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STUDY OF THE ACOUSTIC OSCILLATION OF FLOW OVER CAVITIES. PART 2. SOUND REDUCTION

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ABSTRACT

We present experimental results obtained with a deep cavity, like an Helmholtz resonator, excited by an airflow. The resonance under the action of the vortices generated in the shear layer is well described and quantified. The mounting of actuators, based on a few piezo-electric elements, allows to generate a series of two-dimensional vortices forced at a frequency which is different than the natural resonance frequency. The sound level in the cavity is strongly decreased and the broadband noise of the turbulence only remains.

INTRODUCTION

In many areas and especially in the domain of transportation, one can often find deep cavities submitted to an airflow. The resulting noise and its reduction is an interesting challenge. This paper is the extension part of a more general paper, Amandolèse (2002) and is specifically devoted to the sound reduction for the deep cavity. The problem is similar to the so-called Helmholtz resonator and a typical application is a car vehicle with an opened roof. In such case the geometry of the cabin makes the cavity response more complex than a single rectangular box, Kang, Lee & Kim (2000). When the cabin volume of the car is large enough, the pressure level at very low frequency (infrasound range), can reach more than 130 dB and therefore a control has to be performed.

Many works have been conducted in the past on this problem, for example Naudascher & Rockwell (1994), Sunyach & Béra (1997), Luca (1995) and more recently Massenzio (1997) and Noger (1999). The flow produces in the neck a shear layer which exhibits a Kelvin-Helmoltz instability leading to periodic shedding of vortices. Some researchers, Ziada & Rockwell (1982), Rockwell (1983), have isolated a feedback mechanism which appears when the vortices shed by the upstream lip are impinging the downstream one: a pressure wave is then sent back upstream and triggers further shedding. When the frequency is close to that of the acoustic mode of the cavity, the acoustic pressure level becomes very high due to the resonance.

In the car industry, two kinds of passive techniques are mainly used in order to reduce the sound, both based on the addition of an upstream deflector whose purpose is to modify the mean wind velocity field. The first technique is quasi two-dimensional and creates a secondary air flux which is directed partially inside the cavity: the velocity gradient is then reduced in the shear layer. The second method consists in introducing a three-dimensional perturbation in order to produce a decrease in the correlation of the vortices in the spanwise direction. However the efficiency of these deflectors is far from perfect. They are also not very elegant from a design point of view because they must be mounted high enough over the boundary layer on the roof.

In this context we study experimentally a resonant cavity and we then introduce a semi-active system of sound reduction based on piezo-electric actuators. The term "semi-active" is used here since our reduction scheme is more a forcing technique than a standard active control which is usually carried out with a feed-back loop. The present paper extends the preliminary results given by Amandolèse et al. (2002).

EXPERIMENTAL SETUP

Pressure measurements

The model of the cavity has been mounted in a small acoustic wind tunnel of the Institut Aérotechnique as in the part 1 of the paper. The original feature is that it allows to measure the outside acoustic pressure P_e even when the wind is blowing: this kind of measurement is performed by plane waves intensimetry.

The principle of this technique consists in writing the pressure along the test section longitudinal direction x as

$$P(x,f) = A(f)\, e^{-jk_i x} + B(f)\, e^{jk_r x} \tag{1}$$

where the incident and reflexive wave numbers are $k_i = \dfrac{2\pi\ f}{c+U_0}$ and $k_r = \dfrac{2\pi\ f}{c-U_0}$ respectively. The hypothesis of plane waves leads to coefficients independent of the position in the test section. By measuring the pressure with two microphones separated by a distance d_1, and inverting the relation (1) we obtain

$$A(f) = \frac{P_2(f)\, e^{-jk_r d_1} - P_1(f)}{e^{-jk_r d_1} - e^{jk_i d_1}} \quad B(f) = \frac{P_1(f) - P_2(f)\, e^{jk_r d_1}}{e^{-jk_r d_1} - e^{jk_i d_1}} \tag{2}$$

Then from relation (1) it is possible to recombine the pressure at a given position of the test section, and then obtaining P_e . This technique implies here a measurement frequency range between 80 and 660 Hz. In practice a more complex signal processing is performed, involving adaptative filtering for instance. More details on the measurement system and its validation are available in Hémon (2000).

