THE INFLUENCE OF RECIRCULATION INTENSITY ON THE CO AND NO EMISSIONS

Gustavo Ronceros Rivas
Instituto Tecnológico de Aeronáutica – Pça. Mal. Eduardo Gomes, 50, São José dos Campos – SP – Brazil, 12228-900
gustavo@ita.br

Pedro Teixeira Lacava
Instituto Tecnológico de Aeronáutica – Pça. Mal. Eduardo Gomes, 50, São José dos Campos – SP – Brazil, 12228-900
placava@ita.br

Abstract. *The present work shows an experimental investigation about the influence of the presence of an internal recirculation zone (IRZ) in natural gas combustion process. Basically the experiments were conducted to verify the influence of the IRZ on the CO and NO emissions. A special burner was designed to allow changes in IRZ intensity, and the experiments were conducted in a refrigerated horizontal combustion chamber of laboratorial scale. The IRZ intensity was characterized by the swirl number and the Reynolds number of the natural gas jet, both calculated using the burner geometry and the reactants flow conditions. The results show the tendency of NO reduction when the IRZ intensity increases and that pollutant emission is not highly influenced by the fuel jet Reynolds number. However, the Reynolds number has an important influence in the CO emission; higher Reynolds numbers increase the mixing processes between combustion air and the fuel jet by entrainment effects. But increasing the IRZ intensity to minimize the NO emissions can produce higher CO levels. Therefore, the results pointed that the ideal IRZ to conciliate the NO and CO emissions is a compromise solution.*

Keywords*: swirler, pollutant emissions, natural gas combustion, swirl number.*

1. Introduction

Nowadays, the industrial applications that use the thermal energy realized by the exothermic combustion reactions require basically two things: high efficiency, and low pollutant emissions. The natural gas is a fuel that helps to get that conciliation, and it is economically viable. A lot of research works have demonstrated natural gas combustion may have lower emissions of partial oxidation pollutants (ex: carbon monoxide (CO), soot, and unburned hydrocarbons) than some others hydrocarbons, for example: coal, diesel fuel, heavy oil, and liquefied petroleum gas (LPG).

The use of natural gas does no mean low pollutant emission, it is necessary some operational strategy related to the combustion to really been possible to have an industrial process with less environmental impact. The low emission situation depends on the physical and chemical process that happen into the combustion chamber, which means, how the reactants are injected (Lyons, 1982). Moreover, in combustion equipments it is fundamental the correct design of the air distribution system and the flame holder, due their influences on the flame dimensions, fuel consumption, emissions of CO and NO_x (Kuo, 1986).

Swirlers have been frequently utilized in modern industrial plants and gas turbines as substitute of other kinds of flame holders. When the combustion air flow passes through the axial swirler a rotational movement is introduced, which outside the swirler creates a radial pressure gradient. Therefore, if rotation is sufficiently intense, the adverse pressure gradient creates an internal recirculation zone (IRZ) (Oliveira, 2001, Chigier, 1981). This kind of flame holder have been preferred than bluff bodies, because, in spite of its higher efficiency, there are satisfactory analytical formulations to design and to predict the behavior, in contrary of the bluff bodies, which depends on the empirical formulation.

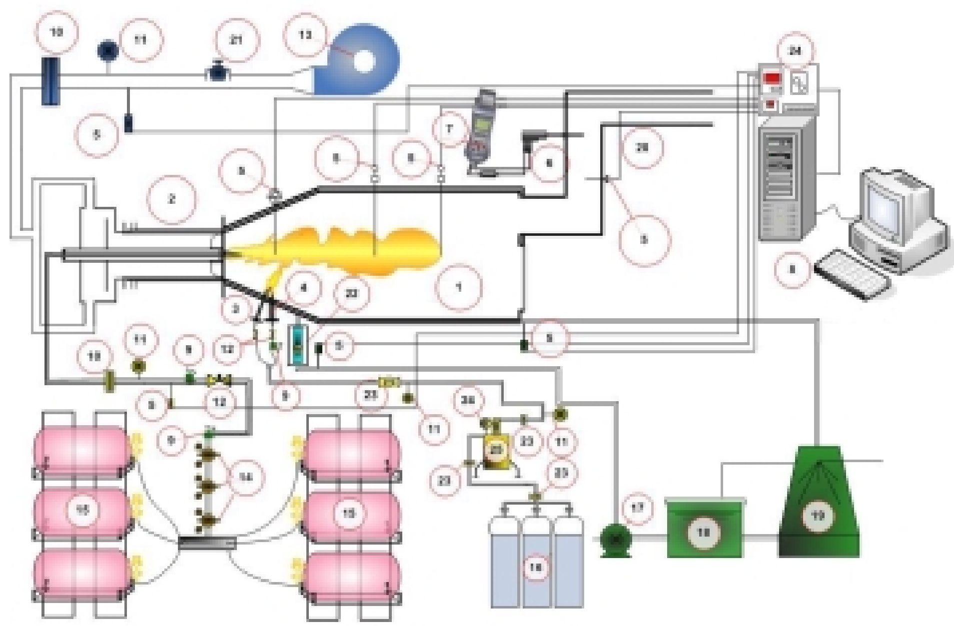
The intensity of the IRZ established in front of the burner exit will depend on the swirler geometry (number of blades, passage area, blades angle, etc.), combustion air that flows through the swirler, and the fuel jet velocity (Muniz, 1993). Muniz et al (2000) have shown the interaction between the IRZ and the Reynolds number of LPG fuel jet changes considerably the flame shape and the temperature across the flame. In this case, the variation of IRZ intensity was done modifying the blades angle, and Reynolds number changing the orifice diameter of the gaseous fuel injection.

