encit 2020

ABCM
Associação Brasileira de Engenharia e Ciências Mecânicas

18th Brazilian Congress of Thermal Sciences and Engineering
November 16-20, 2020 (Online)

# ENC-2020-0086
# INTERCHANGEABILITY OF FUELS IN A PREMIXED RADIANT WALL BURNER – A CFD APPROACH
# 2020

**Rodolpho da Cunha Marques**
FIGENER – rua Bela Cintra, 986, cj. 73, Consolação, São Paulo - SP
rodolphocmarques@gmail.com
**Laiete Messias Soto, Msc.**
PACTO Engenharia e Meio Ambiente Ltda – São Paulo - SP
laiete@uol.com.br
**Lea Soledar dos Santos, MSc.**
Braskem – III Polo Petroquímico - Unidade de Químicos – Triunfo - RS
Lea.santos@braskem.com

***Abstract.*** *Premixed venturi burners are commonly encountered in refineries where different process gases can be used as fuel in furnaces to avoid flaring them. With a huge range of fuel compositions, the interchangeability is an important point to be evaluated. This case study presents a fuel interchangeability analysis in a premixed burner using two approaches: Wobbe Index and CFD (Computational Fluid Dynamics) modelling. Analysis with Wobbe Index are very simple and a quick way to get an answer for the interchangeability capacity on a burner, however due to its simplicity it cannot be done any inference about strategies to make the interchangeability feasible. Then, equipment modifications are proposed and tested by CFD using an Open Source Software (OpenFOAM). The model is validated by experimental data and then it was applied to predict new conditions, varying geometry of fuel nozzle with an evaluation of the limiting effect of burner geometry and, thus, it was concluded wall roughness restricted the possibility of interchangeability.*

***Keywords:*** *Premix, CFD, Burner, Wobbe Index, OpenFOAM*

## 1. INTRODUCTION

A furnace in an ethylene unit can be considered a complex system of heat exchangers (Kiese, 2013). The furnace consists of the feed route (and the decoking gas route), the cooling i.e. steam production system, the flue gas system and the fuel gas / firing system. All these subsystems are influencing each other.

According with Baukal (2004) radiant wall burners have been used in many ethylene-cracking heater applications. These furnaces typically operate in temperature range between 1.050 °C e 1.250 °C. The heat transfer from the refractory side walls is the dominant heat transfer mode. In order to obtain the best performance in cracking process, and maintain under control the hydrocarbon coking inside the tubes, the total heat input process is divided in many burners (> 150) equally spaced (~1 x 1 m) along the furnace refractory walls.

As explained by Foote (2004), a burner is a fuel and air metering device. Its purpose is to provide an environment for the proper fuel and air mixture ratio to mix and react. Much time and energy are spent to find the proper tip drillings, tile design, venturi design (pre-mixed burners), and air inlet design. The burner tile is the main part responsible on shaping the flame and providing a means for air and fuel mixing. Current burner technology for Ethylene Cracking Furnaces is generally designed for 10% excess air operation. Through the years furnace manufacturers and burner vendors have worked closely to the burner specifications for capacity match the furnace specifications and devoted considerable research efforts to determine the optimum heat flux profiles in various furnace-burner configurations to optimize products yields. This ensures that the burners will operate in a range where the excess air is controllable, taking into account the reduction the emissions of atmospheric pollutants, specially the NOx.

The work of Guégués (2020) present a numerical simulation of an industrial partially premixed radiant wall burner using a CFD model as a tool for heat and mass transfer simulation. The chemical NOx mechanisms kinetics and radiation heat transfer mechanisms were introduced to obtain, beyond the contours of temperature and velocities in a different location in the wall close to the burner positions, the NOx previsions.

Adams (2014) and his coworkers examines the usefulness of CFD modelling in evaluating burner in a furnace performance in new or retrofit pyrolysis furnaces.

Baukal points out also that in many refinery applications, the fuel gas composition that a radiant wall burner is required to fire can vary widely because it is made up of various gas streams from different processes that change within the time. Considering this aspect, the purpose of the present work is to present a method to evaluate an industrial burner using a CFD model using Open Foam. The main objective is to define if it is possible to burn two different combustible gases: Fuel 1 (a fictitious light combustible gas, such as Natural Gas) and Fuel 2 (a fictitious heavier combustible gas, such as LPG) with the same maximum heat release, keeping the excess of oxygen in 2%, attending the safety operation condition.

## 2. METHODS AND BASIC CONCEPTS

In the study it is assumed that all combustion air necessary for burning the fuel introduced in the burner is induced into the body considering the following common operating conditions for a cracking furnace:

- $O_2$ content in the combustion product gases: 2% (dry and volume basis); and,
- Internal furnace pressure: -5.0 mmwc for the lowest burners row and -7.0 mmwc for the highest one.