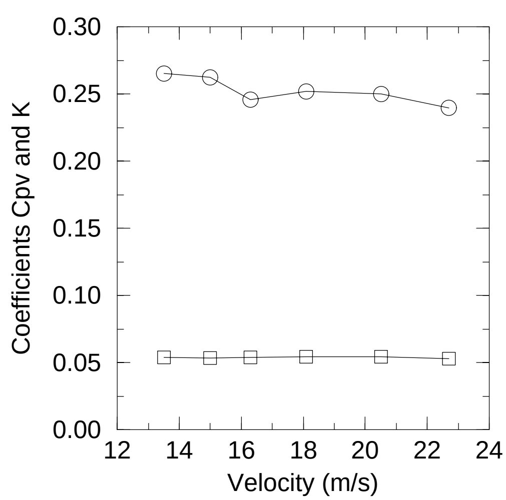

Figure 1. Mean pressure coefficients versus U_o (○: K, □: Cp_v)

The pressure P_v in the cavity is measured by a microphone and another static pressure transducer was mounted to measure the mean pressure level. It was found that in the range of these experiments, the mean pressure level was not influenced by the acoustic resonance. The mean pressure coefficient Cp_v (the mean pressure in the cavity normalized by the dynamic pressure $Q_o = \rho\, U_o^2 / 2$) and the pressure loss coefficient K (the pressure difference between upstream and downstream normalized by Q_o) are given in Figure 1 versus wind velocity. They are both almost constant in the range of these experiments.

