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Thermal simulation of a furnace for production of frits and comparison of costs using air and pure oxygen

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Abstract. *This study is initially about the numerical simulation of an operational furnace from the industry to producing ceramic frits. The software Ansys CFX solves the main phenomena found in the heat transfer and combustion in the furnace. Four chimney positions are tested to evaluate the furnace's performance. Natural gas and pure oxygen are used for combustion. The results from the current furnace's configuration are validated with experimental values. Numerical results indicate that the chimney's positions do not strongly influence the energy distribution. An analytical study involving hypothetical combustion with air is conducted and compared to the current oxy-combustion. In this case, it was considered that the energy consumed by material in the furnace should be the actual value (606 kW) to result in a similar performance. With unheated air, a natural gas flow rate of 2.43 times the current value will provide the performance with pure oxygen. In this case, 70% of the energy goes out with the flue gas; therefore, it has a high potential to preheat the air. When analyzing the combustion costs with pre-heated air, they can be significantly lower than the oxy-combustion.*

Keywords: *heat transfer, oxy-combustion, thermal simulation, frits production, ceramic kiln*

1. INTRODUCTION

Industrial furnaces are equipment intended to obtain high-temperature fields by burning gaseous, solid, or liquid fuels. They are widely applied in the production of glass, where natural gas and air are burned, in the production of ceramic frits, where natural gas with air or pure oxygen are burned, and in the production of crystals, by burning natural gas and air, and many others process.

The ceramic and glass segment requires intensive energy use and is under strict environmental legislation. This sector is historically competitive and heavily dependent on the national economy. Since the fuel price consumes a significant part of a company's revenue in this field, precise control of the ceramic manufacturing process is essential for the company's longevity.

With the expansion of computational processing, numerical simulations have become an essential tool for engineering progress. By mathematically describing several physical-chemical phenomena, the simulations can evaluate the performance of equipment in almost real operating conditions and allow estimating the performance of these systems under new geometries at a relatively low cost.

The study in question deals with the numerical simulation of a furnace dedicated to producing ceramic frits under the conditions encountered during its operation. Currently, the furnace uses pure oxygen as an oxidizer because it has advantages such as better control over the process and higher temperatures. In addition to the current combustion process with natural gas and pure oxygen, three new positions for the furnace's chimney are tested to verify if the heat transfer can be more efficient. Also, an analytical study comparing combustion with air and pure oxygen is conducted, focusing on costs.

The study is dedicated to the internal part of the combustion chamber, considering combustion with natural gas and pure oxygen. Heat transfer to the furnace walls are added through the boundary conditions. Experimental data measured during the furnace's operation is used to validate numerical results obtained via the commercial software Ansys CFX 14, available at the Laboratory of Combustion and Thermal Systems Engineering (LABCET) at the Federal University of Santa Catarina (UFSC).

2. Bibliography Review

When comparing oxy-combustion and air–combustion, greater temperature levels are found with the former (Stelzner *et al.*, 2017), reducing fuel consumption. The N_2 from the air does not contribute to the air-combustion process and carries away power when it flows with the flue gas through the chimney (Jr., 2000).

Although several studies have contributed to this work, two principal studies are responsible for guiding its progress: Possamai *et al.* (2012) and Possamai *et al.* (2015). Possamai *et al.* (2012) modeled a 1.0 MW ceramic frits melting kiln with oxy-firing combustion. The geometric domain consisted of the internal cavity of the kiln, excluding the chimney and walls. The CFD resolution is coupled to a three-dimensional heat conduction code along the kiln walls to determine the external temperature distribution. The thermal problem is composed of the combustion of natural gas with oxygen, the internal turbulent flow of exhaust gases, the energy loss by convection and radiation to the environment through the walls and the radiation within the kiln cavity with participating media. The energy balance estimated that 32.9% of the furnace's input was carried away by the flue gases, 40.8% left with material, and 29.9% left through walls.

Later, Possamai *et al.* (2015) studied an industrial kiln to produce sodium silicate. The 1.2 MW furnace is operated by firing natural gas and pre-heated air in a non-symmetrical kiln. The model is based on the Finite Volume method and solves the phenomena of turbulence, combustion, and radiation, through the RANS (Reynolds Averaged Navier Stokes), EDM (Eddy Dissipation Model), and DTRM (Discrete Transfer Radiative Model) models, respectively. The model involves the entire furnace: cavity with the burning of natural gas, raw material melting bath, and the furnace walls. The distribution of energy resulted in 66.6%, 21.4%, and 12% of the total energy input left the furnace with the flue gas, load, and walls.