In this condition, the primary and secondary air flows induced into the burner body are one of the factors limiting the maximum thermal capacity of the burner, based on the premise that the flames produced by these devices must have specific characteristics regarding stability and behavior radiation suitable for the heating process developed in the cracking furnace.

The internal geometry of the mixing tube was obtained through measurements made on a specimen already used in campaign. The specimen was sectioned in parts so that it was possible to access the survey with the necessary precision of the internal geometry of the mixing tube (convergent and divergent segments) and the discharge section of the mixture.

The flow of primary entrained air, which constitutes the largest portion of the total induced air, was determined from the modeling and simulation of the internal flow in the air + gas mixture tube using computational methods (CFD - Computational Fluid Dynamics), using the OpenFOAM package, whose premises, boundary conditions and turbulence models adopted are described in a specific item.

The flow of the secondary air portion through the adjustment valves, and the annular passages around the mixing tube and burner refractory block, were estimated based on an empirical model elaborated from geometry, and empirical coefficients available in the specialized literature (Idel'čik, 2005), assuming that this portion of the air flow is established due to the pressure differential between the internal and external sides of the furnace.

The validation of the model is based on the measurements previously made in a combustion laboratory.

Considerations are made about the interchangeability of the gases in view of the characteristics of the gases and the design data of the burner analyzing the Wobbe Index. **The data published in this article are only for means of illustrating the applied methods and do not correspond to real process data.**

## 3. BURNER CHARACTERISTICS

The burners mounted on the side walls of the furnaces are of the pre-mix type that produce flat flames ("Radiant Wall Burners") suitable for the furnace heating process, that is, the greater temperature uniformity of the wall and, therefore, the flow of radiant heat generated from the oven wall.

The original set of the burner is basically made up of cast iron parts and brass gas injection nozzles.

The refractory block is an integral part of the burner assembly and has the function of stabilizing the flames in addition to thermal insulation and protection of the side wall of the furnace.

The silencer is a non-original device of the burner that was introduced as a mitigation measure for noise generated in the flow of combustion air induced into the body of the burner.

The diagrams in Fig. 1 and Fig. 2 illustrate, respectively, the installation of the original burner refractory and block mounted on the side wall of the oven and the installation of the silencer.

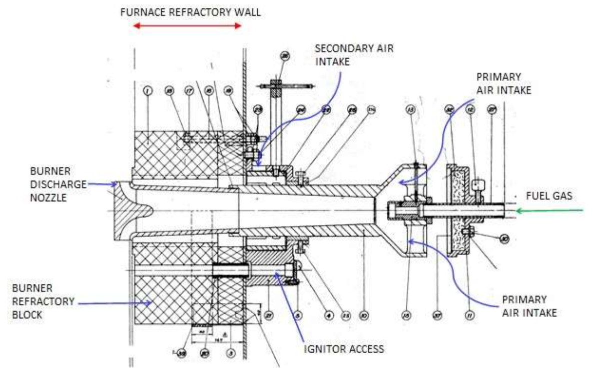


Figure 1. Burner assembly on the furnace wall (original scheme without silencer).

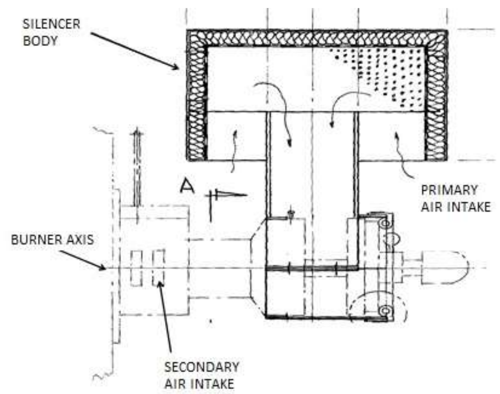


Figure 2. Assembly detail of the silencer and burner.

## 4. GAS FUEL CHARACTERISTICS AND WOBBE INDEX ANALYSIS

Fuel 1 was selected in the present work to illustrate the burner operation with a light fuel gas and Fuel 2 is intended to be used in burner as alternative, representing an operation with a heavier fuel gas such as LPG. The characteristics of both hypothetic fuels are shown in Tab. 1.

