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Vibration prediction in combustion chambers by coupling finite elements and large eddy simulations

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Abstract

To decrease NO_x emissions from combustion systems, lean premixed combustion is used. A disadvantage is the higher sensitivity to combustion instabilities, leading to increased sound pressure levels in the combustor and resulting in an increased excitation of the surrounding structure: the liner. This causes fatigue, which limits the life time of the combustor. This paper presents a method to calculate these liner vibrations. The time-dependent pressures on the liner are calculated using large eddy simulation (LES) for both steady combustion and combustion with a pulsated fuel flow. These pressures are interpolated on a finite element grid and used in a transient analysis with a finite element model of the liner structure. The calculated vibrations agree well with experiments made on a 500 kW test rig.

1. Introduction

Air pollution has become an important subject in the last few decades. An important component of the pollution is nitrogen oxide or NO_x, which is, amongst other things, formed during combustion of fossil fuels in gas turbines. To decrease this formation, lean combustion is applied in gas turbines, using an excess of air to combust the fuel. This leads to lower NO_x emissions, but also to increased acoustic pressure levels in the combustion chamber [1]. Due to the high temperatures in the combustion chamber, the surrounding structure, the so-called liner, has to be thin and is therefore flexible. The acoustic pressure perturbations induce vibration of this structure, leading to fatigue problems.

Previous work [2] discussed a method to predict liner vibrations based upon an acousto-elastic finite element model in which the flame as source of sound is modeled as an acoustic volume source. This model has full coupling between structural and acoustic behavior. From CFD calculations only a temperature field is used to obtain correct acoustic properties for the combustion chamber. The current paper presents an alternative approach. The excitation pressures on the liner are calculated using large eddy simulation (LES) with hard

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walls. These pressures are then used as loads in a structural finite element model to calculate the liner vibrations.

The main difference between the two methods lies in the coupling between combustion, acoustics and vibration. The finite element based approach can describe two-way interaction between acoustic and structural behavior, but it has only one-way coupling between combustion and acoustics (the flame as acoustic source is not influenced by the acoustics). In the approach presented in this paper, the behavior of the flame is fully coupled with the acoustic behavior, but there is only one-way coupling with the structure (because no information from the structural model is used in the LES).

Compressible LES methods such as the one used here bring an important improvement over the use of Reynolds averaged codes for fluid structure interaction in combustion chambers because they solve explicitly for the unsteady pressure field (which is not the case in classical Reynolds averaged methods [3] where additional equations for the RMS pressure are needed and where phases must be reconstructed using ad hoc models). It should be noted though that the LES signal is spatially filtered and therefore the pressure signal obtained by LES probably misses high-frequency components. Nevertheless, being able to explicitly solve for the unsteady pressure on walls during a combustion oscillation is a major improvement compared to the state of the art offered by Reynolds averaged codes.

The paper will first present the experimental setup used, which is also the geometry used in all calculations. Subsequently, the modeling procedure is outlined. Experimental and numerical results are then compared for steady combustion as well as for combustion using a pulsated fuel flow.

2. Experiments

The experiments were performed on a test rig at the University of Twente, of which a cross-section is depicted in Fig. 1. The combustion chamber is a square tube of 150×150 mm with a length of approximately 1.8 m. It consists of three main sections. The left part is the combustion section, consisting of the plenum (1, the last chamber upstream of the burner), a generic burner (2, in which the air gets a swirling motion and fuel is injected) and the first part of the combustion chamber (3, where the combustion takes place in a partially premixed flame). The middle section is the structural section, in which the vibration of the liner can be measured using laser vibrometry through windows (4) in the pressure vessel. In the combustion section and the structural section a relatively thin structure, called the liner (5), is present within a much thicker pressure vessel (6). Cooling air flows between the liner and the pressure vessel. In the structural section, the liner consists of thicker end parts (4 mm) and a thinner part (7, 1.5 mm) in between. Structural vibrations are measured on this thinner part. Acoustic pressures in the combustion chamber are measured with pressure transducers p_2 and p_3. The sensor signals are amplified and then, together with the laser vibrometer signal, acquired using a SIGLAB data acquisition system at a sample frequency of 2.56 kHz. The most right part is the water cooler in which the cooling air is mixed with the combustion air and the hot gasses are cooled down by water injection to temperatures that the throttle valve can withstand. The combustion channel in the structural section and the water cooler are separated by a tube which separates the acoustics of the water cooler and the rest of the system to a certain degree, because acoustic waves hardly travel from the structural section through the tube (8) to the water cooler and back. The test rig is operated at a thermal power of 150 kW at a mean pressure of 1.5 bars. The air factor is 1.8, meaning there is 80% more air available than strictly needed for the combustion. For further information on the test rig reference is made to Refs. [2,4].

