# ANALYSIS OF GAS TURBINE POWER PLANT WORKING WITH OXYGEN-ENHANCED COMBUSTION

**Leonardo Zimmer, leo.zimmer@mecanica.ufrgs.br**
**Amir Roberto De Toni Junior, detoni@mecanica.ufrgs.br**
**Wagner da Silva Boeira, wagner.boeira@ufrgs.br**
**Cristiano Frandalozo Maidana, cristiano.maidana@ufrgs.br**
**Paulo Smith Schneider, pss@mecanica.ufrgs.br**
Universidade Federal do Rio Grande do Sul – Departamento de Engenharia Mecânica
Rua Sarmento Leite 425, Porto Alegre, Rio Grande do Sul, Brazil. 90050-170

***Abstract.*** *The analysis of the performance of a gas turbine running under Oxygen-Enhanced Combustion (OEC) conditions is presented in this paper. The oxygen content on the oxidant stream starts at 21% for atmospheric composition and ranges up to 30%. A gas turbine is modeled following a thermodynamic approach, and several options of heat recovery for efficiency improvement are proposed. Five different assemblies are presented running at variable oxygen contents in the oxidant stream. Cycle overall efficiency and oxygen consumption are evaluated according to variable oxygen feed concentration under stoichiometric combustion and with oxidant excess. All chemical species are taken as ideal gases. An oxygen supply scenario for the studied case was evaluated, considering cryogenic air separation, adsorption process and membrane technology. The main results obtained are the direct relation of the cycle efficiency with the increase in the oxygen proportion in the oxidant feed, for all cases. The complete assembly displayed the highest cycle efficiency, around 45%, upon 32% of the base cycle.*

***Keywords****: Oxygen-enhanced combustion, oxy-fuel combustion, gas turbine, air separation units*

## 1. INTRODUCTION

Oxygen-enhanced combustion (OEC) is the process of burning a fuel using an oxygen-rich stream instead of air as the primary oxidant. Since the nitrogen component of air is reduced, it is possible to achieve great reduction in fuel consumption and higher thermal efficiencies. The viability of this system, mainly, depends on the availability of an oxygen-rich stream, that is usually a very expensive and energy intensive product. The objective of this paper is the theoretical analysis of the performance of a gas turbine running under OEC conditions, with the oxidant ranging from atmospheric composition up to 30% in oxygen content.

The adopted methodology is a thermodynamic modeling of a gas turbine, considering the integration of heat exchangers to improve its efficiency. Five different assemblies have been studied, including intercoolers and regenerators, in order to check the behavior of the cycle coupled to these equipments and with variable oxygen content in the oxidant stream. Cycle overall efficiency and oxygen consumption were evaluated according to variable oxygen feed concentration under stoichiometric combustion and with oxidant excess. Additionally, an oxygen supply scenario for the studied case was evaluated, considering cryogenic air separation, adsorption process and membrane technology.

## 2. LITERATURE REVIEW

Poola *et al.* (1996) carried out studies of OEC in internal combustion engines for locomotives, and reported a 13% increase in thermal efficiency for naturally-aspirated engines and a 4% increase for turbocharged engines. According to Bisio *et al.* (2002), the barrier to couple oxygen to power cycles continues to be the high cost of oxygen production in cryogenic plants, but the use of membranes technology to obtain air enriched with 30-45% oxygen may offset the costs of oxygen implementation with the fuel saving obtained.

Wu *et al.* (2010) studied the influence of oxygen concentration from 21 to 30% in the combustion of natural gas in the heating and furnace-temperature fixing tests. The most attractive gain is that the fuel consumption at 30% $O_2$ was reduced by 26.1% if compared to atmospheric concentrations (21% $O_2$) when the furnace temperature was at 1493 K. In addition, the retrofitting of power plants to OEC is much easier and less expensive compared to the oxyfuel retrofit, being possible to enrich the air to 28% of oxygen without any modification in the combustion chamber, Baukal (1998).

The authors of this paper, Maidana *et al.* (2010) carried out simulations of a conventional open Brayton cycle in OEC with enrichment up to 30%, considering stoichiometric combustion. The results showed a decrease of 8.25% in fuel consumption and a reduction of 33.2% in flue gas volume, along with higher adiabatic flame temperatures. The study on the performance of a more realistic gas turbine is the goal of the present work, by adding some conventional heat recovery devices as regenerators and intercoolers, in order to improve the overall efficiency.

