Large Eddy Simulation of Mild Combustion

R. Vicquelin[1,2,3], B. Fiorina[2], O. Gicquel[2], G. Lartigue[3], T. Poinsot[4]

[1]CERFACS, 42 Av. G. Coriolis, F-31057 Toulouse, France.

[2]EM2C, CNRS-UPR288, Ecole Centrale Paris
Grande Voie des Vignes F-92295, Châtenay-Malabry, France.

[3] Pôle CCMF, Direction de la Recherche, Gaz de France
F-93211 La plaine Saint Denis, France.

[4]IMF Toulouse, UMR CNRS/INP-UPS 5502,
Av. C. Soula, F-31400 Toulouse, France.

1 Introduction

Mild combustion [1], also referred as flameless combustion, is characterized by low temperature increase in combustion process. To achieve this regime, fresh gases are preheated and diluted with burnt gases before entering into the reaction zone. Temperature peaks that directly impact on the NO_x formation are then considerably reduced compared to standard combustion regimes. In industrial furnaces, this regime is obtained by means of large internal recirculations of burnt gases cooled by walls. Therefore, the temperature becomes almost homogeneous in the combustion chamber and no visible emission of the reaction zone are detectable.

In addition to reducing pollutant formation, hot products recirculation also improves the flame stability. In premixed combustion devices, such introduction of hot products in fresh gases may cause unwanted mixture auto-ignition and flashback phenomena, while in non-premixed systems the flame position remains controlled by the fuel and oxidizer mixing. Therefore mild combustion has more practical application in non-premixed combustion devices [2].

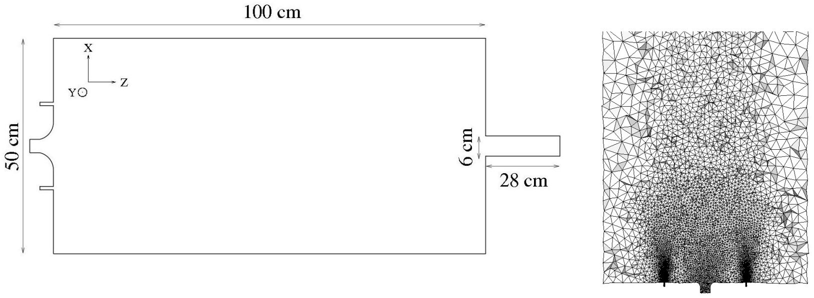
Even though mild combustion processes are used in industrial applications, flame structure and pollutant formation have not been yet fully understood in such regimes. To numerically investigate mild combustion properties, Large Eddy Simulation (LES) is a good candidate. Indeed, high-fidelity simulations of non-premixed turbulent combustion regimes requires an accurate description of the fuel and oxidizer mixing that can not be achieved under steady assumptions (RANS). LES of a non-premixed mild combustion chamber designed at CORIA [3, 4] is performed. In this preliminary study, a simplified representation of the chemistry is assumed. Discussions of the flame structure are proposed.

2 Numerical procedure

The investigated configuration is a 20 kW combustion chamber designed at CORIA [3, 4] in such a way that large recirculation zones made of burnt gases are present. A 2-D view of the combustor geometry is shown on Fig. 1(a). The furnace has a rectangular cross section. The burner consists of a central preheated air jet and two separated methane jets. A high flow velocity at the inlets and a small exit section cause the formation of a large burnt gases recirculation. The global equivalence ratio for the present case is 0.93. Walls are cooled and maintained at a constant temperature of 1300 K in order to

Correspondence to : ronan.vicquelin@em2c.ecp.fr represent industrial furnace conditions.

Large Eddy Simulation is performed with the AVBP solver [5]. This code solves the fully compressible Navier-Stokes equations. The second-order finite volume scheme Lax-Wendroff and a second-order Runge Kutta explicit time stepping are used. Navier-Stokes characteristic boundary conditions [6] are prescribed at inlet and outlet boundary conditions. Turbulence is injected at the three inlets, assuming imposed mean and fluctuating velocity profiles [7]. The bulk velocities of methane and air inlet are respectively 43 ms^{-1} and 38 ms^{-1}. Isothermal walls and wall functions are prescribed. An optically thin model is used to take into account radiation effects [8]. The 3-D geometry is meshed into 900,000 tetrahedra (Fig. 1(b)). All turbulence and turbulence/combustion interaction models are exactly the same as in other studies [8-11] without any adjustement for mild combustion so that the present LES can be viewed as a test of these models in this specific regime.