Table 1. Gas fuel characteristics and burner expected thermal energy

| PARAMETER | UNIT | Fuel 1 | Fuel 2 |
|---|---|---|---|
| Molecular mass | kg/kgmol | 13,18 | 42,12 |
| Density | kg/Nm$^3$ | 0,59 | 1,88 |
| LHV (Lower Heating Value) | kcal/kg | 12.500 | 10.900 |
| | kcal/Nm$^3$ | 7.356 | 25.634 |
| | kJ/kg | 52.325 | 45.627 |
| | kJ/Nm$^3$ | 30.792 | 85.796 |
| Wobbe Index | kcal/Nm$^3$ | 10.874 | 18.956 |
| | kJ/Nm$^3$ | 45.519 | 79.350 |
| Wobbe Index Variation | % | | 35,89 |
| Total Combustion Air / Fuel Mass Ratio (stoichiometric) | kg/kg | 18,01 | 15,49 |
| Total Combustion Air / Fuel Mass Ratio ($O_2$ = 2%vol dry basis) | kg/kg | 19,81 | 17,09 |
| Minimum thermal energy | kcal/h | 160.000 | |
| Normal thermal energy | kcal/h | 260.000 | |
| Maximum thermal energy | kcal/h | 310.000 | |

To analyze this question, the principle of interchangeability of combustible gases is considered, having as reference the values of the Wobbe index of gases defined by Eq. (1) where $LHV_{vol}$ is the Lower Heating Value (kJ/Nm³ at normal conditions) and $SG$ is the Specific Gravity at 0 °C and 101.325 Pa based on air at same condition. The thermal power released from burner is described by Eq. (2), where Area refers to the passage area from burner discharger orifice in m², $K$ is the dimensionless orifice discharge coefficient and $\Delta P$ is the pressure difference between nozzle chamber and burner in Pa.

$$W = \frac{LHV_{vol}}{\sqrt{SG}} = Wobbe\ index\ [kJ/Nm^3] \tag{1}$$

$$P = \left(\frac{LHV_{vol}}{\sqrt{SG}}\right) \cdot Area \cdot K \cdot \sqrt{\Delta P} = Thermal\ Power\ [kW] \tag{2}$$

Close Wobbe index values with different fuels in the same burner (without any change in geometry) ensure the minimum interchangeable conditions for the replacement of gaseous fuels. As a general decision criterion, it is assumed that two combustible gases are interchangeable when the variation of the Wobbe index is within the range of ± 5%.

Table 1 shows that the condition of interchangeability between the two gases is not satisfied, as the gases are quite different, especially with regard to molecular weight and, therefore, Fuel 2 has values significantly higher for density and heating value (volume basis) resulting in a significant variation in the Wobbe index.

In other words, considering the terms from Eq. (2) of thermal power released, maintaining the area of the gas discharge holes (3 x Ø 2.8 mm) and the respective discharge coefficient, it is necessary to operate with injection pressure of noticeably lower gas when operating with Fuel 2. The differences in the chemical composition of the gases (C/H) result in lower entrained air/fuel ratios for Fuel 2 in relation to Fuel 1.

On the other hand, the differences in the chemical composition of the gases (C/H ratio) result in lower entrained air/fuel ratios for the emergency gas in relation to the original fuel.

The sum of these factors and their implications for the fluid dynamics of the internal flow to the mixing tube are those considered in the CFD modeling and simulation described below, particularly focused on the Fuel 2.

## 5. MODELLING AND BURNER AIR FLOW SIMULATION

It is assumed as a basic premise that the maximum thermal capacity of the burner, which is the object of this study, is associated with the capacity of the burner to induce enough air into the body for combustion to result in combustion products with at least 2% of oxygen (volume and dry basis) with no formation of CO (complete combustion).

There are two distinct compartments of the burner body available to air flow, they are: the internal one is the mixing pipe where the combustible gas is discharged entraining primary air by Venturi effect, and the other is external, between the outside of mixing tube and the burner refractory block forming an annular passage for the secondary air.

The flows that are established in these compartments are mixed inside the furnace in the combustion region, in front of the discharge nozzle of the premix pipe, therefore, outside the region of analysis of this study.

The analysis and modeling carried out below consider the two compartments separately, observing that the secondary air portion is significantly smaller than that of primary air, and can even be canceled when the manual air flow control lever is closed. It is worth mentioning the maneuvers of this lever is usually carried out according to the operation's own criteria of visual observations of the individual flames serving as fine tuning of the combustion air stoichiometry.

The process of pre-mixing the fuel gas with the portion of primary air that occurs is dominant in the performance of the combustion process and, therefore, in the behavior of the flame, which is why it has a more detailed approach in this study.

### 5.1 Secondary Air Flow

The flow of secondary air through the burner is basically established due to the pressure difference existing in the atmosphere of the interior and the external environment, that is, the secondary air is induced to the interior as a result of the negative pressure in the furnace atmosphere and the flow can be adjusted by manual positioning of the secondary air regulation lever in the diagram of Fig. 1.

To compute this flow to determine the burner maximum thermal capacity, the passage areas were considered as the dimensions of the secondary air manifold and the annular space between the premix pipe and the burner refractory block. The pressure loss coefficients (local and distributed) along the path were estimated from empirical data collected in specialized technical literature (Idel'čik, 2005).

