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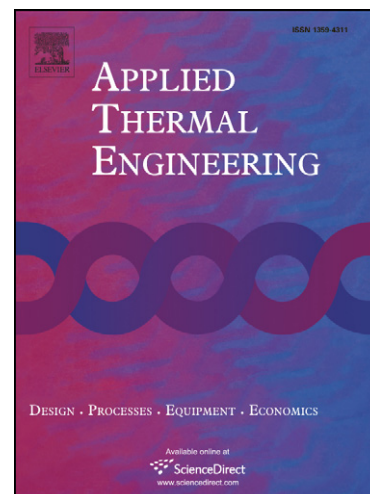
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LOW COMPUTATIONAL COST CFD ANALYSIS OF THERMOACOUSTIC OSCILLATIONS

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ABSTRACT

The numerical analysis of thermoacoustic oscillation phenomena by means of time-dependent CFD simulations usually requires a great computational effort, which may not be reasonable in industrial design. On the other hand, CFD tools provide the only approach that includes all the physical and chemical aspects involved in the thermoacoustic coupling between flame heat release and the acoustic modes of the burner/combustion chamber system. This paper presents some guidelines to reduce the computational effort required to perform a CFD analysis of the thermoacoustic oscillations with commercial codes. These guidelines are organized in a procedure that can be followed to analyze thermoacoustic coupling conditions that actually lead to unstable oscillations or are identified as potentially critical in the design phase. This procedure is also illustrated by an example of application, the partially-premixed flame type burner of a real 10MW industrial boiler which shows noisy pressure fluctuations at a low frequency.

Keywords: thermoacoustic oscillations, combustion instabilities, industrial burners, CFD analysis

1. INTRODUCTION

Combustion chamber designers often have to deal with unexpected problems such as flashback, quenching and oscillations. In particular, combustion instabilities can seriously affect the safety of

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combustor operation, since the self-sustained oscillations arising from the thermoacoustic coupling between flame heat release and the acoustic modes of the burner/combustion chamber system might even lead to the destruction of the combustor itself.

Unstable thermoacoustic oscillations and their control by passive or active means have been widely studied in the literature in the last years [1-6], because these phenomena are recurrent in gas turbine combustors when lean premixed flames are used to reduce NO_x emissions. Nevertheless, thermoacoustic coupling can affect the stability of diffusion or partially-premixed type flames as well.

A wide variety of tools have been proposed and applied to study the behaviour of unstable thermoacoustic oscillations:

- simple analytical models with few lumped parameters [7-9]. These models are used to describe the growth of the unstable oscillations at a very rough level without taking into account the spatial distribution of the acoustic modes and of the flame front;
- numerical models with a network of acoustic elements characterized by transfer matrices [10-15]. These models use radical assumptions on flow and combustion in order to simulate the flow with its fluid dynamic losses and to obtain the frequencies and the growth rates of the acoustic modes of interest (the complexity of which may vary from one-dimensional longitudinal modes up to fully three-dimensional modes). The presence of the flame is modelled by reproducing its effects on pressure and velocity fields through a flame transfer function. Both linear and non-linear dynamics can be used to analyze the coupling between the heat release at the flame front and the pressure fluctuations of the acoustic modes;
- transient CFD simulations of the reacting flow [16-19]. Different levels of detail in the analysis of the thermoacoustic coupling can be achieved by making appropriate choices about grid refinement and the physical models describing turbulence, combustion kinetics (reaction rates) and their interaction.

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It is apparent that only a transient CFD approach is potentially able to take into account all the chemical and physical aspects involved in the coupling between flame heat release and acoustic modes. However, this approach can be extremely expensive in terms of computational effort when

- grids are made of a high number of cells (up to millions of cells);
- computational demanding physical models are used;
- very small time steps are required to capture the propagation of pressure waves or the oscillations of high frequency acoustic modes;
- a large number of time steps is required to observe the onset of self-sustained unstable oscillations.

In this paper the attention is focused on low computational cost CFD tools, and the aim is to provide some guidelines to reduce as much as possible the computational effort required to perform a CFD analysis of the thermoacoustic oscillations with commercial codes. The steps of such an analysis are also discussed and can be summarized in the following procedure:

- 1) a non-reacting (“cold”) flow acoustic analysis of the burner/combustion chamber system is performed to identify the acoustic modes that are potentially critical for the thermoacoustic coupling or the modes corresponding to the most significant frequencies in an experimental spectrum, if the latter is available;
- 2) the computational domain is simplified and reduced by taking advantage of the symmetry of the system and of the symmetry and the nodes of the acoustic modes considered;
- 3) a stationary CFD simulation of the reacting flow is performed on the reduced computational domain to evaluate the time distribution of the heat release and check whether it is in phase with pressure oscillations;
- 4) the same CFD simulation is switched to transient mode, applying at start time an appropriate acoustic perturbation at a boundary to excite the considered acoustic mode and verify its stability.

The ways of performing these steps with the least computational effort are discussed for an example application, the partially-premixed-flame burner of a real 10MW industrial boiler which shows noisy pressure fluctuations at low frequency (32Hz).

The paper is organized as follows. In the next section a short background is given about unstable thermoacoustic phenomena, with particular reference to Rayleigh's stability criterion and the description of flame transfer function as the time delay between the moment in which fuel is injected and the moment in which it reaches the flame front and releases its thermal energy through oxidation. Then, the features of the burner/combustion chamber system used as example of application are introduced in Section 3. Finally, each of the four steps of the procedure outlined above are discussed in separate sections, in which all the details about the guidelines to reduce the computational effort of the CFD simulations are given.

2. BACKGROUND

Two categories of combustion oscillations exist: forced oscillations and self-sustained oscillations. Forced (or resonant) oscillations occur as a response of the burner/combustion chamber system to an external driver that generates a harmonic signal in resonance with one of the acoustic modes of the system. They are not the subject of this paper and will not be discussed further. On the contrary, self-sustained oscillations do not need an external source and may arise from infinitesimally small initial perturbations of the heat release at the flame front. Under stable operation these perturbations result in a white combustion noise, whereas the spectra of the pressure signals produced by combustion instabilities feature single frequencies that can be easily identified because of their significant amplitudes and correspond to the unstable acoustic modes.

Self-sustained oscillations are the outcome of a variety of mechanisms that are often specific to the features of burner configuration (see, e.g., [20] about industrial burner design and [21] for a

review of a number of burner applications). In fact, the perturbations in heat release can be generated by variations in fuel/air mixing, variations in the supply of fuel and/or air, vortex shedding, and other causes. In some applications more than one mechanism can contribute at the same time to the growth of unstable oscillations, and/or different mechanism may intervene with different strength at different operating conditions. Whatever the mechanism, self-sustained oscillations are always originated by a feedback between the variation of the heat release at the flame front and the acoustic modes of the burner/combustion chamber system (hence they are called “thermoacoustic” oscillations).

