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FLUID STRUCTURE INTERACTION TO PREDICT LINER VIBRATIONS IN AN INDUSTRIAL COMBUSTION SYSTEM

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Abstract

To decrease NOx emissions from a combustion system lean premixed combustion in combination with an annular combustor is used. The disadvantage is that sound pressure levels in the combustion system become higher which excite the liner (the surrounding structure). This limits the life of the combustor, because it will fail earlier due to fatigue. This problem is studied in the European project DESIRE (Design and demonstration of highly reliable low NOx combustion systems for gas turbines).

A simplified model is studied first. This model consists of a rigid rectangular box which has a flexible plate on one of the sides. Sound is injected into the box using a tube coming from a rectangular box. A loudspeaker generates sound inside this box. A fully coupled finite element model has been made of the structure and the acoustic cavity. The results of the model are compared to the measurements performed on the actual setup. Both the structural mode shapes (measured using a laser vibrometer) and the transfer function from pressure to displacement are used for this purpose. The results match very well.

INTRODUCTION

To decrease NOx emissions from an industrial gas turbine lean premixed combustion (using a surplus of air) in combination with an annular combustor is used.[3] The leaner combustion has the side effect of higher sound pressure levels in the combustion system. On the other hand the annular liner (the surrounding plate-like structure around the combustion chamber) does not have a high stiffness and therefore the sound pressures severely excite the liner. This liner can vibrate excessively which limits the life of the combustor and the range of operability, because it will fail due to fatigue. There are two approaches to decrease this problem. The most common approach is to decrease the pressure levels in the combustor.[2] The other approach is to directly decrease the vibration level of the liner by, for instance, increasing the damping. This requires investigation of the fluid structure interaction between the liner and the combustion chamber.

The European DESIRE project is concerned with this interaction. The interaction will be studied in a complex test-rig using a 500 kW burner at 5 bar. It essentially consists of a square liner surrounded by a square pressure vessel. The combustion takes place inside the liner and therefore the exiting acoustic field is generated there. Between the pressure vessel and the liner flows cooling air through a shallow cavity. In this paper a more simple setup is discussed first, consisting of an acoustic cavity covered by a flexible plate. After discussion of the setup the finite element model will be explained. This is followed by a comparison of measured and calculated results.

EXPERIMENTAL SETUP

The experimental setup is shown in figure 1 and photos are given in figure 2. It consist of a stiff aluminum box covered by a flexible aluminum plate with a thickness of 1.1 mm. The plate is fixed using aluminum beams of thickness 5 mm, which are clamped with bolts. The part of the plate that can vibrate is 160x210 mm. The depth of the cavity behind the plate is 140 mm. The material properties are Young's modulus 70.5 GPa, Poisson's constant 0.3 and density 2700 kg/m^3. All bolts are tightened with a torque of 20 Nm. The properties of the air are density 1.22 kg/m^3 and speed of sound 343 m/s.

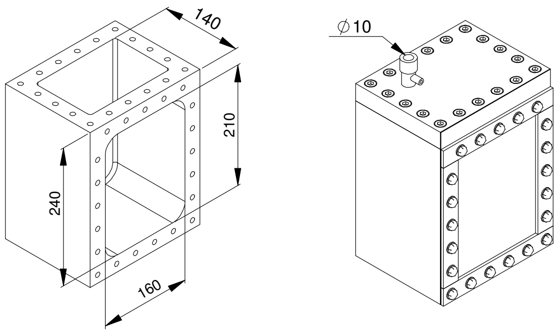
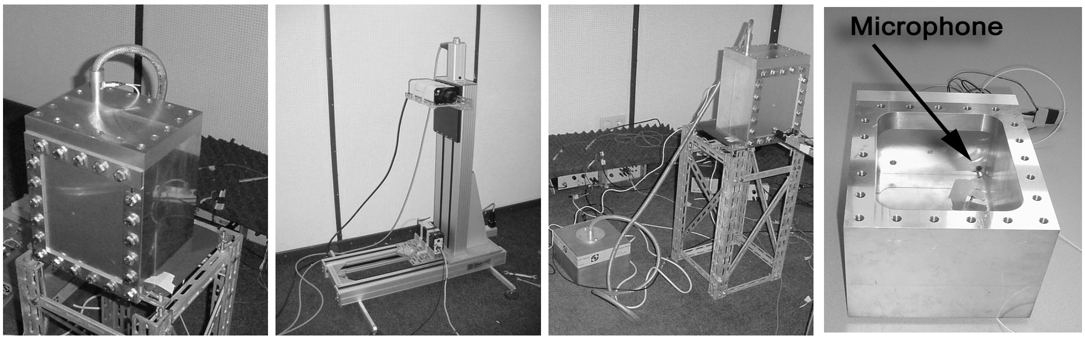


