NUMERICAL SIMULATION OF A PREMIXED TURBULENT V-SHAPED FLAME

by

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Original scientific paper
UDC: 662.76:533.6.011:66.011
DOI: 10.2298/TSCI090120012E

In this paper we simulate a turbulent premixed V-shape flame stabilized on a hot wire. The device used is composed of a vertical combustion chamber where the methane-air mixture is convected upwards with a mean velocity of 4 m/s. The flow was simulated running Fluent 6.3, which numerically solved the stationary Favre-averaged mass balance; Navier-Stokes equations; combustion progress variable, and k-ε equations on a two-dimensional numerical mesh. We model gaseous mixture, ignoring Soret and Dufour effects and radiation heat transfer. The progress variable balance equation was closed using eddy break up model. The results of our simulations allow us to analyze the influence of equivalence ratio and the turbulent intensity on the properties of the flame (velocity, fluctuation, progress variable, thickness of flame).This work gives us an idea on the part which turbulence can play to decrease the risks of extinction and instabilities caused by the lean premixed combustion.

Key words: *premixed turbulent combustion, numerical simulation, V-flame*

Introduction

The combustion is now one of the major processes to produce energy, whether it is starting from coal, oil or gas. Combustion intervenes in the fields of transport (rocket motor, planes and automobiles), of the electrical production (thermal power station) or thermal device (boilers and industrial furnaces, domestic hearths…). The growing energy demand of both local and international level implies the need to improve combustion efficiency while preserving the environment by reducing pollutants emissions.

In most practical applications, combustion takes place within a turbulent flow where the phenomena of transfer (mass, energy ...) are more intense than in the laminar regimes. The control of turbulent combustion is therefore fundamental to all current combustion systems. That is why the turbulent combustion is the subject of much research whose main concern determining the reaction rate, different speeds flames, the stability and extinction criteria or polluting emissions.

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Lean premixed combustion is a very promising way to reduce the nitrogen oxide pollutant emissions. Unfortunately, this operating mode leads to local extinction, source of unburnt residue and combustion instabilities. Many experimental and numerical studies are led to the laboratory in order to understand these phenomena [1-3]. The effects of the mixture and turbulence on the premixed flames are studied experimentally by Errard [4], Renou [5], Escudie *et al*. [6], and Degardin [7]. In the other hand, Bell *et al*. [8], Hauguel *et al*. [9], and Robin *et al.* [10] have presented a numerical work which is exclusively interested in the numerical methods and their fields of validities and in the behaviour of the flames with regard, separately, to the variation of the flow, the composition and turbulence. In our knowledge there are no more numerical studies which treat these parameters simultaneously.

The aim of this work consists of a parametric study of the effects of the equivalence ratio and turbulence intensity on the form and the thickness of a premixed V-shaped flame. This study is carried out using the numerical simulations with Fluent [11]. Our simulations correspond to a real configuration which was studied in some experiments at the CORIA laboratory (France) [4-7].

Description of the flow configuration

The physical configuration used in this paper is a vertical, two-dimensional flow which allows a parametric approach of the characteristics of turbulence and composition of gases, in order to study the influence of each quantity on the properties of the flame. The device used consists of a vertical combustion chamber of 230 mm in length and 80 × 80 mm^2 section where the premixture methane-air is convected with a mean velocity of 4 m/s, fig. 1. A hot wire (diameter equal to 0.8 mm) is placed at 90 mm downstream of the entrances of the gas.

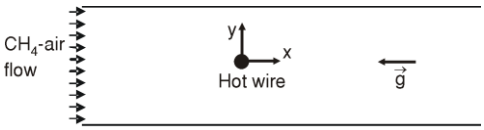

Figure 1. Flow configuration

A V-shape flame is obtained when a premixed flame is stabilized on a hot wire. In this case, the combustion is initiated by the energy released by the wire; the most localized burning kernel serves to stabilize a premixed flame that develops downstream. In a laminar flow, the reaction layer propagates against the incoming fluid and a premixed V-shape flame is obtained. When the flow is turbulent, the two wings of the flame are wrinkled by the fluctuations of the velocity and the V shape of the flame is recovered on mean.