Lord Rayleigh [22] stated a fundamental necessary condition, the well-known Rayleigh criterion, for the growth of thermoacoustic oscillations. If, for sake of simplicity, only the time dependence of the oscillations is considered, and $q'(t)$ denotes the fluctuating heat release while $p'(t)$ denotes the fluctuating pressure inside the combustion chamber, Rayleigh observed that the thermoacoustic oscillations are likely to grow if the variations of $q'(t)$ and $p'(t)$ are in phase. On the contrary, the oscillations are certainly dampened if $q'(t)$ and $p'(t)$ are out of phase. Putnam [20] extended this elementary observation into the following integral condition

$$\int_0^T q'(t)p'(t)dt > 0 \quad (1)$$

so that single frequency oscillations may be sustained if the phase shift is smaller than 90° (a quarter of the acoustic period T). This is still a necessary condition, however, because the surplus of energy supplied by the unsteady heat release may be overcome by the additional energy dissipated by damping processes even when the instability condition is satisfied. In fact, even the growth in the amplitude of the unstable oscillations eventually comes to an end (the so-called “limit cycle”) because the damping processes are strengthened by the fluctuations themselves.

Rayleigh criterion has been nevertheless the reference point for all the literature about unstable thermoacoustic oscillations. The phase shift between $p'(t)$ and $q'(t)$ is obtained considering the

effects on the flame heat release induced by variations of the pressure field in the combustor (flame frequency response function). This relationship is usually approximated in a very simple way by the time lag between $p'(t)$ and $q'(t)$. Chemical kinetics, fluid dynamic convection and mixing contribute to the total lag, but in highly turbulent combustors the convection time plays a dominant role, so in elementary cases the time lag τ can be roughly estimated as the distance between the point of fuel injection and the flame front divided by the average flow velocity. Accordingly, pressure and heat release oscillations are nearly in phase if

$$\tau/T = \tau f = n - c (\pm 0.25) \quad n = 1, 2, 3, \dots \quad (2)$$

where c is a constant that depends on burner configuration (it is clear that maximum driving occurs when $p'(t)$ and $q'(t)$ are perfectly in phase, that is τ/T is exactly equal to $n-c$). For example, in diffusion or partially premixed flame type burners a negative pressure fluctuation in the region downstream the fuel nozzle causes a momentary increase in the fuel flow. This higher amount of fuel reaches the flame front with a time delay τ that is characteristic of the burner/combustion chamber system. If this surplus of fuel produces an immediate increase in the heat release, thermoacoustic oscillations will be most likely when one of the successive peaks of the pressure fluctuation is in phase with this increase in the heat release. This occurs when

$$\tau/T = \tau f = 1/2 (\pm 0.25), 3/2 (\pm 0.25), 5/2 (\pm 0.25) \dots \quad (3)$$

that is in this case $c=1/2$. In premixed flame type burners fuel flow through the injectors is choked, so air flow oscillations determine the amount of fuel that is convected to the flame front and the constant c depends on the upstream acoustic boundary condition (see [23,24]).

In practical applications, however, the space dependence of the oscillations of pressure field and heat release strongly affects combustion stability and cannot be neglected. As a consequence, Rayleigh integral condition has to be evaluated all over the combustor volume:

$$\int_V \int_0^T q'(x, y, z, t) p'(x, y, z, t) dV dt > 0 \quad (4)$$

The space dependence of pressure field oscillations is obtained from the characteristics of the acoustic modes involved, whereas the space dependence of the heat release oscillations is strictly related to the shape of the flame front and to the portions of the injected fuel that are oxidated in the different regions of it. The latter aspect in particular complicates the flame frequency response function and results in a distribution of time lags rather than a single value τ , as highlighted by many authors (see, e.g., [10,11]).

It appears that CFD tools are the most suitable ones for the detection of potentially unstable operating conditions, because they are able to describe with a sufficient level of detail the phenomena that characterize the reacting flow and to determine, as a result, the space dependence of pressure and heat release oscillations. In other words, CFD simulations are essential for the analysis of both fully three-dimensional acoustic modes of the burner/combustion chamber system and the trajectories along which the injected fuel is delivered to the flame front, in order to determine the time distribution of its heat release. This is the basic motivation for the procedure discussed in the following sections.

3. THE EXAMPLE OF APPLICATION

The burner used as example of application in this paper is shown in Figure 1. A single fuel supply axial line, which do not appear in the figure, feeds two rows of injection holes after a spider leg bifurcation. The portion of the duct that continues after the bifurcation leads to a row of ten small radial holes that are close to burner axis. The small amount of fuel that is injected through these holes (pilot holes) burns in a diffusion flame surrounded by the air jets coming from the holed plate upstream. This high-temperature flame is simply used for burner ignition. Most of the fuel is injected through the other row of ten large radial holes (main holes) that are arranged at the end of the ten spider leg ducts departing from the axial fuel supply line. The fuel jets exiting these holes

have the time to partially mix with the main air flow, which comes from the external annular bypassing section of the burner, before reaching the flame front. This is why the main flame cannot be considered a diffusion type flame, but is rather a partially-premixed type flame.

This prototype burner, which was designed to work in a 10MW industrial boiler, showed an unstable behaviour when it operated nearly at full load (around 9.3MW) in the atmospheric test facility shown in Figure 2. This facility is made of a 10m long cylindrical combustion chamber (internal diameter 1.2m) and a 4m long chimney (internal diameter 0.8m) that discharges the flue gases to the ambient. A throttling device, which is used to set combustion chamber pressure during the tests, divides the combustion chamber in two parts at an axial position located just before chimney axis. The facility is located in the research and development laboratory of the manufacturer.

An experimental spectrum of the pressure signal registered at the burner end of the combustion chamber with a differential pressure transducer (full scale 10000Pa, sensitivity 8mV/V, accuracy 1% FSO, sampling frequency 10kHz) is shown in Figure 3. It is apparent that a limit cycle resulting from self-sustained combustion oscillations is present and involves an acoustic mode of the burner/combustion chamber system at a frequency around 32Hz. The other acoustic modes produce much smaller peaks, among which one at a frequency around 10 Hz is clearly visible. Thus, in the following sections the attention will be focused on the unstable acoustic mode at 32Hz.

4. COLD FLOW ANALYSIS OF THE ACOUSTIC MODES

The purpose of a modal analysis of the burner/combustion chamber system is to obtain the frequencies and the spatial distribution of the acoustic modes of interest, i.e. the ones that are potentially critical for the thermoacoustic coupling or those that are already known to be unstable from experimental data. If this analysis is performed using CFD tools it requires a transient

simulation of the acoustic domain and an elaboration of the pressure signals obtained at various locations through Fast Fourier Transforms (FFTs). Since this simulation focuses on acoustic aspects only, i.e. on pressure wave propagation at local sound speed, several expedients may help reduce the computational effort:

- it is not necessary to consider the chemical transformations due to the reacting flow and an appropriate temperature field (which is to be verified later after the stationary CFD simulation of the reacting flow) can be imposed instead. Temperature can be even considered as constant (e.g. ambient temperature) all over the domain if this is not expected to alter the spatial distribution of the modes significantly (the obtained frequencies must then be scaled according to the chosen value of the constant temperature);
- if flow velocities are expected to be much lower than local sound speed, then a quiet domain can be considered as the initial condition of the transient simulation instead of the actual pressure and velocity fields;
- non-essential geometry can be simplified (e.g., ducts with irregular shapes can be rectified, but features causing relevant acoustic reflections, such as sharp bends, cannot be removed);
- the choice of the turbulence model is not really significant in this kind of analysis, so low computational cost unsteady Reynolds-Averaged Navier-Stokes (RANS) models can be used;
- the intervals for space and time discretization can be chosen as large as possible according to the maximum frequency of the modes of interest. As usual, the two intervals are linked by the stability considerations arising from the well-known criterion by Courant, Friedrichs, and Lewy [25] (the CFL number must be lower than one, i.e., the characteristic dimension of grid cells should be higher than the product of local sound speed and the time step of the transient simulation, so that pressure waves cannot travel more than one cell in one time step).