Figure 1: The test setup, left is the open box, right is the box covered by a flexible plate

A laser vibrometer (Polytec OFV-303 optics head combined with an OVD-01 velocity decoder) mounted on a traverse system is used to measure the vibration of the flexible plate (figure 2(b)). This allows for automated measurement of the vibration shapes. The pressure inside the box is measured using an electrec microphone which is mounted in the upper right corner in the back (figure 2(d)). No acoustic mode shape of the box has a nodal plane at this location and therefore the microphone should be able to detect all acoustic mode shapes. The signal is amplified using a custom made amplifier with an amplification factor of 100. The acoustic field is excited by a point source. To create this source a speaker is mounted in a box (figure 2(c)). A hose transfers the sound towards the box with the vibrating plate. Close to the point where the hose ends on the box a Kulite pressure pickup is mounted to measure the excitation pressure. The signals from the laser vibrometer and the microphones are digitized using a Siglab 20-24 data acquisition unit and postprocessed using Matlab. The speaker is driven by an amplifier which is connected to an output channel of Siglab.



(a) Box with plate (b) Laser scanner (c) Acoustic source (d) Position microphone

Figure 2: Some details of the test setup

NUMERICAL MODEL

The acoustic and structural eigenfrequencies are calculated using the finite element package ANSYS. The model is made from a combination of linear shell (SHELL63) and acoustic (FLUID30) elements. An unstructured mesh made of triangular and tetrahedral elements is used. The clamping is assumed to be perfect and therefore all translational and rotational degrees of freedom are suppressed on the clamped border of the plate. The acoustic and shell elements are coupled using special elements that include fluid structure interaction. These elements ensure that the acoustic velocity normal to the surface is the same as the structural normal velocity.

RESULTS

Acoustic and structural eigenfrequencies

The calculated and measured eigenfrequencies of the coupled system are listed in table 1. The accompanying mode shapes are depicted in figure 3. It can be seen that the first seven eigenmodes match quite well. The structural response due to the acoustic modes

number	mode	calculated	measured	difference
1	$(1,1)_s$	307.5 Hz	306.2 Hz	0.42 %
2	$(2,1)_s$	506.9 Hz	505.0 Hz	0.37 %
3	$(1,2)_s$	702.5 Hz	702.5 Hz	0.00 %
4	$(1,0,0)_a$	736.3 Hz	742.5 Hz	-0.84 %
5	$(3,1)_s$	850.2 Hz	851.3 Hz	-0.13 %
6	$(2,2)_s$	885.5 Hz	895.0 Hz	-1.07 %
7	$(0,1,0)_a$	1080.0 Hz	1086.2 Hz	-0.57 %

Table 1: Eigenfrequencies of the setup, s denotes structurally dominated, a denotes acoustically dominated

is also predicted well, which is an indication that the spatial prediction of the pressure field is also correct.

The first acoustic eigenfrequency can be approximated by taking the box as a rectangular cavity. The acoustic eigenfrequencies then become[1]

$$f_n(k,l,m) = \frac{c_0}{2}\sqrt{\left(\frac{k}{a}\right)^2 + \left(\frac{l}{b}\right)^2 + \left(\frac{m}{c}\right)^2} \tag{1}$$

In which c_0 is the speed of sound, k, l and m are the number of half waves in the three directions and a, b and c are the accompanying dimensions. Applying this formula gives an eigenfrequency of 714.6 Hz for the $(0,1,0)_a$ mode. The finite element model and measurements predict slightly higher frequencies which is due to the beam at the top (see also the mode shape in figure 6).