Mathematical model

For the simulations undertaken in this study, it is necessary to simplify the governing equations. We adopt a standard set of assumptions that are well justified for many gaseous combustion systems and have been used in many previous studies. Accordingly, the following phenomena are neglected in this study: the Soret and Dufour effects and radiation heat transfer.

The governing equations used for this study can be written according [12, 13]:

– the mass equation can be written as follows:

$$\frac{\partial \overline{\rho}}{\partial t} + \frac{\partial}{\partial x_i}(\overline{\rho}\tilde{u}_i) = 0 \tag{1}$$

– the momentum conservation equations:

$$u_i'' u_j'' \quad \frac{\partial \overline{\rho}\tilde{u}_i}{\partial t} + \frac{\partial \tilde{u}_i \tilde{u}_j}{\partial x_i} = -\frac{\partial \overline{p}}{\partial x_i} - \frac{\partial \overline{\rho} u_i'' u_j''}{\partial x_j} + \frac{\partial \overline{\tau}_{ij}}{\partial x_i} + \overline{\rho \mathrm{g}}_i \tag{2}$$

– the final major equation completing the formulation is the equation of state for ideal gases:

$$\overline{p} = \frac{\overline{\rho} \mathrm{R} \tilde{T}}{\tilde{M}}; \quad \frac{1}{\tilde{M}} = \frac{\tilde{Y}_i}{M_i} \tag{3}$$

Turbulent model

The Reynolds stress tensor represents correlations between fluctuating velocities. It is an additional stress term due to turbulence. This term is unknown and the number of unknowns in the equations system – eqs. (1), (2), and (3) – became larger than equation number. So we need a model for $u_i'' u_j''$ to close the equations system.

The standard k-ε (proposed by Launder *et al.* [14, 15]) is model based on model transport equations for the turbulence kinetic energy (k) and its dissipation rate (ε). The equation for k is derived from the exact equation, while the equation for ε was obtained using physical reasoning and bears little resemblance to its mathematically exact counterpart. In the derivation of the k-ε model, the assumption is that the flow is fully turbulent, and the effects of molecular viscosity are negligible.

The turbulence kinetic energy and its rate of dissipation are obtained from the following equations:

$$\frac{\partial}{\partial x_i}(\overline{u}_i \overline{k}) = \frac{\partial}{\partial x_j}\left(\frac{\nu_\mathrm{t}}{\sigma_k}\frac{\partial \overline{k}}{\partial x_j}\right) - \overline{u_i' u_j'}\frac{\partial \overline{u}_j}{\partial x_i} - \overline{\varepsilon} \tag{4}$$

$$\frac{\partial}{\partial x_i}(\overline{u}_i \overline{\varepsilon}) = \frac{\partial}{\partial x_j}\left(\frac{\nu_\mathrm{t}}{\sigma_\varepsilon}\frac{\partial \overline{\varepsilon}}{\partial x_j}\right) + C_1\left(-\frac{\overline{u_i' u_j'}(\partial \overline{u}_i)}{\partial x_i}\frac{\overline{\varepsilon}}{\overline{k}}\right) - C_2\frac{\overline{\varepsilon}^2}{\overline{k}} \tag{5}$$

$$\overline{u_i' u_j'} = -\nu_\mathrm{t}\left(\frac{\partial \overline{u}_i}{\partial x_j} + \frac{\partial \overline{u}_j}{\partial x_j}\right) + \frac{2}{3}\overline{k}\delta_{ij}; \quad \nu_\mathrm{t} = \frac{\overline{k}^2}{\overline{\varepsilon}} \tag{6}$$

where C_μ = 0.09, C_1 =1.44, C_2 =1.92, σ_ε =1.30, and σ_k =1.00.

In this paper we opt for the choice of the standard k-ε model for its robustness, economy, and reasonable accuracy for a wide range of turbulent flows.