Choosing a large time step is convenient if the transient simulation has to be prolonged in time in order to have a good resolution in the frequency domain;

- in this context, an accuracy of the same order of the truncation error in the implicit scheme used for time discretization can be accepted. As a consequence, the algorithms achieving a high accuracy in time integration (such as PISO) are not convenient, and non-iterative time advancement schemes (e.g., [26]) can be used instead if the time step is properly limited.

Evidently, particular attention should be paid to the boundaries chosen for the acoustic domain, because both the position of domain boundaries and the type of condition imposed (e.g., pressure node, velocity node, non-reflecting surface) are critical for the results of the modal analysis. Moreover, these limits are likely to be the same limits of the computational domain in the CFD simulation to be performed in the successive steps of the thermoacoustic analysis.

In the example of application, the boundaries of the acoustic domain are the burner end of the combustion chamber (velocity node) and chimney outlet section (pressure node). At time $t=0$ s of the transient simulation the domain is filled with air in quiet at ambient pressure and temperature (101325Pa and 300K). A time step of 0.1ms should be sufficient to depict the modes up to 1000Hz (ten sampling time steps per period) and then the characteristic dimension of the hexahedral cells of the block structured grid shown in Figure 4 is 100mm. A pressure impulse is given at chimney outlet section at time $t=0$ s to perturb the acoustic domain and the simulation is carried out until $t=2$ s, so that the frequency resolution and the lowest frequency value are 0.5Hz, while pressure signals are registered at the various locations along combustion chamber and chimney axes shown in Figure 4. The CPU time required to perform the simulation is about one hour on an old 1.8GHz Pentium IV single-core processor desktop computer.

The first four peaks of the spectra obtained by these virtual sensors are at 5Hz, 17.5Hz, 30.5Hz and 42Hz, and the shapes of the corresponding modes are shown in Figure 5. The first mode (5Hz) is the fundamental (quarter wave) mode on the sum of lengths of the first part of the combustion

chamber and the chimney. The second mode (17.5Hz) can be interpreted either as the second harmonic (three quarter wave) mode on the same length or as the fundamental (half wave) mode on the length of the whole combustion chamber. The third mode (30.5Hz) is the second harmonic (one wave) mode on the length of the whole combustion chamber and is partially disturbed by the fourth mode (42Hz), which is the fundamental (half wave) mode on the length of the chimney only. After the frequencies of the four peaks are scaled by means of the different sound speeds in the cold flow and in the reacting flow (the mean temperature of the reacting flow is about 1250K according to the experimental data), it is apparent that the second mode is the one involved in the self-sustained thermoacoustic oscillations. In the same way, the other peak highlighted in the experimental spectrum (10Hz) corresponds to the first mode obtained.

5. SIMPLIFYING THE COMPUTATIONAL DOMAIN

The computational domain of the reacting flow CFD simulations can be greatly simplified and/or cut in order to reduce the computational effort, according to the geometry of the burner/combustion chamber system and the information obtained about the shape of the acoustic modes of interest. As mentioned in the previous sections, the description of the details that do not alter the overall acoustic behaviour can be dropped. Domain symmetry and periodicity are of great help, as usual, but even non symmetrical or non periodical domains can be simplified by substituting the non symmetrical or non periodical elements with acoustically equivalent geometry in the reduced domain. These simplifications must naturally comply with the shape of the acoustic mode of interest as well. For example, a domain reduction due to symmetry cannot be considered if the acoustic mode of interest is azimuthal, and a domain reduction due to periodicity allows to reproduce azimuthal modes with the same periodicity only. On the other hand, the information about the shape of the acoustic modes is particularly valuable because of the location of the

stationary pressure nodes. In fact, the computational domain can be cut at these locations and adequate pressure conditions can be imposed at the new boundaries. The acoustic behaviour (i.e. mode frequency and shape) of the resulting domain is equivalent to the original as far as the mode of interest is concerned. However, the cut of the domain at pressure nodes must not interfere with the reacting zone (i.e. with the shape and position of the heat release source) in order to guarantee the same thermoacoustic coupling.

In the example of application, the symmetry of the test facility (Figure 2) and the shape of the acoustic mode of interest (longitudinal, in this case) would suggest to halve the computational domain along the plane containing the axes of the combustion chamber and of the chimney. However, a major reduction of the computational domain can be achieved if burner periodicity (36° , see Figure 1) is taken into account in spite of the chimney. The latter element can be substituted with ten equivalent chimneys having the same length and one tenth of the section to respect the periodicity. Another major reduction can be achieved if, within a tenth of the original domain, the symmetry along the plane containing the axis of one spider leg duct is taken into account as well. Finally, according to the shape of the mode involved in the thermoacoustic oscillations, the computational domain can be further reduced by cutting it at the section that corresponds to the pressure node (see Figure 5) at about 5m from the burner end of the combustion chamber. The resulting domain is the one used in the reacting flow stationary and unsteady simulations described in the following sections.

6. CFD STATIONARY SIMULATIONS AND THE HEAT RELEASE ANALYSIS

The stationary CFD simulation of the reacting flow should be prepared keeping in mind that the same domain, computational grid, boundary conditions and physical models will be used in the successive unsteady simulation, so the strategy to reduce the computational effort should be

planned accordingly. The basic idea behind this strategy is that in the simulation of combustion oscillations acoustic phenomena are by far more important than the fluid dynamic details of the flow field [27-29]. This leads to a reduction of the number of cells in the computational grid (they can be limited down to a range from 50,000 to 100,000), which in turn allow the unsteady simulation to be performed with commercial CFD codes in reasonable CPU times. Grid roughness also impose the use of low computational cost RANS turbulence models instead of the more demanding Large Eddy Simulation (LES) ones. Combustion chemistry can be taken into account with one or few reactions, the kinetics of which is controlled using combined Arrhenius/eddy dissipation models.

The purpose of the stationary simulation is not only to obtain the initial condition for the transient simulation. In fact, the results obtained can be exploited for a preliminary flame heat release analysis that is of great importance in the design phase to identify the frequency ranges in which unstable thermoacoustic oscillations are likely to arise. Clouds of virtual particles without mass are “injected” from the exit sections of the fuel supply system and their trajectories are evaluated at a very low computational cost using the velocity field obtained from the stationary simulation. The relationship between the reaction rate and the time elapsed from injection is derived along the trajectory of each particle to determine statistically the time at which the injected fuel releases its thermal energy across the flame front. This Lagrangian time profile $Q(t)$ of the heat release associated with the fuel injected at a generic instant is then compared to the sinusoidal time evolution of an acoustic mode at a given frequency (a cosine function in time). The time shift between the two functions depends on the acoustic boundary conditions of the specific burner configuration (see the discussion about the constant c in Eq. 2). If most of the heat release takes place in proximity of a maximum of the cosine function in time (i.e. most of the area bounded by the Lagrangian time profile of the heat release $Q(t)$ is concentrated close to this maximum), then the necessary condition expressed by Rayleigh criterion is satisfied for that frequency. On the contrary,

thermoacoustic oscillations are dampened at that frequency if most of the heat release takes place in proximity of a minimum of the cosine function in time, or are very unlikely to occur if the heat release spans across several maxima and minima of the cosine function in time (because either the heat release is distributed almost uniformly over a relatively long time or mode frequency is relatively high).