The values obtained according to this procedure were compared and validated with experimental data measured during the experimental tests carried out at specialized combustion laboratory, as can be seen in Tab. 2.

Although the results refer to a gas pressure condition, it is estimated that the secondary air flow through the burner is little impacted by this parameter, but strongly dependent on the depression in the oven and the opening position of the secondary air passage whose adjustment and positioning is done manually.

Table 2. Measured data and modelled values of secondary air flow

| PARAMETER | UNIT | SECONDARY AIR OPENING | | | |
|---|---|---|---|---|---|
| | | 0 mm | 10 mm | 20 mm | 40 mm |
| **Measured Data (Furnace pressure -5,5 mmwc)** | | | | | |
| Total air / Fuel ratio | kg/kg | 18,10 | 19,40 | 20,60 | NA |
| Fuel mass flow | kg/h | 20,00 | 20,00 | 20,00 | NA |
| Total air flow | kg/h | 362 | 388 | 412 | NA |
| Secondary air mass flow | kg/h | 0 | 26,00 | 50,00 | NA |
| Secondary air / fuel ratio | kg/kg | 0 | 1,30 | 2,50 | NA |
| **Modeled values (Furnace pressure -5,5 mmwc)** | | | | | |

| | | | | | |
|---|---|---|---|---|---|
| Secondary air mass flow | kg/h | 0 | 26,18 | 52,06 | - |
| Secondary air / fuel ratio | kg/kg | 0 | 1,31 | 2,60 | - |
| Deviation (measured x calculated) | % | - | 0,70 | 4,12 | - |

**Modeled values (Furnace pressure -5,0 mmwc)**

| | | | | | |
|---|---|---|---|---|---|
| Secondary air mass flow | kg/h | 0 | 25,04 | 49,63 | 81,24 |

**Modeled values (Furnace pressure -7,0 mmwc)**

| | | | | | |
|---|---|---|---|---|---|
| Secondary air mass flow | kg/h | 0 | 32,44 | 54,83 | 95,20 |

*NA: Not Available

### 5.2 Primary Air Flow

The internal flow in the mixing tube is dominant in the combustion process and fundamental in the behavior of the flame that starts immediately at the outlet of the discharge nozzle and develops close to the surface of the burner refractory block with the specific characteristics desired (flat flame).

In this case, the previous mixing of the fuel gas with the combustion air is the result of interaction process between fuel gas jets and the primary air flow entrained. The primary air flow is induced into the convergent segment of the mixing tube due to the expansion of the fuel jet.

The process of expansion and mixing of jets in a confined environment is very complex, being governed by phenomena associated with the transfer of momentum at the microscopic level (turbulent velocity diffusion), involving not only the physical-chemical properties of fluids, but also the geometry of the enclosure containing the jet as interfering factors in the process.

The computational methods (CFD) constitute the most appropriate tool for this study where the main objective is to obtain the entrained air/gas flow ration entrained for a given fuel gas composition (emergency gas) and fuel pressure, considering the discharge of the mixture in the furnace at pressure equals to -5 mmwc.

### 5.3 CFD Model

The first step is to determine the geometry and the domain in which the mesh will be generated, therefore, it should be as simple as possible to describe the phenomenon without the need to use meshes with many elements. So, the geometry comprises the following elements in Fig. 3: burner (red), tip (yellow), nozzle (brown), nozzle support (green), fuel pipe (blue), muffler cover (white), muffler (orange) and soundproof blanket (pink).

The number of mesh elements was defined as a consequence of the complexity of the geometry and the physical phenomenon to be described, that is, regions of interest have a larger number of cells as can be seen in Fig. 4, where hot colors regions have smaller cells size than cold colors regions. The total number of cells are around 1 million and the boundary layer cells were characterized according to the value of the variable "$y+$" so that the wall functions behave properly.

The boundary conditions for velocity field were chosen as zero gradient for the primary air inlet and domain frontier inside the furnace.

Constant atmospheric total pressure was used at the air inlet and for the discharge section of the mixture inside the furnace it was defined as constant static pressure -5 mmwc.

The boundary conditions for the orifices of the fuel gas discharge nozzle are determined by a mathematical verification if at the mass flow desired to be simulated the flow through nozzle orifices is chocked. If it is chocked, the flow occurs with the speed of sound and velocity boundary condition is already determined, so the pressure is set in a manner to make density the value necessary to reach the mass flow desired. If it is not chocked, the pressure is set fixed to 101.000 Pa and the velocity is calculated based on the mass flow.

Other useful properties needed to modelling are listed in Tab. 3.