3. Modeling

The LES solver used, AVBP [5], simulates the fully compressible multi-species (variable heat capacities) Navier–Stokes equations on hybrid grids. Subgrid stresses are described by the classical Smagorinsky model [6]. A two-step chemical scheme is fitted for lean regimes on the GRI-Mech V3 reference [7] using genetic algorithms. The objective of the fit procedure is that the two mechanisms (two-step and GRI) must produce the same flame speeds and maximum temperatures for laminar premixed one-dimensional flames. The flame/turbulence interaction is modeled by the dynamic thickened flame (DTF) model [8] and allows handling both mixing and combustion (which are crucial in partially premixed flames). The numerical scheme uses

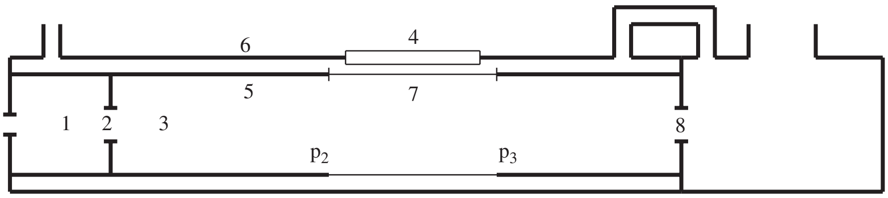

Fig. 1. Cross-section of the test rig, 1 = plenum, 2 = burner, 3 = combustion chamber, 4 = window, 5 = liner, 6 = pressure vessel, 7 = thin liner part, 8 = exit tube, p_2 and p_3 are pressure sensor locations.

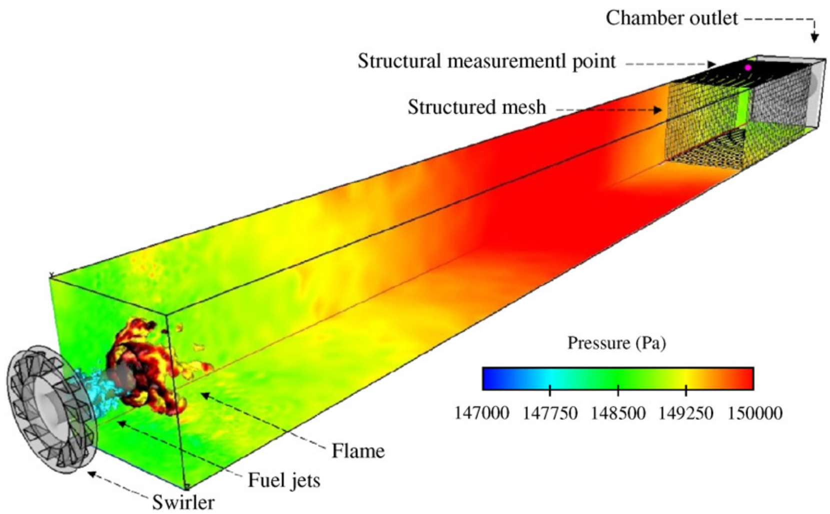

Fig. 2. Structural mesh, the dot denotes the approximate location of the structural measurements.

third-order spatial and third-order explicit time accuracy [9]. The LES computational domain includes all parts from the air supply chamber (upstream of the plenum) to the outlet flange. This is necessary to have the right acoustic impedance for the combustion chamber, to predict accurately the chamber acoustic modes and to minimize the uncertainties on boundary conditions. The boundary condition treatment is based on a multi-species extension [10] of the NSCBC method [11], for which the acoustic impedance can be controlled [12]. In all the simulations, the linear relaxation method [12] on the outlet boundary condition ensures that the acoustic waves of interest (over 100 Hz) are only reflected by the geometry at the downstream end of the combustion chamber (the flange) and not spuriously by the LES boundary condition on the exit tube. The exact acoustic boundary condition at this tube is unknown, because of the turbulence and water injection downstream of the tube. Nevertheless, it is expected to be fairly anechoic and therefore this boundary is made non-reflecting. At the upstream end (upstream of the plenum chamber) the boundary is made fully reflecting, because the flow is nearly choked there. Consequently the LES are performed in a 'stand-alone' mode and mimic the experiment in both terms of aerodynamic and acoustic conditions. The walls are handled using a dynamic logarithmic law-of-the-wall formulation for velocity and temperature: the thermal treatment mimics the convective losses due to the cooling channel with a conjugate approach, imposing a heat resistance and prescribing the cooling air temperature. Moreover, a simple gray gases model is employed to account for radiation. A good agreement with experimental evaluation of global heat losses is achieved [13]. Typical runs are performed on grids of the order of 2.7 millions tetrahedra on several massively parallel computer architectures (*SGI origin 3800*, *Compaq alpha server*, *Cray XD1*) with a very efficient speedup [14]. The objective of this paper is not to analyze the CFD results, a full description can be found in Ref. [13]. Even so, the instantaneous flame shape and pressure at the walls is shown in Fig. 2.