In the example of application, the reduced domain defined in the previous section is decomposed in about 50,000 unstructured cells (see some details on the computational grid in Figure 6), the maximum cell dimension being around 50mm. The grid is refined in the regions where high gradients of thermal, chemical, or flow quantities are expected, and cell thickness at walls is kept within the limits recommended for wall functions. Pressure inlet boundary conditions are imposed at both air and fuel inlets, and a pressure outlet boundary condition is imposed at the section corresponding to the pressure node of the unstable acoustic mode (domain outlet). The shear stress transport (SST) $k-\omega$ turbulence model is used, as it combines the advantages of the $k-\omega$ and the $k-\epsilon$ approaches, and the kinetics of the one step methane-oxygen reaction is determined combining the Arrhenius and eddy dissipation models. Since combustion chamber optical thickness is greater than one, radiation from the high-temperature flame is well approximated using the P-1 model, and the local absorption coefficient is evaluated using the weighted-sum-of-gray-gases model. The CPU time required to perform the steady-state reacting flow simulation is about three hours on the same old 1.8GHz Pentium IV single-core processor desktop computer.

The numerical results of the stationary simulation are validated by means of the available direct or indirect experimental measures. In particular, the choices about the kinetic and radiation models are confirmed by the mean temperature of flue gases and the optical estimation of the length of the flame front, while the chosen turbulence model reproduces correctly the pressure losses across the burner. The resulting temperature field is shown in Figure 7. Two high temperature zones clearly indicate the two flames that are present within the combustion chamber, as expected from the

burner design presented in Section 3. The stoichiometric pilot flame is the one with the highest temperature and is located in front of the central holed plate, whereas the leaner main flame surrounds the methane jets exiting the spider leg ducts. It is apparent that the latter flame is “free” (i.e. not attached to a flame holder) and then particularly prone to oscillate when the flow field is perturbed.

The analysis of flame heat release allows to explain why the second mode identified in Section 4 is involved in the unstable thermoacoustic oscillations, the frequency of which should be around 36Hz according to the mean temperature of the reacting flow. The Lagrangian time profile $Q(t)$ of the heat released by the fuel injected through the main radial holes at a generic instant is obtained with the method described above and is compared in Figure 8 with the sinusoidal pressure fluctuation $p'(t)$ of this acoustic mode. The delay between the first pressure peak, set at $t=0s$, and the start time of the heat release profile is equal to half of the period of the acoustic oscillation ($T=0.02778s$ for $f=36Hz$), because the phases of injected fuel mass flow rate and pressure fluctuations are opposite in this particular burner configuration (see Section 2). In other words, the first pressure peak causes a peak in the injected fuel mass flow rate when the pressure fluctuation $p'(t)$ is at its minimum. According to Figure 8, most of the fuel injected at that moment releases its thermal energy near the next (second) peak of $p'(t)$, and this results in a driving of the thermoacoustic oscillations. Figure 9 shows that this does not happen with the first mode identified in Section 4 (the frequency of which should be around 10Hz). In fact, the largest part of the area bounded by the heat release time profile (that is unaltered) is far from the second pressure peak, time delay being still half of the period corresponding to 10Hz ($T=0.1s$ for $f=10Hz$). If the frequency domain is systematically scanned, the critical range for potential unstable modes is found to lie between 30Hz to 70Hz.

7. CFD TRANSIENT SIMULATIONS

In the last step of the procedure for the analysis of thermoacoustic oscillations, the stationary CFD simulation of the reacting flow is switched to transient mode and the onset of self-sustained fluctuations is helped by applying at start time an appropriate perturbation of the initial pressure field. This operation has to be set up carefully according to the shape of the mode to be excited, but may show itself unnecessary if the driving of the thermoacoustic oscillations is strong enough. It is not easy to establish in advance whether the pressure perturbation could be helpful or unnecessary: however, a particularly strong driving of the thermoacoustic oscillations may be diagnosed by a difficult convergence of the stationary simulation. In order to reduce the computational effort, the time step of the simulation is still chosen as the largest allowed by the propagation speed of pressure waves (sound speed) and the sampling of the pressure fluctuations associated with the frequency of acoustic mode of interest. The simulation is usually prolonged until the growth of the thermoacoustic oscillation comes to an end and the limit cycle is reached.

In the example of application, the time step is fixed at 0.1ms according to the dimension of grid cells (the same grid of the stationary simulation is used). This choice allows computational time to be saved while reaching convergence within each time step, although the estimated sound speed, about 620m/s, and the average cell size in the combustion chamber, 50mm, would suggest a maximum time step of about 0.07ms to keep the CFL number lower than one. The limit cycle is reached after nine periods of the unstable oscillations without the need of applying an impulsive perturbation at the computational domain outlet. The CPU time required to simulate 0.3s of transient operation is about 36 hours on the same old 1.8GHz Pentium IV single-core processor desktop computer.

The frequency of these oscillations is equal to about 38Hz. Figure 10 shows the fluctuations of:

- the pressure at the burner end of the combustion chamber,

- the instantaneous thermal power (obtained by integrating the methane reaction rate all over the volume),
- the fuel mass flow rate through the holes of the spider leg ducts.

It appears that the phases of pressure and fuel mass flow rate oscillations are opposite, as supposed in the previous section, whereas pressure and thermal power oscillations are nearly in phase, as predicted by the analysis of flame heat release. Figure 11 illustrates a picture of the temperature field during the limit cycle. A sequence of high and low temperature zones downstream the main flame clearly shows the effects of the heat release fluctuations. The amplitude of the limit cycle (4600Pa) is noticeably higher than the experimental one (1400Pa) and the difference is due to the cut of the computational domain at the pressure node. In fact, if on one hand the shape of the acoustic mode that is considered is altered in the reduced domain, on the other hand its acoustic damping is lowered for two main reasons:

- energy dissipation due to viscosity in the regions excluded from the reduced domain is not taken into account;
- the energy dissipated by the lower frequency stable modes, which cannot be represented in the reduced domain, is instead redistributed among the acoustic modes of the reduced domain (note that the mode that is considered becomes the fundamental one of the reduced domain, so it receives the largest quota of this energy).

Accordingly, a relevant portion of the overall acoustic energy associated with the heat release, that otherwise would be absorbed, remains available for the unstable fluctuations.

8. CONCLUSIONS

Reacting flow CFD simulations are the most suitable tools for the comprehension of the mechanisms underlying combustion oscillations, and an insight of the driving mechanism is of

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course fundamental for the control of undesirable or even dangerous self-sustained oscillations due to the thermoacoustic coupling. These techniques are often set aside because of the high computational effort required by a detailed description of the flow field, which also results in the necessary adoption of very small time steps in unsteady simulations.