Turbulent combustion model

Progress variable

The progress variable is defined as a normalized sum of the product species,

$$c = \frac{\sum_{i=1}^{n} Y_i}{\sum_{i=1}^{n} Y_{i,eq}} \tag{7}$$

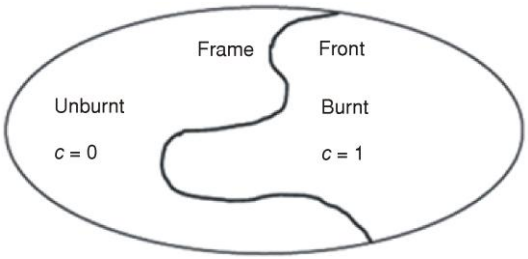

Figure 2. Progress viriable

Relying on this definition we can say that: $c = 0$, where the mixture is cool and $c = 1$, when the mixture is burned, fig. 2.

The value of c is defined as an initial condition, it is usually specified as 0 (unburnt) or 1 (burnt).

Eddy break up model

To simulate the kinetic reaction rate of the combustion phenomena, we have considered the eddy break up mode.

This model is based on a phenomenological analysis of turbulent combustion. The reaction zone is viewed as a collection of fresh and burnt gases pockets. Following the Kolmogrov cascade, turbulence leads to break down of fresh gases structures. Accordingly, the mean reaction rate is mainly controlled by the turbulent mixing time τ_t. When oxidizer is in excess, the mean reaction rate is expressed as:

$$\dot{\omega} = -C_{\mathrm{EBU}}\overline{\rho}\frac{\sqrt{Y''^2}}{\tau_t} \tag{8}$$

where Y''^2 denotes the fuel mass fraction fluctuations and C_{EBU} is a model constant of the order of unity [16]. The turbulent mixing time, τ_t is estimated from the turbulence kinetic energy k and its dissipation rate ε according to: $\tau_t = k/\varepsilon$, as an approximation of the characteristic time of the integral length scales of turbulent flow field.

The reaction rate may be recast in terms of progress variable, c, as:

$$\dot{\omega} = -C_{\mathrm{EBU}}\overline{\rho}\frac{\sqrt{c''^2}}{\tau_t} \tag{9}$$

Mass fraction fluctuation Y''^2 (or progress variable fluctuation c''^2) must be modeled and may be estimated from a balance equation. Assuming infinitely thin flame front, $\widetilde{c}''^2$ is easily estimated because $c^2 = c$:

$$\overline{\rho c''^2} = \rho\overline{(c-\tilde{c})^2} = \overline{\rho}(c^2 - \tilde{c}^2) = \overline{\rho}\tilde{c}(1-\tilde{c}) \tag{10}$$

The square root has been introduced from dimensional reasons in eqs. (8) and (9) but, unfortunately, eqs. (9) and (10) lead to inconsistencies because the $\widetilde{c}$ derivative of $\dot{\omega}$,

$\mathrm{d}\tilde{\dot{\omega}}/\mathrm{d}\tilde{c}$, is infinite both when $\tilde{c}=0$ and when $\tilde{c}=1$ (Borghi, 1999, private communication). Then a correct version of the eddy break up model, without the square root, is used for practical simulation:

$$\tilde{\dot{\omega}} = -C_{\mathrm{EBU}}\bar{\rho}\frac{\varepsilon}{k}\tilde{c}(1-\tilde{c}) \tag{11}$$

Initial conditions

Table 1 summarizes the turbulence condition in the middle of the combustion zone and the combustion parameters for the different flames presented in this paper. Three cases of turbulence and four cases of chemistry conditions were investigated in terms of turbulence intensity $I = u'/U$ and equivalence ratio variations φ. Values of turbulence intensity vary from 4% to 12.5% and the equivalence ratio varies from φ = 0.55 to 1 for stoichiometric case.

These initial conditions were chosen such a ways to compare them with the experimental results realized in CORIA, [4].

