Spontaneous Oscillations in LNGT Combustors: CFD Simulation

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INTRODUCTION

The development of modern gas turbine combustors is linked to the goal of low NOx emissions. Low NOx Gas Turbine (LNGT) combustors operate at lean conditions, thus ensuring low flame temperature, and are, then, adequate to accomplish this requirement. Unfortunately at lean conditions, just before blow-out, spontaneous oscillations of pressure and temperature can be observed [1-4]. These oscillations are unwanted as they cause acoustic oscillations and mechanical and thermal stresses which can damage the combustor and the engine.

In order to understand the mechanisms which drive the instabilities in lean premixed combustion, many studies, both experimental and theoretical, were carried out, but up to now it is still unclear if the occurrence of the oscillations has to be related to instability of the flame (intrinsic instabilities), or whether there is a feedback mechanism between acoustics and heat release from the flame (system instabilities).

The intrinsic instabilities may have different origins: intrinsic kinetic instability [5, 6], thermal-diffusion instability and hydrodynamic instability [7, 8].

Many authors relate the observed oscillations to system instabilities and more precisely to the coupling between the oscillating heat produced by the chemical reaction and the acoustics [2-4, 9-12]. The heat release is assumed to oscillate due to the periodic variations of the air and/or fuel flow rates and, then, of the equivalence ratio. Most of these models are zero-dimensional or 1D and do not show spontaneous oscillations: they assume that the pressure field is oscillating and coupled with the heat release due to the reaction. Consequently, it is still unclear what is exciting the oscillating mode. Spontaneous oscillations were simulated by means of multidimensional models, in which coupling between combustion and fluid-flow is taken into account. This occurrence was addressed to the unsteady heat release at the flame front, which can be in phase with the acoustic resonance of the system, depending on the distance between fuel injector and flame zone [9-11].

In a previous paper [13] we showed, according to the experimental results of Richards et al. [1], that combustion of lean mixtures may give rise to spontaneous oscillations near the blow-out points. The dynamic features of lean premixed combustion were studied by modelling the combustion zone as perfectly mixed and the stability analysis was performed by means of the bifurcation analysis. The satisfactory comparison between the experiments and the theoretical results allowed us to illustrate that the origin of the oscillating behaviour can be thermo-kinetic, i.e. related to the coupling between the stabilising effect of heat produced by chemical reaction and the destabilising effect of heat losses, enhanced at lean conditions. This represents an intrinsic instability rather than a system instability.

The simplified model adopted was able to point out the genesis of the dynamic regimes, but it was unable to take into account any effect of coupling with the fluid flow and spatial non uniformities.

With the specific purpose of confirming the previous analysis in a realistic configuration, in this paper we present the results of CFD simulations showing the dependence of the dynamic behaviour of a lean premixed combustor on the presence of heat losses.

MATHEMATICAL MODEL DEVELOPMENT

Numerical simulations of the LNGT combustor, used by Richards and Janus [14] in their experimental work, were carried out. A 2D axisymmetric CFD model was developed. The computational domain, shown in Figure 1, starts at the inlet plane of the premixer, where a swirling air flow enters and ends at the exit plane of the neck duct.

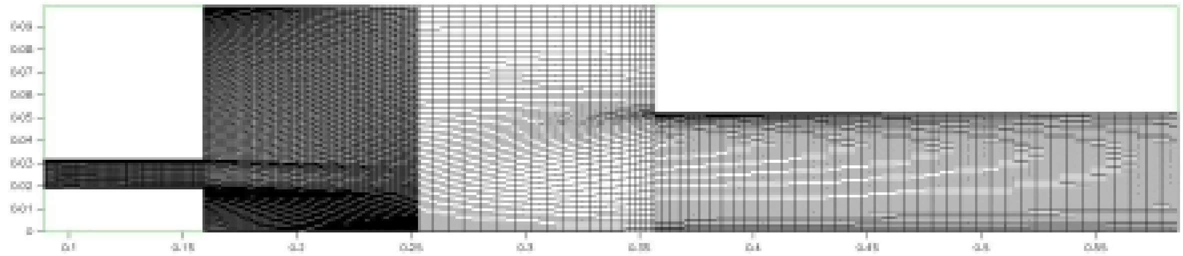

Fig. 1. Computational domain and mesh adopted for all the computations.