3. Methodology

3.1 Geometry

The baseline geometry for the numerical simulation is a simplified model from the current operational furnace found in a company at Santa Catarina state, Brazil, as shown in Fig. 1, where the chimney's position is at the upper rear of the furnace.

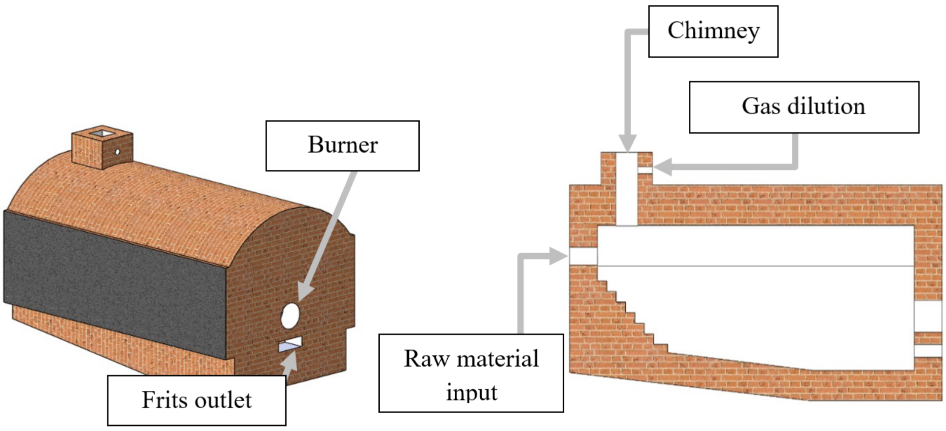

Figure 1. Baseline geometry of the furnace.

In addition to this default geometry with a chimney at the upper rear, three new hypothetical positions for the chimney are evaluated, located in the upper front, upper central, and lateral. The computational domain of combustion chamber is shown in Fig. 2, showing all the chimney's positions simultaneously in the mesh. However, only one is activated in each case. In this same figure, there is a more detailed view of the burner region. In the inner annular region (black) is the fuel injection, slightly pre-mixed. The green region is for the oxygen flow, and the red region separates the fuel injection from oxygen.

There is also a mesh for the load of material being processed inside the furnace. In this domain, raw material enters from behind, rolls over the internal ramp, and ceramic frits leave the furnace in the liquid phase at the outlet. Its width and length dimensions are identical to the combustion chamber, while its height is constant and equal to 150 mm.

These domains are solved separately, and their results are coupled. Initially, the conservation equations at the combustion chamber are solved with a temperature field prescribed at the interface with the load. After convergence, the heat flux on the interface is used as a boundary condition at the load's domain. Then after simulating the load, the new temperature field at the interface of the tank is set at the combustion chamber. These steps are repeated until the variable fields of both domains.