Table 1. Numerical conditions for methane/air flame corresponding to the different simulations

u'/U	L_0 [mm]	φ	ρ_u	ρ_b	S_L [ms^{-1}]
4%	3	0.55	1.1435	0.2174	0.061
		0.6	1.1411	0.2050	0.095
		0.7	1.1363	0.1849	0.175
		1	1.2250	0.1490	0.383
9%	6.5	0.55	1.1435	0.2174	0.061
		0.6	1.1411	0.2050	0.095
		0.7	1.1363	0.1849	0.175
		1	1.2250	0.1490	0.383
12.5%	6	0.55	1.1435	0.2174	0.061
		0.6	1.1411	0.2050	0.095
		0.7	1.1363	0.1849	0.175
		1	1.2250	0.1490	0.383

Numerical simulation

We have simulated methane-air flames stabilized behind a hot wire in a rectangular channel performed in different cases characterized by different equivalence ratio and turbulence intensity. The mean inlet velocity was equal to 4 m/s. The results were simulated by running Fluent 6.3, which numerically solved the stationary Favre-averaged mass balance; Navier-Stokes, combustion progress variable, and k-ε equations on a two-dimensional numerical mesh consisting of 340 × 60 nonuniformly distributed nodes in x (axial) and y (transversal) directions, respectively. The nodes were concentrated in the near-field region behind the wire.

Results and discussion

The mean fields of the progress variable c are presented in figs. 3 and 4. The generic V-shaped flame is observed. This allows verify that the simulations are indeed stationary in mean.

We noticed that the angle of the V-flame, is not only function of rate flow but it increases with the equivalent ratio and turbulence intensity.

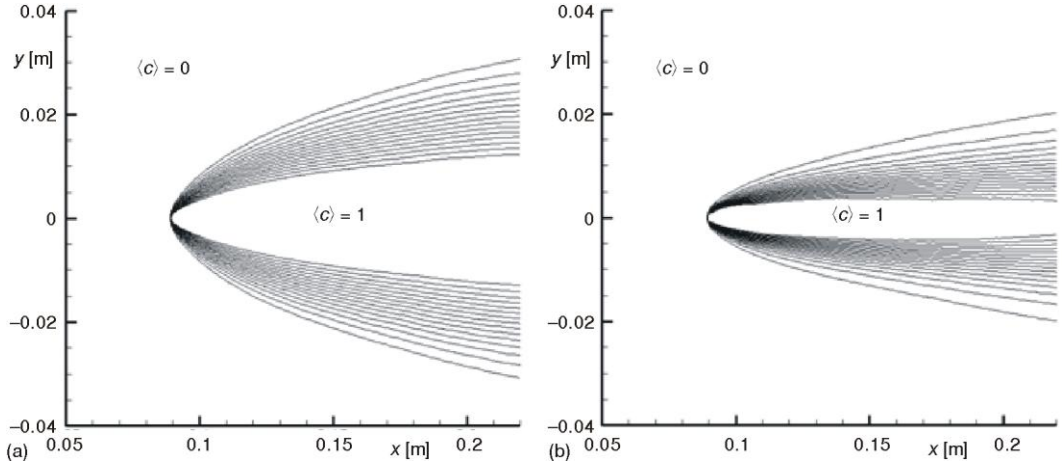

Figure 3. Field of mean progress variable for turbulence intensity I = 9% and for two value of equivalence ratio, respectively, φ = 1 (a) and φ = 0.55 (b)