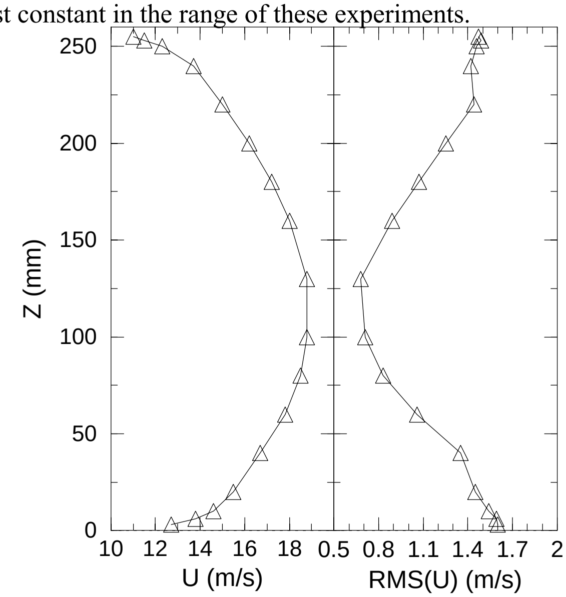

Figure 2. Profiles of horizontal velocity *U* at X=-2 *L* and U_o=15.7 m/s

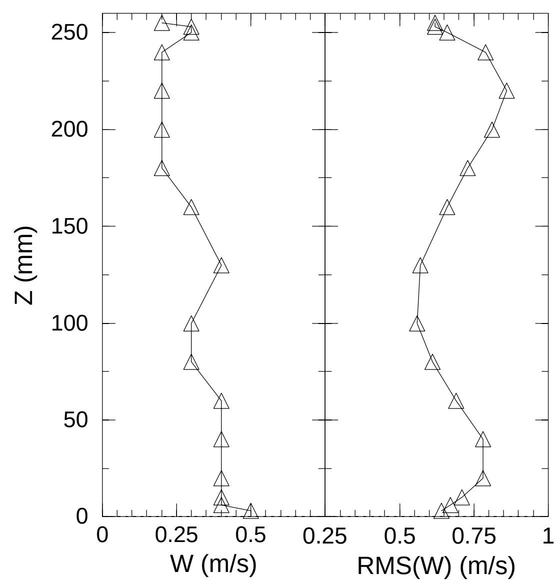

Figure 3. Profiles of vertical velocity *W* at X=-2 *L* and U_o=15.7 m/s

Upstream velocity profiles

The pressure measurements are completed by velocity measurements using hot wire anemometry. The profiles of the incoming flow, taken at an upstream position 2 L in front of the cavity lip, are given in Figure 2 for the horizontal velocity U, and in Figure 3 for the vertical component W. The mean flow velocity U_o was 15.7 m/s. The profiles of the root-mean-square velocity (RMS) are also given in each case. The vertical velocity has a very small mean value, close to the accuracy of the measurement chain and its value is not very significant. The horizontal component is not perfectly symmetrical due to a difference in wall roughness between the bottom wall (where the cavity is mounted) and the roof.

RESULTS WITHOUT SOUND REDUCTION

Cavity natural resonance

When there is no wind, the classical theory of Helmholtz resonators (see Doria (2000) for an example) is valid and it is now well admitted that the natural frequency of such a cavity is given by

$$f_v = \frac{c_o}{2\pi}\sqrt{\frac{A_c}{\left(H_g + 0.8\sqrt{A_c}\right)V}} \tag{3}$$

Applying this formula to the present model, we find a frequency f_v=235 Hz. However, when the cavity is excited by a white noise, the natural frequency can be determined experimentally and we obtain here 263 Hz (± 0.5), which differs from formula (3) by 11 %. In fact, the difference comes from the term $0.8\sqrt{A_c}$ which represents an empirical correction in order to determine an effective thickness of the neck section. Fitting the formula with the experimental result leads to replace the previous term by $0.63\sqrt{A_c}$.

Pressure levels with the flow

When the wind is blowing over a cavity, the dimensionless frequencies of the oscillations of the shear layer are well approximated by Rossiter's formula (1964)

$$St = \frac{n-\gamma}{M + \dfrac{U_o}{U_c}}, \tag{4}$$

where the Mach number M can be neglected in our case, n is the order of the mode and γ a parameter linked to the shape of the lips and usually equal to 0.25 for sharp edges and rectangular cavities (see Tam & Block (1978) for a review). The ratio of the convection velocity U_c to the freestream velocity U_o is almost universal and equal to 0.57. Then, according to formula (4), the first Strouhal number should be 0.41.

Nevertheless, the experiments showed that the frequency of resonance between the volume of the cavity and the oscillation in the shear layer was obtained for a Strouhal number of 0.32, as can be seen in Figures 4 and 5.

In Figure 4, the horizontal dashed lines on the frequency curve are the frequencies which correspond to the natural frequency of the cavity, for the first one, and to one acoustic mode frequency of the test section for the second one. Each resonance case leads to a high pressure level in the cavity, and we focus the present paper on the first peak. The different sets of triangles correspond to the same test which was made twice.

Figure 5 presents the dimensionless frequencies that are significant for the excited Helmholtz resonator. Triangles represents the reduced frequency f_r given by the measurements, the dashed line is the cavity volume natural frequency, and the continuous line the Strouhal number given by Rossiter's formula, empirically corrected by setting γ=0.41 for the first mode (n=1).

It is interesting to see that the measured frequencies do not exactly follow the evolution of the frequency of the cavity volume, but markedly differ from the shear layer frequency predicted by Rossiter. The system responds as if the volume of the cavity was dominant but not completely. It responds between the single fluid instability of the shear layer and the acoustic mode of the resonator.

