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CFD MODELING OF A LEAN-LEAN-PREMIXED-STAGED COMBUSTION CHAMBER.

D. Valentini, A.A. Barresi and E. Carresa

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CFD Modeling of a Lean-Lean Premixed Staged Combustion Chamber

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INTRODUCTION

Lean premixed combustion is applied to gas turbines combustion systems as a proven DLN (dry low NOx) method [1].

In the present work, the influence of the combustion chamber geometry on pollutants (CO and NOx) emission is investigated. The system under analysis is based on a new mixer and swirl generator characterized by the presence of two sequential stages as shown in Fig. 1.

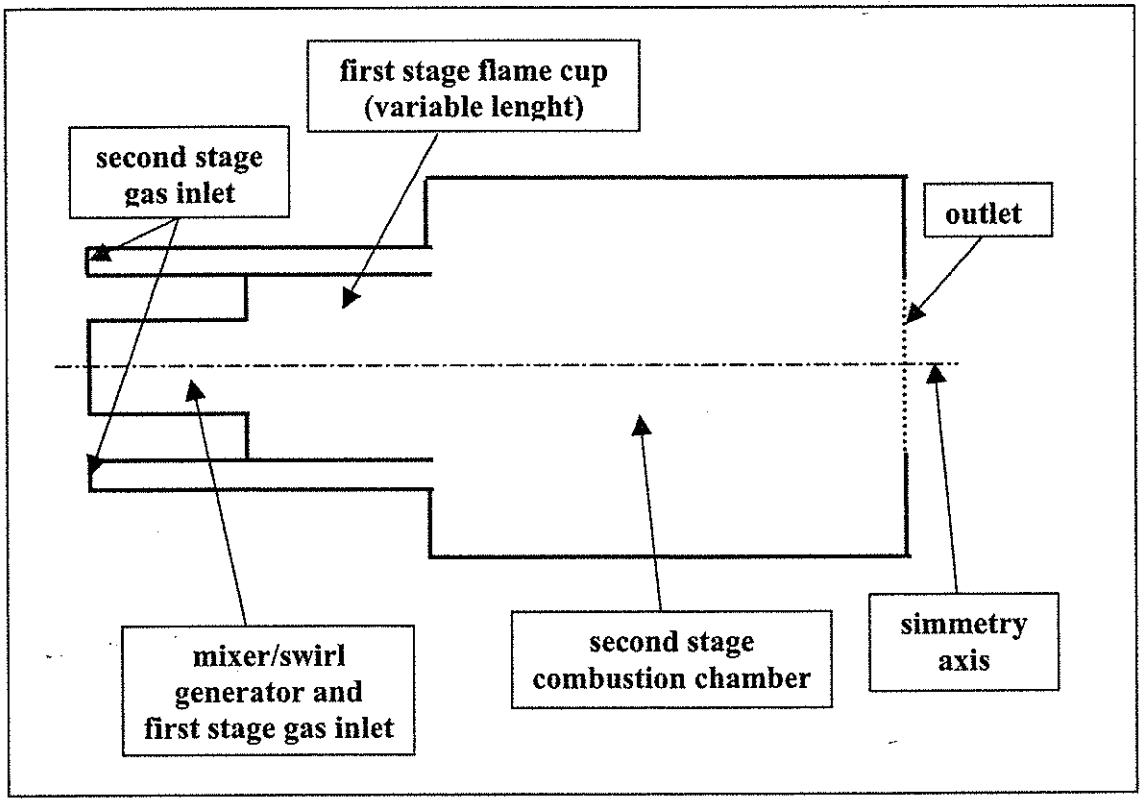

Figure 1. Scheme of the lean-lean premixed staged combustion chamber

The first stage operates at constant temperature, while the second stage operates at variable temperature depending on the load conditions, to keep a stable flame with low NOx emissions over a wide operation range. In the first stage the expansion ratio between the swirl generator and the flame cup allows a recirculation zone to be established to assure stable ignition and combustion in the first stage before cross-ignition with the second stage occurs.

The combustion simulations reproduce experimental tests carried out at atmospheric pressure. The correct reproduction of experimental data will allow the model to be used for further simulations as a reliable prediction tool.

CFD SIMULATIONS APPROACH

The system described above has been experimentally investigated at atmospheric pressure under a wide set of operating conditions [2]. This work reports the results of a CFD simulation approach for the same system; some of the experimental runs have been selected for the comparison. Simulations have been carried out using Fluent 6. A bi-dimensional, axial-symmetric grid has been considered. Measured velocity and mixture concentration profiles have been used as inlet conditions 90 mm upstream the swirler exit into the first flame cup.

Cold flow simulations: mesh and turbulence model validation

Mesh and turbulence model have been validated via comparison with experimental data of axial velocity measured with LDA method in a water channel [2]. The geometry considered in the simulations has a different scale, compared to the one used in the cold flow measurements, and the fluid considered was air instead of water. The comparison here attempted is based on the Reynolds similitude. Different turbulence models have been tested: standard k-ε , RNG k-ε and Reynolds Stress Model. The choice of the standard k-ε model is justified by the correct reproduction of shape and position of the vortex breakdown bubble.

Combustion simulations: choice of the combustion model

Combustion of the methane-air mixture has been modelled considering a lumped two-steps mechanism :

$$CH_4 + 3/2\ O_2 \leftrightarrow CO + 2\ H_2O$$
$$CO + 1/2\ O_2 \leftrightarrow CO_2$$

The "finite rate/eddy dissipation" model has been used [3].

Modified kinetic coefficients, derived by genetic algorithms, have been used [4]. NOx post-processing considers exclusively the thermal NOx formation mechanism.

