A FINITE VOLUME SOLUTION FOR NON-PREMIXED COUNTERFLOW FLAMES INSIDE A FLAT PLATE CHANNEL

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Abstract. Non-premixed counterflow flames inside a parallel flat plate channel are studied by a finite volume method. In the present work, the oxidant and the fuel enter in the channel through different slots positioned on each channel wall. In the entrance region of the channel, a counterflow is formed. The channel walls are adiabatic. The thermo-physical properties are assumed constants. The model is based on mass, momentum and mixture fraction conservation equations considering the Simple Chemical Reaction mechanism. The numerical solution uses a WUDS interpolation in a structured regular non-staggered grid. The SIMPLEC algorithm is employed to solve the pressure field. An implicit scheme is used. The proposed model considers a laminar incompressible flow formulation for the combustion problem. The limitations of the incompressible flow approach are discussed. A parametric study is conducted for Reynolds number, Peclet number, stoichiometric mixture fraction and heat of reaction. The influences of the parameters on flame behavior are analyzed. The results show the flame shape, temperature, streamline and mixture fraction fields.

***Keywords**: non-premixed counterflow, flame, finite volume, flat plate channel*

1. INTRODUCTION

The process of diffusion flame is present in many types of equipment as ovens, diesel engines, gas turbines and bipropellant rocket engines. Furthermore, nowadays, the environment and the energy problems highlight the combustion studies. The combustion science and engineering search the optimum exploitation energy of fuels, reducing the emission of pollutants. The knowledge in the combustion field is also important to develop technologies to combat forest fires and industrial fires.

The slot burner is used in industry and in research laboratories. In this burner, a gaseous fuel jet is opposed to an oxidant jet, creating a counterflow. Such configuration is relevant in industrial applications where rapid and uniform mixing of two or more gas/liquid streams are required without the action of mechanical mixers. "Opposing jet impacting head-on provides a simple in-line mixer configuration with potential industrial applications for rapid mixing of viscous fluids" (Wang *et al*., 2005). "Due to their favorable mixing features opposing jet configurations have found applications in several different processes […] and combustion" (Hosseinalipour and Mujumdar, 1997).

The opposing jets configuration is also employed in the study of diffusive combustion phenomena as: vortex/flame interaction (Santoro *et al*., 1999, Amantini *et al*., 2006), extinction (Pellet *et al*., 1998, Daou and Liñán, 1999), the effect inhibitors on the diffusion flame (Tsuji, 1982), the transition of the diffusion to pre-mixed flame (Frouzakis *et al*., 2002) and the multi-fuel diffusion flame (Fachini, 2001).

Roseira Jr (2005) studied the non-premixed counterflow flames inside a parallel flat plate channel, describing the transient flame behavior under the effects of Reynolds number, jet velocities and nozzle thickness. In Roseira Jr (2005) a finite difference method was applied with the transient vorticity-stream function formulation. The incompressible flow approach was adopted and the infinity fast chemical reaction model is considered.

The present work is devoted to the numerical investigation of the non-premixed flames in laminar confined opposing jets inside a parallel flat plate channel. Two different chemical species, the fuel and the oxidant, enter through different slots (nozzles) positioned on each channel wall. The proposed model considers a laminar incompressible flow formulation. A parametric study is conducted for Reynolds number, Peclet number, stoichiometric mixture fraction, jet velocities, nozzle thickness, inlet temperature ratio and heat of reaction. The influences of the parameters on flame behavior are analyzed. The results show the flame shape, temperature, streamline and mixture fraction fields.

2. PHYSICAL AND MATHEMATICAL MODELS

The physical and the mathematical model involve the combustion process of two initially separate streams of different reacting chemical species in two-dimensional confined laminar opposing jets. The fuel and the oxidant jets enter in the channel through different slots positioned on each channel wall. In the entrance region of the channel, the opposing jets create a counterflow. The thermo-physical properties are assumed constant. The model is based on the mass, momentum and mixture fraction conservation equations.

"The Burke–Schumann flame-sheet model is adopted. Thus, equal diffusivities of heat and matter are assumed, and combustion is described by a single irreversible and infinitely fast one-step reaction of the form

$$F + \nu \, Ox \rightarrow P \tag{1}$$

where F denotes the fuel, Ox the oxidizer, and P the products. The stoichiometric coefficient ν represents the moles of oxidizer consumed per mole of fuel" (Daou and Rogg, 1998).

The figure 1 shows the simplified geometry of the proposed problem, representing the entrance and exit regions of the channel, the parallel flat plate channel walls, the symmetry plane and the mathematical domain.