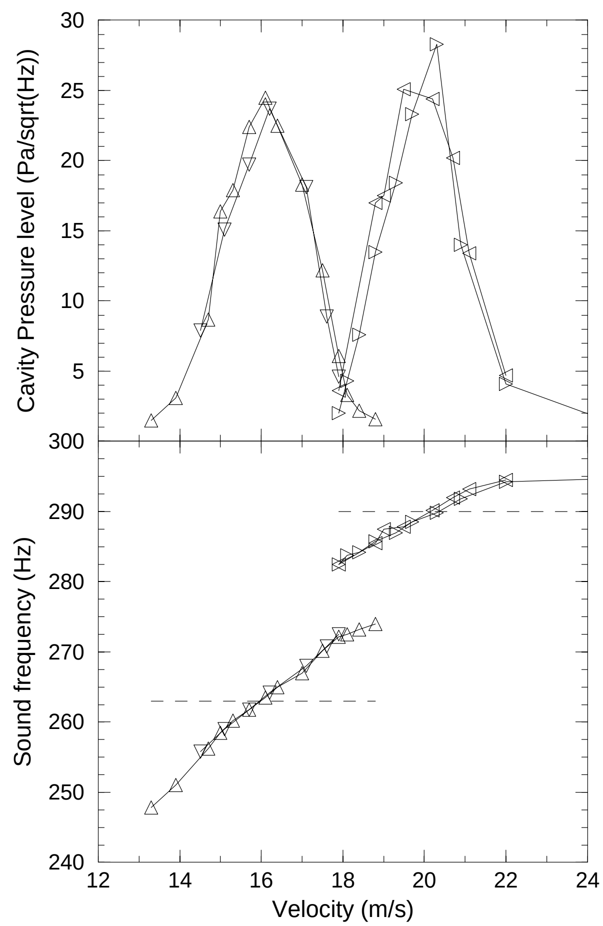

Figure 4. Cavity pressure level and corresponding frequency versus U_o for the deep cavity

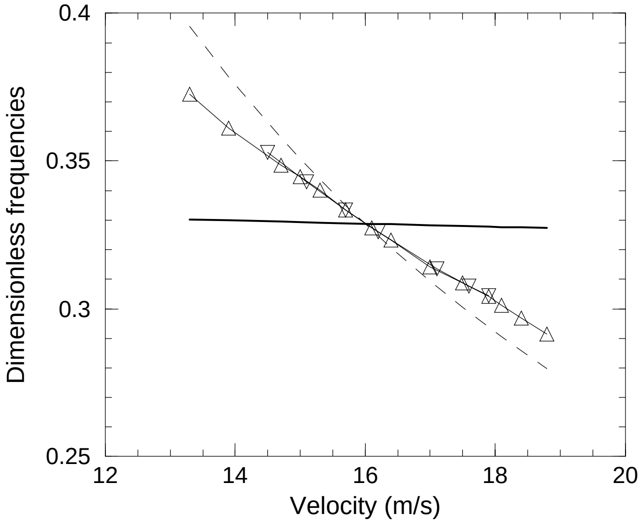

Figure 5. Dimensionless frequencies versus U_o (△: measurements, - - - - -: natural cavity frequency, ——: Rossiter's formula)

Transfer function measurements

In order to better understand the phenomenon, we measure using intensimetry the external pressure P_e, around the resonance velocity with the cavity. This is performed simultaneously with measurements of the internal pressure P_v. Then we plot the transfer function P_v/P_e in amplitude (modulus) and phase angle in Figure 6.