## 3. PROBLEM DEFINITION AND MODELING

A simple industrial gas turbine with OEC (Maidana *et al,.* 2010) was first taken as a reference cycle with the purpose of building a set of standard data to the analysis of this work. This base cycle was an open Brayton cycle (Fig. 1, left) composed by regular devices found on industrial gas turbines, as a multigroup compressor, an expansion turbine, an electric generator and a combustion system. Regular combustion systems are composed by a combustion chamber and auxiliary devices, as an air splitter and a gas mixer. For OEC, an oxygen injector was added before the combustion chamber input, along the oxidant stream, stage 4 at the same figure.

The base cycle and the full assembly cycle proposed in this work are shown in the Figure 1.
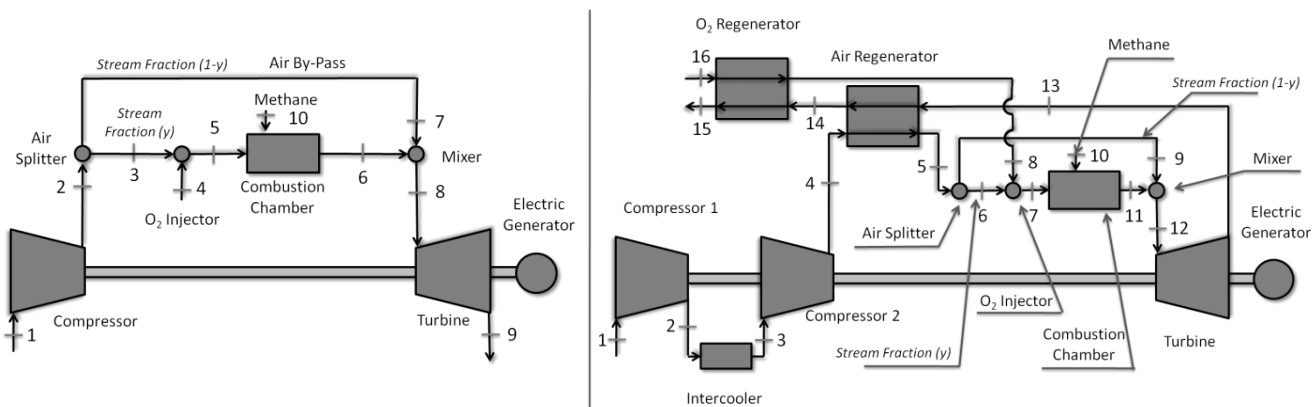


Figure 1 – a) Industrial gas turbine base cycle adapted for OEC; b) and complete proposed assembly.

For this base gas turbine, air was the working fluid and its operation followed a conventional Brayton cycle, besides the addition of an oxygen injection system to promote the oxygen enrichment. The complete assembly, depicted at Fig. 1, right, incorporated an air regenerator along stages 4 and 5, to recover heat exhausted from the flue gas stream (points 13 and 14), an intercooler at points 2 and 3, placed along the compressors. Finally, a second regenerator was proposed to exchange heat from flue gases (points 14 and 15) to the oxygen feeding stream (points 16 to 8). This kind of heat exchanger is seldom used to improve cycle efficiency. Four combinations of these heat exchangers were proposed, starting with the base cycle and ending up a complete assembly, as exposed in the Table 1.

Table 1: Proposed assemblies for the OEC gas turbine base cycle

| ASSEMBLY | DESCRIPTION |
|---|---|
| 1 | Base cycle |
| 2 | Base cycle + Intercooler |
| 3 | Base cycle + Air regenerator |
| 4 | Base cycle + Intercooler + Air regenerator |
| 5 | Base cycle + Intercooler + Air regenerator + $O_2$ regenerator |

All cases were modeled through mass, species and energy balances, and simulated with the aid of the Engineering Equation Solver (EES) software (www.fchart.com/ees), a Newton-Raphson based equation solver which provides an extensive thermodynamic properties database.

Compared to the original cycle proposed by Maidana *et al.* (2010), the isentropic efficiencies of the compressor stages and turbine has been changed from 60% to 80%, and from 80% to 86%, respectively. This modification has been introduced to follow values usually found in regular equipments as well as in the CGAM case, Bejan *et al.* (1996).

For all five cases the oxygen content at the feeding stream was ranged from 21%, atmospheric condition, to 30%. For oxygen/fuel ratios at stoichiometric limits, the combustion temperature was left free to evolve. In an alternative situation, for constant combustion temperature, an oxidant excess was promoted. As a standard to all cases the net power was set to 30 MW.

The main simplification hypothesis were the following: 1) steady state regime; 2) all chemical species are considered ideal gases; 3) pressure drop due to friction flow in piping and equipments is negligible; 4) heat transfer from piping and equipments to the environment is negligible; 5) fuel was composed by pure $CH_4$; 6) chemical dissociations during combustion, which led to the formation of traces of substances, are negligible; 7) compressor and turbine efficiencies are considered constant; 8) stagnation properties of the flow streams are negligible.