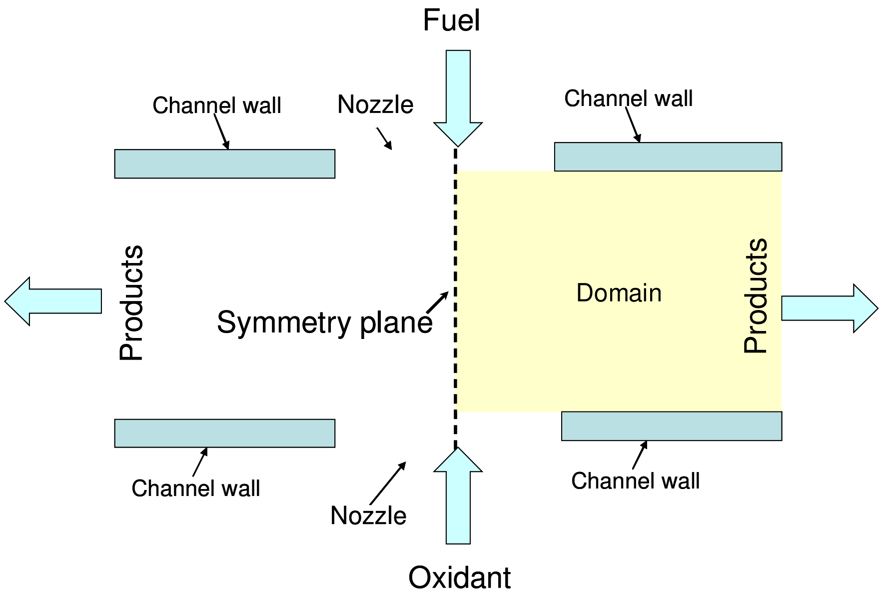

Figure 1. Simplified geometry of the problem.

The geometry adopted to represent the mathematical domain is described in Fig. 2. The left boundary represents the flow field symmetry plane and the nozzle separation width (L^*), the right boundary is the outflow region, as well as the truncation position in the horizontal direction. The top and bottom boundaries (P^*) include the nozzles thickness, respectively, A_i^* and D^*, and the adiabatic solid walls.

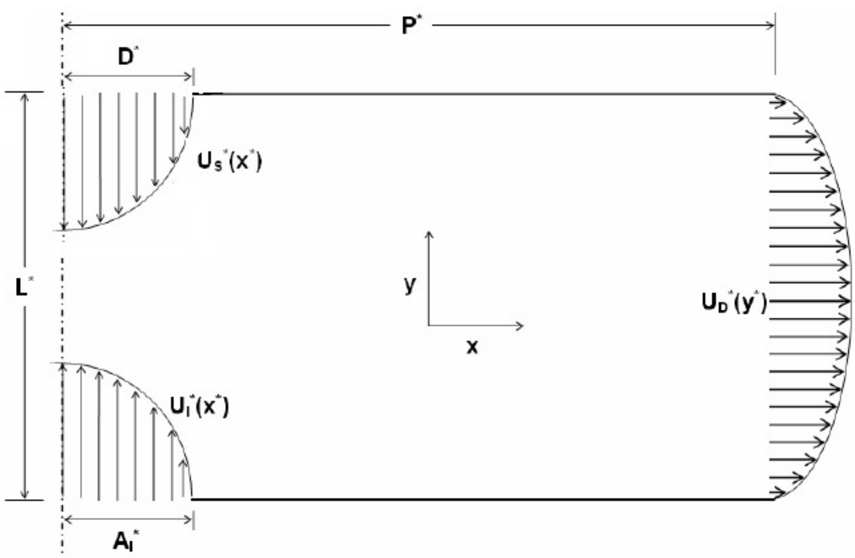

Figure 2. Geometry of the physical domain.

The transient, laminar and incompressible flow formulation is presented below. However, in the present work only the steady-state results will be investigated.

Mass conservation equation:

$$\nabla \cdot \boldsymbol{u} = 0 \tag{2}$$

Momentum conservation equation:

$$\frac{\partial \boldsymbol{u}}{\partial t} + \boldsymbol{u} \cdot \nabla \boldsymbol{u} = -\nabla p + Re^{-1}\, \Delta \boldsymbol{u} \quad (3)$$

Mixture fraction conservation equation:

$$\frac{\partial Z}{\partial t} + \boldsymbol{u} \cdot \nabla Z = (\, LePe\,)^{-1}\, \Delta Z \quad (4)$$

In the conservation equations ∇ and Δ denote the nondimensional gradient and the Laplace operator, respectively. In addition Z is the mixture fraction.

Eq. (5) gives the definitions of the dimensionless variables and parameters employed in the mathematical model.

$$x = \frac{x^*}{L^*} \quad y = \frac{y^*}{L^*} \quad P = \frac{P^*}{L^*} \quad p = \frac{p^*}{\rho^2 \left(U^*_{DMAX}\right)^2} \quad \boldsymbol{u} = \frac{\boldsymbol{u}^*}{U^*_{DMAX}}$$

$$Re = \frac{U^*_{DMAX} L^*}{\upsilon^*} \quad Pe = \frac{U^*_{DMAX} L^*}{\alpha^*} \quad R_b = \frac{A^*_i}{D^*} \quad R_v = \frac{R_b\, U^*_{I\,MAX}}{U^*_{S\,MAX}} \quad R_{DL} = \frac{D^*}{L^*} \quad (5)$$

$$Pr = \frac{Pe}{Re} \quad Le = \frac{\alpha^*}{D^*_{ij}} \quad q = \frac{q^*}{c^*_p\, T^*_{ox,i}} Z_{st} \quad e = \frac{T^*_{fu,i}}{T^*_{ox,i}} \quad Z_{st} = \frac{Y_{ox,i}}{n + Y_{ox,i}}$$

In Eq. (5) U^*_{DMAX} is the maximum outlet fluid velocity, $\boldsymbol{u}^*$ is the fluid velocity vector, L^* is the height of the channel, q is the nondimensional heat of reaction, Z_{st} is the stoichiometric value of the mixture fraction, n is the stoichiometric coefficient (mass of oxidant consumed per mass of fuel) and e is the inlet temperature ratio. The thermo-physical properties of the fluid are the thermal diffusivity (α^*), the kinematic viscosity (υ^*), the density (ρ^*), the specific heat at constant pressure (c_p), the heat of reaction (q^*) and the species diffusivity (D_{ij}^*). The dimensionless numbers are the Reynolds number (Re), the Prandlt number (Pr), the Peclet number (Pe), and the Lewis number (Le). Others parameters are defined: the nozzle thickness ratio (R_b), the confinement ratio (R_{DL}) and the inlet jet velocities ratio (R_v). In the present work, the Lewis number is considered constant and equal to 1.