The procedure discussed in this paper suggests the use of CFD calculations for their capability on acoustic aspects rather than on fluid dynamic details. The guidelines given to lower the computational effort by restraining domain boundaries or by increasing cell dimensions and time steps are nevertheless able to reproduce the physics of the thermoacoustic phenomena. A flame heat release analysis based on the results of the stationary simulation only can also be used in the design phase to identify the frequency ranges that are potentially critical for the arising of unstable thermoacoustic oscillations.

REFERENCES

- [1] P. Berenbrink and S. Hoffmann, Suppression of dynamic combustion instabilities by passive and active means, 2000, ASME Paper No. 2000-GT-0079.
- [2] G. Kelsall, C. Troger, Prediction and control of combustion instabilities in industrial gas turbines, *Applied Thermal Engineering* 24 (2004) 1571–1582.
- [3] M.P. Auer, C. Gebauer, K.G. Mösl, C. Hirsch, T. Sattelmayer, Active instability control: feedback of combustion instabilities on the injection of gaseous fuel, ASME paper GT2004-53055
- [4] T. Scarinci, C. Freeman, I. Day, Passive control of combustion instability in a low emissions aeroderivative gas turbine, ASME paper GT2004-53767.
- [5] S. Kokanović, G. Guidati, S. Torchalla, B. Schuermans, Active combustion control system for reduction of NO_x and pulsation levels in gas turbines, ASME paper GT2006-90895.
- [6] A.S. Morgans, S.R. Stow, Model-based control of combustion instabilities in annular combustors, *Combustion and Flame* 150 (2007) 380–399.
- [7] S. Hubbard, A.P. Dowling, Acoustic resonances of an industrial gas turbine combustion system, *ASME J. Eng. Gas Turbines Power* 123 (2001) 766–773.
- [8] C.A. Arana, B. Sekar, M.A. Mawid, C.B. Graves, Determination of thermoacoustic response in a demonstrator gas turbine engine, *ASME J. Eng. Gas Turbines Power* 124 (2002) 46–57
- [9] Z.M. Ibrahim, F.A. Williams, S.G. Buckley, C.Z. Twardochleb, An acoustic energy method for estimating the onset of acoustic instabilities in premixed gas-turbine combustors, *ASME J. Eng. Gas Turbines Power* 130 (2008) 051506-1-16.
- [10] D.E. Hobson, J.E. Fackrell, G.Hewitt, Combustion instabilities in industrial gas turbines - Measurements on operating plant and thermoacoustic modelling, *ASME J. Eng. Gas Turbines Power* 122 (2000) 420-428.
- [11] P. Flohr, O.C. Paschereit, B. van Roon, B. Schuermans, Using CFD for time-delay modeling of premix flames, 2001, ASME Paper No. 2001-GT-0376.
- [12] C. Pankiewicz, T. Sattelmayer, Time domain simulation of combustion instabilities in annular combustors, *ASME J. Eng. Gas Turbines Power* 125 (2003) 677-683.
- [13] M.A. Macquisten, M. Whiteman, A.J. Moran, Application of low order thermo-acoustic network to DLE staged combustor, ASME paper GT2004-54161.
- [14] V. Bellucci, B. Schuermans, D. Nowak, P. Flohr, C.O. Paschereit, Thermoacoustic modeling of a gas turbine combustor equipped with acoustic dampers, *ASME J. Turbomachinery* 127 (2005) 372-379.
- [15] V. Bellucci, D. Nowak, W. Geng, C. Steinbach, On the use of thermoacoustic analysis for robust burner design, *ASME J. Eng. Gas Turbines Power* 130 (2008) 031506-1-6
- [16] S.M. Cannon, V. Adumitroaie, C.E. Smith, 3D LES modeling of combustion dynamics in lean premixed combustors, 2001, ASME Paper No. 2001-GT-0375
- [17] S.J. Brookes, R.S. Cant, I.D.J. Dupere, A.P. Dowling, Computational modeling of self-excited combustion instabilities, *ASME J. Eng. Gas Turbines Power* 123 (2001) 322-326.
- [18] G. Lartigue, U. Meier, C. Bérat, Experimental and numerical investigation of self-excited combustion oscillations in a scaled gas turbine combustor, *Applied Thermal Engineering* 24 (2004) 1583–1592.
- [19] S. Roux, G. Lartigue, T. Poinso, U. Meier, C. Bérat, Studies of mean and unsteady flow in a swirled combustor using experiments, acoustic analysis, and large eddy simulations, *Combustion and Flame* 141 (2005) 40–54.
- [20] A.A. Putnam, *Combustion-Driven Oscillations in Industry*, American Elsevier Publishers, 1971.

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- [21] S.M. Candel, Combustion instabilities coupled by pressure waves and their active control, in: The Twenty Fourth (International) Symposium on Combustion, 1992.
- [22] J.W.S. Rayleigh, The explanation of certain acoustical phenomena, Royal Institute Proceedings VIII (1878) 536-542.
- [23] T. Lieuwen, B.T. Zinn, Theoretical investigation of combustion instability mechanisms in lean pre-mixed gas turbines, 1998, AIAA paper AIAA-98-0641.
- [24] T. Lieuwen, H. Torres, C. Johnson, B.T. Zinn, A mechanism of combustion instability in lean premixed gas turbine combustors, ASME J. Eng. Gas Turbines Power 123 (2001) 182-189.
- [25] R. Courant, E. Isaacson, M. Rees, On the solution of non-linear hyperbolic differential equations by finite differences, Commun. Pure Appl. Math 5 (1952) 243-249.
- [26] H.M. Glaz, J.B. Bell, P. Colella, An analysis of the fractional-step method, Journal of Computational Physics, 108 (1993) 51-58.
- [27] S. Tiribuzi, CFD modelling of thermoacoustic oscillations inside an atmospheric test rig generated by a DLN burner, 2004, ASME paper GT2004-53738.
- [28] S. Tiribuzi, Very rough grid approach for CFD modelling of thermoacoustic oscillations inside an annular premixed combustor, ASME paper GT2006-90055.
- [29] S. Tiribuzi, CFD simulation of noise in gas turbine combustors by means of turbulence re fluctuation method, ASME paper GT2008-50241.

FIGURE CAPTIONS

Figure 1 – The burner of the 10MW industrial boiler used as example of application

Figure 2 – The combustion chamber of the test facility.

Figure 3 – Spectrum of an experimental pressure signal measured at the burner end of the combustion chamber.

Figure 4 – The hexahedral unstructured grid for the cold flow analysis of the acoustic modes of the test facility and the location of the virtual pressure sensors used to reconstruct mode shape.

Figure 5 – The shape of the first four acoustic modes obtained with the cold flow analysis of the test facility (line segments show the relative amplitude of local pressure fluctuation).

Figure 6 – Details of the unstructured grid of the reduced domain used in the stationary and transient reacting flow simulations (about 50,000 cells).

Figure 7 – The temperature field [K] obtained from the stationary CFD simulation.

Figure 8 – Timing of the pressure fluctuation due to the acoustic mode at 36Hz and of the heat released by the fuel injected through the main radial holes (if the heat is released in the gray zones, the necessary condition stated by Rayleigh criterion is satisfied and thermoacoustic oscillations at this frequency are likely to grow).

