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Adaptive control of acoustic intensity with turbulent flow

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ABSTRACT

Generally, studies on the noise active control use a minimization of acoustic pressure thanks to a real time adaptive algorithm.
Nevertheless, applied to a complex pressure field with forward and backward-moving waves, this approach can led to the modification of field without spatial reduction or even to a noise increase if the location of microphone(s) is unsuited.
A way to avoid this phenomenon is to increase the number of sensors and to minimize, e.g., the quadratic sum of measured pressures or to couple the minimization to a modal model.
An other approach consists in active control of quantities linked to energy propagation. So, to minimize locally the active intensity in direction of acoustic waves can allow the decrease of pressure level downstream the sensors, without significant spillover phenomenon. In practice, the "cost function" can be the "Instantaneous Intensity" whose time average is the active intensity.
The purpose of this paper is to show an application of active control of instantaneous intensity to reduce noise (tones) propagated in a circular duct in presence of turbulent flow (Mach 0.15), before and after cut-off-frequency of the first azimuthal mode (800-1300 Hz). The algorithm requires microphones at two close locations and theoretical formulations to determine intensity (e.g. Euler and Morfey equations).
The efficiency of intensity control is noticed at downstream microphones by comparison with a classical pressure minimization.
This activity takes place as part of French project COMBE in a bench intended to reproduce propagation of "fan noise" in multi-modal configuration.

1. INTRODUCTION

After Audrain et al.[1], Noël et al.[2] have used Instantaneous Structural Intensity, instead of local acceleration, as cost function of a real-time controller to reduce the propagated energy, and therefore the vibration, all over a highly damped cantilever beam. The "energy minimisation" has been led, thanks to a time-domain FXLMS algorithm, on the moment part of instantaneous structural intensity, measured by a two lightweight accelerometer probe. The vibration shape has appeared after "intensity" minimisation more constant than with an "acceleration" minimization and the vibration level at the beam end was lower. A generalisation of this approach has been applied to a 2D-wave propagation configuration[3]. The propagation has been generally well organised and the velocity field more homogeneous. Therefore, the intensity control appeared mainly more interesting as regards the vibration. Then, an active control of the instantaneous acoustic intensity (thanks to two microphones) downstream from a porous liner has been suggested and tested in duct without flow[4]. Supposing the presence of a source of noise upstream liner, minimizing the acoustic intensity downstream reverts maximizing the

intensity entering liner and thus improving its absorption. The same type of algorithm as previous authors[2-3] has been used. The only difference has lied in the input signals (pressure for the acoustic intensity, acceleration for the intensity of structure).
In this paper, the influence of a flow in an algorithm of intensity control is taken into account. The new algorithm is applied with a «virtual» adaptive control from data acquired in an aeroacoustic bench.

2. THEORY

A. Theory of acoustic intensity

Without flow, instantaneous acoustic intensity vector produced by acoustic waves is defined as following:

$$\vec{i}(t) = p(t)\vec{u}(t) \tag{1}$$

with *p* and *u* respectively acoustic pressure and velocity at time *t*.
Active intensity vector, $\vec{I} = \langle p.\vec{u} \rangle$, time average of instantaneous acoustic intensity, is representative of the exact power flow in frequency domain.

When the fluid is moving, its definition is no longer clear. Nevertheless, if we assume an isentropic and irrotational flow, we can apply the Morfey[5] formulation, used by Munro and Ingard[6], that verifies the energy conservation law in a given volume without internal source:

$$\vec{I} = \left\langle p\vec{u} + (\vec{u}_0.\vec{u})\left(\frac{\vec{u}_0}{c_0^2}p\right) + \rho_0(\vec{u}_0.\vec{u})\vec{u} + \left(\frac{p^2}{\rho_0 c_0^2}\right)\vec{u}_0 \right\rangle \tag{2}$$

with $\vec{u}_0$ mean flow velocity, ρ_0, mean density and c_0 sound speed.