The boundary conditions are written in Eqs. (6-11). These equations represent the symmetry plane in the left boundary of the domain Eq. (6); the developed condition of the flow in the right boundary of the domain Eq. (7); the inlet jet condition in the upper and lower nozzles, respectively, Eq. (8) and Eq. (10), and the no-slip and impenetrability boundary condition on the adiabatic solid walls, Eq. (9) and Eq. (11). The inlet jet conditions consider a parabolic function with maximum velocity in the symmetry plane and null velocity at the solid wall.

$$\frac{\partial u}{\partial x} = 0 \;\;, \frac{\partial v}{\partial x} = 0 \;\;, \frac{\partial Z}{\partial x} = 0 \;;\; x = 0, \quad 0 < y < L \quad (6)$$

$$\frac{\partial u}{\partial x} = 0 \;\;, \frac{\partial v}{\partial x} = 0 \;\;, \frac{\partial Z}{\partial x} = 0 \;;\; x = P, \quad 0 < y < L \quad (7)$$

$$u = 0 \;\;, v = -\frac{(\, R_{DL}^{\;2} - x^2\,)}{R_{DL}^{\;3}(\, 1 + R_v\,)} \;\;, Z = 1 \;;\; 0 < x < R_{DL} \,,\, y = L \quad (8)$$

$$u = 0 \;\;, v = 0 \;\;, Z = 0 \;;\; R_{DL} < x < P \,,\, y = L \quad (9)$$

$$u = 0 \;\;, v = \frac{R_v \left[(R_b R_{DL})^2 - x^2 \right]}{(R_b R_{DL})^3 (1 + R_v)} \;\;, Z = 0 \;;\; 0 < x < R_b R_{DL} \,,\, y = 0 \quad (10)$$

$$u = 0 \;\;, v = 0 \;\;, Z = 0 \;;\; R_b R_{DL} < x < P \,,\; y = 0 \quad (11)$$

3. NUMERICAL PROCEDURE

The finite volume method was used for the numerical solution of the governing equations previously described. The semi-infinite physical domain was truncated at $x = L$ where the outlet boundary conditions were applied. Numerical tests were performed for different values of L in order to guarantee the solution independence of the truncated domain length. It was observed that a good criterion to truncate the domain is

$$P = 0.2\ Re\ . \tag{12}$$

The Eq. (12) is based on the fluid dynamic development problem inside a parallel flat plate channel. It is a modified expression for the fluid dynamic development length (Kays and Crawford, 1980). The finite volume method is employed in a non-staggered grid (Maliska, 1995) and the pressure-velocity coupling is solved by the SIMPLEC algorithm (Van Dormaal and Raithby, 1984). The WUDS (Raithby and Torrance, 1974) is used as the interpolation function in an implicit scheme. The linear systems of equations are solved by the GMRES algorithm (Press *et al.* 1992). A uniform structured mesh with 120 x 50 volumes is adopted. Numerical tests were performed for different number of volumes to guarantee that maximum relative discrepancy is 1% in the velocities and mixture fraction solutions. The pseudo-transient approach is considered until the steady-state condition is reached.

4. RESULTS

The numerical solution of Roseira Jr (2005) for the heat transfer problem depicts the confined counterflow of one hot and one cold stream. This reference solution of a confined opposing jet configuration inside a parallel flat plate channel is used to validate the proposed numerical procedure. It is important to note that the heat transfer problem described by Roseira Jr (2005) and the present mixing problem are mathematically the same when the Lewis number is equal to 1. Nevertheless, the dimensionless procedures in the present work and in Roseira Jr (2005) are different. So, the results of Roseira Jr (2005) are transformed to the same dimensionless form adopted here.

From the mixture fraction variable is possible to compute temperature, fuel concentration and oxidant concentration, employing the Eq. (13-15) (Daou and Rogg, 1998, Williams, 1985).

$$\theta = \begin{cases} 1-(1-e)Z + \dfrac{q}{Z_{st}} Z & , Z \le Z_{st} \\ 1-(1-e)Z + q\dfrac{(1-Z)}{(1-Z_{st})} & , Z \ge Z_{st} \end{cases} \tag{13}$$

$$Z \le Z_{st} \begin{cases} Y_{ox} = \dfrac{Z_{st}-Z}{Z_{st}} Y_{ox,i} \\ Y_{fu} = 0 \end{cases} \tag{14}$$

$$Z \ge Z_{st} \begin{cases} Y_{ox} = 0 \\ Y_{fu} = \dfrac{Z-Z_{st}}{1-Z_{st}} Y_{fu,i} \end{cases} \tag{15}$$

In Eq. (13) $\theta \equiv T^*/ T^*_{ox,i}$ is the nondimensional temperature and $T^*_{ox,i}$ is the inlet oxidant temperature. Furthermore, in Eq. (14) and in Eq. (15), Y_{ox} is the oxidant concentration, Y_{fu} is the fuel concentration, $Y_{ox,i}$ is the inlet oxidant concentration and $Y_{fu,i}$ is the inlet fuel concentration.