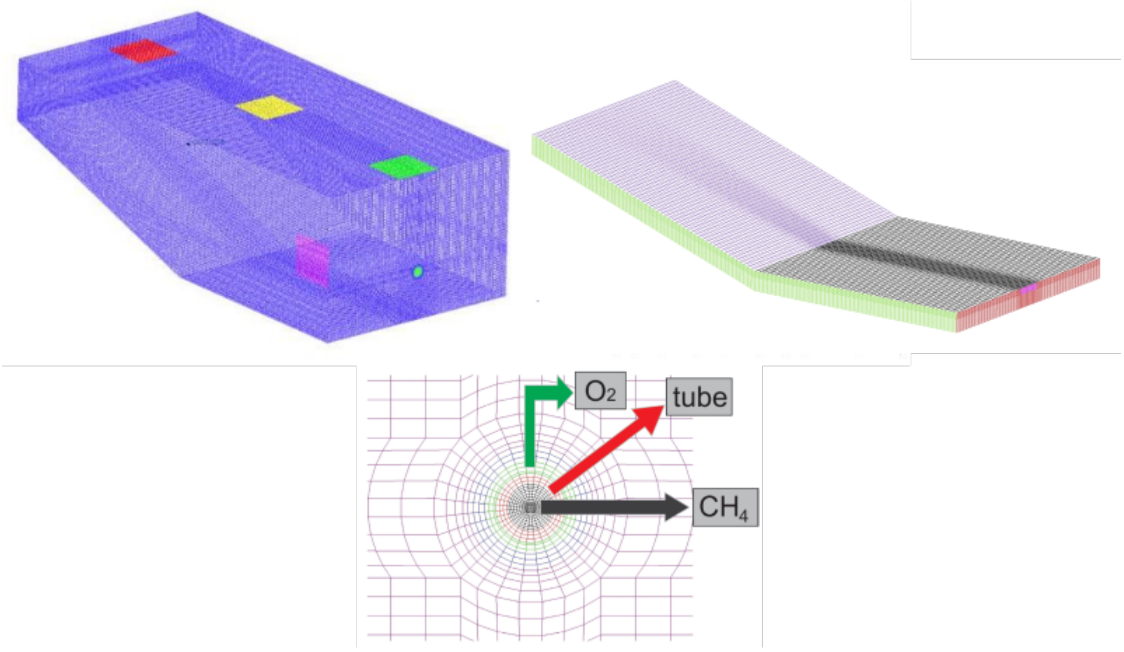

Figure 2. Meshes of combustion chamber (top left), load (top right), and the detail in the burner (bottom).

3.2 Conservation equations

The combustion, heat transfer, and flow are solved in the furnace's domain. In the mesh for this domain, part of the energy input is transferred to a load of ceramic frits ($\dot{E}_{load}$), and a fraction goes through the walls ($\dot{E}_{wall}$), and the remaining leaves with the exhaust gas at the chimney ($\dot{E}_{gas}$).

Combustion, kinetic mechanism, turbulence, radiation, and participant medium are the main phenomena presented in this problem. An EDM (Eddy Dissipation Model) formulation is used to solve combustion together with the 1-step WD kinetic mechanism (Galletti *et al.*, 2013; Yin *et al.*, 2011). The traditional $k-\epsilon$ model with a turbulent intensity of 5% is employed to solve turbulent flow (Launder and Spalding, 1974). DTRM model (Discrete Transfer Radiative Model) solves the radiation-related equations (Lockwood and Shan, 1981). Finally, the participant medium model based on the Weighted Sum of Gray Gases (WSGG) and proposed by Kangwanpongpan *et al.* (2012) is used to describe the influence of CO_2 and H_2O gases on radiation heat transfer (Hottel and Sarofim, 1967). The load in this work is solved as laminar flow with prescribed thermodynamics properties as a function of temperature.

3.2.1 Boundary conditions

The work of Morsch Filho *et al.* (2020) is used as a benchmark to validate the numerical results. That work discusses the parameters of the same furnace studied here and experimental measurements from its operation. The conditions used as boundary conditions in this work are in Table 1. Except for the chimney's position, the remaining values are identical among all the cases.

3.3 Analysis for combustion with air and oxygen

An estimate of the furnace consumption was carried out if the burning occurred with air. An excess of 20% oxidizer was adopted in this work, a value commonly found in this industry. Therefore, the combustion equivalence ratio ϕ is 0.833. The adiabatic flame temperature, in this case, is around 2200 K. However, this is an ideal maximum value, not reached in real engineering problems. When compared with other works available in the literature, it is assumed that the value of the exhaust gases is 1230 °C.

In order to make a comparison between the consumption for this hypothetical case and the present configuration found in the company, it is considered that the energy consumed by the load ($\dot{E}_{load}$) must be identical in both cases so that the production performance of the furnace is the same. Therefore, as obtained experimentally, $\dot{E}_{load}$= 606 kW (Morsch Filho *et al.*, 2020). Furthermore, it is assumed that the heat losses through the walls ($\dot{E}_{wall}$) are identical in both cases, that is, $\dot{E}_{wall}$= 130 kW.

The input energy is given by Eq. (1):

$$\dot{E}_{in} = \dot{m}_{NG} LHV + \dot{m}_{air}\left(h_{T_{in}} - h_{T_{ref}}\right) \quad (1)$$

Table 1. Boundary conditions and parameters of the numerical model.