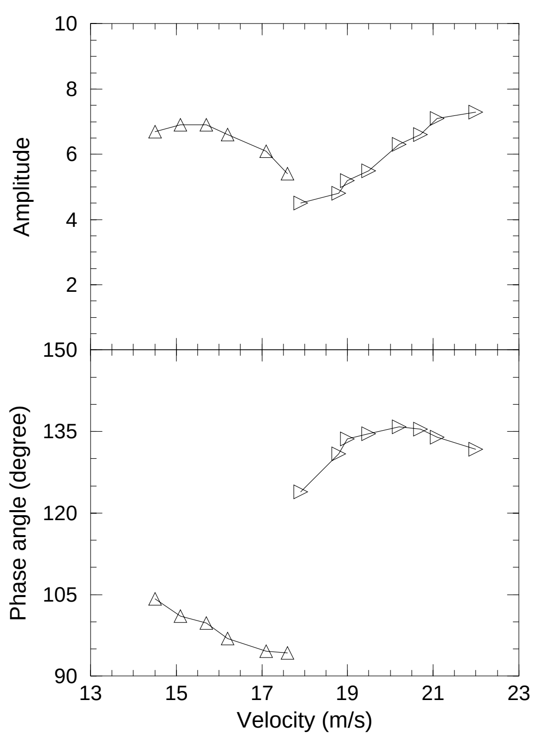

Figure 6. Transfer function P_v/P_e versus U_o for the deep cavity

The main point of these data is the phase angle which is found to be 97° (± 2°) at the point of resonance for U_o=16 m/s. It means that the system is not only a simple resonator which is excited by a shear layer tuned in frequency. Indeed, the confinement of the flow over the cavity produces a damping which contributes to limit the pressure level in the cavity.

From the classical linear theory of the Helmholtz resonator, assuming that the air is a perfect gas and the cavity volume is adiabatic, we can write

$$\frac{V\, H_g}{c_o{}^2 A_c}\ddot{P}_v + P_v = P_e + \varepsilon\, P_{sl}\ . \tag{5}$$

The coefficient ε is a dimensionless unknown amplitude of the excitation term P_{sl} produced in the neck by the shear layer instability. It may be noticed that the eigenfrequency of the left-hand side is given by expression (3) in which the empirical correction term $0.8\sqrt{A_c}$ was added. The external pressure P_e appears as a forcing term for the cavity pressure. Since the response is harmonic at the resonance point, it may be written as

$$P_e = \alpha\, P_v + \beta\, \dot{P}_v \tag{6}$$

This expression assumes linearity as a first approximation. The phase angle ϕ of the transfer function P_v/P_e which is found experimentally leads to a negative coefficient β which indeed represents a damping term in system (5). However, the result is not an exact quadrature response (i.e. 90°) and the small difference, between 90° and the measured 97°, means that a phenomenon of added aerodynamic stiffness (or mass, the phase being the same) is also present.

In the case of the second peak, due to the resonance with an acoustic mode of the duct, the damping term due to the confinement is lower, which is physically logical, the confinement being itself the reason for this resonance.

Nevertheless, it must be recalled that the excitation is physically the instability of the shear layer, which is modelled by $\varepsilon\, P_c$ and can be estimated from the measurements of the other parameters of system (5). Indeed, this system has to be in equilibrium, i.e. at a zero balance of energy over one period. Then at resonance, the shear layer instability is tuned with the frequency of the cavity, therefore the excitation term might be rewritten as

$$\varepsilon\, P_{sl} = \mu\, \dot{P}_v\ . \tag{7}$$

By replacing (6) and (7) in (5) we can identify μ as:

$$\mu = -\beta = \frac{1}{2\pi\, f}\left|\frac{P_e}{P_v}\right| \sin\phi\ , \tag{8}$$

where all the terms are given by the measurements. This leads to $\beta = -9.1\,10^{-5}$ s/rad for the resonance point at U_o=16 m/s. We have also $\alpha = -1.85\,10^{-2}$ by using

$$\alpha = \left|\frac{P_e}{P_v}\right| \cos\phi\ . \tag{9}$$

SOUND REDUCTION

Actuators

The setup for reducing the noise is mounted on the surface of the upstream lip. The actuators are piezo-electric bimorph elements (two layers of PZT with a middle layer metal sheet, provided by Piezomechanik) which are assembled in order to produce a bending movement such as a cantilever beam. Their active length is 20 mm, and their width 6 mm. Eight actuators have been mounted spanwise with a distance of 20 mm between their axis. Figure 7 gives a sketch of the setup.