In the premix duct, four injectors allow the fuel (methane) inlet. Fuel and air are mixed in the premix barrel downstream the methane inlet. The burnt gas, leaving the combustion chamber, enters the neck duct. It is worth saying that the combination of the combustion chamber and the exhaust neck is an Helmotz resonator [15], but in our simulations the acoustic instability is avoided by using appropriate boundary conditions, since this instability should hide the thermo-kinetic nature of the oscillations we are here investigating.

The model consists of the unsteady Navier-Stokes equations coupled with the conservation equations of mass and energy. Simulations were performed by solving the Reynolds Averaged Navier-Stokes equations (RANS). The renormalization group (RNG) k-ε model for turbulence was chosen [16]. This model is able to give good time-accurate solutions, being the turbulent time scale k/ε of the same order or less than the time scale of the large scale motion. The Eddy-Break-Up model was adopted to model the combustion rate. The reaction rate was combined with a kinetic, one step reaction [17].

An inlet boundary condition of fixed velocity was specified for both air and fuel. In particular, to take into account the effect of the swirler, not included in the computational domain, specific profiles of all the three velocity components were assigned at the inlet section of the premixer. At the exit of the neck a fixed static pressure was specified. By fixing the inlet velocity, it is possible to eliminate the flow rate fluctuation due to the impinging of the acoustic waves on the inflow boundary. This fluctuation leads to the coupling between the heat produced by the chemical reaction and the acoustics. On the other side, at the outlet, assigning constant static pressure means that the pressure perturbations vanish at the outflow [18].

The external walls of the combustor were assumed as cooled by fixing the wall temperature, for the base simulation, at a value equal to 400 K. The CFD-GEOM code [19] was used to develop the 2D axisymmetric structured grid, with 18820 computational cells in total, shown in Figure 1. The time integration was performed by using the second order Crank-Nicholson scheme, with a time step equal to 1.0E-5 s, for temporal differencing and a second order scheme for spatial differencing. The resulting system of non-linear algebraic equations was solved by means of the CFD-ACE+ code [19].

RESULTS

The base case was run starting from the combustor filled with air at 533 K and feeding methane which spontaneously ignites. The CPU time required to reach the stationary solution was about 500 hr on an AMD 1700 MHz single processor workstation. The operative conditions are: velocity in the premixer equal to 30 m/s; pressure P_0 = 500000 Pa; air and fuel inlet temperature equal to 533 K; nominal fuel equivalence ratio equal to 0.77.

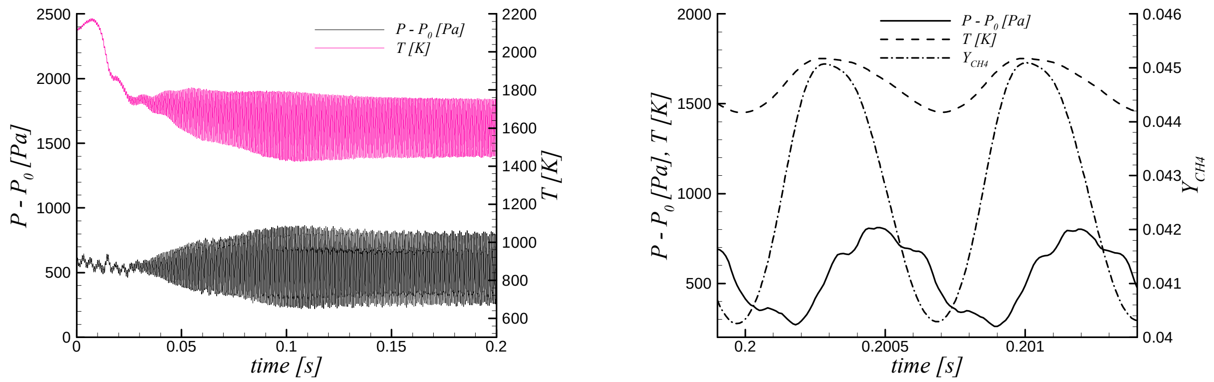

Fig. 2 *Time profiles of the oscillations close to the flame tip for the base case: reduced pressure, $P - P_0$, and temperature, T (left); two entire cycles of the oscillations of reduced pressure, temperature and fuel mass fraction, Y_{CH4} (right).*

In Figure 2 (left), reduced pressure, $P - P_0$, and temperature, T, as function of time in a position close to the flame tip, calculated starting from the time required to ignite the mixture, are shown. It appears that after a transient phase, self-sustained oscillations stabilise with frequency equal to about 1400 Hz. In the same figure (right) two entire cycles of the oscillations of reduced pressure, temperature and fuel mass fraction are shown. It is worth noting that fuel mass fraction and temperature oscillations are in phase in agreement with our previous finding [13].
Being the acoustic instability of the system suppressed by the specified boundary conditions, the found dynamic behaviour cannot be addressed to the acoustics of the simulated combustor, whose natural frequency, in the adopted operative conditions, is about 300 Hz.