The non-dimensional relation that quantifies the IRZ intensity is the swirl number, which expresses the ratio between the axial component of the flux of angular momentum and the axial component of the flux of linear momentum (Beér and Chigier, 1972, Wall, 1987). For swirl number inferiors than 0.6 ($S' < 0.6$) the IRZ intensity is considered weak, and long flames will established, but the presence of this recirculation is important to hold the flame. For stronger swirl numbers, there is a tendency of flames partially confined into IRZ. Therefore, the intensity of the swirl number interferes on the mixing process between the gaseous fuel and combustion air, changing the physical and chemical processes that happen in the combustion region. So, there is a direct relation between flame structure as consequence the IRZ intensity, and the pollutants formation.

The objective of the present work is to evaluate experimentally how the presence and the intensity of a recirculation zone affect the natural gas combustion, mainly on the pollutants emission CO e NO, into a refrigerated laboratorial furnace. The motivation was, in spite of the increase of the Brazilian consumption of natural gas, there is a clear necessity for more knowledge about the natural gas combustion for industrial processes.

2. Experimental Setup

The experiments were conducted into a circular modular combustion chamber, with refrigerated wall, and positioned horizontally. The first module is conical with extreme diameters of 0.20m and 0.455m, length of 0.40m, and a swirl burner is coupled at the 0.20m diameter face. The second module is cylindrical with diameter of 0.455m and length of 1.0m, and at its upper side is positioned the exit of the combustion gases. The Fig. 1 presents a diagram of the combustion chamber and the experimental setup utilized. All chamber modules are refrigerated by water recirculation through the jackets around the chamber walls. The water refrigeration was done by closed system, as presented in Fig. 1, and the water mass flow rate was measured by rotameter and the inlet and outlet temperatures by thermocouples.



(1) combustion chamber	(2) burner	(3) pilot burner	(4) spark
(5) thermocouple	(6) probe for sample gases	(7) gases analyzer	(8) data acquisition system
(9) solenoid valve	(10) orifice plate flow meter	(11) manometer	(12) control flow valve
(13) radial compressor	(14) pressure regulator	(15) natural gas cylinders	(16) LPG cylinders
(17) centrifugal pump	(18) water reservatory	(19) refrigeration system	(20) combustion gases duct
(21) air control flow valve	(22) rotameter flow meter	(23) blockage valve	(24) temperature indicator
(25) LPG electrical heater			

Figure 1. Experimental setup.

The combustion air is admitted into a chamber by two radial inlets, and in the sequence it is forced to pass through fixed plates, which conducts it to a duct where the flow is established preferentially at axial direction before the swirler blades, where the flow acquires the tangential velocity component. The swirler blades angle (α) could be changed from 0 to 70^o, steps of 5^o. Therefore, it was possible to change the recirculation zone intensity by the angle α, keeping the fuel and the air mass flow rates constants. At the center of the burner was positioned the fuel gas injector. Three different diameters of fuel injection orifices were utilized to change the Reynolds number of the fuel jet, 4mm, 5mm, and 7.1mm. The natural gas flow rate was 1 g/s for all experiments; then, the Reynolds numbers verified were 30,449, 24,978, and 18,435, respectively. The Figure 2 presents the swirl burner.

Two additional channels for fuel and combustion air injection were predicted to capacity the burner for fuel or air staged combustions, which are strategies to control the NO emission. However, in the present work, the results for staged combustion is not presented, due the focus of the present work is juts the influence of the recirculation zone on the natural gas combustion, mainly on the CO and NO emissions.

The combustion gases were sampled at 0.62m after the strangeness in the gas exit duct. It was verified the homogenous radial distribution of O_2, CO and NO. To obtain the volumetric percentage of these three gases it was utilized an electrolytic cell analyzer Unigas 3000$^+$. The combustion gases were sampled with a heated probe, after they were filtered, dried, and then directed to continuous analyser. The figures for gas concentrations are, therefore, presented on dry basis, and to correct the air excess dilution effect, the CO and NO emissions were converted to the base of 7% of O_2. The analyser accuracies are: O_2: ±0.1% vol.; CO: ±10 ppm <300 ppm, ±4% rdg up to 2000 ppm, ±10% rdg > 2000 ppm; NO: ±5 ppm < 125 ppm, ±4% rdg >2000 ppm.