The temperature, fuel concentration and oxidant concentration fields obtained by Roseira Jr (2005) are compared with the present results in Fig. 3 and in Fig. 4. These cases consider the following parameters values: $R_{DL} = 0.5$, $R_v = 1$, $R_b = 1$, $Re = 60$, $Pr = 0.7$. The concentration results in Fig. 3 and the temperature results in Fig. 4 are evaluated at $x = 0.4$. From these figures is possible to observe that there are not relevant discrepancies between the present results and the Roseira Jr (2005)´s results.

A parametric study of the problem is conducted in order to identify the influences of the Reynolds number (Re), the Peclet number (Pe), the nozzle thickness ratio (R_b), the inlet jet velocities ratio (R_v), the nondimensional heat of reaction (q), the stoichiometric value of the mixture fraction (Z_{st}) and the temperature ratio (e) on the flame behavior and on the mixture fraction and streamlines fields.

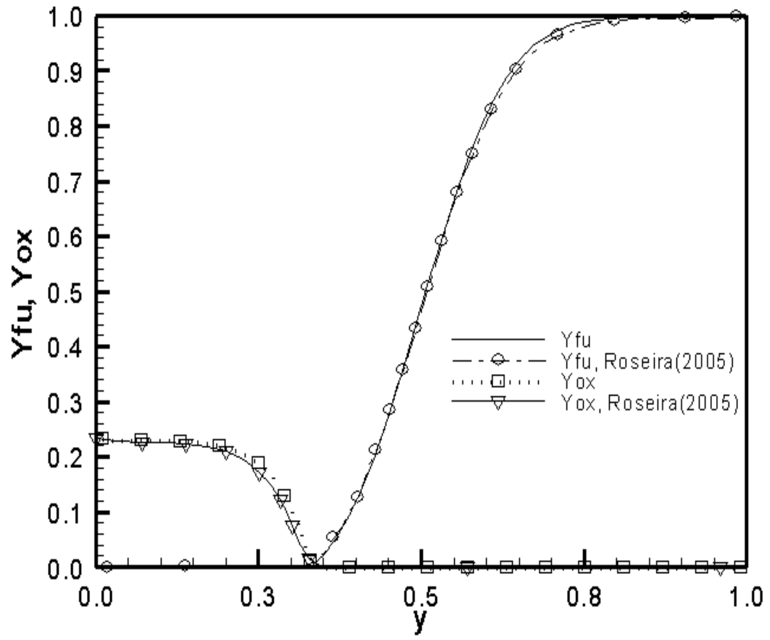

Figure 3. Fuel and oxidant concentration profiles at $x = 0.4$.

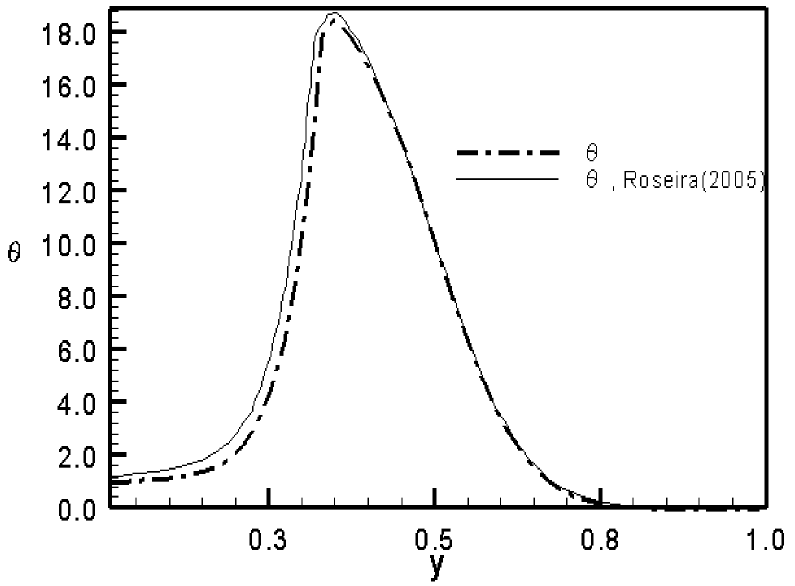

Figure 4. Temperature profiles at $x = 0.4$.

One reference case is chosen to perform the parameter analysis. Then, in the following results each parameter above described will be modified in relation to the reference case, permitting the evaluation of each parameter influence on the solution. The values of the reference case parameters are presented in Tab. 1. The values of the physical and geometric parameters shown in Tab. 1 are defined with basis on the Roseira Jr (2005) work.

Table 1. Values of the reference case parameters.

Re	Le	Pe	R_b	R_v	R_{DL}	q	Z_{st}	e
60	1	42	1	1	0.5	2	0.2	1

The streamline and mixture fraction field for the reference case are presented in Fig. 5 and in Fig. 6. In these figures, the symmetry of the isolines in relation to a longitudinal line positioned at $y = 0.5$ is verified. This behavior is promoted by the symmetry of the boundary conditions. At the channel exit the mixture fraction reaches an equilibrium value approximately equal to 0.5. These results are consequence of the equal inlet mass condition for the oxidant stream and for fuel stream. The equilibrium condition reveals the ideal mixture fraction.