The following sections present the detailed modeling of the main devices in the complete assembly and its simplification hypothesis.

### 3.1. Compressor and intercooler

The intercooler was placed along points 2 and 3 (Fig. 1.b), in between the two groups of the air compressor. The main simplification hypothesis for both groups of the compressor were: 1) the same isentropic efficiency, $\eta_{ci}$ ; 2) the same compression ratio, $r_{pi}$.

The compression ratio for each group, $r_{pi}$, was defined by Eq. (1), while the total compression ratio, $r_p$, was given by Eq. (2):

$$r_{p,1} = r_{p,2} = \frac{p_2}{p_1} = \frac{p_4}{p_3} \tag{1}$$

$$r_p = r_{p1} r_{p2} \tag{2}$$

where $p_i$ is the pressure [kPa] at state $i$.

The setting of the same compression ratio for both compression groups is based on Boyce (2002) which states that the maximum efficiency for a two stage compressor is obtained when both compression ratios are equal to $\sqrt{p_4/p_1}$.

The mass balance for both groups was defined by Eq. (3):

$$\dot{n}_1 = \dot{n}_2 \qquad\qquad \dot{n}_3 = \dot{n}_4 \tag{3}$$

where $\dot{n}_i$ is molar flow rate [kmol/s].

The power consumption, $\dot{W}_{ci}$ [kW], and specific work, $w_i$ [kJ/kg], at each compression group were given by Eqs. (4) and (5), while the total power consumption in the compressor , $\dot{W}_c$ [kW], and the total specific compression work, $w_c$ [kJ/kg], were defined by Eqs. (6) e (7):

$$\dot{W}_{c1} = \dot{n}_1(\bar{h}_1 - \bar{h}_2) \qquad\qquad \dot{W}_{c2} = \dot{n}_3(\bar{h}_3 - \bar{h}_4) \tag{4}$$

$$w_{c1} = \frac{\dot{W}_{c1}}{\dot{n}_1 M_1} \qquad\qquad w_{c2} = \frac{\dot{W}_{c2}}{\dot{n}_2 M_2} \tag{5}$$

$$\dot{W}_c = \dot{W}_{c1} + \dot{W}_{c2} \tag{6}$$

$$w_c = w_{c1} + w_{c2} \tag{7}$$

where $M_i$ [kg/kmol] is the molecular mass of mixture.

The properties of all gas mixtures were evaluated as:

$$\bar{h}_i = \sum_{k=1}^{N} x_{k,i} \bar{h}_{k,i} \tag{8}$$

$$\bar{s}_i = \sum_{k=1}^{N} x_{k,i} \bar{s}_{k,i} \tag{9}$$

$$\dot{n}_i = \sum_{k=1}^{N} x_{k,i} \dot{n}_{k,i} \tag{10}$$

$$M_i = \sum_{k=1}^{N} x_{k,i} M_{k,i} \tag{11}$$

taking $k$ as the k-nth component of the mixture ($k = O_2, N_2, CO_2, H_2O, CH_4$) and $N$ as the number of pure substances in the mixture: $x_{k,i}$ as the molar fraction of the k-nth component of the mixture at state $i$; after that, $\bar{h}_{k,i}$, $\bar{s}_{k,i}$, $\dot{n}_{k,i}$ and

$M_{k,i}$ are, respectively, the specific enthalpy [kJ/kmol], the specific entropy [kJ/kmol K], the molar flow rate [kmol/s] and the molecular mass [kg/kmol] of the k-nth component of the mixture at state $i$.

The intercooler was modeled as a heat exchanger with ambient air as the cold stream. There's no interest in the outlet condition of the cold stream, thus the modeling of the intercooler is very simplified, Eqs (12) and (13):

$$\eta_{it} = \frac{\bar{h}_2 - \bar{h}_3}{\bar{h}_2 - \bar{h}_1} \tag{12}$$

$$\dot{Q}_{it} = \dot{n}_2(\bar{h}_2 - \bar{h}_3) \tag{13}$$

where $\eta_{it}$, is the intercooler efficiency and $\dot{Q}_{it}$ [kW] is the heat exchanged with the environment.