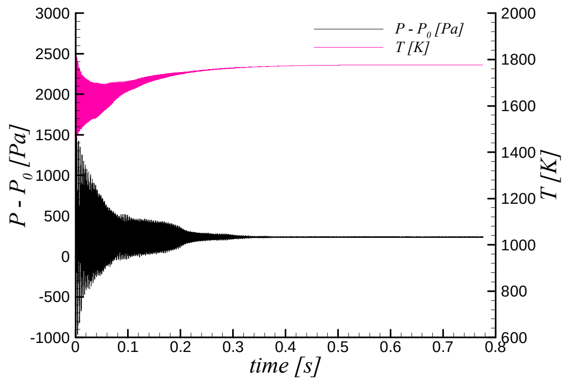

Fig. 3 *Reduced pressure, $P - P_0$, and temperature, T, as function of time, in a position close to the flame tip for the adiabatic case.*

On the ground of the results presented in [13], knowing the role played by heat losses in driving combustion instabilities, we carried out a simulation in which, leaving unchanged any other parameter, the combustor walls were assumed adiabatic. In Figure 3, reduced pressure and temperature, as calculated in the same probe position of Figure 2, are plotted as function of time. After an oscillating transient phase, a stable steady state point is reached.
In Figure 4 (left) the snapshot of the reaction rate in the base case at t = 0.2 s is shown. Figure 4 (right) shows the steady reaction rate profile in the adiabatic case. The irregular location of the leading edge of the flame is due to the adoption of the simple Eddy-Break-Up model, that is well know to overestimate the burning rate in the highly strained regions like the wall

zones [20].

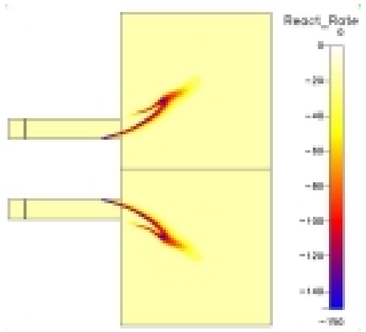

Fig. 4 *Reaction rate field profile [*kg/(m^{3}s)*]: base case at t = 0.2* s *(left); adiabatic case at t = 0.78* s *(right).*

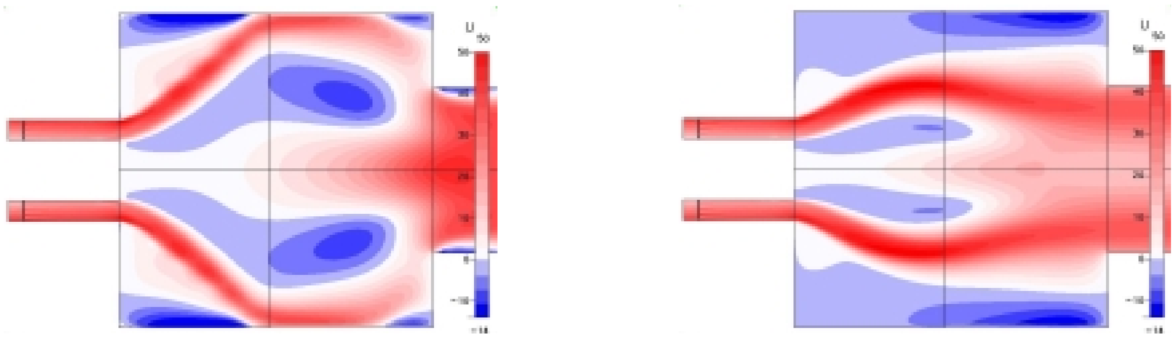

Fig. 5 *Axial component of velocity field profile [*m/s*]: base case at t = 0.2* s *(left); adiabatic case at t = 0.78* s *(right).*