Table 3. Fluid properties

| | Air | Fuel 1 | Fuel 2 |
|---|---|---|---|
| **Molar mass (kg/kgmol)** | 28,8 | 18,31 | 42,08 |
| **Specific heat capacity (J/kg.K)** | 1005 | 2034 | 1548 |
| **Viscosity (Pa.s)** | 1,8e-5 | 1,1e-5 | 9e-6 |
| **Speed of sound (m/s)** | - | 422,0 | 261,8 |

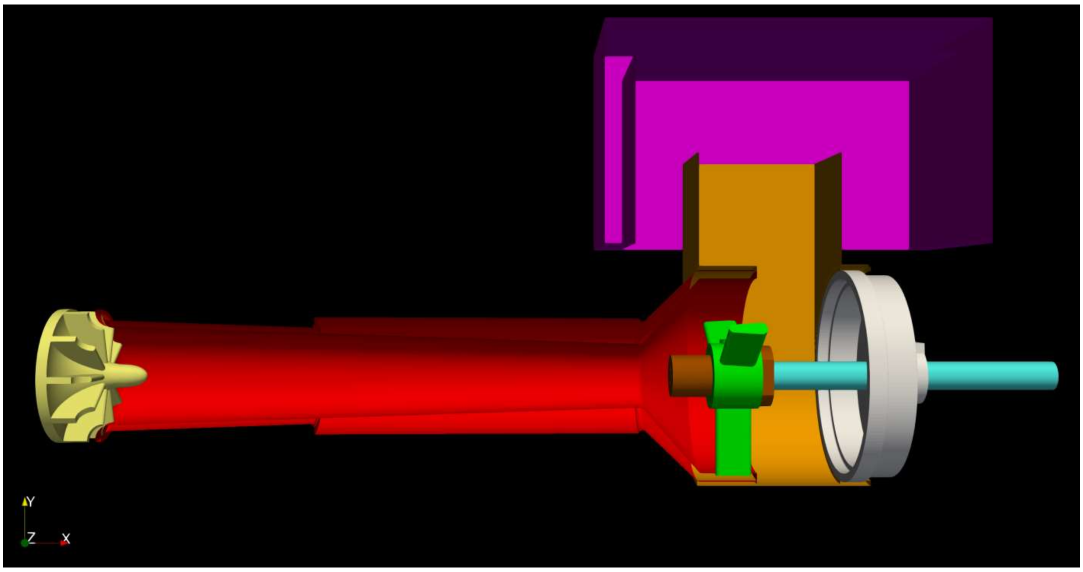

Figure 3. Geometry description

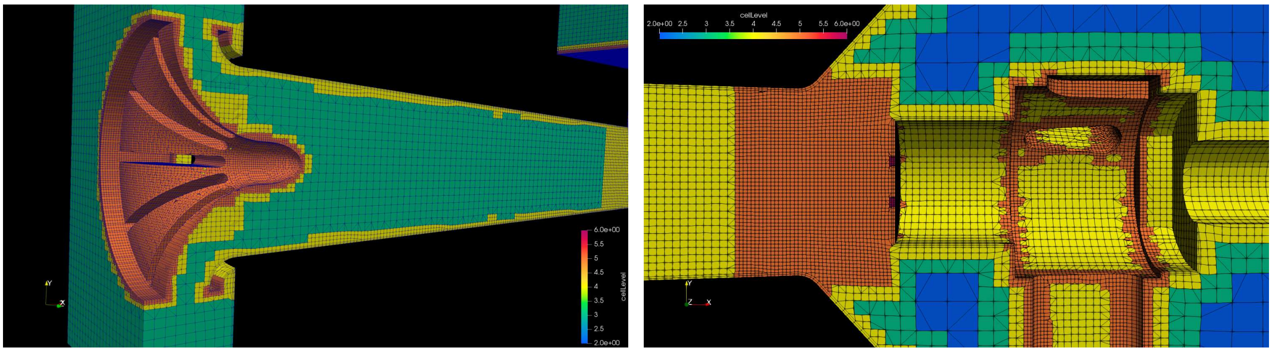

Figure 4. Mesh cells in the tip and nozzle regions

Navier-Stokes and Continuity Equation are solved for compressible fluid, there is no reacting flow and heat exchange. Four RANS turbulence models were tested: standard k-Epsilon, RNG k-Epsilon, k-Omega and k-Omega SST. K-Omega and k-Omega SST does not represent well the experimental data as the models based on k-Epsilon. Standard k-Epsilon and RNG k-Epsilon models showed different results in tests with coarser meshes, but with refined meshes (meshes with a greater number of cells) the models showed similar results. k-Epsilon model was chosen because it is widely used in industry simulations, the authors have already tested extensively this model, numerical convergence is easier and it is less sensitive to the initialization values than k-Omega. The last one is an important point because the fuel injection in this CFD model begins in front of the fuel orifices so it is difficult to estimate k and epsilon values for this region. Thompson (Thompson et al., 2014) tested different turbulence models for a premix burner and have encountered k-Epsilon as the best model to represent this physics too.