The operation conditions of this equipment, without gas extraction and neglecting flow friction, led to:

$$\dot{n}_2 = \dot{n}_3 \qquad\qquad p_2 = p_3 \tag{14}$$

### 3.2. Air and oxygen regenerators

The air regenerator efficiency, $\eta_{reg}$, and the air and flue gas mass balances are expressed by Eqs (15) and (16), Kakaç and Liu (1997):

$$\eta_{reg} = \frac{T_5 - T_4}{T_{13} - T_4} \tag{15}$$

$$\dot{n}_4 = \dot{n}_5 \qquad\qquad \dot{n}_{13} = \dot{n}_{14} \tag{16}$$

The energy balance is given by Eq. (17):

$$\dot{n}_{13}(\bar{h}_{13}^* - \bar{h}_{14}^*) = \dot{n}_4(\bar{h}_5 - \bar{h}_4) \tag{17}$$

The oxygen regenerator modeling is very similar to the air regenerator. Its efficiency, $\eta_{reg,O_2}$ mass balances and energy balance are analogous to Eqs. (15) to (17), respectively.

In the Eq. (17), $\bar{h}_i^*$ [kJ/kmol] is the flue gas enthalpy at state $i$, due to its temperature variation only:

$$\bar{h}_i^* = \sum_{k=1}^{N}(x_{k,i}\bar{h}_{k,i} - x_{k,i}\bar{h}_k^0) \tag{18}$$

where, $\bar{h}_k^0$ [kJ/kmol] is the formation enthalpy of the k-nth component at 298.15 K and 1 atm.

This approach is necessary due to: 1) there is no chemical reaction inside the regenerator, thus the heat exchange is driven only by temperature difference; 2) the equation solver applied in this analysis is unable to evaluate formation enthalpy and state enthalpy distinctively; in addition, many substances found in the gas mixtures are reported at different reference temperatures, as well as the formation enthalpy of $O_2$ and $N_2$ are null at ambient temperature.

According to third simplification hypothesis:

$$p_4 = p_5 \qquad\qquad p_{13} = p_{14} = p_{15} \qquad\qquad p_8 = p_{16} \tag{19}$$

### 3.3 Splitter, Oxygen Injector, Combustion Chamber and Mixer

The combustion chamber, the splitter and the mixer were modeled according to Carvalho (2006), besides the addition of an oxygen injector to enrich the oxidant stream. The chemical species balances were modified to achieve an oxygen enhanced combustion analysis.

After Figure 1, left, the compressed air stream coming from the compressor discharge is separated into two flows, at the splitter (point 2). The stream fraction *y* goes to the combustion chamber to react with the fuel, while the second one *(1-y)* bypasses the combustion chamber to be mixed to the flue gas at the dilution zone. This assembly is able to keep the flue gas temperature at the turbine inlet at constant 1373.15, Boyce (2002). The splitter modeling was also based on the mass and energy balances of Eqs. (20) to (23):

$$\dot{n}_6 = y\dot{n}_5 \tag{20}$$

$$\dot{n}_9 = (1 - y)\dot{n}_5 \tag{21}$$

$$\dot{n}_5\bar{h}_5 = \dot{n}_6\bar{h}_6 + \dot{n}_9\bar{h}_9 \tag{22}$$

$$p_5 = p_6 = p_9 \tag{23}$$

At the oxygen injector the air flow for combustion, $\dot{n}_6$, is enriched. This equipment was also modeled by energy and mass balances, Eqs. (24) to (26), with pressure remaining constant:

$$\dot{n}_7 = \dot{n}_6 + \dot{n}_8 \tag{24}$$

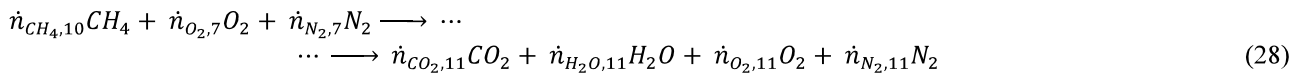$$\dot{n}_7\bar{h}_7 = \dot{n}_6\bar{h}_6 + \dot{n}_8\bar{h}_8 \tag{25}$$

$$p_6 = p_7 = p_8 \tag{26}$$

In addition, the oxidant stream composition is modified by the enrichment with oxygen and the molar fraction of the components in that stream is given by Eq. (27):

$$x_{N_2,7} = 1 - x_{O_2,7} \tag{27}$$

The combustion process was modeled by chemical equilibrium, considering the adiabatic behavior of the combustion chamber and the complete combustion of the fuel, avoiding the formation of pollutant traces, regardless the high temperatures that may occur during combustion reaction.