Fuel inlet	Velocity	$39.39\hat{i} - 6.94\hat{k}$ m/s
	Temperature	300 K
	Mass composition	$0.7CH_4$+$0.3O_2$
	Turbulent intensity	5%
Oxidizer inlet	Velocity	$38.82\hat{i} - 6.84\hat{k}$ m/s
	Temperature	300 K
	Mass composition	100% O_2
	Turbulent intensity	5%
Gas outlet	Manometric pressure	0 Pa
	Other variables	Neumann condition
Inner wall	Emissivity	0.5
External wall	Convection coefficient	7 W/m$_2$K
	External temperature	300 K
	External emissivity	0.8
Interface at the combustion chamber	Emissivity	0.9
	Emission of CO_2	0.0037 kg/sm$_2$
	Velocity	Field from load
	Temperature	Field from load
Interface at load	Velocity	Free slip
	Heat flux	Field from combustion chamber
Raw material inlet	Velocity	0.00039 m/s
	Temperature	300 K
Frits outlet	Pressure	0 Pa
	Other variable	Neumann condition

where $\dot{m}_{NG}$ is the mass flow rate of natural gas and LHV its lower heating value (45,397 kJ/kg). If the air is preheated, there is an extra energy input to the system proportional to the mass flow rate of air ($\dot{m}_{air}$), and the enthalpy difference of air between the inlet temperature ($h_{T_{in}}$) and the reference temperature ($h_{T_{ref}}$).

Therefore, the load energy will be given by the energy balance in Eq. (2):

$$\dot{E}_{in} = \dot{E}_{out} \rightarrow \dot{E}_{in} = \dot{E}_{load} + \dot{E}_{gas} + \dot{E}_{wall} \rightarrow \dot{E}_{load} = \dot{E}_{in} - \dot{E}_{wall} - \dot{E}_{gas} \tag{2}$$

A parameter M will be defined as the ratio between the normalized flow of natural gas when burning hypothetically with air ($\dot{V}_{NG,Air}$) and the current burning condition with O_2 ($\dot{V}_{NG,O_2}$= 99.7 Nm^3/h), according to Eq.(3):

$$M = \frac{\dot{V}_{NG,Air}}{\dot{V}_{NG,O_2}} \tag{3}$$

For the case of combustion with natural gas and air, ignoring other parts, the production cost will be given exclusively by the consumption of natural gas. On the other hand, when the combustion process uses natural gas and pure oxygen, there is an extra cost associated with the oxygen. Therefore, the cost between air-combustion and oxy-combustion is calculated by Eq. (4):

$$\Delta\text{Cost} = \dot{V}_{NG,Air}P_{NG} - \dot{V}_{NG,O_2}P_{NG} + \dot{V}_{O_2}P_{O_2} \tag{4}$$

where P_{NG} and P_{O_2} are the prices of natural gas and oxygen, respectively. Substituting $\dot{V}_{NG,Air}$ from Eq. (3):

$$\Delta\text{Cost} = M\dot{V}_{NG,O_2}P_{NG} - \dot{V}_{NG,O_2}P_{NG} + \dot{V}_{O_2}P_{O_2} \tag{5}$$

Compared to the current cost of production, a percentage variation can be calculate according to Eq. (6):

$$\Delta\% = \frac{\Delta\text{Cost}}{\dot{V}_{NG,O_2}P_{NG} + \dot{V}_{O_2}P_{O_2}}100 \tag{6}$$

A positive value in this previous equation indicates that the combustion with pure oxygen is less expensive than the combustion with air, while a negative value means that the combustion with oxygen is more expensive.

4. Results

4.1 Numerical simulation

Firstly, the temperature fields along the furnace are showed in Fig. 3. In the hottest region, the flame temperatures are around 2,250°C and extend up to the beginning of the furnace's inclination. In the chimney region, it is verified that the gases are approximately at the temperature of 1460 °C. The case with the chimney in the middle had the flame most inclined, closer to the load. However, as stated earlier, the energy distribution was significantly different among the cases. The flame was not symmetric for all the cases and always tilted to the same lateral.