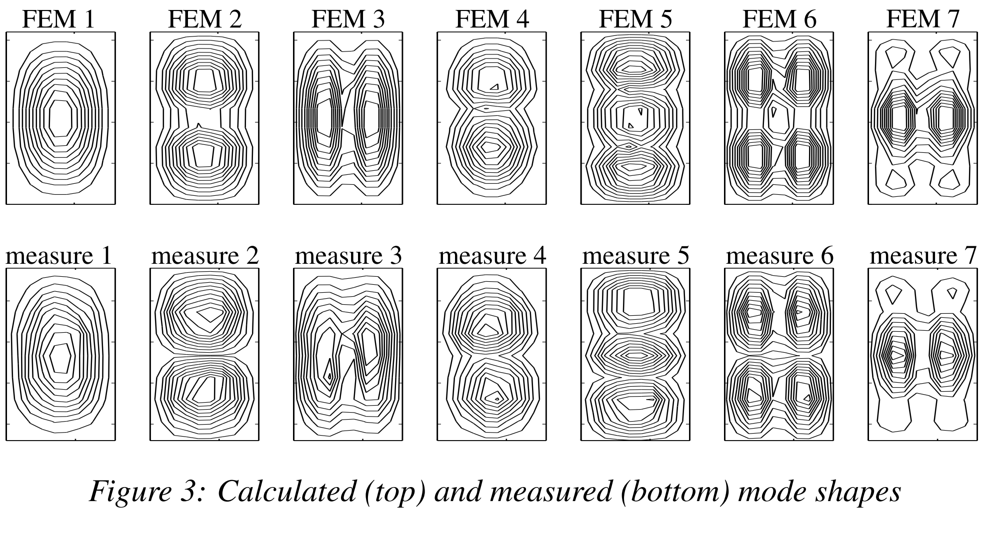


Figure 3: Calculated (top) and measured (bottom) mode shapes

Pressure signals

The sound pressure level at the end of the supply tube is depicted in figure 4. It can be observed that there are peaks at intervals of 62 Hz. These are the resonances of the supply tube. The tube has an open end on both sides and therefore it resonates at

$$f = \frac{n\,c_0}{2L} \tag{2}$$

In which n is an integer and L is the length of the tube. With the length of the tube (2.67 meter) this results in the peaks seen in the measurements. Figure 4 shows the transfer from an applied pressure at the beginning of the tube to the pressure at the location of the sensor. It was calculated using the Transfer Matrix Method[4] and shows the same resonances. The measured and calculated results cannot be compared directly, because the speaker produces an unknown pressure level at the inlet of the tube. It can also be seen that the acoustic excitation is between 95 and 115 dB, with the pressures becoming lower at higher frequencies. This is most likely due to the limited frequency range of the speaker used as sound source. The pressure produces enough excitation over the entire frequency range to measure structural vibrations on the plate using the laser vibrometer.

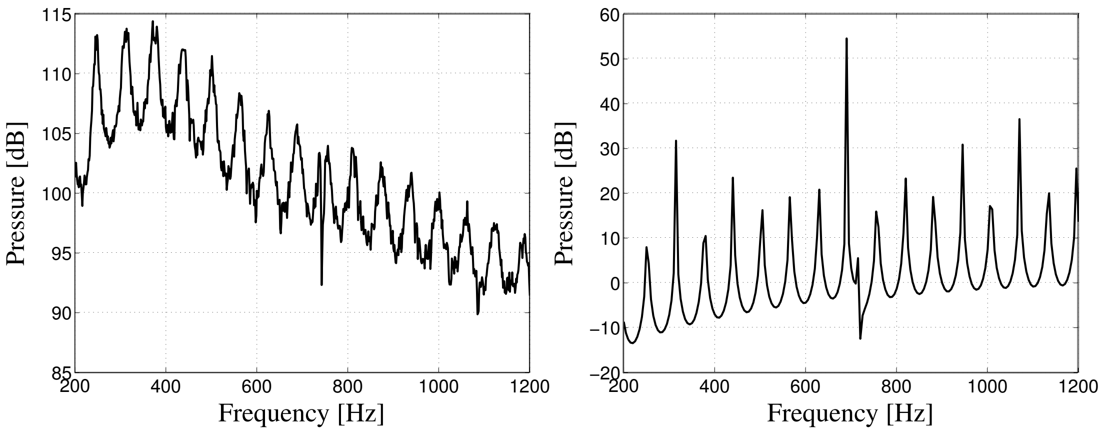


Figure 4: Measured (left) and calculated (right) pressure at the end of the supply tube

The transfer from the pressure in the supply tube to the pressure in the box is depicted in figure 5. The results of the harmonic finite element calculations are also given in the same figure. The results match very well. The transfer varies between 0 and -60 dB with an average somewhere around -30 dB. This strong decrease in pressure amplitude from the supply tube to the box is due to acoustic reflection against the open end of the tube. This can also be approximated using a 1D acoustic model and the Transfer Matrix Method of which the result is depicted in figure 5. This figure only shows the vertical acoustic resonance of the cavity, because the one dimensionality cannot describe other acoustic resonances or structural resonances. The transfer is of similar magnitude as found in the measurements, although the 1D model is a very crude approximation.