In Fig. 7 the results for *Re* = 150 are reported. There are no relevant qualitative differences between the streamlines behavior in Fig. 7 and in Fig. 5 where the streamlines of the reference case are shown. However, the flow field is different and the convection effects are more important for *Re* = 150. In other words, the difference is quantitative and cannot be observed in the streamline figures present in this work.

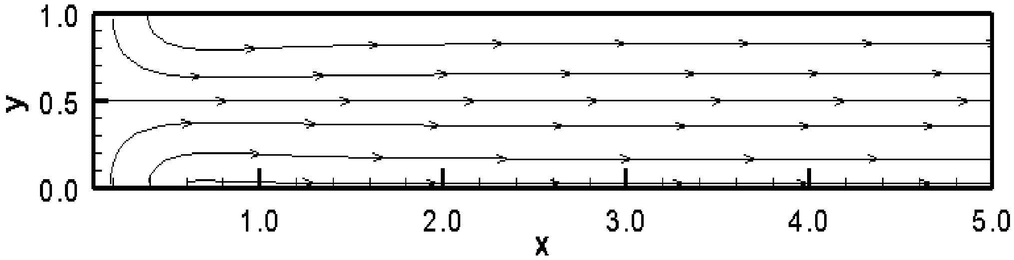

Figure 5. Streamlines (reference case).

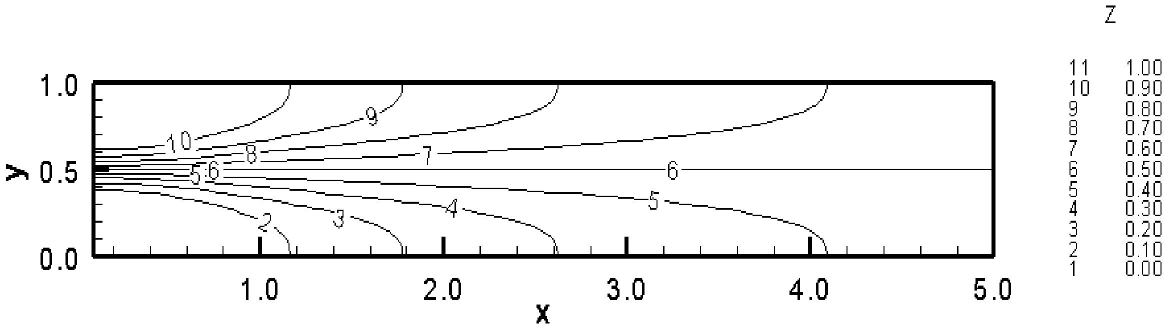

Figure 6. Mixture fraction field (reference case).

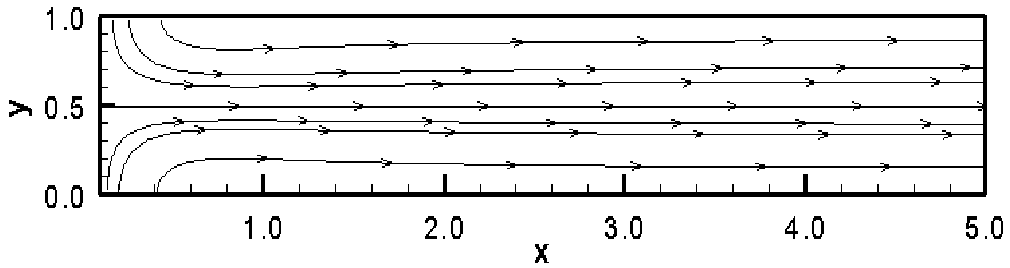

Figure 7. Streamlines (*Re* =150).

The effect of the Reynolds number parameter on the mixture fraction field can be analyzed comparing Figure 6 and 8. In Fig. 8 the results for *Re* = 150 case are shown. The increase in the Reynolds number implies in the increase in the mixture fraction development region. For high values of the Reynolds number the flow drags the species downstream, disturbing the mixing process. So, large channel lengths are required to reach the equilibrium mixture fraction condition. However, the symmetry of the figure is preserved.

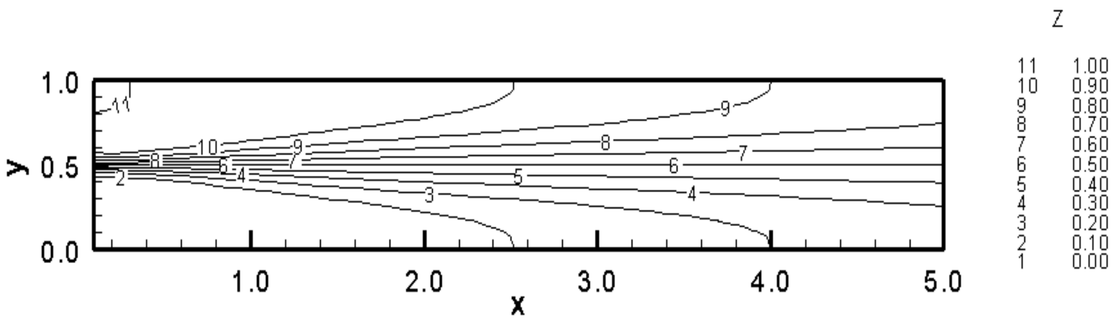

Figure 8. Mixture fraction field (*Re* =150)

In Fig. 9 the effect of the *Pe* parameter is depicted. Comparing, Fig. 6 with Fig. 9 is possible to note that decreasing the Peclet number the development region of the mixture fraction field is reduced. The mixture fraction reaches the equilibrium condition in short lengths of the channel. The species diffusivity effects become more important in the case presented in Fig. 9, improving the mixing process. The symmetry of the Fig 9 is kept as observed in Fig. 6 and Fig. 8.

