application To A Heat And Power Cogeneration Plant*, ECOS'98, vol.II, Nancy, France, 1998, pp. 617-632

[17] Bejan, A., *Entropy Generation Minimization*, CRC Press, Boca Raton, Florida, USA, 1996

[18] De Oliveira. S., Jr., *Exergy Anlysis Of Cogeneration And Combined Cycle Plants*, ECOS'98, volume II, 1998, Nancy, France, pp. 759-765

[19] Tsatsaronis, G., *On The Efficiency Of Energy Systems*, ECOS'95, Istanbul, Turkey, 1995, pp. 53-60

[20] Descombes, G., Pluviose, M., Jullien, J., Parkinson, N., *Analysis Of Energy Conversion Within A Radial Turbine Stage*, ECOS'98, volume II, Nancy, France, 1998, pp. 685-689

[21] Saraç, H.I., Güven, N., I., Sözbir, N., Uysal, ü., Çalli, I., *A New Algorithm For Solution Of The Non – Linear Equations System Occurring In The Design Of Air Compression Systems*, Energy, volume 19, no. 10, 1994, pp. 1067-1075

[22] Saad, M.A., Cheng, D.Y., *The New LM2500 Cheng Cycle For Power Generation And Cogeneration,* Energy Convers. Mgmt, volume 38, no. 15-17, Pergamon Press, 1997, pp. 1637-1646,

[23] Bathie, W.W., *Fundamentals Of gas Turbines*, John Wiley & Sons Inc. 1984

[24] Saad, M.A., Cheng, D.Y., *Steam Injection Technology For Utility – Size Gas Turbines*, ECOS'98, volume II, Nancy, France, 1998, pp. 733-738

[25] Sonntag E., R., Van Wylen G., *Introduction to Thermodynamics, Classical and Statistical*, (Second Edition) John Wiley & Sons, New York, 1982

[26] ASHRAE, *Fundamentals Handbook*, Chapter 7, Cogeneration Systems, 1985

[27] Horbaniuc B., Marin O., Dumitraşcu Gh., Charon O., *Oxygen-Enriched Combustion in Supercritical Steam Boilers*, Energy – The International Journal, Elsevier Science Ltd., 29 (2004), 427-448