Figure 2. Swirler burner.

3. The Swirl Number Model

The non-dimensional parameter that characterizes the intensity of the swirler, used here to quantify the intensity of recirculation zone, is called swirl number (S´), which corresponds to the ratio between the axial component of the flux of angular momentum and the axial component of the flux linear momentum. The model utilized to calculate the swirl number is basically the same presented in Wall (1987) with the consideration of blockage described by Muniz (1993) and Couto et al (1995). The Fig 3 presents a general scheme of an axial swirler and its geometry.

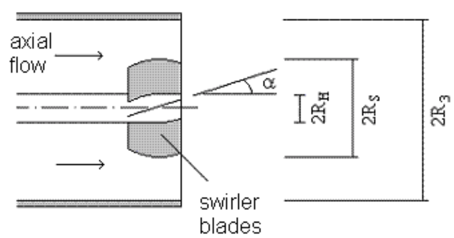


Figure 3. General scheme of an axial swirler.

The swirl number may be defined based on velocity distribution across the swirler (Wall, 1987), and for the present case where just the secondary flow (combustion air) passes through swirler blades it may be written as (Couto et al, 1995):

$$S' = \frac{S_s'}{(1+Mr)}, \tag{1}$$

being S_s' is the swirl number of the secondary flow and Mr is ratio between the axial component of the flux of linear momentum of the primary flow (jet of gaseous fuel) and axial component of the flux of linear momentum of the secondary flow (combustion air). S_s' and Mr may be calculated as:

$$S_S' = \frac{G_\phi}{(R_3 - R_H).G_x'}, \tag{2}$$

$$Mr = \frac{(\rho/m^2)_S.(m^2/\rho)_P.(R_3^2 - R_H^2)}{R_I^2}, \tag{3}$$

ρ is the density, m is the mass flow rate, R_I the radius of the fuel injection, and the subscriptions p and s are referents to the primary and secondary flows, respectively. The ratio between the axial components of the flux of angular momentum and of the flux of linear momentum is given by:

$$\frac{G_\phi}{G_x'} = \frac{2.(CB).\tan g(\alpha).(R_s^3 - R_H^3)}{3.(R_3^2 - R_H^2)}, \tag{4}$$

where α is the angle between the swirler blades (see Fig. 3), and CB is the blockage coefficient due the finite thickness of the directional blades. The calculate of the blockage coefficient was done following the methodology presented by Couto et al (1995), which considers, yonder the finite thickness of the blades, the influence of Reynolds number despicable. Therefore, the blockage coefficient depends on the geometrical dimensions of the blades in the straight line perpendicular to the duct axis, calculated as:

$$CB = \frac{1}{(1-\sigma)}, \tag{5}$$

and σ is the blockage factor :

$$\sigma = \frac{(A_S - A_{ef})}{(A_{3a} - A_S)}, \tag{6}$$

being: A_{3a} the annular area between the swirler and duct wall, A_s the circular area of the swirler exit, and A_{ef} the effective area of the flow through the swirler blades, which is determined geometrically by:

$$A_{ef} = Z.(R_S - R_H).(K - 2T).\cos\alpha\,, \tag{7}$$

where K is determined by:

$$K = \cos(\pi / 2.Z).[R_s.\sin(\pi / Z) + R_H.\tan(\pi / Z)], \tag{8}$$

being Z the numbers of swirler blades and T their thickness. For the present case, the swirler dimensions are: $R_3 = R_s =$ 77.5mm, R_H = 20mm, R_I = 4mm, 5mm, and 7,1mm (depending the fuel jet Reynolds number desired), α between 0^o and 70^o, Z = 8, and T = 1,5mm). For the burner used here $R_3 = R_s$, so the experiments were realized at the condition of maximum blockage for a determined swirler angle α.

The experimental conditions may be resumed as: 1) fuel flow rate: kept constant at 1g/s for all experiments; 2) equivalence ratio (ϕ): changed by the combustion air flow rate; 3) fuel jet Reynolds number: changed by the diameter of the orifice of natural gas injection, 4) swirl number: changed by the angle α between the swirler blades (for a fixed Reynolds number and equivalence ratio). The Figures 4, 5 and 6 present the swirl number as function of equivalence ration for different swirler angle α, and for natural gas jet orifice of 4mm, 5mm, and 7.1mm, respectively.

4. Results

The Figures 7, 8 and 9 present the CO emissions corrected to $O_{2(7\%)}$ for the three different injection orifice diameters, 4mm, 5mm, and 7,1mm, respectively.

For the orifice of 4mm and equivalence ratio inferior of 0.7, the results are in the same order of the analyzer accuracy, ± 10 ppm; then, the variation on CO emission up to this point of equivalent ratio is quite small. However, for the equivalence ratio of 0.75 the emission increased to 17.80 ppm for S' = 0.91 ($\alpha = 65^o$), and to 39.19 for S' = 1.44 ($\alpha = 70^o$). In any case these CO emissions can be considered very low when compared to the emissions for the others injection orifices.