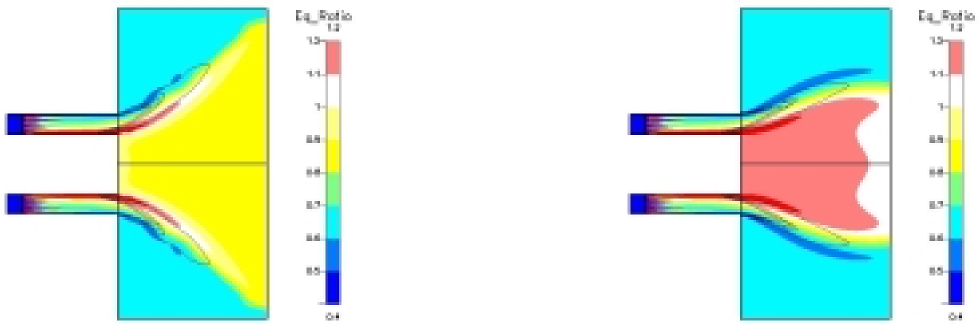

Fig. 6 *Fuel equivalence ratio field profile whit superposed isolines of the reaction rate (corresponding the inner to -100, the outer to -10 [kg/(m^{3}s)]): base case at t = 0.2* s *(left); adiabatic case at t = 0.78* s *(right).*

In both cases the chemical reaction takes place along the inner and the outer shear layer where the cold premixed reactants and the hot products are mixed and burnt. In the base case the unstable flame is clearly unable to attach to the combustor wall, while in the adiabatic case the flame is anchored to the wall.

Moreover, in the base case the swirling flow entering the combustion chamber is attracted by the cold combustor wall, consequently it opens up, creating inner and outer recirculation zones, as it is shown in Figure 5 (left). In the adiabatic case (Figure 5 right), being the combustor cooling absent, the flow expands only slightly, but the two recirculation zones are still present: the outer one is shifted ahead, while the inner one is reduced.

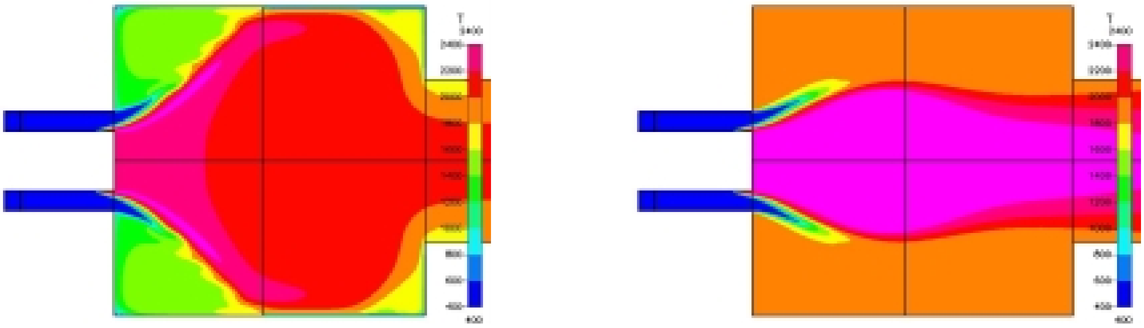

Fig. 7 *Temperature field profile [*K*]: base case at t = 0.2* s *(left); adiabatic case at t = 0.78* s *(right).*

In Figure 6 the maps of the fuel equivalence ratio are shown for both cases. In the inner side of the flame, a fuel richer mixture is present, while in the outer side the mixture is leaner: this condition, recognizable by the isolines of the reaction rate also reported, occurs for both cases. These profiles are due to a partial mixing established in the premixer by the swirling flow.

Temperature maps (Figure 7) show a strong difference between the two cases. The inner recirculating gases are always hotter than the outer ones. In the base case, the outer burnt gases, before mixing whit the fresh reactants, lick the cold combustor walls, lose heat and, then, cool the outer flame zone. In the adiabatic case they remain hot. As a consequence, in the outer side of the flame, which is the leaner mixture, a cooling of the gases occurs in the base case. In both cases the inner side of the flame is steady being stabilised by the rich mixture and the heat exchange with hot recirculating gases. The instability of the flame is produced in the outer side, where a different behaviour between the two cases is observed. In the adiabatic case, the flame is able to self-sustain, though lean, being surrounded by the hot recirculation stream. On the contrary, in the non-adiabatic case, the coupling between lean conditions and heat losses leads to the spontaneous oscillations.