The general combustion equation for OEC was adapted from Baukal (1998), Eq. (28):

$$\dot{n}_{CH_4,10}CH_4 + \dot{n}_{O_2,7}O_2 + \dot{n}_{N_2,7}N_2 \longrightarrow \cdots$$
$$\cdots \longrightarrow \dot{n}_{CO_2,11}CO_2 + \dot{n}_{H_2O,11}H_2O + \dot{n}_{O_2,11}O_2 + \dot{n}_{N_2,11}N_2 \tag{28}$$

Carrying out the chemical species balances for carbon, hydrogen, oxygen and nitrogen, the following equations between reactants and products were obtained, Eqs. (29) to (33):

$$\dot{n}_{CH_4,10} = \dot{n}_{CO_2,11} \tag{29}$$

$$4\dot{n}_{CH_4,10} = 2\dot{n}_{H_2O,11} \tag{30}$$

$$2\dot{n}_{O_2,7} = 2\dot{n}_{CO_2,11} + \dot{n}_{H_2O,11} + 2\dot{n}_{O_2,11} \tag{31}$$

$$2\dot{n}_{N_2,7} = 2\dot{n}_{N_2,11} \tag{32}$$

$$\dot{n}_{11} = \dot{n}_{CO_2,11} + \dot{n}_{H_2O,11} + \dot{n}_{O_2,11} + \dot{n}_{N_2,11} \tag{33}$$

The energy balance for the combustion process is described by Eqs. (34) to (36):

$$H_r = H_p \tag{34}$$

$$H_r = \dot{n}_7\bar{h}_7 + \dot{n}_{10}\bar{h}_{10} \tag{35}$$

$$H_p = \dot{n}_{11}\bar{h}_{11} \tag{36}$$

where, $H_r$ [kW] is the enthalpy rate of reactants; $H_p$ [kW] is the enthalpy rate of products. Considering the process to be adiabatic, those two quantities must be conserved, Eq. (34).

For OEC processes an important parameter is the oxygen-fuel ratio, $S$, instead an air-fuel ratio, as usually adopted on conventional combustion processes. According to Baukal (1998), $S$ is defined by Eq. (37):

$$S = \frac{Oxygen\ molar\ flow\ in\ the\ oxidant\ stream}{Fuel\ molar\ flow} = \frac{\dot{n}_{O_2,7}}{\dot{n}_{CH_4,10}} \tag{37}$$

This definition differs slightly from the one commonly used in industry in which the stoichiometry is usually defined as the total oxidizer flow divided by the fuel flow. The problem with the definition commonly used in industry is that the stoichiometry must be recalculated whenever the oxidizer composition changes and that stoichiometric conditions change for each oxidizer composition. This is not a concern if air is always used as the oxidizer, which is the case for the vast majority of combustion processes. The benefit of the definition used here is that stoichiometry is independent of the oxidizer composition, so stoichiometric conditions are the same for any oxidizer composition. The value of $S$ for the stoichiometric case is 2, since its definition involves oxygen and fuel amounts, instead of air solely.

At the mixer, the bypass air is mixed with the flue gas to maintain the turbine inlet temperature, $T_{12}$, constant. This process is modeled using mass, energy and pressure analogous to the previously presented ones.

## 3.4 Turbine

The turbine was modeled in analogy to the compressor, except for having only one stage. The main parameter was the isentropic efficiency, $\eta_t$. The compression ratio is equal to the compressor one, thus:

$$p_{13} = p_{12}/r_p \tag{38}$$

The mass balance, the turbine power output, $\dot{W}_t$ [kW], and the turbine specific work, $w_t$, [kJ/kg] are given by Eqs. (39) to (41), respectively:

$$\dot{n}_{12} = \dot{n}_{13} \tag{39}$$

$$\dot{W}_t = \dot{n}_{12}\left(\bar{h}_{12} - \bar{h}_{13}\right) \tag{40}$$

$$w_t = \frac{\dot{W}_t}{\dot{n}_{12}M_{12}} \tag{41}$$

The net power output, $\dot{W}_{net}$ [kW] and the cycle efficiency, $\eta_{cycle}$, are evaluated by Eqs. (42) and (43):

$$\dot{W}_{net} = \dot{W}_e - \dot{W}_c \tag{42}$$

$$\eta_{cycle} = \frac{\dot{W}_{liq}}{\dot{n}_{10}M_{10}PCI} \tag{43}$$

## 4. OPERATION CONDITIONS

The operation conditions for each component in the cycle are shown below in the Table 2.