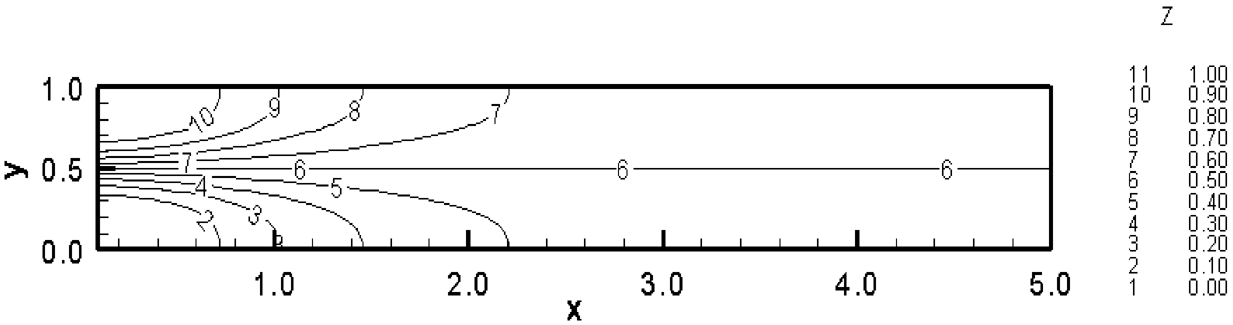

Figure 9. Mixture fraction field (Pe = 21).

The streamlines for Pe = 21 case is the same represented in Fig. 5 because for the incompressible flow the variations in the Peclet number does not influence the fluid dynamics.

The inlet jet velocities ratio (R_v) is investigated in Fig. 10 and Fig. 11. Fig. 10 shows the streamlines and Fig. 11 shows the mixture fraction field. When the inlet jet velocities ratio is not equal to 1, the inlet amount of fuel and of oxidant is different. In consequence, downstream of the channel, the mixture fraction equilibrium condition is modified and the value of the mixture fraction is not equal to 0.5. The ideal mixing condition represented by the mixture fraction isoline $Z = 0.5$ touches the symmetry plane and the upper channel wall in vicinity of the $x = 2.5$. So, increasing the inlet jet velocities ratio parameter the ideal mixing condition is obtained in short length of the channel. The other hand, a higher oxidant concentration is found downstream. The symmetry discussed above is not preserved for $R_v \neq 1$. In addition, a small eddy zone is formed near the lower nozzle.

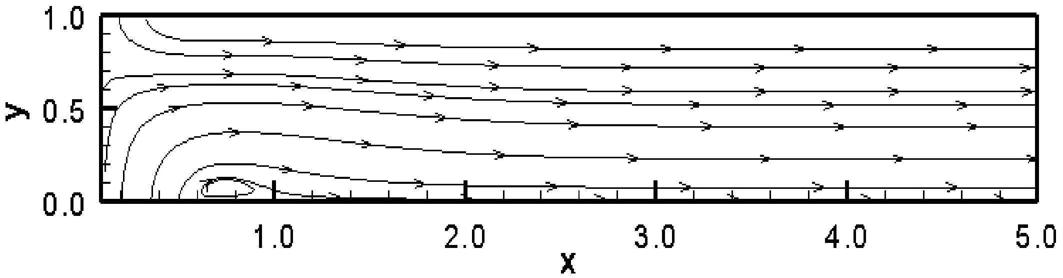

Figure 10. Streamlines (R_v = 2).

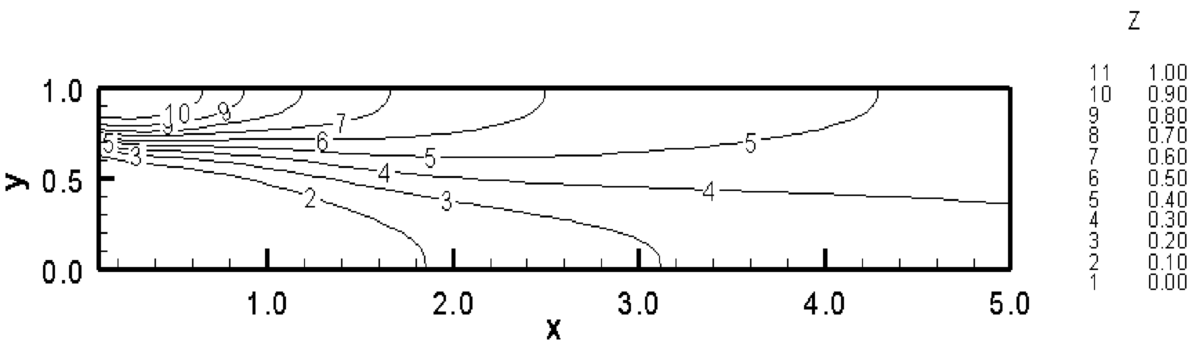

Figure 11. Field of mixture fraction (R_v = 2)

The effects of nozzle thickness ratio parameter (R_b) on the streamlines and on the mixture fraction are presented in Fig. 12 and 13, respectively. The symmetry of the streamlines of the mixture fraction field observed in Figs. 5-9 are not verified when the value of R_b is reduced. In Fig. 13 a strong deformation of the isoline $Z = 0.1$ is verified. It is related with the intense vortex zone formed in the vicinity of the inferior nozzle, which can be seen in Fig. 12.