Practically, if the mean flow and acoustic waves follow the x direction (1D), the instantaneous intensity is as following:

$$i(t) = (p + \rho_0 u_0 u_x)(u_x + \frac{u_0}{\rho_0 c_0^2} p) \tag{3}$$

It is conventional to determine acoustic velocity thanks to pressure measurements.
With flow, acoustic velocity u_x is obtained by:

$$\begin{cases} \frac{1}{c_0^2}\left(\frac{\partial p}{\partial t} + u_0.\frac{\partial p}{\partial x}\right) + \rho_0 \frac{\partial u_x}{\partial x} = 0 \\ \rho_0\left(\frac{\partial u_x}{\partial t} + u_0.\frac{\partial u_x}{\partial x}\right) + \frac{\partial p}{\partial x} = 0 \end{cases} \tag{4}$$

and so, by integration over time :

$$u_x = \frac{M_0}{\rho_0 c_0} p - (1 - M_0^2)\frac{1}{\rho_0}\int \frac{\partial p}{\partial x} dt \tag{5}$$

with M_0 mach Number

that can be approximated with two pressure measurements, spaced of a distance *d* (pressure gradient).

After Fourier transform, intensity, for monochromatic waves at pulsation ω becomes[6]:

$$I = -\frac{(1-M_0^2)(1+3M_0^2)}{\omega\rho_0 d}\mathrm{Im}(G_{12}) + \frac{M_0(1+M_0^2)}{2\rho_0 c_0}(G_{11}+G_{22}+2\mathrm{Re}(G_{12}))$$
$$+\frac{M_0(1-M_0^2)^2 c_0}{\rho_0(\omega d)^2}\frac{[G_{22}-G_{11}]^2+4\{\mathrm{Im}(G_{12})\}^2}{G_{11}+G_{22}+2\mathrm{Re}(G_{12})} \tag{6}$$

with G_{11}, G_{22} and G_{12} respectively autospectra and cross-spectrum for microphones 1 and 2. Without flow, intensity only takes $\mathrm{Im}(G_{12})$ into account, as classical formulation.

If acoustic waves are propagated in 3D, to measure separately intensity in the 3 directions of an orthonormal coordinate system and a vector reconstruction don't supply the "exact" intensity (even without intrinsic phase difference of the microphone signals). The error increases with Mach Number.
Moreover, two acoustic fields[6] with same pressures and local pressure gradients can have different local acoustic velocities, because dispersion relation with flow:

$$c_0|\mathbf{k}| = |\omega - \mathbf{u_0}\cdot\mathbf{k}| \tag{7}$$

That is the reason why the formulation (6) must be used cautiously, particularly, for high Mach number and for acoustic fields with waves in several directions.
Consequently, it is suited to propagation of plane waves in duct, with two microphones flush with the wall, configuration that allows to determine easily the acoustic power propagated through a section.

B. Intensity cost function

The minimization of acoustic intensity requires a two-channel diagram (figure 1) and concerns the instantaneous intensity $i(n)$ expressed from pressure signals of microphones 1 and 2 ($e_1(n)$ and $e_2(n)$), measured at sample *n* for a given sampling frequency:

$$i(n) = \left(\frac{1+M_0^2}{2}\cdot\gamma(n) + \frac{c_0.M_0(1-M_0^2)}{2.\omega^2.d}\cdot(3.\delta(n-2)-4.\delta(n-1)+\delta(n))\right)\cdot$$
$$\cdot\left(\frac{M_0}{\rho_0.c_0}\cdot\gamma(n) + \frac{(1-M_0^2)}{2.\omega^2.d.\rho_0}\cdot(3.\delta(n-2)-4.\delta(n-1)+\delta(n))\right) \tag{8}$$

with $\gamma(n) = e_1(n) + e_2(n)$
$\delta(n) = e_2(n) - e_1(n)$

which can be simplified, without flow, by :

$$i(n) = \frac{\gamma(n)}{2}.\frac{1}{2\omega^2 d\rho_0}.(3.\delta(n-2)-4.\delta(n-1)+\delta(n)) \tag{9}$$

This last formulation has been used by Minotti et al.[4].