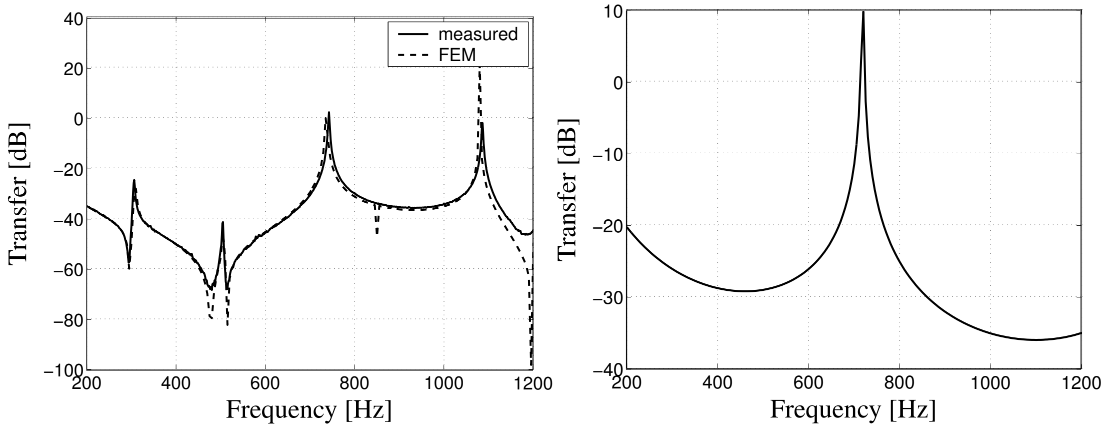
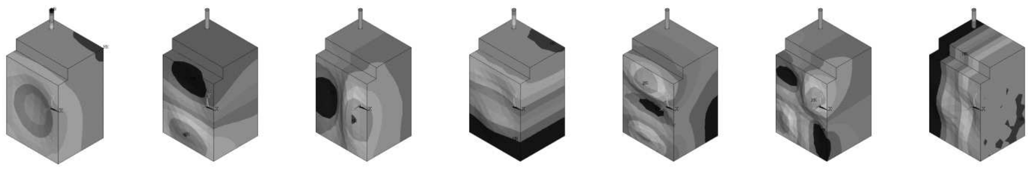


Figure 5: Transfer from supply pressure to pressure in the box (left) and 1D approximation (right)

The acoustic eigenfrequencies are clearly visible in figure 5. Furthermore it can be seen that some structural eigenfrequencies also show up in both the calculated and the measured pressure signal. This is especially true for the first two structural eigenfrequencies. The structure starts to vibrate strongly in these modes and the pressure level is therefore also higher. Finite element calculations (figure 6) show that for higher structural modes the accompanying pressure field is only present near the plate and therefore does not reach the microphone and does not show up in the transfer.



$(1,1)_s$ $(2,1)_s$ $(1,2)_s$ $(0,1,0)_a$ $(3,1)_s$ $(2,2)_s$ $(1,0,0)_a$

Figure 6: Calculated pressure field at resonance

Structural vibration

Figure 7 shows the spatial L2 norm of the transfer from supply pressure to normal velocity of the front plate. The results of the finite element calculations and the measurements again match very well. The graph shows more peaks than the one for the pressure field (figure 5). At the acoustic eigenfrequencies the acoustic pressure becomes high which leads to high excitation levels and therefore high vibration levels. At the structural eigenfrequencies the excitation is rather low, but the structure is very compliant and therefore the vibration levels are still very high. Figure 7 also shows the transfer from the pressure in the box to the vibration of the plate. The acoustic eigenfrequencies have disappeared in this plot because the high vibration response is divided by the high pressure level. In both plots the third structural mode ($(2,1)_s$) does not show up strongly. This is because the pressure field generated is already very similar to the coming $(0,1,0)_a$ acoustically dominated mode. This mode does not couple with the structural vibration and therefore the transfer between pressure and vibration of the plate is diminished.