### 5.4 CFD Model Validation

To determine turbulence model and wall roughness, two experiments from laboratory were used. Table 4 describe the states and results from experiments and CFD model using k-Epsilon and absolute wall roughness value of 800 μm. The parameter wall roughness was used to set the model in agreement with the experiment data.

Table 4. Experimental and calculated data

| **Data** | **State 1** | **State 2** |
|---|---|---|
| Fuel | Natural Gas | Natural Gas |
| Fuel nozzle orifices | 3 | 3 |
| Orifice diameter (mm) | 2,8 | 2,8 |
| Secondary air opening (mm) | 0 | 0 |
| Silencer | No | Yes |
| Furnace Pressure (mmwc) | -7 | -7 |
| **Results** | | |
| Experimental primary air/fuel mass ratio (kg/kg) | 14,60 | 14,15 |
| CFD Model primary air/fuel mass ratio (kg/kg) | 14,50 | 14,30 |

### 5.5 Results

Figure 5 summarizes the results obtained from air/fuel ratio for different Fuel 2 flow values, maintaining the largest secondary air opening possible (40 mm) and standard fuel nozzle (3 orifices and diameter = 2,8 mm). The fuel flow for the desired condition (2% $O_2$ and -5 mmwc) is 13,3 kg/h, producing 144.941 kcal/h which is around 50% of the power released in normal operation with Fuel 1.

In search to achieve the same energy release of Fuel 1 some changes in nozzle geometry of the burner were tested (description of geometry in Fig. 6):

- Distance between throat and nozzle with fuel 2 flow 12,5 kg/h (results in Fig. 7 are function of distance increment (negative values represent distance reduction));
- Number of nozzle orifices keeping the same passage area with fuel 2 flow 12,5 kg/h (results in Fig. 8);
- Orifice distance from center axis with fuel 2 flow 12,5 kg/h (results in Fig. 9);
- Orifice diameter (results in Fig. 10).