These actuators are all excited electrically by the same sinusoidal signal at the frequency f_c= 310 Hz. This is their first bending frequency, for which the obtained displacement is maximum. The ratios with the two resonance frequencies are then 1.18 and 1.07 which is well within the range [0.5 – 2] mentioned by Ho and Huerre (1984). All the actuators are correlated, i.e. they are all in phase. An arbitrary estimate of the displacement amplitude gave an order of magnitude of ± 0.25 mm at their extremity. There is also the possibility to use one actuator out of two, in order to have an estimate of the influence of their spanwise distance.

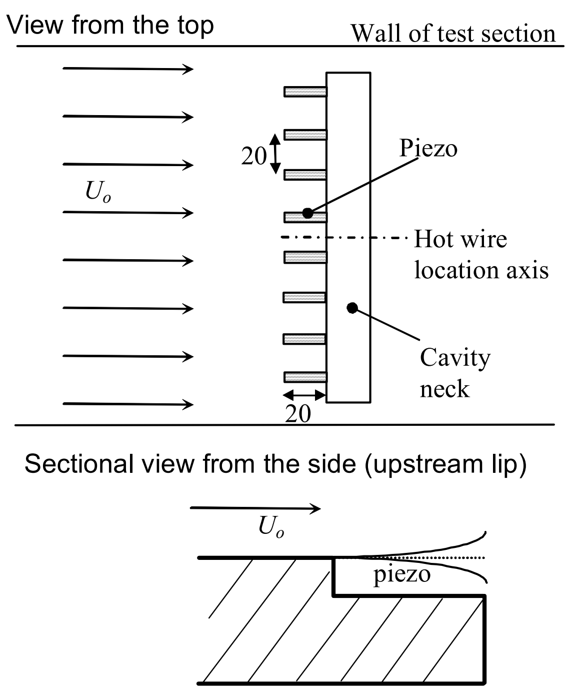

Figure 7. Sketch of the actuators mounting

Results

The pressure level P_v measured without and with the actuators activated are given in Table 1 for the two peaks of resonance observed in Figure 4. The physical unit is a power spectral density and does not depend on the acquisition parameters. The displayed value is for the frequency of each resonance. For a better approach of the physics, note that a pressure level of 25 $Pa/\sqrt{Hz}$ with our frequency resolution is equivalent to 115 dB, which is a very high level for the human ear. Examples of such spectra are given in Figure 8 where the cross cursor is placed at the resonance frequency. The thin band peak at 310 Hz, on the spectrum with the actuators, is due to a vibracoustic perturbation (not aeroacoustic) of the actuators to the measurement microphone.

When all the actuators are active, the sound level is completely buried within the broadband noise of turbulence. The sound reduction is very efficient. When only one actuator out of two is used, the sound is also decreased but not totally.

Number of actuators	Resonance 1 U_o=16m/s ($Pa/\sqrt{Hz}$)	Resonance 2 U_o=20m/s ($Pa/\sqrt{Hz}$)
0	24,5	28,3
4	6,0 to 10,8	-
8	1,4	2,1