Table 2: Operation conditions for the main devices in the proposed assemblies

| Temperatures [K] | $T_1 = T_{10} = T_{16} = 298.15$ | | $T_{11}{}^* = 2000.15$ | $T_{12} = 1373.15$ | |
|---|---|---|---|---|---|
| Absolute pressures [kPa] | $p_1 = 101.3$ | | | $p_{16} = 1013$ | |
| Compression ratios | rp1 = rp2 = 3.16 | | | | |
| Molar fractions | $x_{O2,1} = x_{O2,5} = x_{O2,6} = x_{O2,9} = 0.21$ | | $x_{N2,1} = x_{N2,5} = x_{N2,6} = x_{N2,9} = 0.79$ | | $x_{O2,16} = 1.0$ |
| Isentropic efficiencies | $\eta_{C1} = \eta_{C2} = \eta_t = 0.86$ | | | $\eta_{it} = 0.80$ | |
| Regenerator efficiencies | $\eta_{reg} = \eta_{regO2} = 0.80$ | | | | |
| Net power [MW] | $W_{net} = 30$ | | | | |
| Oxygen-fuel ratio | $S = 2.0^†$ | | | | |

\* Only in the excess of oxidant condition

$^†$ Only in the stoichiometric condition

The air composition at the admission (point 1 at Figure 1) for all cases was simplified as a mixture of 21% of $O_2$ and 79% of $N_2$, in a way that others components, such as $CO_2$, $H_2O$, and others gases were negligible. The only points of the cycle that were modified in molar fraction and/or chemical reaction are the points 7, 11 and 12 of the Fig. 1, right. These $O_2$ e $N_2$ proportions at point 1 are also used in the points 2 to 6 and 9.

The content of oxygen at the injection stream (point 8) was a 100%, despite the fact that commercially it is available at a range of 88% to 99.5%. The oxidant stream that feeds the combustion chamber, point 7, is dosed by Eq. (27). If the $O_2$ fraction is enriched with oxygen, point 8, then $N_2$ fraction must be reduced to maintain the molar balance. It is important to stress that the inlet turbine temperature is fixed, 1373.15 K, leading to an unusual cycle efficiency loss in

the first proposed assembly, with intercooler only. In the excess of oxidant condition the temperature of the combustion chamber outlet is limited to 2000.15 K, then the oxygen-fuel ratio, and as a consequence the excess of oxidant, is calculated by energy balance. Similarly, for the stoichiometric condition, the oxygen-fuel ratio is assigned and the temperature of the combustion chamber is calculated by energy balance.

## 5. RESULTS

Simulations were done to evaluate the assembly efficiencies as a function of oxygen content in stoichiometric combustion, compared to the base cycle behavior. The results are presented in Fig. 2.
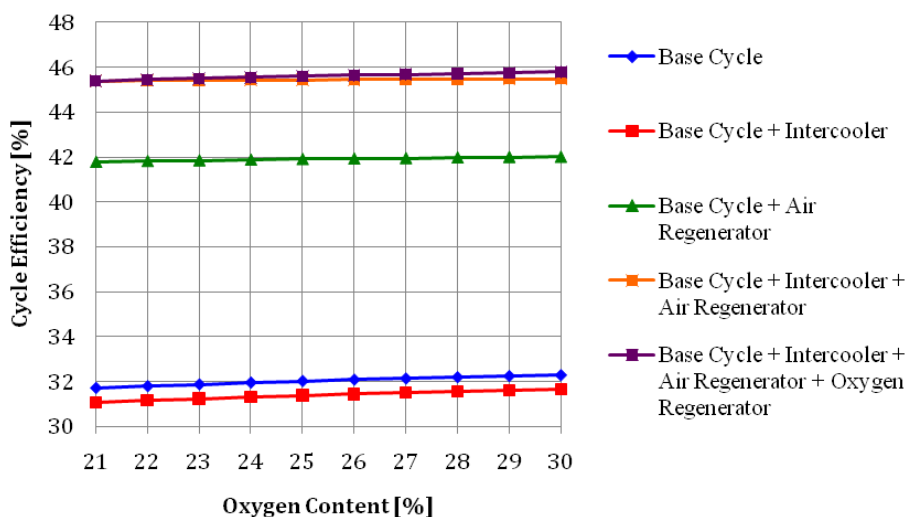


Figure 2 – Cycle efficiencies for the five different assemblies of the industrial gas turbine as a function of the oxygen content in the oxidant, in stoichiometric combustion.

As it can be verified, the oxygen enrichment causes the cycle efficiency to increase for three of the four proposed assemblies. The assembly with the intercooler only presented a decrease in the cycle efficiency compared to the base cycle, due to the impact of the decrease of the air temperature in the combustion chamber inlet, leading to a higher fuel consumption. However, the assemblies with intercooler and air regenerator present an increment in efficiency due to the combination of volumetric efficiency gain in the compressor, related to the intercooler, and heat regeneration to the combustion chamber inlet.

Also, it is possible to notice the decrease of the oxygen enrichment effect in cases which only the air regenerator is included, without oxygen regenerator. This effect can be justified due to the fact that oxygen is added at ambient temperature, resulting in a decrease on the combustion temperature. Thereby, the higher the enrichment level the lower the combustion chamber inlet temperature, reducing the effect on the efficiency gain.