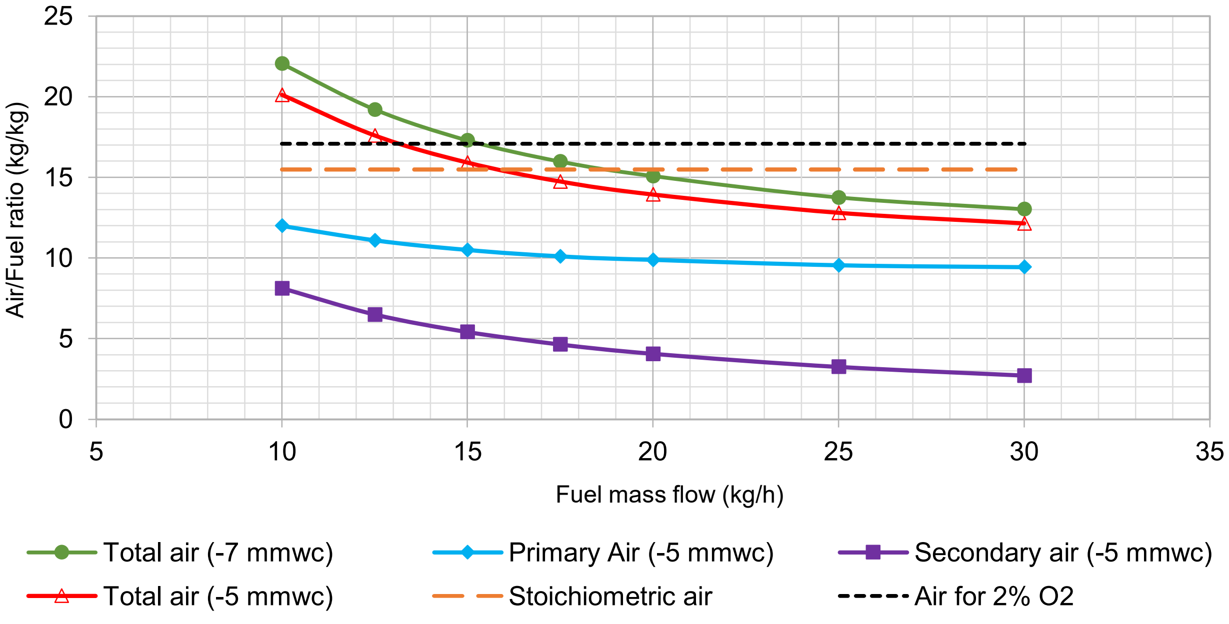


Figure 5. Burner operating flowchart working with Fuel 2

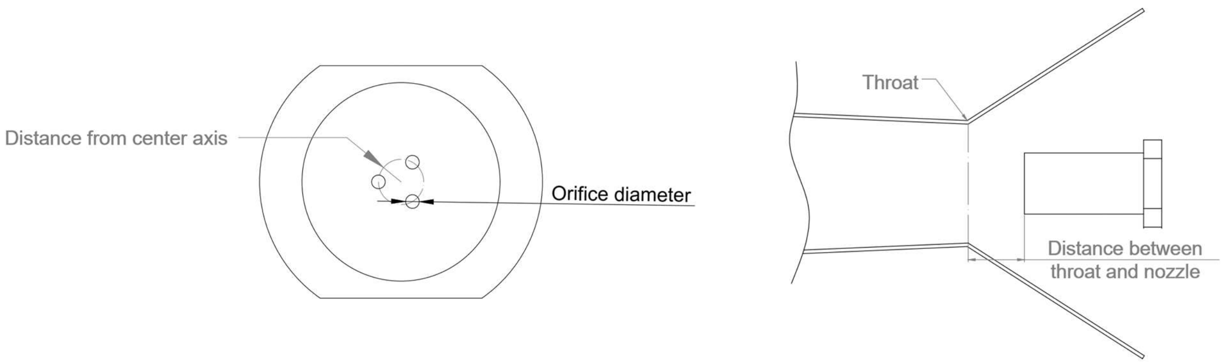


Figure 6 - Nozzle geometry proposed modifications

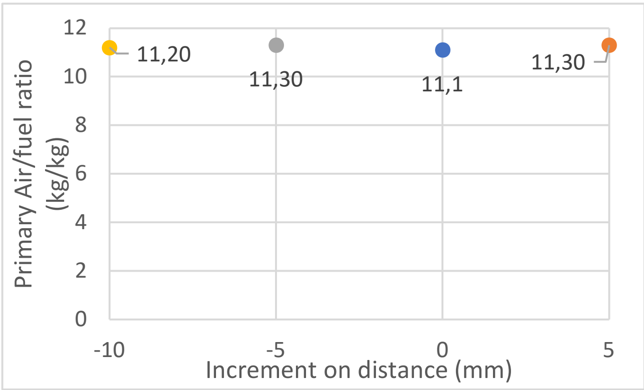


Figure 7 - CFD results for variation in distance nozzle from throat

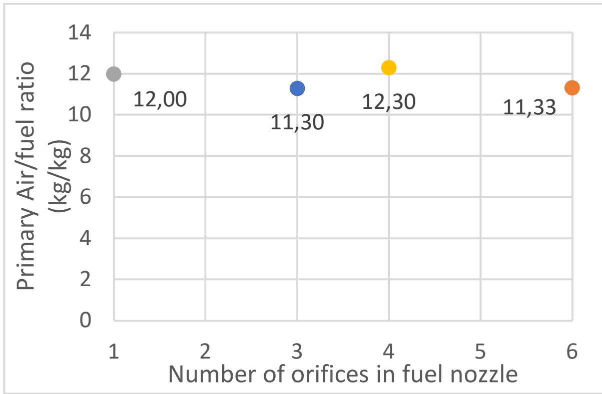


Figure 8 - CFD results for different orifices number

Changes in the number of orifices are responsible for create more area between the fuel and the entrainment air but it was not sufficient to improve primary air/fuel ratio, so interface area wasn’t the limiting variable in momentum exchange between fuel jet and air.

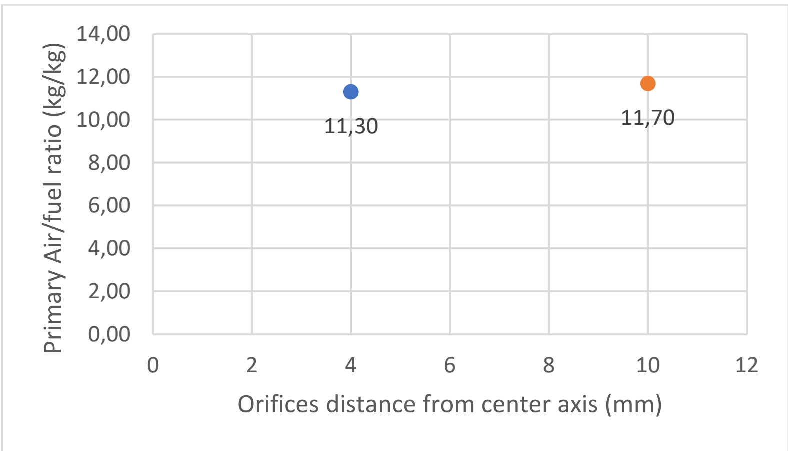


Figure 9 - CFD results for different orifices distance

Distance from center axis makes the fuel jet wider flowing through the throat but the results shows practically none improvement in primary air entrainment.