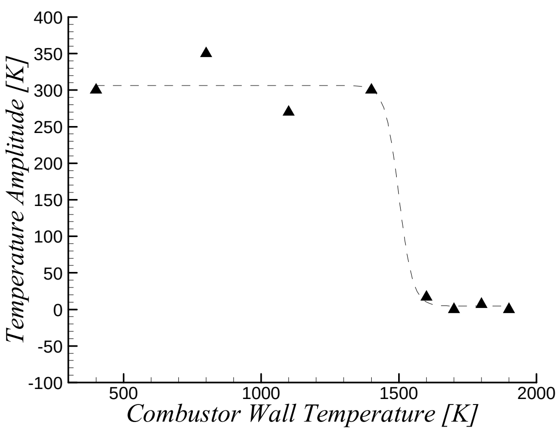

Fig. 8 *Bifurcation diagram: temperature amplitude as a function of the combustor wall temperature.*

In order to study the effect of the cooling medium temperature, in Figure 8 the amplitudes of the temperature oscillations near the flame tip are shown as a function of the wall temperature. It appears that the oscillations are present up to a combustor wall temperature equal to 1400 K and disappear thereafter, showing a bifurcational behaviour. This result points out that the coupling between lean conditions and heat losses is the relevant mechanism in driving the onset of the oscillations, thus suggesting that the results we previously found with the bifurcation analysis [13] are confirmed even in presence of a strong coupling with fluid-dynamics.

CONCLUSIONS

The study of the dynamics of an LNGT combustor has been performed by developing a 2D axisymmetric CFD model. The acoustic instability of the simulated system was suppressed by using adequate boundary conditions. The occurrence of a stable dynamic behaviour in the non-adiabatic conditions and the absence of oscillations in the adiabatic case point out the role played by heat losses in driving the combustion instability. The comparison between flow, reaction rate, fuel equivalence ratio and temperature profiles evidences that the main differences between the adiabatic and the cooled cases can be observed in the outer side of the flame, where mixing between the leaner premixed reactants and the cooler recirculating burnt gases occurs only in the cooled case. This result suggests that a thermo-kinetic mechanism originates the oscillations in the combustor. Coupling with acoustics is expected to simply furnish, as already found in [9-11], also a quantitative agreement with the amplitude and frequency of the oscillations measured in the experimental set-up.
The obtained results seem to confirm the results found with a previous simplified model by means of the bifurcation analysis.

REFERENCES

1. Richards, G. A., Morris, G. J., Shaw, D.W., Keeley, S.A., Welter, M. J.: *Combust. Sci. Tech.,* **94**:57 (1993).
2. Shih, W.P., Lee, J.G., Santavicca, D.A.: *Proceedings of the Combustion Institute*, **26**:2771 (1996).
3. Lieuwen, T., Neumeier, Y., Zinn, B.T.: *Combust. Sci. Tech*., **135**:193 (1998).
4. Lieuwen, T., Zinn, B.T.: *Proceedings of the Combustion Institute*, **27**:1809 (1998).
5. Kalamatianos, S., Park, Y.K., Vlachos, D.G.: *Combust. Flame*, **112**:45 (1998).
6. Park, Y.K., Vlachos, D.G.: *AIChE J*.: **43**:2083 (1997).
7. Clavin, P.: *Prog. Energy Comb. Sci.*, **11**:1 (1985).
8. Clavin, P.: *Proceedings of the Combustion Institute*, **28**: 569 (2000).
9. Smith, C., Cannon, S.: *AIAA Paper* 99-0714 (1999).
10. Steele, R.C., Cowell, L.H., Cannon, S.M., Smith, C.E.: *J. Eng. Gas Turbines Power*, **122**:412 (2000).
11. Cannon, S.M., Adumitroaie, V., Smith, C.E.: *ASME Paper* 2001-GT-0375 (2001).
12. Park, S., Annaswamy, A., Ghoniem, A.: *Thirty-Ninth AIAA Aerospace Sciences Meeting & Exhibit,* Reno, NV, January (2001).
13. Di Benedetto, A., Marra, F.S., Russo, G.: *Combust. Sci. Tech*., **174**:10 (2002).
14. Richards, G.A., Janus, M.C.: *ASME Paper* 97-GT-244 (1997).
15. Richards, G.A., Gemmen, R.S., Yip, M.Y.: *ASME J. Eng. Gas Turbines Power*, **119**:776 (1997).
16. Yakhot, V., Orszag, S.A., Thangam, S., Gatsky, T.B., Speziale, C.G.: *Phys. Fluids*, **4**:1510 (2000).
17. Truffin, K.: *Rapport de stage de D.E.A*., CERFACS, CFD-Combustion, WN/CFD/01-014 (2001).
18. Candel, S.: *Proceedings of the Combustion Institute*, **29**: Hottel Lecture (2002).
19. CFD-ACE+, *CFD Research Corporation*, AL (USA), www.cfdrc.com, (2003).
20. Veynante, D., Vervisch, L.: *Prog. Energy Comb. Sci.*, **28**:193 (2002).