(a) (b)

Figure 1: combustion chamber (a) and mesh view near the inlet (b).

Chemistry is modeled by a two-step mechanism for CH_4, O_2, CO_2, CO and H_2O where the Arrhenius constants have been fitted to reproduce the flame speed of premixed laminar methane/air mixtures. Subgrid scale effects of the turbulence on the chemistry have been in this first approach neglected because flame fronts are expected to be thick and well resolved on the grid for such regimes.

3 Results

Figure 2(a) shows an instantaneous view of the temperature in the centerline plane. The predicted temperature at the combustor exit is homogeneous and equal to 1310 K. This prediction is in good agreement with the experimental value of 1300 K [3]. As expected the maximum temperature reached inside the combustion chamber (1600 K) is considerably lower than the maximum adiabatic temperature observed in non-premixed flames (2500 K) and an important reduction on NO_x levels is expected. Measurements shows that 65% of the heat is transferred to the wall, which is in good agreement with the 57% predicted by the simulation.

A profile of averaged temperature along the furnace axis is shown on Fig. 2(b). Unlike conventional flames, temperature does not show high narrow peak but increases smoothly from the preheating temperature to the equilibrium temperature. This behavior, identified by Wünning and Wünning [12], is characteristic of mild combustion and is recovered by LES.

A 3-D view of the reaction zone is visible on Fig. 3(a) where an isosurface of heat release colored by temperature is plotted. The main reaction zone, wrinkled by the resolved eddies, is clearly visible. This observation illustrates the flame structure in mild combustion regimes and provides information which is not easy to observe experimentally because the hot wall emission hides the flame front structure.

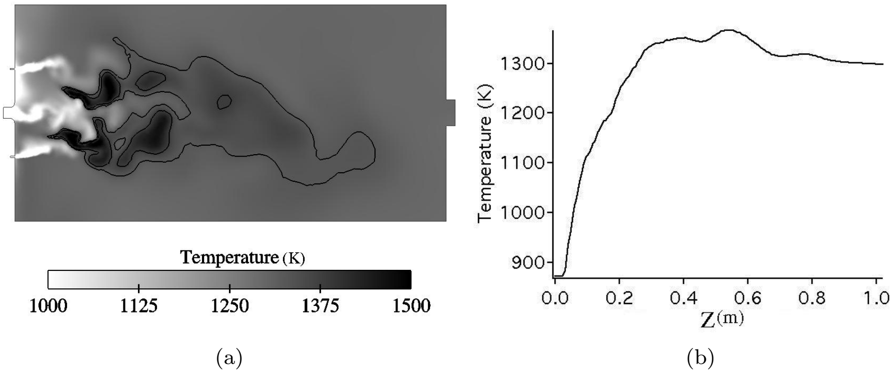
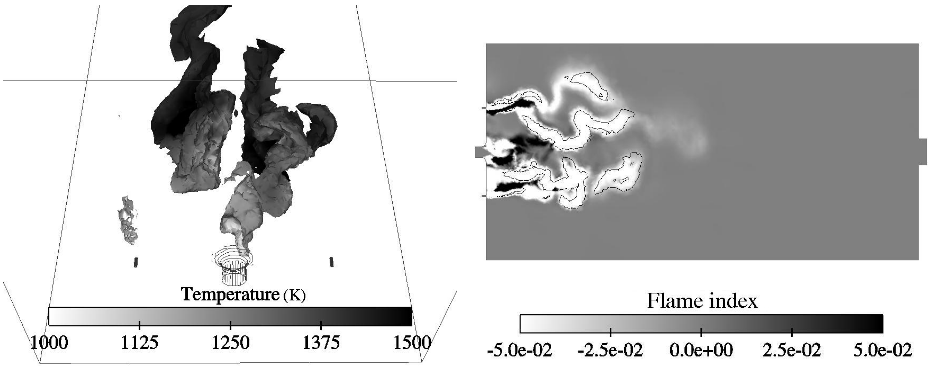


Figure 2: instantaneous temperature field (XZ plane) (a); centerline averaged temperature (b).