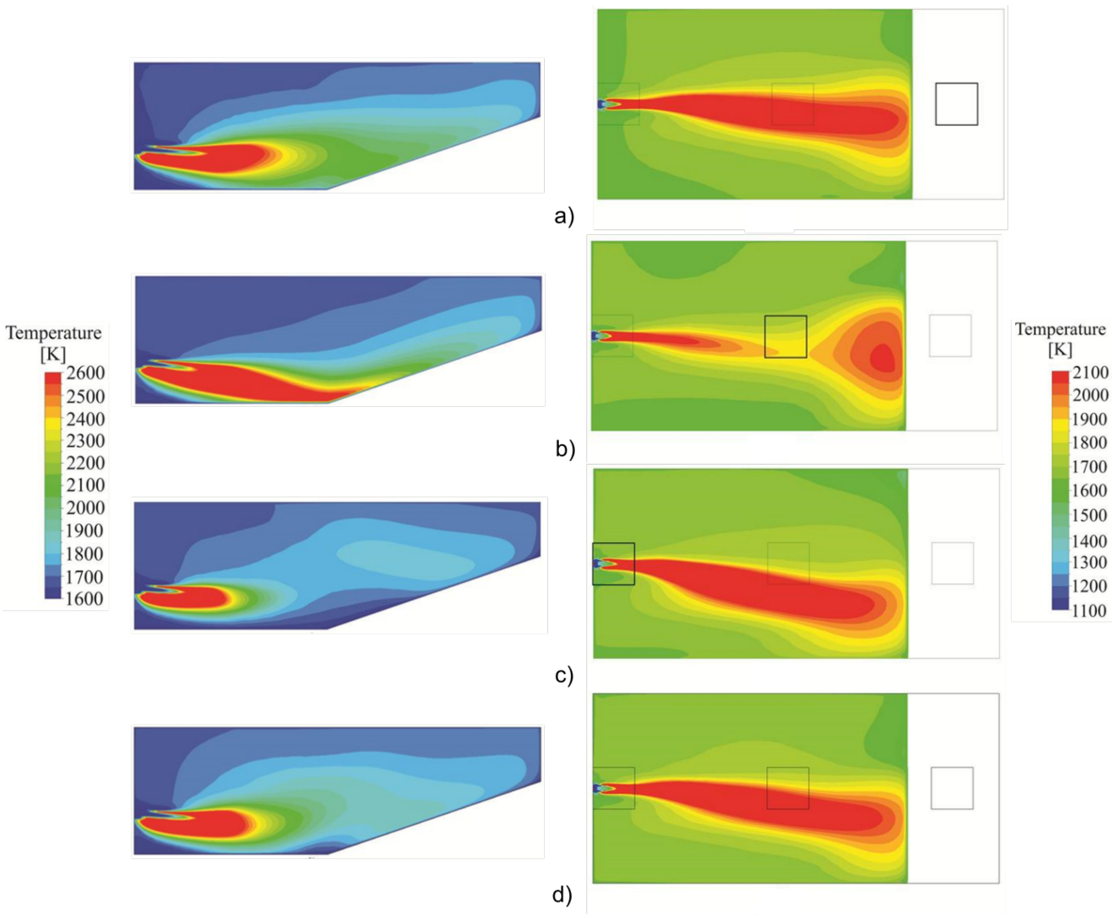

Figure 3. Temperature field in the symmetric plane (left) and at the burner height (right). a) Chimney at upper rear; b) Chimney at upper middle; c) Chimney at upper front; d) Chimney at lateral.

The power input for all the cases is the same, 0.95 MW from the natural gas, and the only difference among them is the chimney's position. Figure 4 shows the energy rates transferred through the walls of the furnace, to a load of ceramic frits, and to the outlet of gas at the chimney for each case. In this same figure, the results obtained with experimental data are presented for the current upper back position of the chimney. Regarding the experimental results, there are uncertainties associated with the measurement process and differences between the numerical geometry and the operational furnace, a reason for a slight discrepancy in its total energy compared to the results from simulations.

The experimental energy distribution for the current operation furnace, with the chimney at the upper rear, agrees with the numerical results from the simulation. It can be assumed that a higher energy rate reaching the load is associated with better performance for the furnace. Maximizing this parameter increases the furnace's efficiency as more energy is used to process the raw material and produces ceramic frits. The graph in Fig. 4 shows that there is no significant variation in the energy distribution among the cases. However, the chimney at the lateral resulted in a slight increment of the energy transferred to the load.