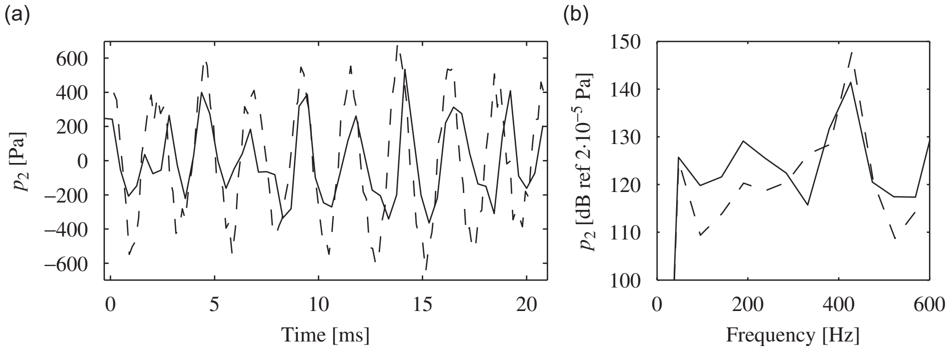

Fig. 3. Calculated and measured pressure for sensor p_2 during steady combustion as function of time (a) and frequency (b). —— experiment, - - calculation.

Fig. 2 also shows the structural finite element model, consisting of approximately 10×10 mm linear shell elements. It consists of only the thinnest part of the structural liner, having dimensions $150 \times 150 \times 400$ mm. The boundaries on both ends are assumed to be clamped, which is accurate for low frequencies [2]. The pressures on the wall calculated with LES are used as loads on the elements of the structural model. The LES computations are performed on a very fine grid (element sizes of the order of 1 mm) and with very small time steps (in the order of 2×10^{-7} s). Structural calculations can use a much coarser mesh and much larger time steps. The LES data are therefore remapped onto a coarser finite element mesh with equidistant nodes spaced 10 mm apart. A pressure load vector is generated of the LES data every 0.15 ms over a total time of 21 ms, giving 140 pressure loads. The very short total time is caused by the high computational costs of LES, which makes it unfeasible to perform calculations for much longer periods. Because the structural transient analysis used a fixed time step of 0.03 ms, the pressure loads were interpolated linearly between the 0.15 ms samples.

4. Results

Results for steady combustion are presented first. The measured and calculated pressure at the location of sensor p_2 (Fig. 1) is compared, as well as results for structural vibrations. Subsequently, results with a pulsated fuel flow are shown.

Fig. 3 shows the acoustic pressure at the position of sensor p_2 from LES and from the measurement. The sample rate for the measurement was only 2.56 kHz and therefore there is a limited amount of samples in the plotted dataset for the measurement. The pressure amplitude of the calculation and the measurement match well, indicating that the calculated pressures on the wall are reasonably accurate. Both show a strong harmonic component near 400 Hz (period 2.5 ms, see also the pressure spectrum), which is related to the second acoustic mode of the combustion chamber. The autospectrum is also shown, but because the physical time of the LES is very short, it has a very low-frequency resolution.

Fig. 4 shows the calculated and the measured vibration velocity at the laser vibrometer measurement point, which is located 50 mm above the lower thick part and 50 mm from the side of the liner (Fig. 2). The figure shows a good match between the calculated and the measured vibration amplitude. Furthermore, it shows that the structure responds mostly to the acoustic mode near 400 Hz, which was also seen in the calculated spectrum in Fig. 3.

The structural response during combustion with a fuel pulsation of 15% of the total fuel mass flow at 300 Hz is also calculated.[1] The time signal for pressure sensor p_2 is depicted in Fig. 5. The first part (about 3 ms) of the LES does not show a clear oscillation because the perturbation is imposed at the location where the fuel enters the swirl passage. This perturbation has to travel to the flame front before it causes a perturbation in the heat release rate and therefore a pressure perturbation [4]. After this delay, both the

[1]The measurements were made with 8.5% fuel flow pulsation and therefore these are multiplied by a factor of 1.76 to obtain a similar forced response. The autonomous response evidently becomes incorrect, but the experimental equipment did not allow a higher excitation level and the computationally very expensive calculations were already performed.