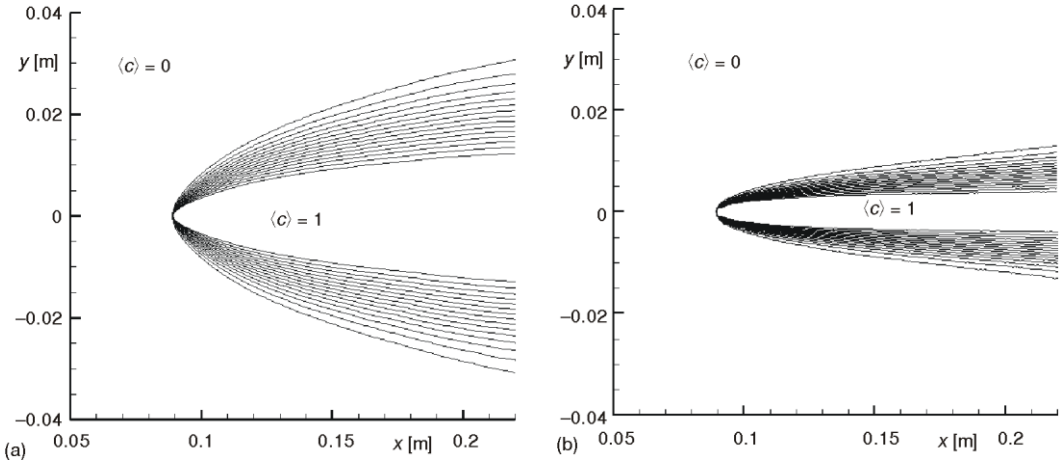

Figure 4. Field of mean progress variable for equivalence ratio φ = 1 and for two value of turbulence intensity, respectively, I = 9% (a) and I = 4% (b)

Figure 5 represents the transverse distributions of mean progress variable for stoichiometric flame at turbulence intensity equal to 9%. It shows that the thickness of the turbulent flame depends on the distance to the hot wire, which is perfectly comprehensible; it is due to the V-shape of the flame.

But there are two other parameters which make increase the thickness of turbulent flame, fig. 6. Shows us that this thickness is maximum for equivalent ratio equal to 1 and becomes increasingly low for lean premix. It also increases when turbulence becomes increasingly important, fig. 7.

To validate our numerical results, we compared them with the experimental results of Erard [4] and we observed a good agreement concerning the profiles of the progress variable for different values the richness, fig. 6.

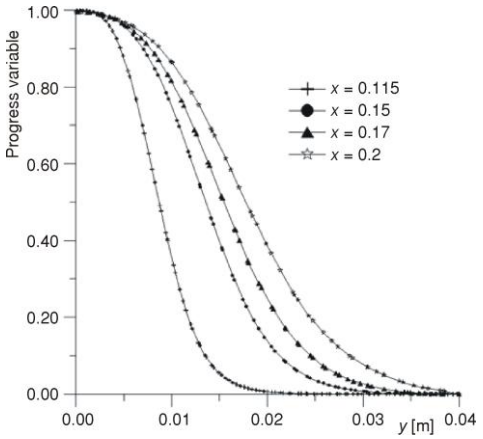

Figure 5. Transverse distributions of mean progress variable for equivalence ratio φ = 1 and for turbulence intensity I = 9%

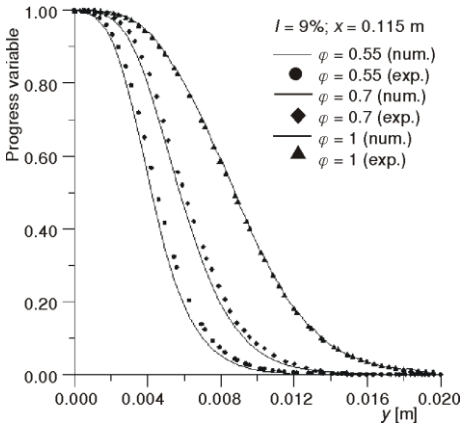

Figure 6. Progress variable at x = 0.115 m for turbulence intensity I = 9%

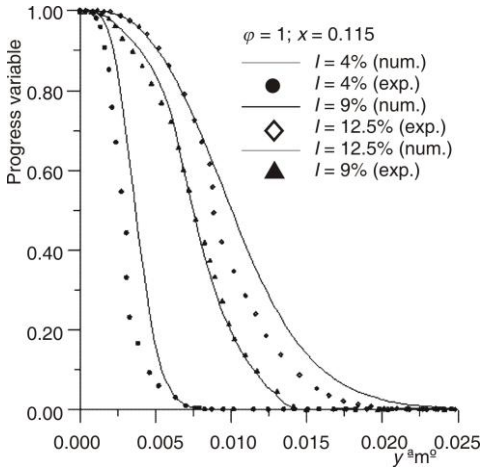

Figure 7. Transverse distributions of mean progress variable for equivalence ratio φ = 1 at x = 0.115 m