According to the simulation results, the complete assembly displayed the highest cycle efficiency, around 45%, against 32% of the base cycle. In addition, it is remarkable that the main improvement in the cycle efficiency is due to the air regenerator. The assembly with air regenerator and intercooler increases the efficiency even further, while the case with intercooler only decreases the efficiency compared to the base cycle.

Fig. 3 displays a comparison of cycle efficiency as a function of the oxygen enrichment in the complete cycle in stoichiometric condition and in excess of oxidant condition. The temperature of the combustion chamber for the case of oxidant excess, 2000.15 K, was chosen according to Boyce (2002), as a way to limit the formation of $NO_x$ to an acceptable standard. According to this reference, for higher temperatures the formation of $NO_x$ increases rapidly and for lower temperatures the quantity of carbon monoxide and hydrocarbons not burnt is considerable.
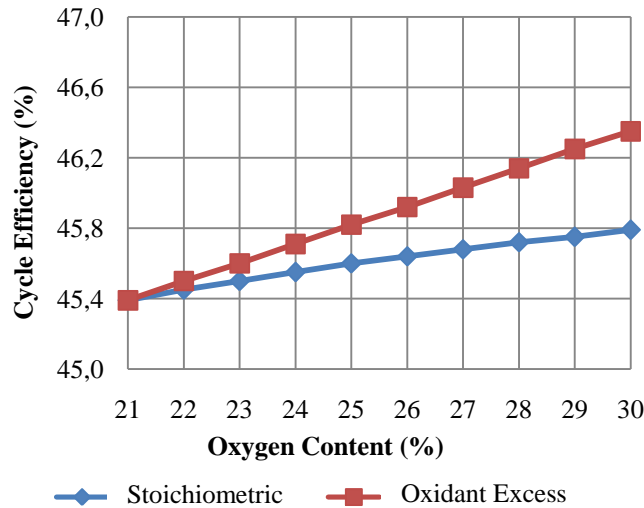
Figure 3 - Variation of cycle efficiency in function of the enrichment with oxygen in the complete cycle in stoichiometric and in excess of oxidant conditions

It is possible to notice that cycle efficiency increases 0.4% for stoichiometric condition and nearly 1% for excess of oxidant condition, in function of the oxygen enrichment. This difference between cases is due to the combination of two effects: the higher enthalpy of the mixture due to the higher amount of pressurized oxygen and the reduction of the compressor molar flow rate, hence the compressor work, as it can be seen in the Fig. 4b.

Fig. 4a displays the oxygen consumption and the Fig. 4b shows the compressor molar flow rate, both as functions of oxygen content in the oxidant, for the complete case in stoichiometric combustion and with oxidant excess combustion. The base case is shown to be compared.
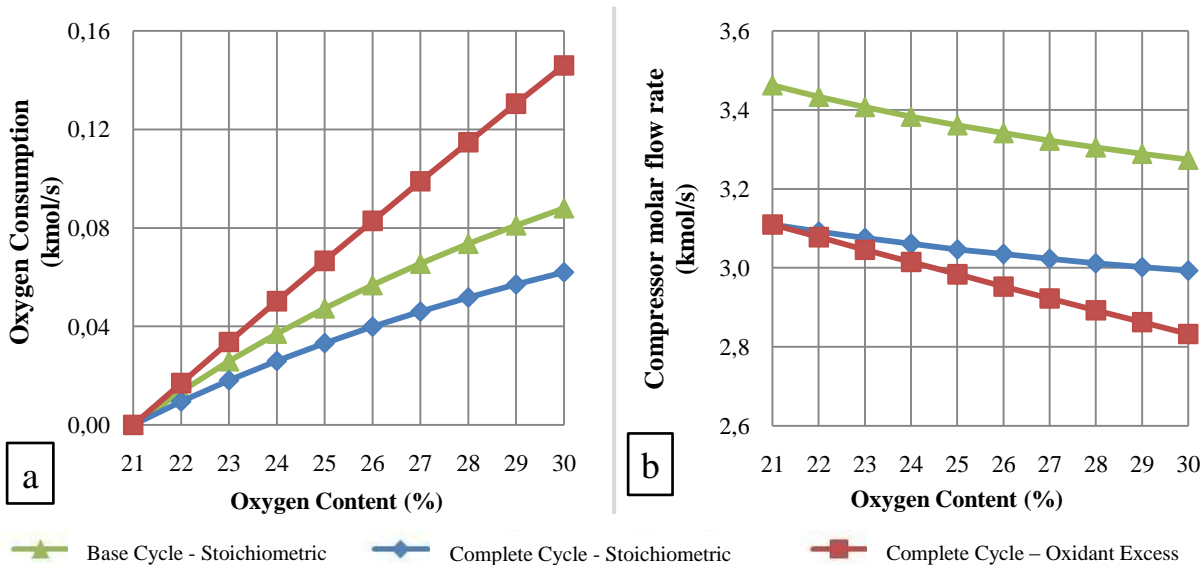