In spite of the recirculation zone stabilized in front of the burner exit, the good results for CO emissions may be a consequence of the higher Reynolds number, Re = 30,449, which intensify the air entrainment in the interface between the fuel jet and oxidant. This effect may influence the local mixing rate of the reactants, accelerating the physical steps of combustion process, bringing forward the CO to CO_2 conversion.

When the Reynolds number of the gas jet decreases to 24,978 by the change of the injection orifice to 5mm of diameter, the CO emissions increased considerably regarding to Reynolds number of 30,449. For any angle α, the CO emission has not an expressive change between the interval of equivalence ratio 0.60 to 0.75. However, after this interval, there is an exponential increase on the CO emission up to equivalence ratio of 0.9. It is also possible to observe that, for any equivalence ratio, the intensification of the recirculation zone by the increase of the angle between the blades increases the CO emission. It is possible that part of the fuel has been captured by the recirculation structure. The increase of swirl number and the reduction of Reynolds jet number may conduct the combustion into recirculation zone to a rich burn, where the presences of CO and Unburned Hydrocarbons are higher. In the present experiment the wall chamber is refrigerated; therefore, it is possible part of the CO formed into this region has not been converted to CO_2, due the lower temperature along the combustion chamber, mainly close to the wall. On average, more then 60% of total energy liberated into the combustor is absorbed by the water or lost by heat transfer.

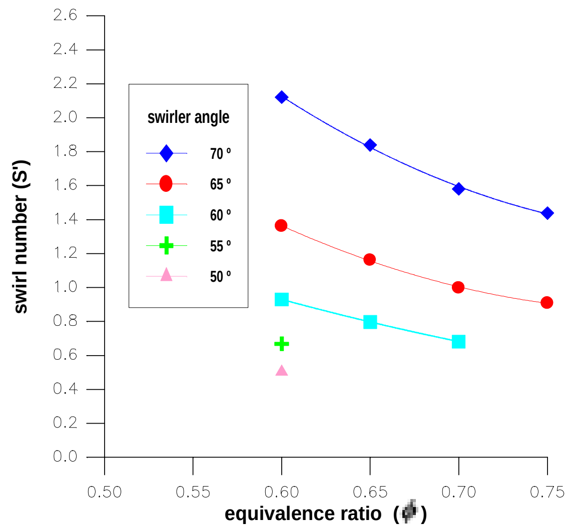


Figure 4. Swirl number as function of equivalence ratio for gas injection diameter of 4mm.

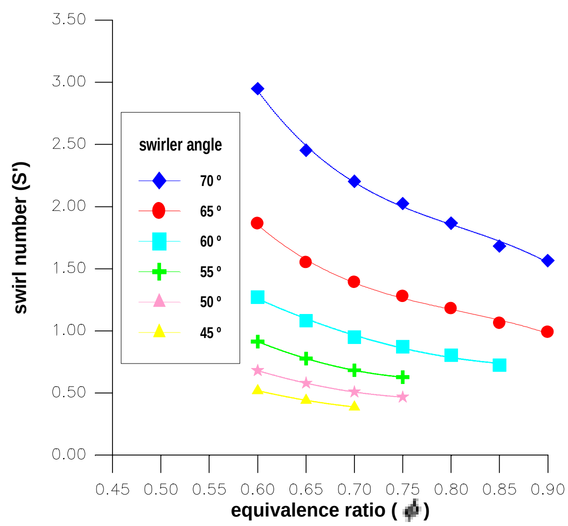


Figure 5. Swirl number as function of equivalence ratio for gas injection diameter of 5mm.

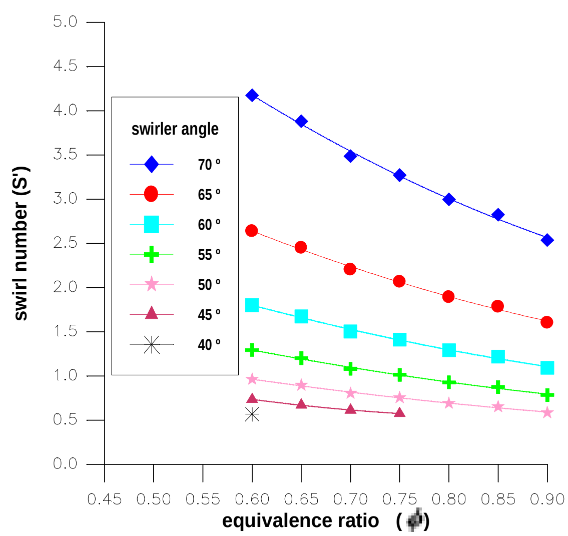


Figure 6. Swirl number as function of equivalence ratio for gas injection diameter of 7.1mm.