A secondary reaction zone that corresponds to the consumption of fuel with the air present in the recirculating burnt gases is visible on the bottom left of Fig. 3(a).

In order to verify whether a diffusion flame structure can still be assumed in this regime, the flame index initialy introduced by Takeno *et al.* is computed [13]. This quantity, defined in term of fuel and oxidizer gradient, is positive in premixed zones and negative in diffusion ones. The centerline plane colored by the flame index is shown on Fig. 3(b). Flame front is localized using an isocontour of the heat release. Premixed zones are identified (dark zones), but combustion mainly occurs under non-premixed conditions (white zones). It means that, in this configuration, flame stabilisation is mainly controlled by diffusion phenomena and not by a premixed kernel propagation.



(a) Isosurface of heat release colored by temperature.

(b) Instantaneous view of flame index and heat release isocontour in XZ plane.

Figure 3: flame structure analysis.

4 Conclusion

A LES performed with a simple two-step chemical scheme qualitatively reproduces the main features of mild combustion. The LES tool used for this work was exactly the same as the one used for multiple other classical turbulent flames. It suggests that as soon as proper mixing and finite rate chemistry effects are included, the main mild combustion characteristics can be captured with LES. Data post-processing shows that the flame structure is mainly controlled by non-premixed regime. Future works will be dedicated to flame chemistry modeling improvements. Comparisons with experimental data of CORIA will be performed.

5 Acknowledgements

We are grateful to CORIA for helpful discussions and to CERFACS and GDF for supporting this study.

References

[1] Cavaliere A. and De Joannon M. (2004). Mild Combustion. Prog. Energy Comb. Sci., 30. 329 : 366

[2] Sobiesiak A., Rahbar S. and Becker H. (1998). Performance Characteristics of the Novel Low-NO$_x$ CGRI Burner For Use with High Air Preheat. Comb. Flame, 115. 93 : 125

[3] Masson E. (2005). Etude expérimentale des champs dynamiques et scalaires de la combustion sans flamme. Ph. D. thesis, Institut National des Sciences Appliquées de Rouen.

[4] Rottier C., Taupin B., Porcheron L., Hauguel R., Boukhalfa A. and Honoré D. (2006). Experimental investigation of flameless (MILD) combustion regime. 31st International Symposium on Combustion, Work In Progress Section, Heidelberg (Germany).

[5] Moureau V., Lartigue G., Sommerer Y., Angelberger C.,Colin O. and Poinsot T. (2004), Numerical methods for unsteady compressible multi-component reacting flows on fixed and moving grids. J. Comp. Physics, 202, 2. 710 : 736

[6] Poinsot T. and Lele S. K. (1992). Boundary conditions for direct simulations of compressible viscous flows. J. Comp. Physics, 101,1. 104 : 129

[7] Smirnov A., Shi S. and Celik, I. (2000). Random flow simulations with a bubble dynamics model. ASME 2000 Fluids Engineering Division Summer Meeting.

[8] Schmitt P., Poinsot T., Schuermans B. and Geigle K. (2007). Large-eddy simulation and experimental study of heat transfer, nitric oxide emissions and combustion instability in a swirled turbulent high pressure burner. Journal of Fluid Mechanics 570. 17 : 46

[9] Selle L., Lartigue G., Poinsot T., Koch R., Schildmacher K.-U., Krebs W., Prade B., Kaufmann P. and Veynante D.(2004). Compressible Large-Eddy Simulation of turbulent combustion in complex geometry on unstructured meshes. Comb. Flame, 137. 489 : 505.

[10] Martin C., Benoit L, Nicoud F. et Poinsot T. (2006) Analysis of acoustic energy and modes in a turbulent swirled combustor. AIAA J., 44, 4. 741 : 750.

[11] Selle L., Benoit L., Poinsot T and Krebs W. (2006). Joint use of compressible LES and Helmholtz solvers for analysis of rotating modes in an industrial swriled burner. Comb. Flame, 145, 1-2. 194 : 205.

[12] Wünning J. A. and Wünning J. G. (1997). Flameless Oxidation to Reduce Thermal NO-Formation. Prog. Energy Comb. Sci., 23. 81 : 94

[13] Yamashita H., Shimada M. and Takeno T. (1996). A numerical study on flame stability at the transition point of jet diffusion flames. 26th Symposium (International) on combustion/The Combustion Institute. 27 : 34