Table 1. Main results of the sound reduction scheme

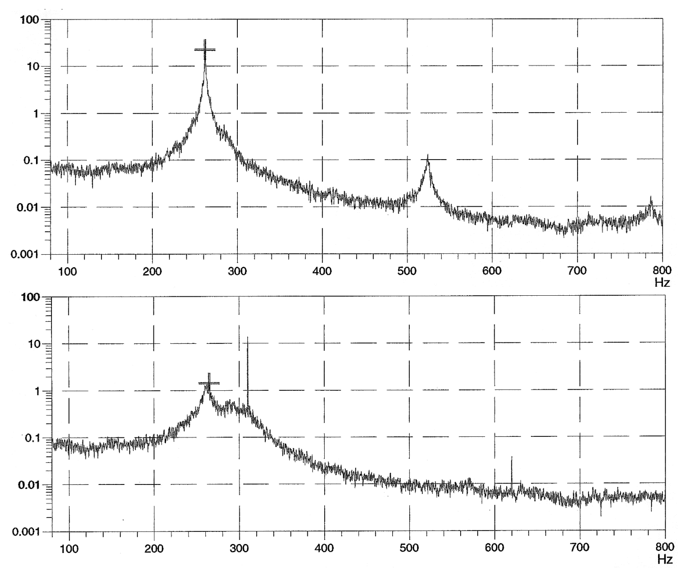

Figure 8. PSD of P_v for the first resonance. (upper: without actuator, lower: with all actuators at 310 Hz)

In order to better understand the action of the piezo elements, the shear layer velocity profile has been measured, with and without actuators. It is given in Figure 9 and was measured at a streamwise position $L/2$ in the middle of the neck, and at the mid-spanwise position between two actuators as it is shown in Figure 7.

It is very interesting to notice that the mean velocity profile is not perturbed by the forcing, and only the fluctuating part is modified. These results were also observed by Cattafesta et al. (1997) and show that the drag of the cavity is not increased by the sound reduction scheme. Without forcing, the maximum fluctuation is concentrated around the vertical position Z=0, corresponding to the floor of the test section for upstream flow. With forcing, this maximum is concentrated at a higher position, just above the neck, which is probably due to the geometrical assembly of the actuators as shown in Figure 7. Another point is the fact that the perturbation generated by the actuator is shown to be spread laterally, which means that the actuators effect is quasi two-dimensional.

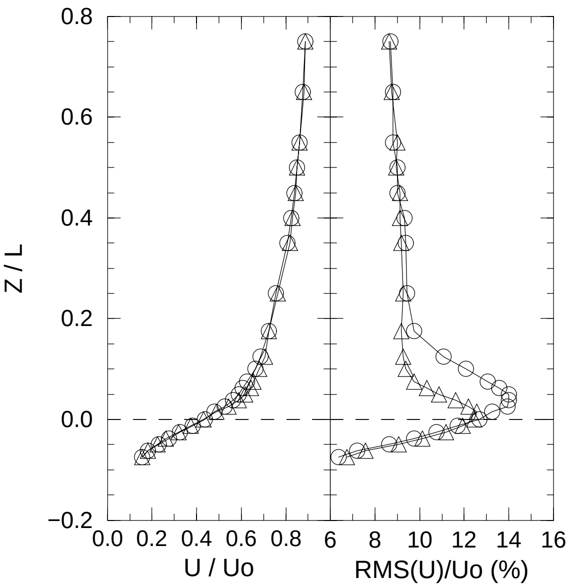

Figure 9. Shear layer profile in the middle of the neck (△: without actuator, ○: with all actuators at 310 Hz)

However, the RMS value is an integration over all frequencies, and it is better for the analysis to select the signals at the frequencies which are dominant. The linear modeling of equation (5) and followings, is only valid at resonance or forcing frequency, where the coherence functions of the pressures are close to 1. Let us define a quantity named *Rpsd* as the ratio given by

$$Rpsd = \frac{PSD(U \ at \ f)^2}{PSD(P_v \ at \ f)} \tag{10}$$

where the power spectral density (PSD) of each quantity is taken at a given frequency. The purpose of this coefficient is first to obtain a quantity close to a dimensionless number, and secondly to smooth the experimental results: indeed the measurement of such a profile is quite long and the mean conditions are weakly modified during the total duration of the test. The unit of *Rpsd* is $1/\sqrt{Hz}$ and the measured profiles are given in Figure 10 for the unforced case (at f_v=263 Hz) and the forced one (at f_c=310 Hz).

We can see more clearly the forced wake of the actuators which is well spread laterally and quasi two-dimensional. Moreover, the amplitude of the fluctuation is really larger than without forcing. The sound reduction is achieved because the forced shear layer is detuned in frequency from the natural resonance frequency of the cavity.