Figure 9 – Timing of the pressure fluctuation due to the acoustic mode at 10Hz and of the heat released by the fuel injected through the main radial holes (if the heat is released outside the gray zones, the necessary condition stated by Rayleigh criterion is not satisfied and thermoacoustic oscillations at this frequency are certainly dampened).

Figure 10 – Time history of a) static pressure at the burner end of the combustion chamber (relative to atmospheric pressure), b) instantaneous thermal power released within the combustion chamber and c) fuel mass flow rate through the main radial holes during the transient CFD simulation.

Figure 11 – The temperature field [K] obtained from the transient CFD simulation during the limit cycle ($t=0.3s$).

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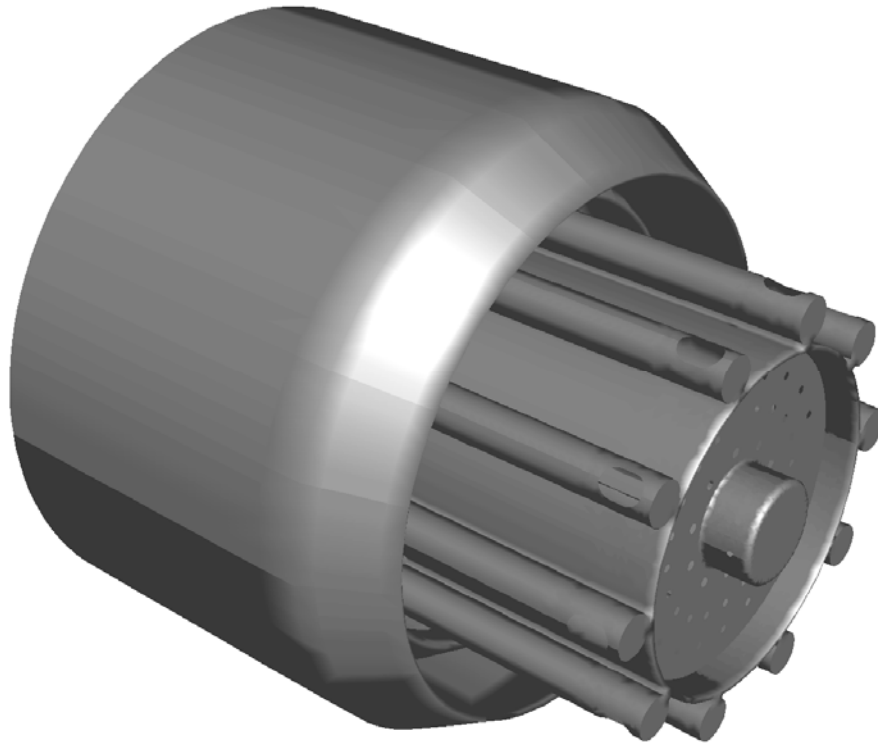


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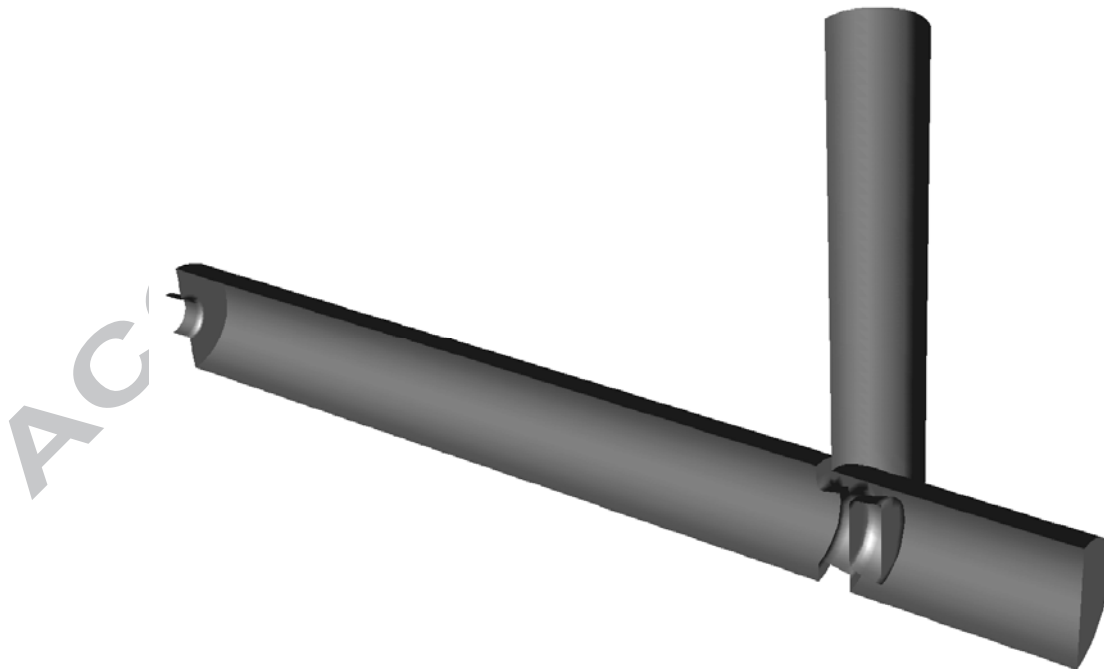


Figure 2 – The combustion chamber of the test facility.

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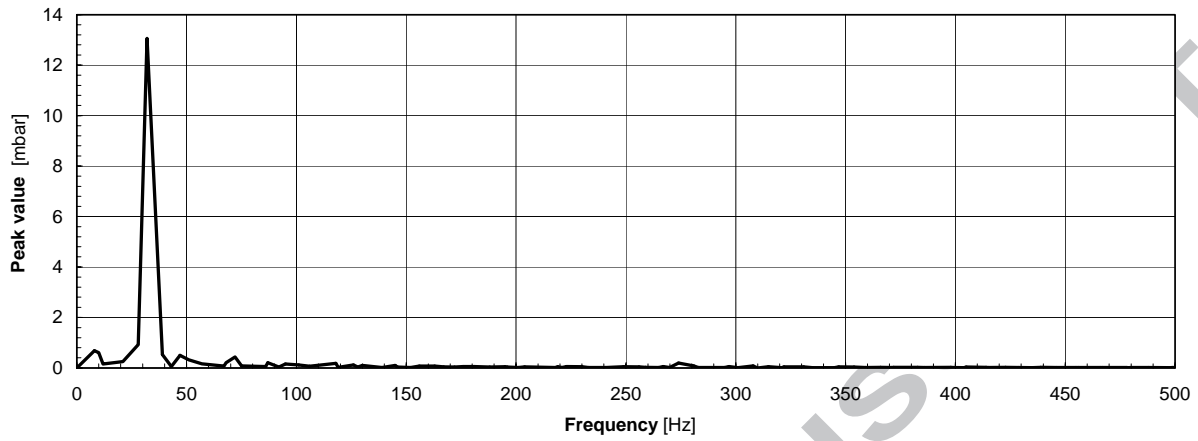