4.2 Operation with air

The analytical balance from Eq. (2) for combustion with air is shown in Fig. 5, as a function of the parameter M given by Eq. (3). It can be seen that, for the load to consume the same power as the furnace burning pure oxygen, the flow rate

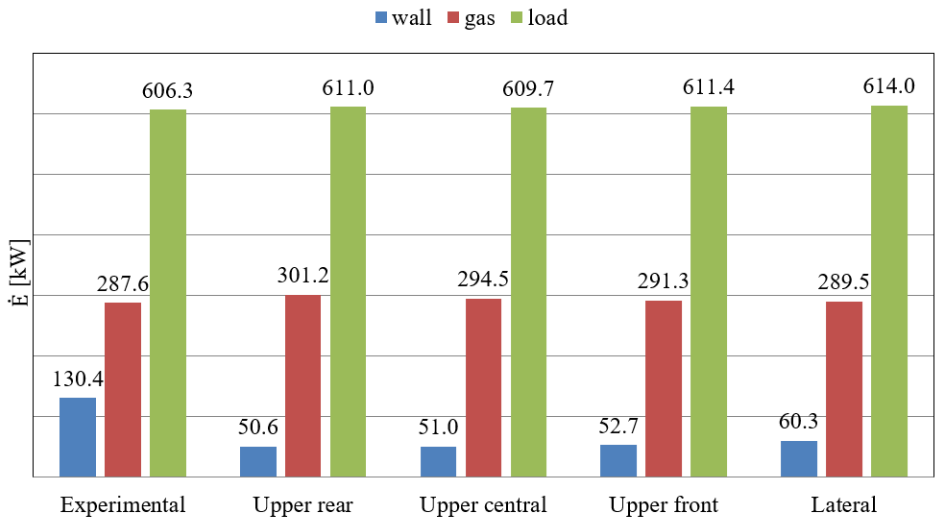

Figure 4. Energy distribution in the furnace for each case.

of natural gas must be 2.43 times the current flow. Below this value, the furnace's performance becomes lower because the load does not receive enough energy (606 kW). For M = 2.43, around 1797 kW of the total energy, or 70%, is lost through the chimney. This energy could be reused to heat the air used in the combustion or another process in the factory.

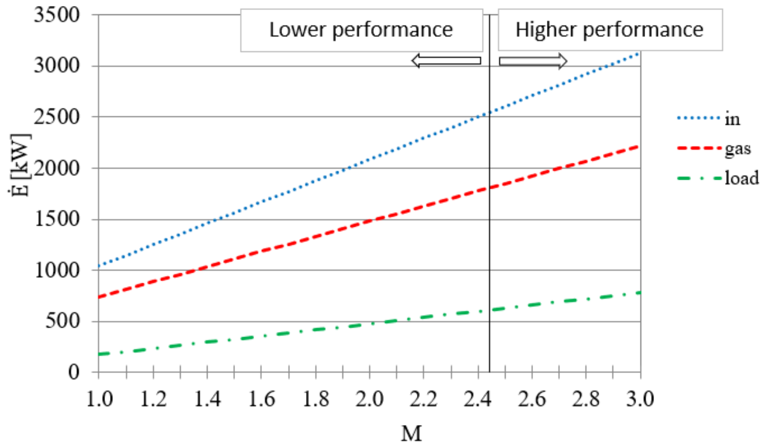

Figure 5. Energy distribution as function of parameter M.

Assuming that the air is preheated before entering the combustion chamber, the consumption of natural gas can be reduced, according to Eq. (1). Employing heat exchangers installed in the chimney, preheating can be carried out at no extra cost (excluding the cost of installing and maintaining the exchanger). This hypothesis was evaluated, assuming that the air inserted in the burner could be heated to a maximum reasonable temperature of around 425 °C. The analysis with the preheating of the air was performed and is shown in Fig. 6. The lines indicate that for a given volume of natural gas, the rate of energy destined for the load increases linearly proportional to the level of preheated air.

It can be seen from the curves in Fig. 6 that the parameter M, consequently the natural gas flow, can be greatly reduced compared to the case without preheating. For the case with M = 1.8, the air temperature should be around 250 °C to provide 606 kW to the load. For the case with M = 1.7, the required temperature for the preheated air rises to 305 °C. If M = 1.6 is used, the required temperature should be 365 °C. Finally, the lowest possible consumption of natural gas with preheated air would be around M = 1.5, that is, 1.5 times greater than what is currently consumed in burning with pure oxygen. In this condition, the preheated air would need to be at 415 °C. For $M \geq 2.7$, there is no need for preheating air to have the minimum required $\dot{E}_{load}$ = 606 kW.