The influence of the swirl number on the CO emission presented here is qualitatively in agreement with other authors: Lacava et al (2000) for diesel fuel (ϕ = 0.77, and S´= 0.5 to 8); Bizzo (1997) for fuel mixture of 70% of LPG (Liquefied Petroleum Gas) and 30% of toluene (ϕ = 0.70 and 0.84, and S´= 0.3 to 1.45); Ballester and Dopazo (1994) and Barreiros et al (1993), both for liquid fuels in several conditions.

The results also show that keeping the swirler blades angle constant; for example 65^o, while the equivalence ratio increases, there is tendency to reduce the swirl number, as presented in Fig.´s 8 and 9; then, the CO emission must reduce too, as commented before. However, there is a reduction in the oxidant quantity into the chamber, making difficult the mixing process between the fuel and the oxidant, increasing the CO emission. Apparently, for the orifice diameter of 5mm, these effects are compensative, and there is not significant modification on CO emissions between the equivalent ratios of 0.6 to 0.75 for the different angles, according to Fig. 8. However, for equivalence ratios higher than 0.75 the CO emission increases considerably; apparently the reactants mixture is deficient, due the reduction on the air flow rate.

Reducing the Reynolds number to 18,435 (orifice diameter of 7,1mm) the CO emission levels increase dramatically, as pointed in Fig. 9. Probably, three factors influenced this result: 1) for the same equivalence ratio and swirler angle, the swirl number increases with the Reynolds number reduction, so that, there is a tendency to rich combustion into recirculation zone; 2) for the Reynolds number lower, there is a tendency to reduce the air entrainment effect; 3) visually it was observed for this particular Reynolds number the flame was more susceptible to convective effects, due the lower momentum in the jet fuel axial direction. This last factor happens because the combustor is positioned horizontally and the combustion gases exit is positioned at the superior part of the burner opposite extremity, as summarized in Fig. 1. Visually was observed that flames which the Reynolds number lower have an inclination to the superior part of the combustion chamber, and in the major part of the cases the flames "touched" the refrigerated wall, probably quenching the conversion process of CO to CO_2. Moreover, it is important observe the particulate material filter reaches the saturated condition more rapidly for the Reynolds number of 18,435 than the others two higher, 24,978 and 30,449.

The Figure 9 also shows for the orifice diameter of 7.1mm that lower equivalence ratios have more CO emission, contradicting the theoretical expectative. Two aspects may explain this fact. Firstly, the convective effects are more drastic for combustion with lower equivalence ratio. In this case, as consequence of lower temperatures reached into the combustion chamber, the velocity of CO conversion is slower and when the reactive flow is close to the refrigerated wall it is easer to quench the conversion reactions. The second point is the residence time, which diminishes when the equivalence ratio reduces (the air mass flow rate reduces), disposing less time to CO conversion. For example, for α = 70^o and ϕ = 0.6 the residence time was 2.7 seconds (calculated assuming atmospheric pressure into the chamber and the average temperature the flue gases temperature), while for ϕ = 0.9 was 3.2 seconds. This comparison is just qualitative, because the temperatures into the chamber are higher than in the flue gases, and if the temperature integration was realized for both cited situation, the residence times will be lower. For the others two higher Reynolds number these two aspects are not important, because for these cases the convective effect has less influence, and lower residence time is compensated by the air entrainment effect.

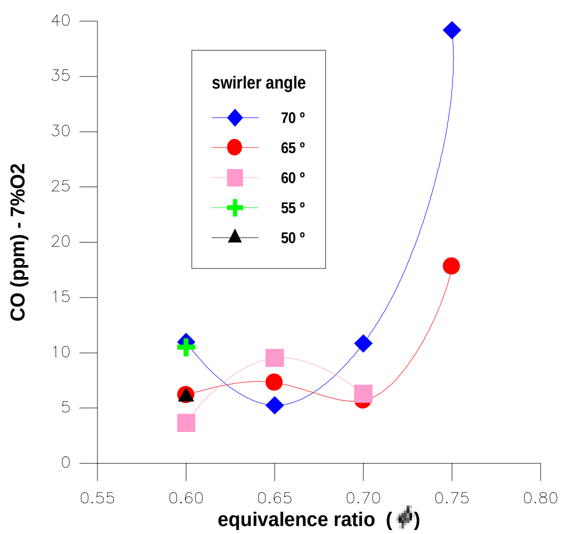


Figure 7. CO emission as function of equivalence ratio for gas injection diameter of 4 mm.

