1 Introduction

Modern gas turbine combustion chambers, such as the advanced sequential combustor of the GT24/26 ALSTOM engines, have little in common with the combustion system used for the first industrial gas turbine at Neuchatel in Switzerland, built in 1939. The thermal combustion intensity has increased from $10MW/m^3$ up to levels of more than $200MW/m^3$. At the same time, the requirement to meet ever increasing environmental standards has brought emission levels down from more than $500ppm$ NOx to below $25ppm$. This dramatic increase in combustion efficiency and decrease in pollutant emission has become possible by the development of lean-premix combustion technology. Here, low emissions are achieved by perfecting the premixing of the fuel-air mixture and by operating at low flame temperatures, i.e. fuel-lean conditions. However, the desire to operate gas turbines in low emission mode over the full engine operating range often brings the combustion process also dangerously close to lean extinction. As a consequence, the combustion at part-load levels is often accompanied by thermo-acoustic instabilities which may deteriorate the combustion process or even reduce the combustor life.

The optimization of the mixing process on the one hand, and the understanding and suppression of thermo-acoustic instabilities on the other hand, have thus become two major drivers for combustor technology and development. In this lecture I will address both these issues from a modeling perspective and explain how CFD tools have been used recently for both, a better understanding of the fundamental processes at hand, as well as supporting the combustor design process.

2 Prediction of flow & mixing in a gas turbine burner

2.1 Introduction

Good mixing as prerequisite for low emission values

Generally, the emission levels increase exponentially for an increase in flame temperature, and this behaviour is shown schematically in the upper part of figure 4.

This implies that any departure from perfect mixing leads directly to an increase in emission behaviour. This is demonstrated in the lower part of figure 4 via a simple thought model: Assuming that the fuel-air mixture is spread according to a Gaussian distribution around a mean fuel concentration $c = 0.05$ (i.e., fuel lean conditions), one can compute an integral emission based on the concentration field with Gaussian spread. The resulting emission behaviour follows again an exponential increase in emission levels if plotted against the variance of the Gaussian distribution of the fuel-air mixture. The lower part of figure 4 indicates that for a Gaussian width of 30% of the mean concentration (which can be considered as realistic for technical mixer devices), the expected relative emission behaviour is increased by a factor of 10. This is an indication for the potential in emission decrease if the fuel-air mixing is perfected, but it also shows the importance to fundamentally understand the mixing process in order to predict the mixing quality with sufficiently high accuracy.

Outline of chapter

We present analytical and numerical modeling work which is aimed at capturing the mixing process in a swirl-stabilized burner in the necessary detail to support the burner development process. In a first step, an analytical model of the mixing process based on inviscid flow is formulated. In a second step, both standard steady-state and advanced unsteady numerical tools are investigated in their capability to capture the flow and mixing fields. We address here the following questions: Are steady numerical turbulence simulations able to capture flow and mixing correctly? What is gained if the unsteady flow is explicitly computed? Finally, can we quantify the effect of unsteady flow on fuel-air mixing? To this end, the large-scale coherent flow structures which are present in this flow are analyzed in detail and an attempt is made to quantify their effect on the mixing process.

2.2 Analytical model of the burner flow & mixing

The Alstom EV burner

A perspective view of the Alstom EV burner is seen in figure 5. Like most modern gas turbine burners, it incorporates swirl. The flow enters the swirler through the two inlet slots on the cone shell in circumferential direction. This builds up a swirl flow inside the cone body. Towards the exit, the swirl number increases, the flow breaks down and forms a recirculation zone near the burner exit, and this recirculation zone is used to stabilize the premix flame at the dump.

The gaseous fuel is injected via several holes which are distributed along the swirler slots. The strong swirl inside the burner cone leads to very rapid mixing such that the fuel-air mixture is very homogeneous at the burner exit.

The burner flow model

An approximation of the flow field inside the conical swirl generator can be derived analytically from the inviscid Bernoulli equation. This formulation is used here to gain a basic understanding of the mixing process inside the burner. A generic sketch of the burner is shown in figure 6. The flow enters the cone via two (or more) slots on the sides of the burner in circumferential direction. The number of slots, their size, the angle of the cone and the exit diameter determines the resulting velocity distribution.

The Bernoulli equation with constant pressure reads

$$p_t = p + \frac{\rho}{2}(u^2 + v^2 + w^2) = \text{const.} \tag{1}$$

Neglecting the radial velocity, the differentiation with respect to the radius yields:

$$\frac{\partial p_t}{\partial r} = \frac{\partial p}{\partial r} + \rho \cdot u \frac{\partial u}{\partial r} + \rho \cdot w \frac{\partial w}{\partial r} = 0 \tag{2}$$

Introducing the radial equilibrium of pressure and centrifugal forces results in the following relation between the axial and circumferential velocity, depending only on the radius:

$$u \frac{\partial u}{\partial r} + w \frac{\partial w}{\partial r} + \frac{w^2}{r} = 0. \tag{3}$$