Considering only the possible scenarios for furnace operation with preheated air ($M \geq$1.5), Fig. 7 indicates the rate of energy that leaves the chimney, as well as the rate of energy recovered in each case to heat the air to the required temperature shown previously, for diverse values of M. The lower consumption of natural gas (lower M), the greater percentage of energy reused from the exhaust gases to heat the air.

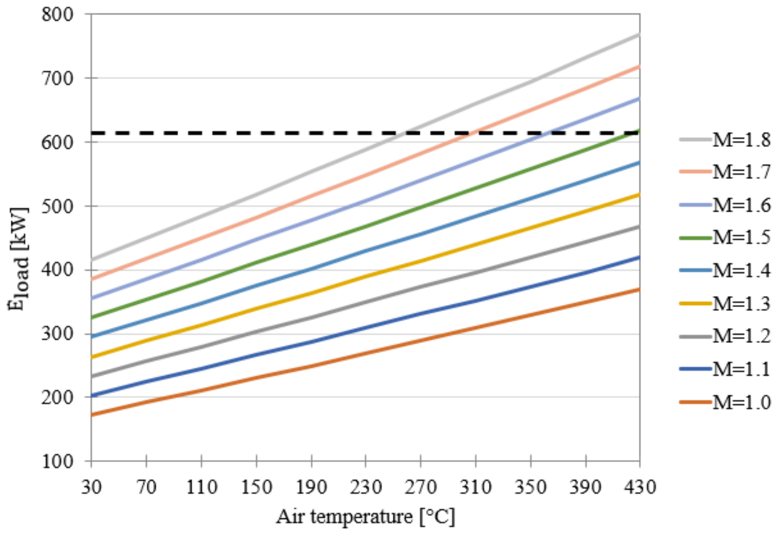

Figure 6. $\dot{E}_{load}$ as function of the parameter M and temperature of preheated air (the dashed line represents the current value found in the operational furnace).

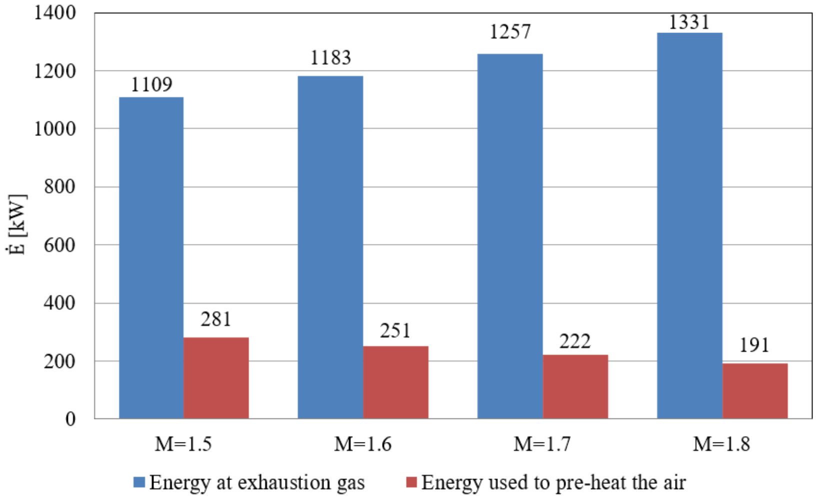

Figure 7. Energy available in the flue gas and the amount required to heat the air for $\dot{E}_{load}$ = 606 kW, for different M.

4.3 Results for the cost

The previous equations related to the cost will be assessed considering two scenarios of oxygen prices[1]:

- Scenario 1: P_{O_2} = \$0.14/Nm^3;
- Scenario 2: P_{O_2} = \$0.26/Nm^3.

The price for natural gas is \$0.29/Nm^3. The remaining parameters to estimate costs are 99.7 Nm^3/h for the consumption of natural gas with pure oxygen ($\dot{V}_{NG,,O_2}$) and 227.1 Nm^3/h for oxygen consumption ($\dot{V}_{O_2}$). The cost analyses in this paper do not consider any extra costs associated with the production or installation of equipment to preheat the air. The results obtained are summarized in Tab. 2.

For Scenario 1, with a lower cost for the oxygen, the air combustion is not advantageous from an economic point of view unless the air is preheated. If the air is preheated and the furnace works uninterruptedly, the savings can be up to \$153,225.00 in a year for the minimum M and maximum preheating temperature. Therefore, it is around 30% less than the actual production costs associated with natural gas and oxygen.