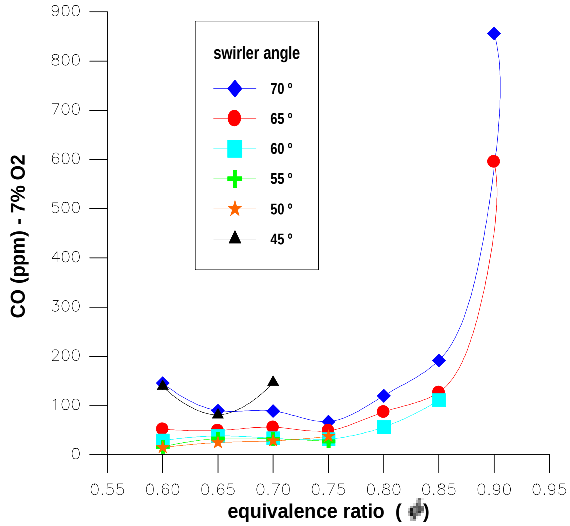


Figure 8. CO emission as function of equivalence ratio for gas injection diameter of 5 mm.

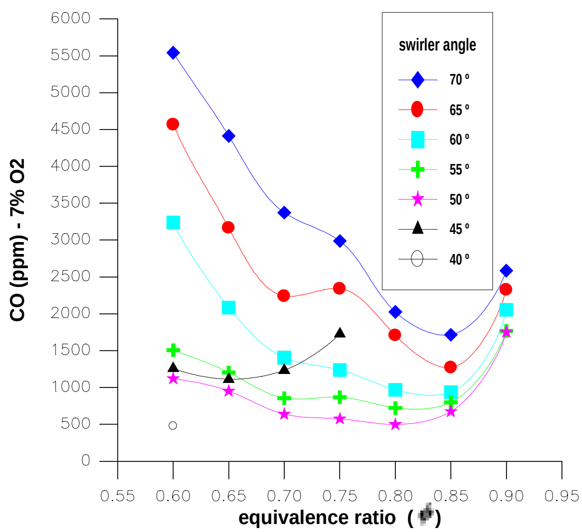


Figure 9. CO emission as function of equivalence ratio for gas injection diameter of 7.1 mm.

The Fig's 10, 11, and 12 show the NO emissions as function of the equivalence ratio for different swirler angles, and for different orifice diameters of fuel injection, 4mm, 5mm, and 7,1mm, respectively. On average, the level of NO emission was relatively low, have never been superior of 35 ppm corrected for 7% of O_2. This behavior is consequence of the fact the combustion chamber wall is refrigerated, where 60% of total energy liberated into the combustor is absorbed by the water or lost by heat transfer, as commented before. Then, the temperature levels reached into the combustion chamber in this case are lower than the expected for a furnace with refractory wall, for example. As the principal mechanism of NO formation is extremely dependent of temperature (Thermal Mechanism), it is foreseeable lower emission of NO in refrigerated chamber.

For the NO emissions observed in the experiment, the analyzer accuracy was ± 5 ppm, and sometimes it was the order of the emission variation when some parameter was changed, complicating the results analysis. However, it is possible to affirm the changes in Reynolds number practically have not modified de NO emission; on the contrary of CO emission.

Apparently the results for 4mm and 5mm show the increase on the equivalence ratio reduces the NO emission. Theoretical calculations of chemical equilibrium or chemical kinetics probably would point the contrary, reducing the air quantity from a condition of lean combustion; the NO emission will increase until the maximum close to the equivalence ratio of 0.85. However, it is necessary to take into account the effects of the flow dynamic in the recirculation region to understand what is happing on the NO emission.

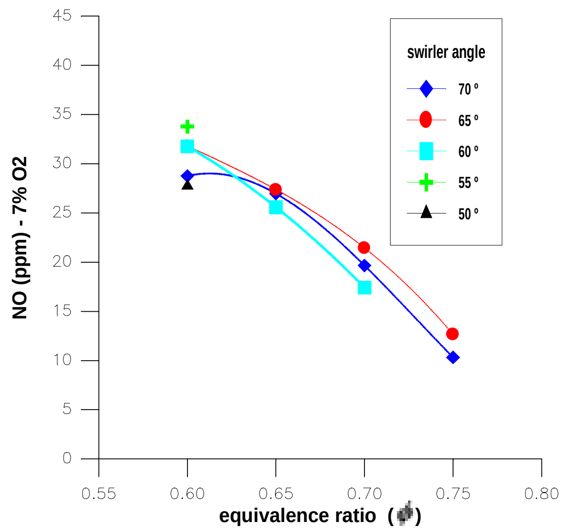


Figure 10. NO emission as function of equivalence ratio for gas injection diameter of 4 mm.

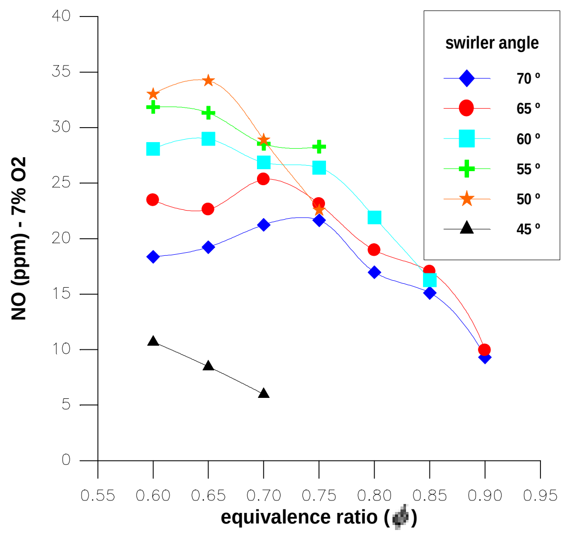


Figure 11. NO emission as function of equivalence ratio for gas injection diameter of 5 mm.