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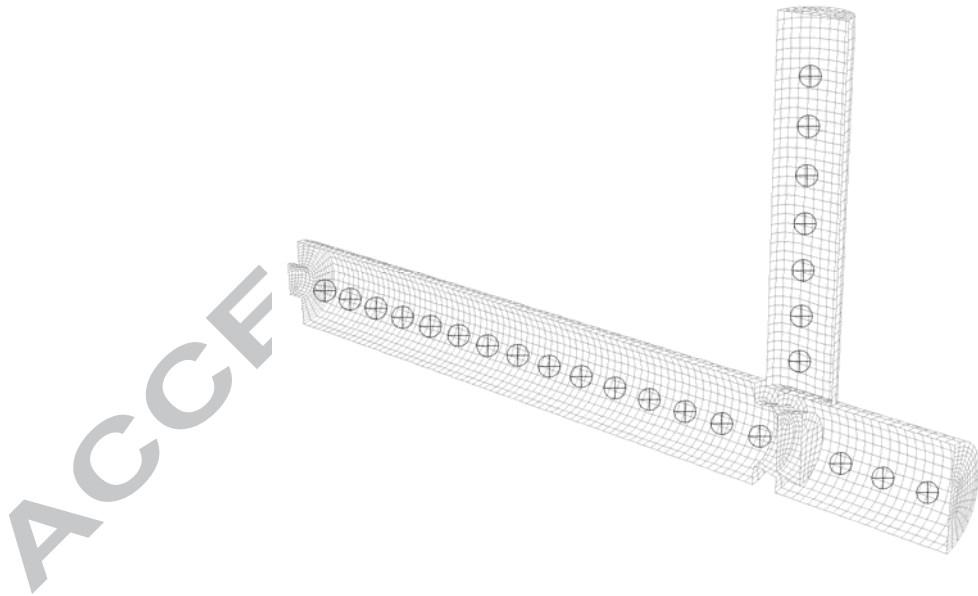


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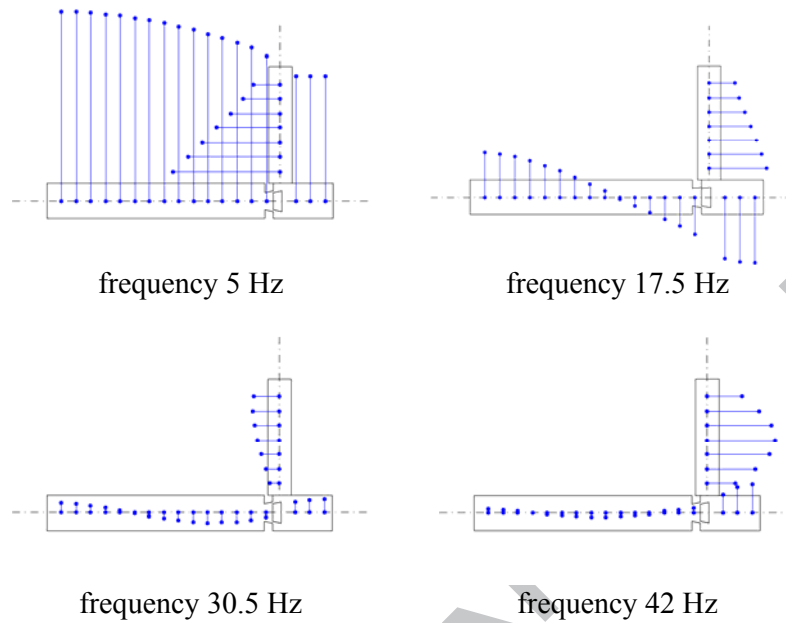


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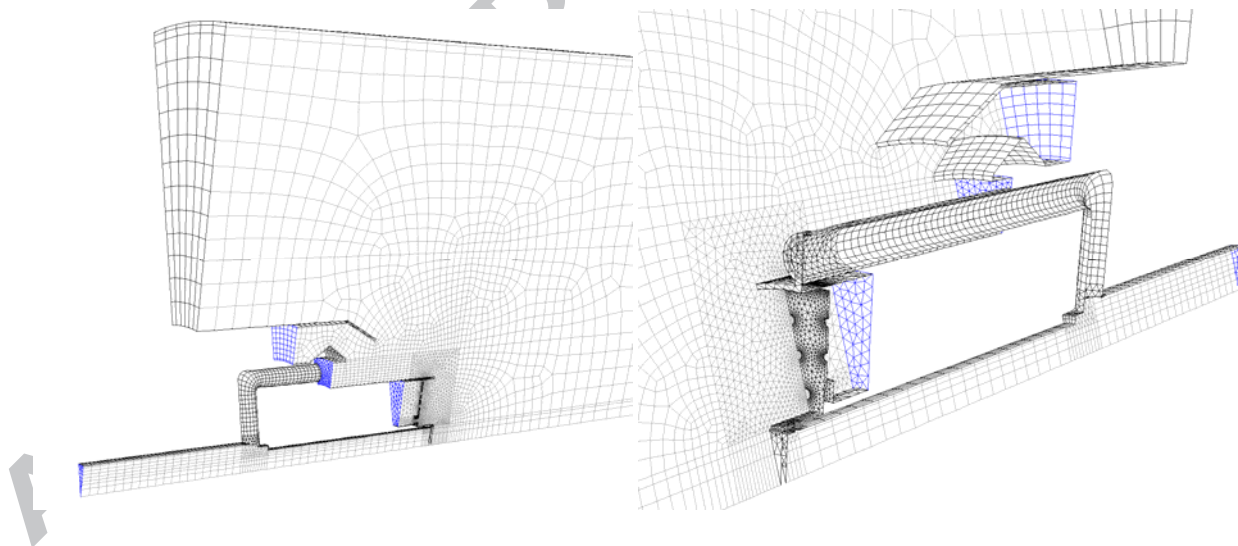


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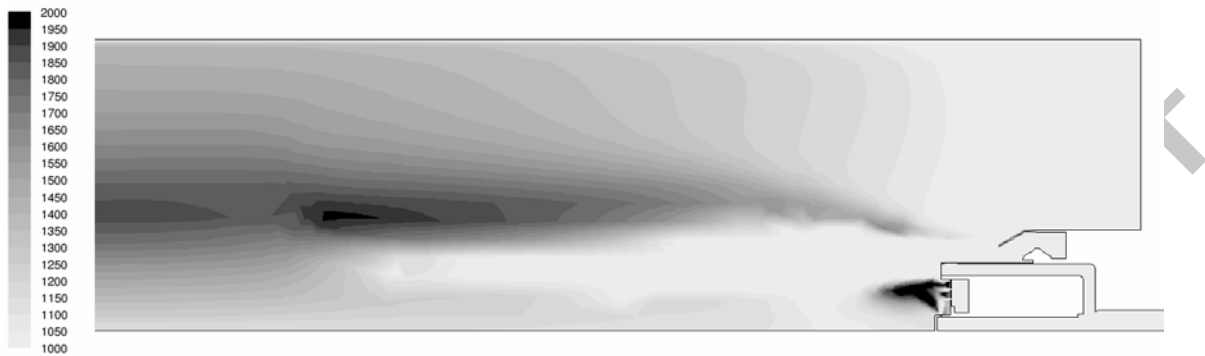


Figure 7 – The temperature field [K] obtained from the stationary CFD simulation.