RESULTS AND DISCUSSION

Figures 2 and 3 show the predicted CO emission trends compared with the experimental data. CO experimental trends are satisfactorily reproduced; but the model underestimates the absolute values (within an order of magnitude).

In the experimental system the cold air coming from the secondary stream has a stronger quenching effect on the CO burnout than in the simulations. This can be related to an uncorrected treatment of the interface between the two different streams by the software and to other reasons discussed in section 4.0.

Fig. 2 compares predictions and experimental CO emissions (dry @ 15% O_2 versus different lengths of the first stage flame cup, L). In this test only air without swirl component is introduced through the second stage. It is possible to observe that the longer residence time (longer first stage flame cup lengths) allows an almost complete CO burnout. At the same time a longer residence time of the gases in a high temperature zone increases the NOx values.

Fig. 3 shows the comparison between predicted and experimental CO emissions at different adiabatic temperatures (dry @ 15% O_2). In this test a methane-air mixture of different compositions (to reach the different adiabatic temperatures desired), is introduced through the second stage. Differently from the previous case, the secondary air flow has a swirl component. At low adiabatic temperatures (corresponding to low fuel quantity introduced through the second stage) there is an under-estimation of the CO values (similarly to what observed in the previous case). On the other hand, for higher adiabatic temperatures CO emissions are overestimated, even if the model predicts correctly the overall trend. It can be observed that for these conditions the contribution of the second stage to the CO

production becomes relevant.

Fig. 4 shows the comparison between predicted and experimental NOx emissions at different adiabatic temperatures (dry @ 15% O_2). Also in this case the trend is correctly reproduced but there is a general under-estimation of the values. At low adiabatic temperature there might be a non-negligible contribution of other NOx formation mechanisms, other than the thermal one, which are not considered by the model. Modeling the second stage as perfectly premixed contributes to the overall NOx underestimation as this does allow to take in account the presence of hot spots due to the non-perfect mixing (as observed in the experimental tests) that causes relevant NOx emissions.

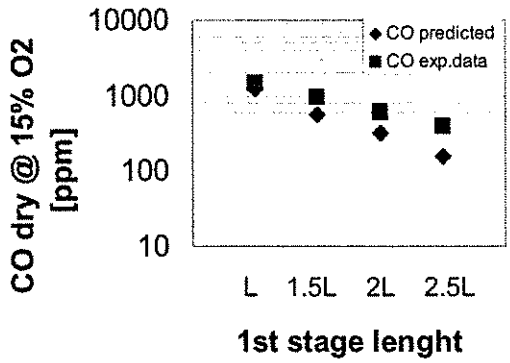

Figure 2. CO emissions versus the flame cup length of the first stage.

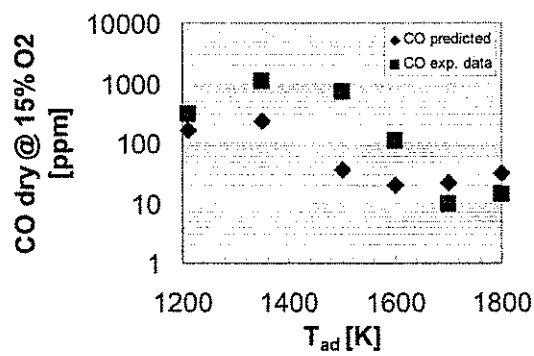

Figure 3. CO emissions versus adiabatic temperature.

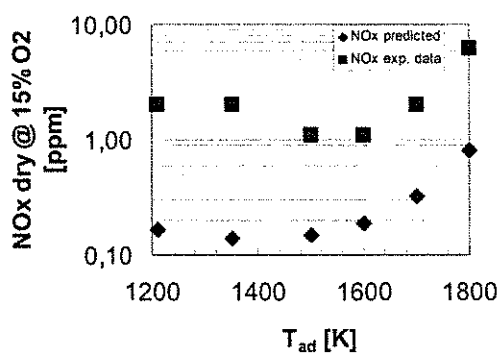

Figure 4. NOx emissions versus adiabatic temperature.

In general it is possible to observe a better agreement between simulations and

experimental data when the swirl component of the secondary flow is present. The high temperature zone shows a shorter length and is extended in the radial direction, indicating a faster mixing, enhanced by the secondary swirling flow.

CONCLUSIONS AND RECCOMENDATIONS FOR FURTHER WORK

A simplified model for the simulation of the combustion process in a lean-lean premixed staged combustor has been considered. The model results have been compared (at atmospheric pressure conditions) with experimental data. CO and NOx emissions have been investigated for different temperatures, lengths of the first stage of the combustion chamber and presence (or absence) of swirl in the second stage. The results confirm that larger residence times have a positive effect on CO burnout, but increase NOx production.

The simplified approach allows good predictions of CO and NOx trends, but a perfect agreement between data and predictions still lacks. In particular quench effect of cold air, and CO production at low adiabatic temperature are underestimated. In the second stage, constant velocity and stoichiometry profiles have been considered, which do not account for the non-ideal profiles established in reality.

Especially regarding NOx emissions values, there is an overall under-estimation of one order of magnitude, that can be caused by the fact that some NOx formation mechanisms are neglected and the formation of hot spots in the second stage is not taken into account.

As concern the kinetics of the combustion process, a simplified 2-step reaction model has been used in this work. In order to improve the performance of the model, envisaged alternatives are:

- to include in Fluent a more detailed kinetic model via a user-defined subroutine implementation;
- to use a statistical approach with the partially premixed model available in Fluent 6 or other user-defined PDF methods;,
- to use a fuel mixture (and relative kinetic model) with a composition similar to that of the natural gas used in the experimental tests (nowadays all simulations consider only the combustion of the methane-air mixture).

Further experimental data are also required in order to set up a complete picture of the system's behaviour. Including these improvements in the model would allow a better agreement with the experimental data to be gained.

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