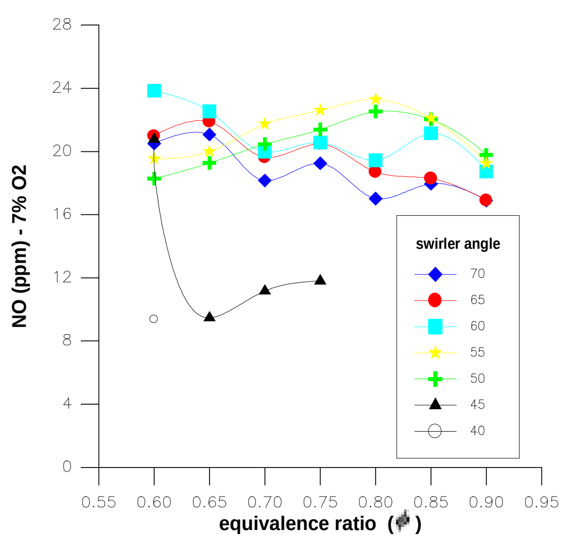


Figure 12. NO emission as function of equivalence ratio for gas injection diameter of 7.1 mm.

Probably this tendency is related to the weakened of the recirculation zone when the air mass flow rate is reduced, and there is a dispersion of region of intense energy liberation. On the contrary situation, the concentration of energy liberation is more appropriated to NO formation, because it promotes regions with higher temperature. On the other side, the dispersion of energy liberation prevents regions with temperature peaks.

For the orifices of 4mm and 7,1mm, the variation of NO emission as function of swirler angle for a determined equivalence ratio is in the range of the analyzer accuracy, and few comments could be done. On the other side, for the orifice of 5mm it is possible to observe the increase on swirl number by the swirler angle adjust reduces the NO emission. For example, the NO emission for the equivalence ratio 0.6 and $\alpha = 50^o$ was $33,0 \pm 5$ ppm; when α increases to 70^o the emission reduces to $18,4 \pm 5$ ppm. This behavior has been reported by other authors. The experiments conducted by Ballester and Dopazo (1994) and Barreiro et al (1993) indicated the increase on swirl number creates a rich combustion into recirculation zone, which is an unfavorable condition to NO formation. Heap et al (1972) observed for natural gas flames stabilized by intense swirl, beyond the rich character in recirculation zone, the swirl flow captures to this region a considerable quantity of burned gases, reducing the temperature in some parts of combustion region, quenching the mechanisms of NO formation.

Lacava et al (2000) have compared the NO emissions in the flame region and in the exit of a non-refrigerated laboratorial furnace for different conditions of swirl number for diesel fuel combustion with air; they have conclude the NO formation by Prompt Mechanism is not despicable, and the recirculation zone whit rich combustion is favorable to convert the Prompt NO to N_2.

The Prompt NO Mechanism involves intermediary radicals of the hydrocarbon oxidation, and the main route for NO formation is the fixation of the molecular nitrogen by fragmented hydrocarbons through the reactions (Syska, 1993):

$$CH + N_2 \Leftrightarrow HCN + N, \quad (R.1)$$
$$CH_2 + N_2 \Leftrightarrow HCN + NH, \quad (R.2)$$
$$N + OH \Leftrightarrow NO + H. \quad (R.3)$$

However, the other products (HCN and NH) will be partially converted to N_2. In the case of rich flame, the NO formed may be reduced to N_2, initially trough the reaction (R.4), which converts NO to HCN, and this last one to N_2. Splithoff et al (1996) also presented for pulverized coal the increase of temperature in rich combustion region accelerates the nitrogen species decomposition, reducing the NO formation. Therefore, the intense recirculation zone creates a region with combustion condition predominantly rich and with high energy liberation, favorable to reduce the NO formed by the prompt mechanism.

$$CH + NO \Leftrightarrow HCN + O. \quad (R.4)$$

Besides Ballester and Dopazo (1994), Barreiros et al (1993), Heap et al (1972), and Lacava et al (2000), others authors have pointed the tendency of NO reduction when the swirl number increase. Hill and Smoot (2000) based on experimental results and simulations show for pulverized coal combustion with air there is an accentuated reduction on the NO emission up to swirl number close to three; after that, the reduction is smooth with progressive increase on the swirl number. This behavior was also observed by Lacava et al (2000) and Barreiros et al (1993), for diesel fuel and heavy oil fuel, respectively. Hsieh et al (1998) have shown the tendency of NO reduction when the swirl number increase for gas fuels in five different furnaces (different scales) and power between 30 kW and 12 MW. Lacava et al (2004), using diesel fuel and LPG as auxiliaries fuel in an aqueous waste incinerator operating with enriched flame (oxidant 50% O_2 and 50% N_2) also identified the NO reduction as consequence of higher swirl number; moreover, the reduction was more pronounced for the LPG.