Figure 4: a) Variation of the oxygen consumption; b) Variation of the compressor molar flow rate, both as function of oxygen content in the oxidant stream

Figure 4a indicates the complete cycle in stoichiometric condition presents a decrease in the oxygen consumption compared to the stoichiometric base cycle, for the entire range of oxygen enrichment. Nevertheless, the complete cycle with excess of oxidant exhibits a steep increase of oxygen consumption with the oxygen enrichment.

In the Fig. 4b, while the increase in the oxygen content reduces the molar flow rate in the compressor, hence the compressor work also increases the oxygen consumption. However, the net work gain provided by the direct injection of pressurized oxygen in the combustion chamber needs to be balanced with the cost of the additional oxygen required.

## 6. OXYGEN SUPPLY SCENARIO

The commercial viability of oxyfuel and oxygen-enhanced combustion strongly depends on the oxygen supply related costs. In a simplified analysis, the oxygen costs will be offset by the decrease in fuel consumption or the increase in generated energy. However, most of the literature in air separation process is concerned with fraction purity and the coupling of product streams with other chemical facilities, lacking of adequate information to perform integration with the studied cycle.

Cryogenic air separation is currently the most efficient and cost-effective technology for producing large quantities of oxygen at high purity, Smith and Klosek (2001). In the literature, the integration of cryogenic facilities with power plants is focused on gaseous fuel production, where the process demands other resulting streams of the cryogenic separation, like nitrogen. Alternative integration is possible for gas-to-liquid coproduction or heat recovery for several chemical facilities. Hence, a cryogenic plant dedicated to oxygen supply for thermoelectric power plant seems unlikely without an extensive integration project.

Alternative air separation methods that were improved in the last decades include adsorption process, using zeolites, and polymeric membrane technology, Castle (2002). Both processes operate in near-ambient conditions, which results in lower energy consumption and a reduced start-up time, compared to cryogenic process. However, the purity of the separated streams is up to 95% to adsorption and lower than 50% to membranes. Besides that, the daily production of these systems is just a fraction of the cryogenic one and the purity of the sub products is considerably inferior.

Additionally, the adsorption beds operate in a batch basis, requiring a system with several beds and accumulation tanks to secure a continuous output of oxygen or enriched air, whilst a membrane can operate continuously. The application of polymeric membranes for OEC was evaluated by Coombe and Nieh (2007), in order to enrich air for soldier portable military power systems.

Considering the technical and economical limitations of those technologies, the enriched air supply through polymeric membranes raises as the most suitable process for integration with the studied cycle. Further studies are necessary to develop a better understanding of this separation process and its equipments.

## 7. CONCLUSIONS

The presented work is a theoretical analysis of the performance of a gas turbine running under OEC conditions, with the oxidant ranging from atmospheric composition up to 30% in oxygen content. The adopted methodology is a thermodynamic modeling of a gas turbine, considering the integration of heat exchangers to improve its efficiency. Five different assemblies were studied, including intercoolers and regenerators, in order to check the behavior of the cycle coupled to these equipments and with variable oxygen content in the oxidant stream.

It has been verified, for all assemblies, an increase in cycle efficiency and oxygen consumption associated to oxygen-enhanced combustion. Considering the base cycle, the complete assembly presents about 40% increase in cycle efficiency, with oxygen content ranging from 21 to 30%. Furthermore, in excess of oxidant condition, the cycle efficiency increases with the oxygen enrichment, compared to the stoichiometric condition, at a cost of higher oxygen consumption. Based on the results obtained in this work, the oxygen-enhanced combustion emerges as an alternative to increase the efficiency of thermal power plants with minor retrofitting.

However, the oxyfuel and oxygen-enhanced combustion strongly depends on the oxygen supply related costs. Cryogenic air separation plants remain the most efficient and cost-effective technology for producing large quantities of oxygen at high purity, although the implementation of this process seems unlikely without an extensive integration project. Aside from adsorption process units, polymeric membranes raises as the most suitable process for enriched air supply for the studied cycle. Additional studies are necessary to evaluate the capital costs related to the air separation facilities and new concepts in plant integration.

## 8. ACKNOWLEDGEMENTS

## 9. REFERENCES

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