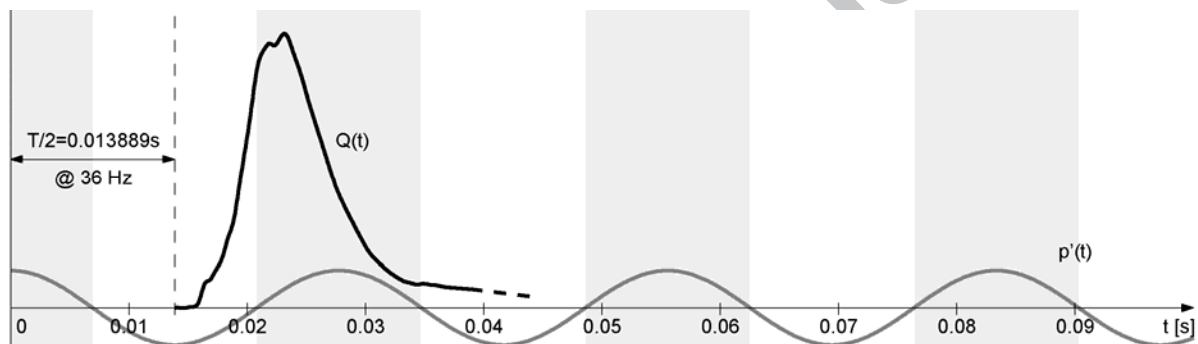


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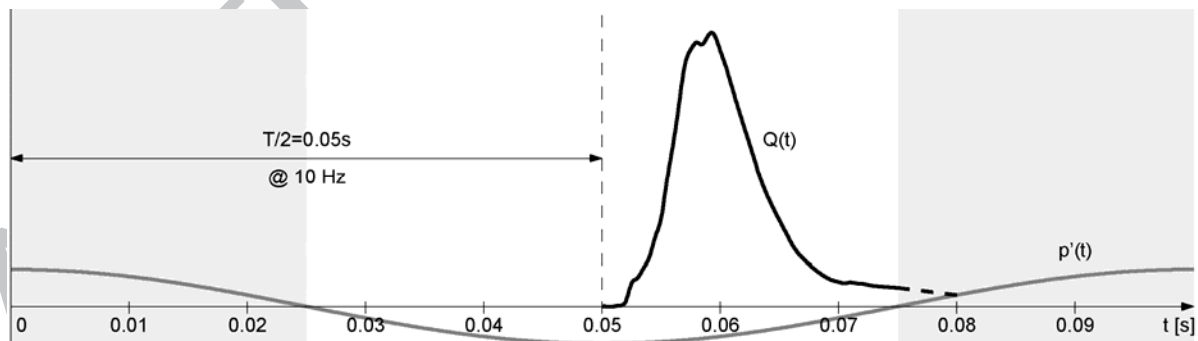


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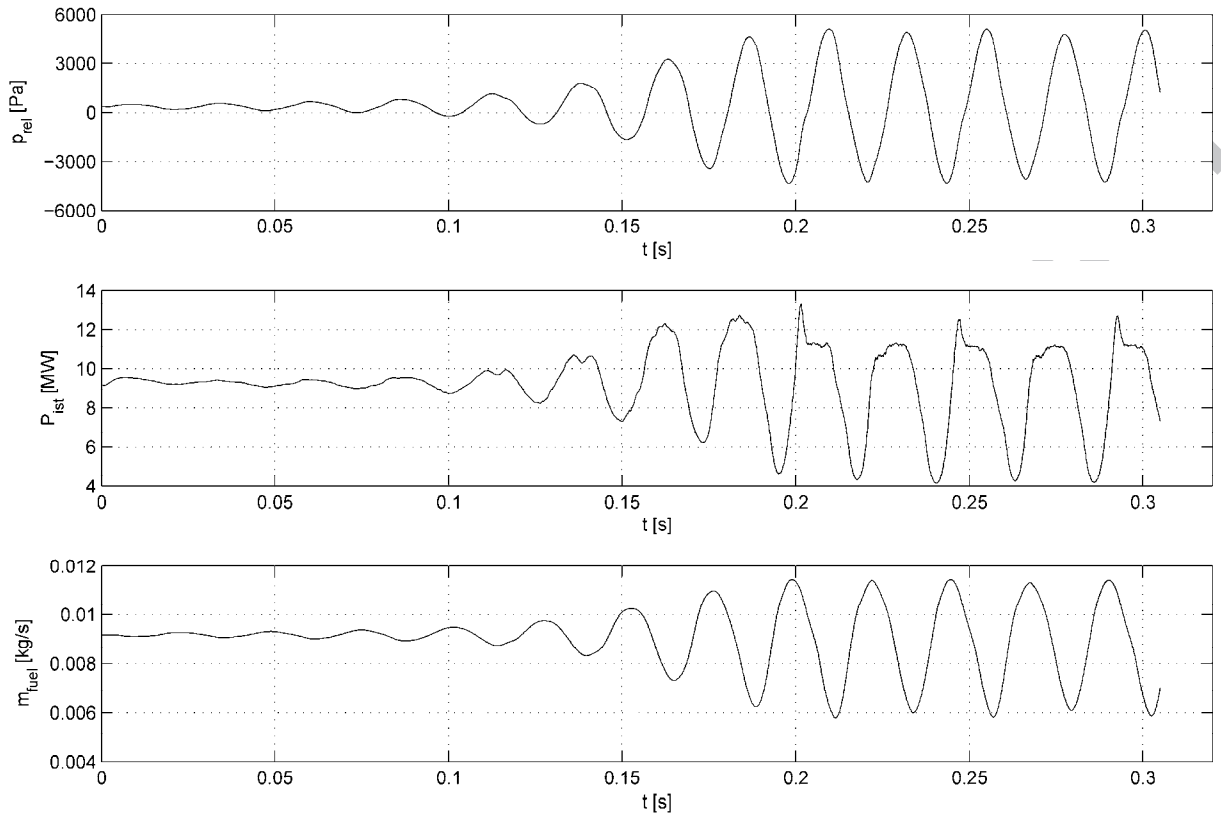


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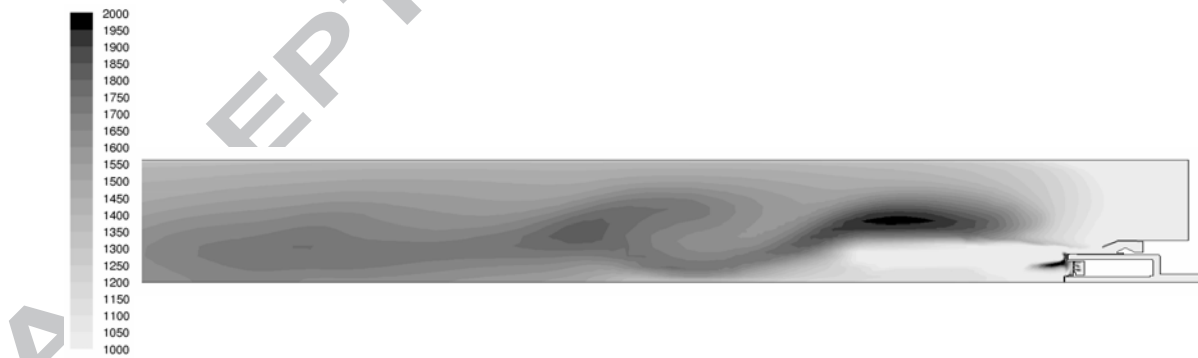


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