5. Conclusions

The present work shows the results for an experimental investigation about the influence of the presence of an internal recirculation zone on the natural gas combustion process, mainly on CO e NO emissions. Therefore, the conclusions based on the results are:

1) In spite of the recirculation zone stabilized in front of the burner exit, for the higher Reynolds number, Re = 30,499, the CO emissions were low, which probably is related with the air entrainment effect.
2) When the Reynolds number diminishes, enhance considerable the CO emission, and the increase of swirl number worsening this pollutant emission.
3) On average, the level of NO emission was relatively low, it have never been superior of 35 ppm corrected for 7% of O_2. This behavior is consequence of the refrigerated wall, where 60% of total energy liberated into the combustor is absorbed by the water or lost by heat transfer.
4) The variation of the Reynolds number of natural gas has not changed significantly the NO emission, just in the range of analyzer accuracy.

5) The increase on swirl number has reduced the NO emission as observed by several authors cited. The intensification of recirculation zone creates a rich combustion region in front of the burner, which is unfavorable to NO formation. Moreover, the recirculation of combustion gases to this region of rich combustion diminishes the temperature and reduces the NO formation. In addition, the presence of radicals from hydrocarbon rich combustion may convert to N_2 the NO formatted.

The results presented and commented in the present work are important to understand the combustion process in natural gas swirl burners, and it helps to take the decision about design parameters. However, the present work just observes the global aspects of combustion, and it did not extend the experiments to details for the flow into the recirculation zone. Therefore, that is the natural recommendation for future works.

6. References

Barreiros, A.; et al. Predictions of near burner region and mensuraments of NO_x and particulate emissions in heavy fuel oil spray flames, **Combustion and Flame**, v.92, p.231-240, 1993.

Beer, J. M; Chigier, N. A. Combustion aerodynamics. New York: John Wiley and Sons, 1972.

Bizzo, W.A. Emissão de monóxido de carbono de hidrocarbonetos totais em câmara de incineração-efeito do número de rotação e composição do combustível, Tese de Doutorado, Universidade Estadual de Campinas, Campinas, 1997.

Chigier, N. Energy, Combustion and Environment. New York: McGraw– Hill, 1981.

COUTO, H. S ; et al. Geometrical parameters for flows across axial swirlers, Proceedings of the third Asian- Pacific International Symposium on combustion and energy utilization, Hong Kong, 11-15 December 1995, Vol. I, p.255-260, 1995.

Heap, M.P; et al. Emission of nitric oxide from large turbulent diffusion flames: International Symposium of Combustion, 13 th, 197?, Pittsburgh . **Proceedings...** Pittsburgh: Combustion Institute, 1972. p. 883-95.

Hill, S.C; Smoot, D.L. Modeling of nitrogen oxides formation and destruction in combustions systems, **Progress in Energy and Combustion Science**, v.26, p. 417-58, 2000.

Hsieh, A.C; et al. Scaling laws for NOx emission performance of burners and furnaces from 30 kW to 12 MW, **Combustion and Flame**, v.114, p.54-80, 1998.

Kuo, Kk; Principles of Combustion. John Wiley & Sons, 1986

Lyons, V. J. Fuel/air nonuniformity – effect on nitric oxide emissions. AIAA journal, v.20, n.5, p.660-5, 1982.

Muniz, W. F. Estudo de um retentor de chama do tipo vortical/axial com pás de ângulos variáveis, Dissertação de Mestrado em ciência espacial/combustão Inpe. São Jose dosCampos, 1993.

Muniz, W.F.; et al. “Flame Holding Performance of Axial Swirlers”, Proceedings of The 5th Asian- Pacific International Symposium on combustion And Energy, 2000.

Oliveira Martins, C. M. Emissões de NOx e partículas de uma fornalha semi-industrial a fuel óleo, INETI Instituto Superior Técnico, Departamento de Engenharia Mecânica, Termodinâmica Aplicada, Março 2001.

Splithoff, H.; et al. Basic effects on NO_x emissions in air staging and reburning at a bench-scale test facility, **Fuel**, v.75, n.5, p.560-64, 1996.

Syska, A. Low NOx staged air recirculation burner undergoing field trials after excellent test performance. Industrial heating, v.60, p.40-3, 1993.

Wall. The combustion of coal as pulverized fuel through swirl burners, in Law, C. J. Ed, Principles of Combustion Engineering for Boilers, Academic Press, 197-335, and 1987.

6. Responsibility notice

The authors Gustavo Ronceros Rivas and Pedro Teixeira Lacava are the only responsible for the printed material included in this paper.