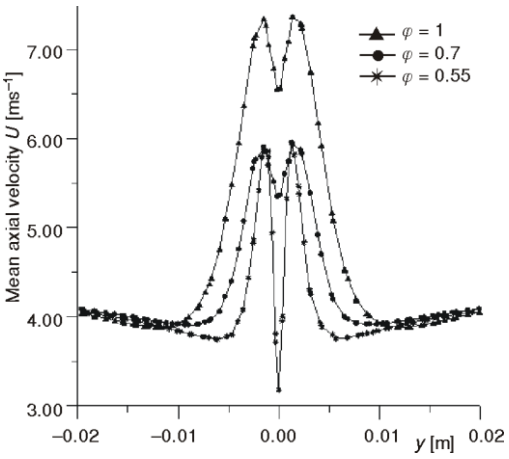

Figure 8. Axial distributions of mean velocity just after the wire for turbulence intensity I = 9%

While on the profiles of the progress variable at different turbulence intensity, fig. 7, the difference between numerical and experimental becomes more remarkable. This may be due to inaccurate experimental method of variation of intensity of turbulence. However, this difference remains within the limits of acceptable and we can say that our results are admissible.

The mean velocity is reduced in wake of wire but it is offset by the acceleration due to thermal expansion of burnt gas, as can be seen on the fuel rich flames, fig. 8. We notice that the velocity of burnt gas increases with equivalence ratio and burnt gases are deflects to the axis of symmetry, where the transversal velocity V is equal to zero.

Figure 9(a) shows that the mean velocity is maximum in the center of flame. This maximum velocity in the burnt gas is related on the heat release and the gradient of density to the crossing of the flame front.

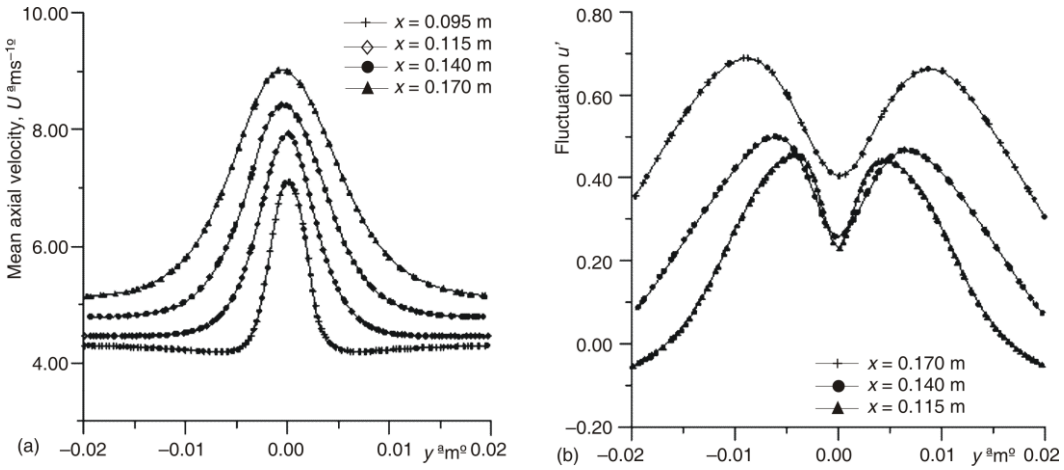

Figure 9. Profiles of the axial mean (a) and fluctuation (b) velocity for φ = 0.55 and I = 9% turbulence

The dynamic profiles calculated for several heights show the spacing of the flame and the acceleration of the burnt gas when moves away from the heated wire. We can see that the jet flame induces good symmetry at axial velocity

The profiles of u', fig. 9(b), show the presence of two maximum centered on the flame front. They also show that the turbulent thickness of the flame increases with the height. These profiles highlight the spacing of the flame with intensities maximum becoming gradually broader at places further away from the origin of the flame, hot wire, since the structures are expanded.