Comparing the isoline $Z = 0.5$ behavior in Fig. 11 and 13 is possible to note that the ideal mixing condition line starts from the symmetry plane deflects and touches the upper channel wall in vicinity of the $x = 4$ for the case with R_b =

2. Then, the ideal mixing condition is improved for both cases depicted in Fig. 11 and 13, but this effect is more intense when the R_v parameter is reduced.

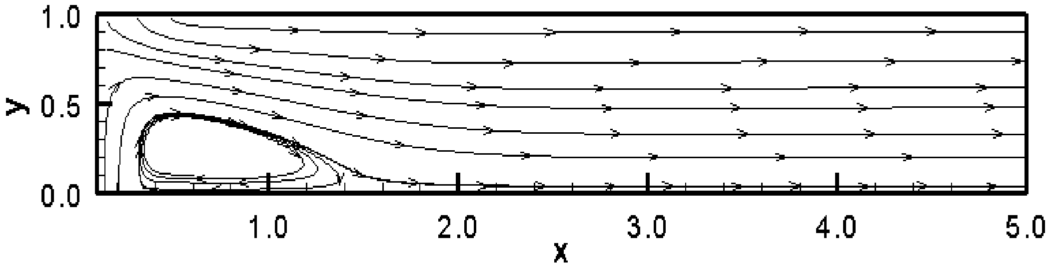

Figure 12. Streamlines (R_b = 0.5).

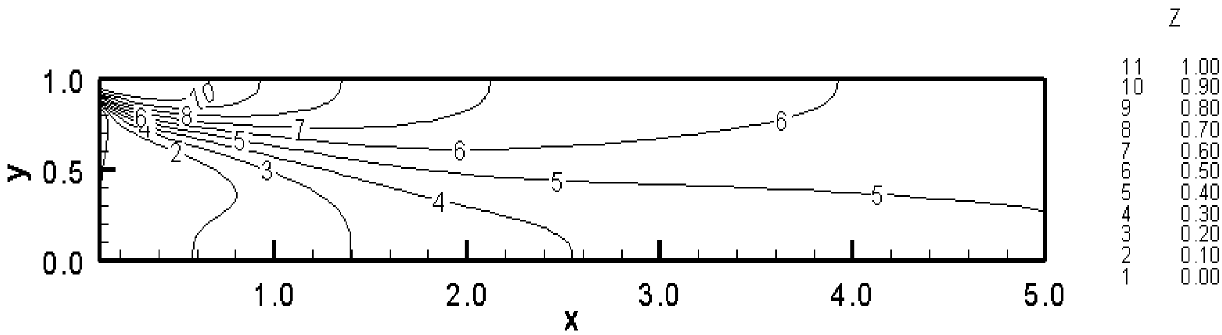

Figure 13. Field of Temperature (R_b = 0.5)

For the incompressible fluid flow hypothesis adopted in this work, the variation on the stoichiometric mixture fraction does not modify the mixture fraction field. Therefore, when the parameter Z_{st} = 0.8 is considered the flame position changes, but the mixture fraction field is the same reported in Fig. 6. For Z_{st} = 0.8 the flame touches the symmetry plane and the upper solid boundary, differing the reference case (Z_{st} = 0.2) where the flame touches the symmetry plane and the lower solid boundary. So, increasing the stoichiometric mixture fraction parameter the flame approaches the fuel inlet nozzle.

In the present model the effects of the heat of reaction parameter and of the inlet temperature ratio acts only in Eq. 12. They do not influence the mixture fraction field and the velocity field, but the effects of these parameters appear only in the temperature field.

In Fig. 14 the temperature field for the reference case is shown. In this figure the high values of temperature identifies the flame. The flame position can also be observed in Fig. 6 in the isoline $Z = 0.2$.

As expected, when the heat of reaction increases, the maximum value of the temperature increases too. However, the flame position is the same for the reference case and for the case with q = 10 which is represented in Fig. 15. In Fig. 16, it is verified that the high temperature region is greater than the reference case (Fig. 14).

In the figure 17 the value of the inlet temperature ratio is reduced and the inlet oxidant temperature is greater than the inlet fuel temperature. The temperature field presented in Fig. 14, for the reference case, and in Fig. 17 is similar. Therefore, the quantity of heat furnished to the flow by the reaction is greater than the quantity of heat furnished to the flow by the inlet oxidant temperature. So, in the present reacting flow the more important source of heat is the chemical reaction found in the flame.