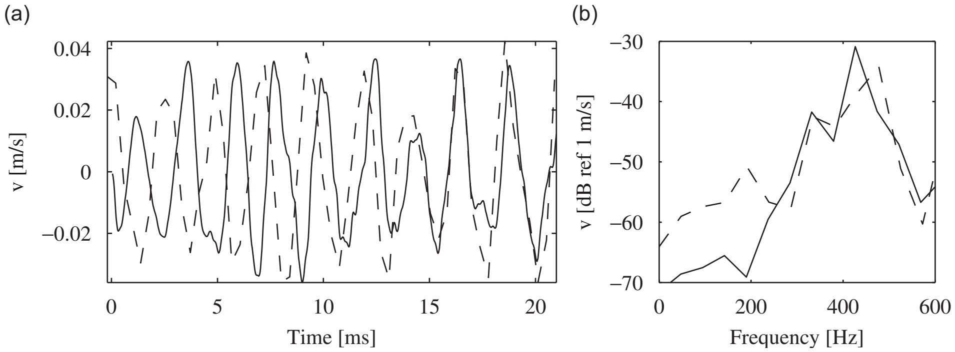

Fig. 4. Calculated and measured velocity for case 15.7 during steady combustion as function of time (a) and frequency (b). —— experiment, - - calculation.

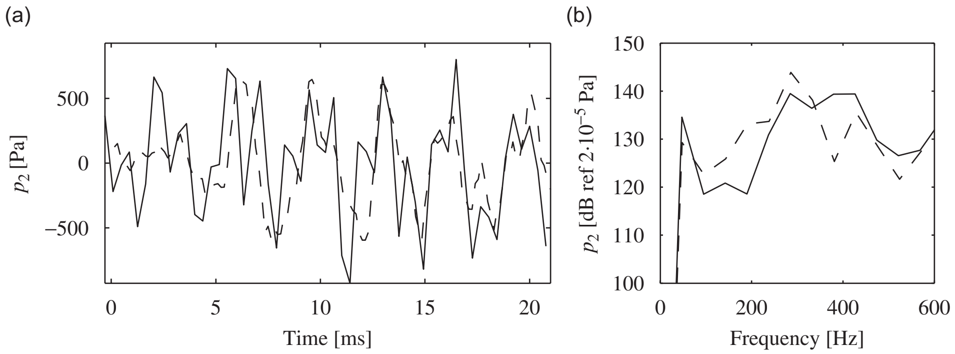

Fig. 5. Calculated and measured pressure for sensor p_2 during combustion with 300 Hz pulsating fuel flow as function of time (a) and frequency (b). —— experiment, - - calculation.

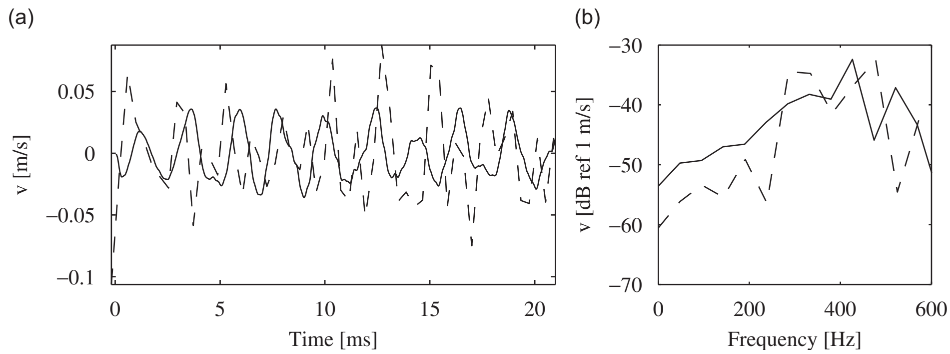

Fig. 6. Calculated and measured velocity for case 15.7 during combustion with 300 Hz pulsating fuel flow as function of time (a) and frequency (b). —— experiment, - - calculation.

measured and the calculated time signal contain a strong 300 Hz response. A second harmonic component is also visible, which is the same as the one found for steady combustion. It can be concluded that the predicted pressure response to the fuel perturbation is very similar to the measured one.

Fig. 6 shows the calculated and measured vibration velocity at the laser vibrometer measurement point during pulsated combustion. It can be seen that the calculated and measured amplitudes are close to each other, which corresponds with the good match at 300 Hz seen in Fig. 4. The LES based calculation starts with a low velocity level, which rapidly increases to a more or less steady vibration amplitude. This is because in the transient finite element calculations no displacement and velocity are used as initial conditions and therefore the vibration has to build up first. In the frequency domain, both measurement and calculation show, as can be expected, a strong increase in vibration level around 300 Hz.

5. Conclusions

A finite element model has been coupled to a LES in a one-way manner by using the pressures on the wall as function of time from LES as loads for a structural finite element model. Liner vibrations in a test rig have been calculated and the results match well with measurements made on the test rig. The method can therefore be used to predict liner vibrations and thus fatigue in combustion systems.

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