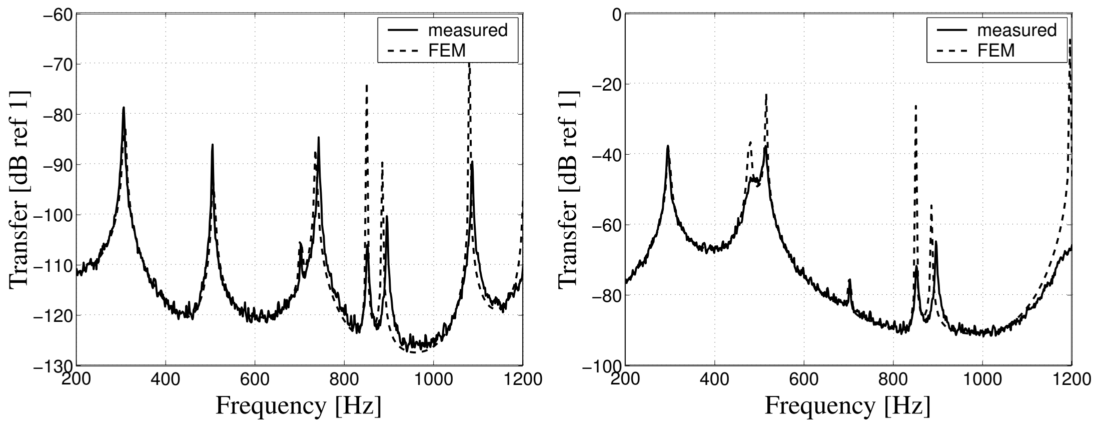


Figure 7: Spatial L2 norm of transfer from pressure to velocities on the plate (left from supply pressure, right from pressure in the box)

The transfer from the pressure in the supply tube to the velocity on a point on the plate is plotted in figure 8. The points to which the transfer is calculated are also shown. Points 1 and 3 show that the transfer is accurately predicted. Point 2 is almost exactly on the vertical middle of the plate. The second structural mode has a nodal line here and therefore this mode is almost invisible in the vibration spectrum. Point 4 in very close to the clamping of the plate. The vibration levels are therefore much lower and there is a lot of noise on the signal, because the measurement accuracy of the laser vibrometer becomes insufficient.

Coupling of acoustics and structure

The finite element model results show that structural modes and acoustic modes can be easily recognized. The pressure field generated by the vibrating flexible structure is mostly present near the structure. It does not really seem to propagate into the box. The acoustic modes on the other hand are present in the entire box. This was also seen in the measurements, where acoustic modes give high broad peaks in the pressure spectrum, whereas structural modes are only present in a very narrow band around the structural eigenfrequency.

Exceptions occur around the $(1,1)_s$ and $(3,1)_s$ modes. These modes are volumetric (pumping) modes and are therefore better able to excite the acoustic field in the entire box and not only that close to the surface. This causes significant acoustic pressure fluctuations in the box for the structural eigenmode. The $(3,1)_s$ is not really seen

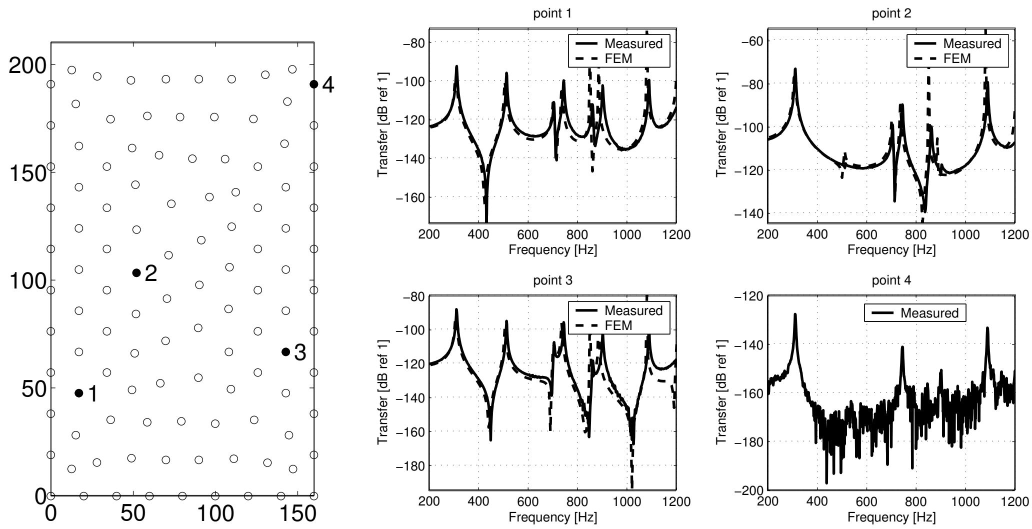


Figure 8: Transfer from supply pressure to velocity on points on the plate

in the acoustic measurements. This is probably because the pressure is not very high in the upper corner where the sensor is located. Furthermore the higher structural modes excite less because volumetric displacement is rather low.

CONCLUSION

A finite element model was made of an acoustic cavity covered by a flexible plate. The FEM model was successfully validated with an experimental setup. A point source was used constructed from a box with a speaker in it to which a long tube is connected which transports the acoustic pressure to the box with the flexible plate. The mode shapes and eigenfrequencies can be accurately predicted. The transfer from the pressure at the end of the tube to the velocities on the plate are also accurately predicted.

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