Analyzing Scenario 2, which has a higher cost for the oxygen, the potential for savings is more significant, even for the air combustion without preheating (M = 2.43). In this case, the savings would be approximately \$17.10 per hour of operation, or around 20% less than the actual scenario, representing a total saving of \$148,709.00 at the end of a year. For the condition of M = 1.5, it is estimated that the savings are 50%, corresponding to \$381,290.00 per year.

[1]These are estimated values based on the dollar value and gas prices found in 2017.

Table 2. Production costs with air-combustion.

Preheating	M	Air-combustion cost [$/h]	Oxy-combustion cost [$/h]	$\Delta\%$ [%]	Cheapest combustion
Scenario 1: P_{O_2} = $0.14/Nm^3					
No	2.43	70.26	60.74	+14.86	O_2
Yes	1.50	43.37	60.74	-29.10	Air
Yes	1.60	46.26	60.74	-24.37	Air
Yes	1.70	49.15	60.74	-19.65	Air
Yes	1.80	52.04	60.74	-14.92	Air
Scenario 2: P_{O_2} = $0.26/Nm^3					
No	2.43	70.26	88.02	-20.17	Air
Yes	1.50	43.37	88.02	-50.72	Air
Yes	1.60	46.26	88.02	-47.44	Air
Yes	1.70	49.15	88.02	-44.15	Air
Yes	1.80	52.04	88.02	-40.87	Air

The frontier value to decide which combustion process is less expensive can be made by setting Eq. (5) to zero and isolating the parameter M. By doing that, the maximum valor for M where air-combustion and oxy-combustion have the same cost, and it is estimated by:

$$M = \frac{\dot{V}_{NG,O_2} P_{NG} + \dot{V}_{O_2} P_{O_2}}{\dot{V}_{NG,O_2} P_{GN}} \tag{7}$$

The results obtained are shown in Tab. 3, for each scenario of prices.

Table 3. Condition for equal production cost with air-combustion and oxy-combustion.

Scenario	M	Air Temp. [°C]	$\dot{E}_{gas}$ [kW]	Heat recovery [kW]
1	2.11	122	1561	95
2	3.04	Ambient temp.	2234	-

For Scenario 1, with a lower price for oxygen, the maximum flow of natural gas for air combustion can be up to 2.11 times greater than the current configuration with oxy-combustion. If the value of M is above, the process is cheaper for oxy-combustion, or then more expensive if M is under 2.11. In this configuration, the air needs to be preheated to a temperature of 122 °C before entering the furnace. The energy carried by the exhaust gases in this scenario is high enough to supply heat for the air through a heat exchanger, for example. Thus, values above this flow rate will benefit from burning with pure oxygen, while values below 2.11 suggest burning with air.

In Scenario 2, the flow rate must be 3.04 for the cost to be identical in both combustion cases. In this configuration, it is not necessary to heat the air, and the available energy for the load is even greater than the required amount of 606 kW. Therefore, the amount of natural gas given by M = 3.04 is beyond the necessary, and the cheapest combustion process will be achieved with air.

5. Conclusions

Numerical simulations of a furnace dedicated to producing ceramic frits were performed considering pure oxygen in the combustion process for different chimney positions. The results did not show significant variations in the energy distribution inside the furnace. Because of this, production is not expected to increase or decrease, varying the position of the chimney.

By conducting an analytical assessment of the costs of combustion with air, it was found that these costs can be significantly reduced, especially when the air is preheated. The analyses considered that the supply of energy to the load with the new combustion process should be equal to that obtained experimentally. The hypothesis is that production performance is the same as the current operation furnace when this happens. For the case of unheated air, a natural gas flow rate of approximately 2.43 times the current value is required to provide the same power to the load as with pure oxygen. In this condition with air, approximately 70% of the energy is lost through the chimney. Therefore it has a high potential for reuse to preheat the air itself or in another factory process. At the upper limit of air preheating, assumed at 425 °C, even flows higher than 1.5 times the current flow of natural gas would reduce current costs.

When estimating the costs of fuel and oxidizer, it can be seen that the potential for savings is significant with air combustion compared to the cost of oxy-combustion. For both scenarios of oxygen price, there is a reduction in costs, especially when using preheated air. Considering the value of R$0.14 Nm^3 for pure oxygen, natural gas consumption of up to 2.11 times the current flow provides a reduction in costs with an air combustion.

6. ACKNOWLEDGEMENTS

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