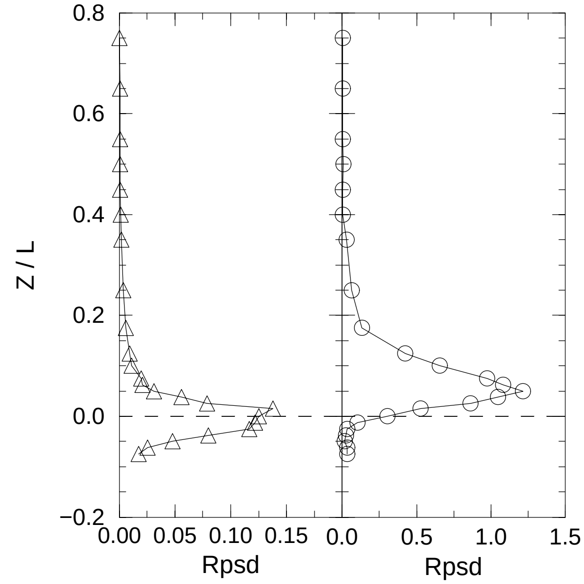

Figure 10. *Rpsd* profiles in the middle of the neck (△: without actuator, ○: with all actuators at 310 Hz)

Transfer function

We give in Table 2 the parameters that are obtained by measurements of the transfer function P_v/P_e following the definition of the coefficients of equations (8) and (9). In the case with the actuators activated, there are two frequencies in the system, f_v and f_c, which we did not find to be coupled, so that the principle of superposition can be applied for each one. Then equations (8) and (9) are valid and used for each frequency. The interesting point is that the forced wake given by the actuators completely brings the pressure P_v and P_e to be out of phase, which leads to the highest possible value of damping β.

Furthermore, we have also modified the volume of the cavity so that its natural frequency was increased up to 310 Hz which corresponds to the frequency of the actuators. The resulting sound was logically increased to more than 50 $Pa/\sqrt{Hz}$, i.e. at least twice the level of natural resonance (see Table 1). The excitation of the shear layer was reinforced in this case.

	Without forcing At f_v=263 Hz	With forcing At f_v=263 Hz	With forcing At f_c=310 Hz
$\left\lvert \frac{P_v}{P_e} \right\rvert$	6.6	6.9	4.0
ϕ	97°	99°	90°
α	-1.85 10^{-2}	-2.27 10^{-2}	0
$\beta = -\mu$	-9.1 10^{-5}	-8.7 10^{-5}	-1.28 10^{-4}

Table 2. Measured parameters of the system

CONCLUSION

We have demonstrated experimentally a very simple method in order to reduce the sound generated by a deep cavity, such as a Helmholtz resonator which is excited by an air flow. The main point is that the system is open loop and it is easier to implement than standard closed loop control, as Cattafesta & al. (1997) or Kikushi & Fukunishi (1999). The forced shear layer creates a control action which is naturally out of phase with the response, but the excitation is completely free from any other constraint.

The shear layer was forced in a two dimensional way by a series of discrete actuators, the forcing frequency being detuned from the natural resonance. However, some points remain unclear and more investigations should be performed. The actuators have to be more carefully studied in order to quantify and then optimize the energy input in the system.

Moreover the sensitivity of the control efficiency to the ratio of the forcing frequency to the natural frequency (f_c/f_v) has to be further investigated.

The influence of the neck thickness H_g has to be studied in order to reach a better fitting of the Rossiter formula with the present low Mach number flows. This should be related to the different neck shapes which were studied by Massenzio (1997). Moreover, simultaneous pressure measurements located at the upstream and downstream lips would be helpful to determine the phase between separation and impingement.

Another objective would be to perform a real test case with a real scale car vehicle on which external flow conditions are more complex, as well as the shape of the cabin.

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