Change the diameter keeping orifices number the same impacts directly in fuel jet momentum which is responsible for increasing air entrainment but at the same time reducing fuel passage area in orifice discharge requires higher pressure in nozzle chamber at the same fuel flow.

Figure 10 demonstrate the improvement in total air/fuel ratio (based on -5 mmwc furnace pressure). For low values of fuel flow the modifications proposed had a positive impact but as the fuel flow is increased the air/fuel ratio reduces rapidly because more air is needed for combustion so the pressure drop across the burner is too great for those fuel flows.

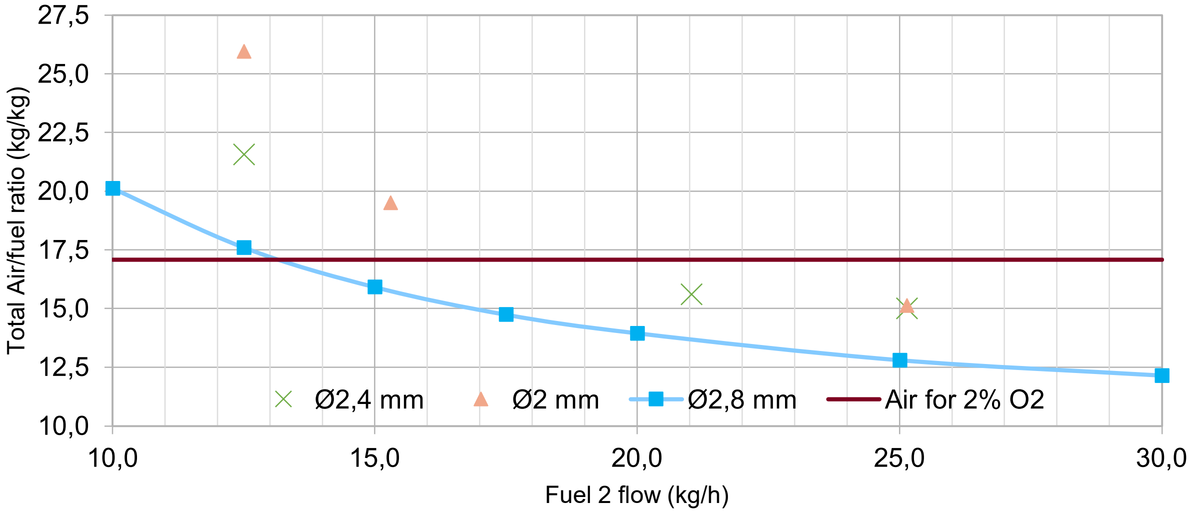


Figure 10 - CFD results for different orifices radius

## 6. CONCLUSION

The Wobbe Index method for interchangeability of fuels is proved to be sufficient to indicate if there is no interchangeability without any modification being made to the burner.

The CFD model, after undergoing validation with laboratory experiments, proved to be a precise tool in the burner analysis and made it possible to make different changes in order to increase the burner power with Fuel 2.

The proposal of greatest impact was to reduce the diameter of the nozzle orifices given the increase in jet momentum. It was not possible to achieve with Fuel 2 the same energy release as the burner with Fuel 1 keeping the excess oxygen of 2%. This fact is due to the burner body geometry and the high wall roughness over the mixing tube. These two factors limit the passage of air, impacting on a substoichiometric air/fuel ratio for Fuel 2 at high fuel flow rates.

## 7. REFERENCES

Adams, B.; Cremer, M.; Use of CFD in Evaluating Pyrolysys Furnace Design; AFRC 2014 Industrial Combustion Symposium..

Baukal Jr., C.E.; Industrial Burners- Handbook, CRC PRESS, 2004.

Foote, D.R., 2004. "Energy Efficient Burner Design and Maintenance in the Olefins Industry". *2004 AIChE.*

Guégués, T.; Herce, C.; Cortés, C.; Numerical Simulation of Industrial Partially Premixed Radiant Wall Burner. www.spire2030.eu/sites/default/files/users/user353/Publications; Acess in 04/27/2020

Idel'čik, I. E. and Steinberg, M. O., 2005. Handbook of Hydraulic Resistance.Jaico Publishing House, 3rd edition.

Kiese, G. 2013. "Furnace Burner Management System Design for New Furnaces". *2013 AIChE Spring Meeting*.

Thompson, J.S., Hassan, O., Carswell, D., Rolland, S., Lavery, N.P., Sienz, J., 2014. "Optimisation of a venturi mixer design to promote efficient combustion in a pre-mix burner". In *Proceeding of 22nd UK Conference of the Associaiton for Computational Mechanics in Engineering*. University of Exeter, Exeter.

## 8. RESPONSIBILITY NOTICE

The authors are the only responsible for the printed material included in this paper.

**The data published in this article are only for means of illustrating the applied methods and do not correspond to real process data.**