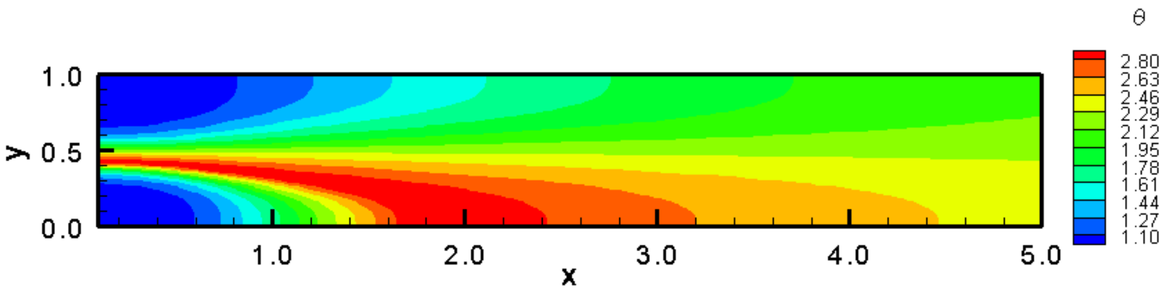

Figure 14. Temperature field (reference case)

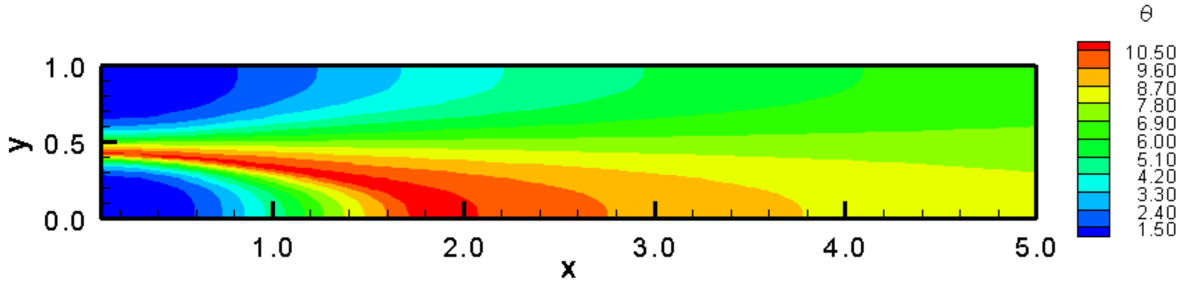

Figure 15. Temperature field (q = 10)

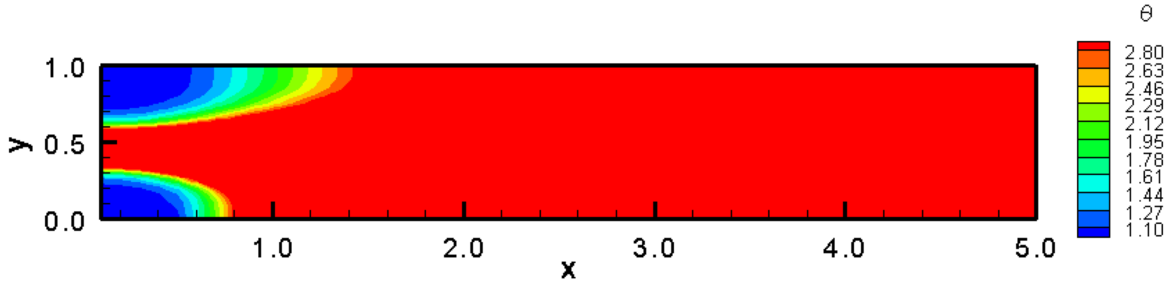

Figure 16. Temperature field (q = 10)

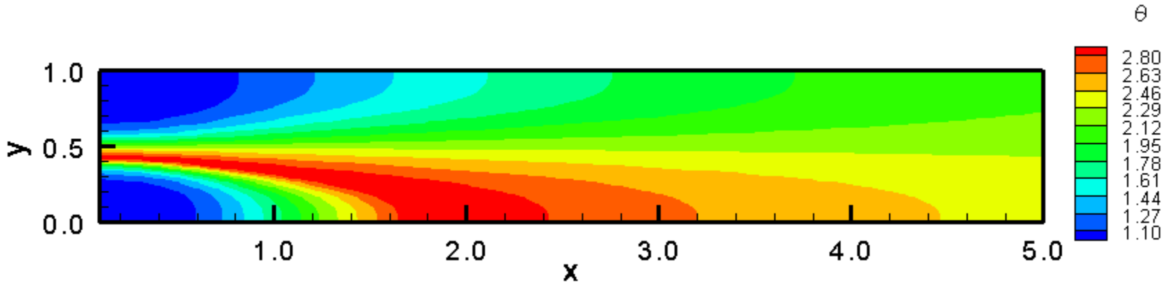

Figure 17. Temperature field (e = 0.5)

5. CONCLUSIONS

The present study shows the results obtained by a finite volume solution for the non-premixed counterflow flame inside a parallel flat plate channel problem. The finite volume solution is compared with a reference solution obtained by Roseira Jr (2005). No significant discrepancies are observed between the proposed solution and the reference solution. A parametric study of the problem is conducted in order to identify the influences of the Reynolds number (Re), the Peclet number (Pe), the nozzle thickness ratio (R_b), the inlet jet velocities ratio (R_v), the nondimensional heat of reaction (q), the stoichiometric value of the mixture fraction (Z_{st}) and the temperature ratio (e) on the flame behavior and on the mixture fraction and streamlines fields. The results show that the fluid flow is more affected by the nozzle thickness ratio and by the inlet velocities ratio, when these parameters are not equal to 1.Under these conditions eddy zones are created, modifying the mixture fraction field. The heat of reaction, the inlet temperature ratio and the stoichiometric mixture fraction ratio does not affect the mixture fraction field, because these parameters are not presented in the conservation equations system. This is a consequence of the incompressible flow hypothesis. For a compressible approach, all of the parameters must be considered in the solution of the conservation equation system, because the equation of state links the temperature field and the density field. However, the incompressible flow formulation is relevant to understand qualitatively the flame behavior.

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