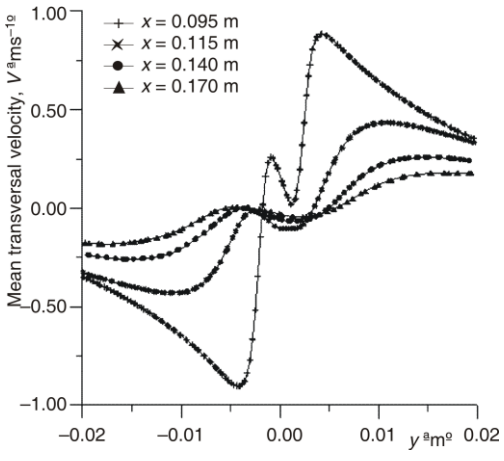

Figure 10. Transverse mean velocity for an equivalence ratio, φ = 0.55 and turbulence intensity, I = 9%

Axial velocity is largest because of main direction flow but the transverse velocity, generated by the flame, although weaker of a factor approximately 10, are also of a great interest for the analysis of combustion. Figure 10 watch that the existence of a non-null transverse velocity upstream of the front and in the flame proves that combustion induces a deviation of the streamlines in fresh gases as in burnt gases. The streamlines are pushed back on both sides of front flame in the outside flow and the greatest deflection is at the approach of the front. The passage in the burnt gases causes a deviation in the opposite direction and transversal velocity changes sign. The burnt gases move towards the center of

the flame. With symmetry, one finds a null transversal velocity in the center and the position of the gradient dV/dx maximum can be regarded as the average position of the front of flame.

Conclusions

In this paper we have simulated a turbulent premixed methane-air flame using the Fluent code. For turbulence used the model k-ε and we model gaseous mixture, ignoring the Soret and Dufour effects and radiation heat transfer is neglected. The progress variable balance equation was closed by using eddy break up model the turbulent combustion model. A numerical procedure is introduced to simulate a configuration in which turbulence interacts with propagating premixed flame front that is stabilized by a hot wire. The results of our simulations allow us to analyze the influence of equivalence ratio and the turbulent intensity on the properties of the flame (velocity, fluctuation, progress variable, thickness of flame).

Initial comparisons of our results to experimentally measured flame indicate that our methodology is sufficiently accurate to model this type of flame.

We have shown that turbulence was a major phenomenon in the combustion. By stretching the flame front, turbulence causes an increase in the surface of this front. It result an increase of flame velocity and thus a faster combustion. This allows us to improve the quality of flame for low values of equivalence ratio. However, the increase in combustion speed by the effect of turbulence must be optimized so as not to fall into the opposite effects.

Nomenclature

C_{EBU} – eddy break up model constant, [–]
c – progress variable, [–]
g – gravitational body force, [m.s^{-2}]
I – turbulence intensity, [%]
k – turbulence kinetic energy, [m^{2}s^{-2}]
M – mixture molecular weight, [gmol^{-1}]
M_i – molecular weight of species i, [gmol^{-1}]
n – number of products, [–]
$u_i''u_j''$ – the Reynolds stresses, [m^{2}s^{-2}]
p – pressure, [Pa]
R – universal gas constant, [kJkmol^{-1}K^{-1}]
S_L – laminar flame speed, [m.s^{-1}]
T – temperature, [K]
t – time, [s]
U – mean axial velocity (Reynolds average), [ms^{-1}]
u – axial velocity, [ms^{-1}]
V – mean transversal velocity (Reynolds averaged), [ms^{-1}]
x – axial co-ordinate, [m]
Y – fuel mass fraction, [–]
y – transversal co-ordinate, [m]
Y_i – mass fraction of species i, [–]
$Y_{i,eq}$ – equilibrium mass fraction of – species i, [–]

Greek symbols

δ – Kronecher delta, [–]
ε – turbulent dissipation rate, [m^{2}s^{-3}]
ν_t – fluid turbulent diffusivity, [m^{2}s^{-1}]
ρ – fluid density, [kgm^3]
σ_ε, σ_k – k-ε model constant, [–]
τ – stress tensor, [Pa]
τ_t – turbulent mixing time, [s]
φ – equivalence ratio, [–]
$\dot{\omega}$ – reaction rate, [kgm^{-3}s^{-1}]

Subscripts

i, j – co-ordinate direction

Superscripts

– – Reynolds average
~ – Favre average
′ – fluctuation

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Paper submitted: September 24, 2009
Paper revised: July 20, 2009
Paper accepted: February 11, 2010