The nomenclature is given in Table 1.

A reference signal $x(n)$ supposed to be monochromatic, is supplied at each *n* to a "primary" actuator. This one generates initial pressure signals $d_1(n)$ and $d_2(n)$.

Reference signal is filtered by an adaptative filter W to produce $y(n)$ in input of the secondary actuator. The pressure signals generated by this actuator are respectively $s_1(n)$ and $s_2(n)$.

To ensure the convergence of this adaptative algorithm, the cost function to minimize has to be a quadratic function of the weights:

$$J = i(n)^2 \tag{10}$$

The coefficients of filter W are updated by a gradient method at each time n (FXLMS algorithm):

$$W_k(n+1) = W_k(n) - \mu \frac{\partial J(n)}{\partial W_k(n)}, \tag{11}$$

with μ convergence rate and

$$\frac{\partial J(n)}{\partial W_k(n)} = 2.i(n).\frac{\partial i(n)}{\partial W_k(n)} = 2.i(n).\left[\begin{array}{l} \left(\left(\frac{(1+M_0^2)}{2}\right).\alpha(n-k) + \frac{c_0.M_0.(1-M_0^2)}{2.\omega^2.d}.(3.\beta(n-2-k) - 4.\beta(n-1-k) + \beta(n-k))\right) \\ .\left(\left(\frac{M_0}{\rho_0 c_0}\right).\gamma(n) + \frac{(1-M_0^2)}{2.\rho_0\omega^2.d}.(3.\delta(n-2) - 4.\delta(n-1) + \delta(n))\right) \\ +\left(\left(\frac{(1+M_0^2)}{2}\right).\gamma(n) + \frac{c_0.M_0.(1-M_0^2)}{2.\omega^2.d}.(3.\delta(n-2) - 4.\delta(n-1) + \delta(n))\right) \\ .\left(\left(\frac{M_0}{\rho_0 c_0}\right).\alpha(n-k) + \frac{(1-M_0^2)}{2.\rho_0\omega^2.d}.(3.\beta(n-2-k) - 4.\beta(n-1-k) + \beta(n-k))\right) \end{array}\right] \tag{12}$$

with $\alpha(n) = x'_1(n) + x'_2(n)$
$\beta(n) = x'_2(n) - x'_1(n)$

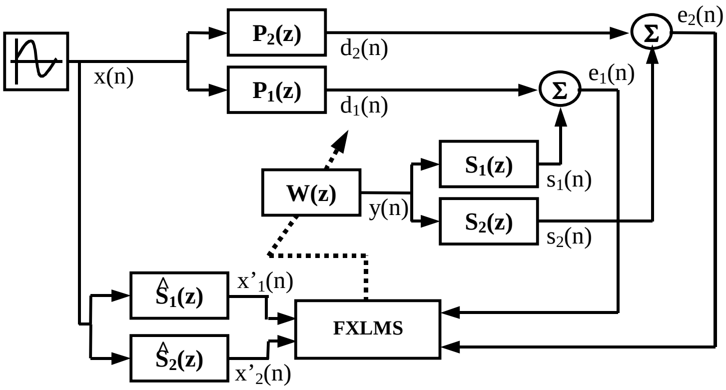

Figure 1: diagram of the FXLMS algorithm with $J = i(n)^2$ as cost function

Table 1: *Nomenclature for the FXLMS algorithm*

$P_j(z)$ = primary path for microphones . #j (j =1..2)
$S_j(z)$ = secondary path for microphones . #j
$\hat{S}_j(z)$ = identified secondary path for microphones. #j (order Q, coefs $\hat{s}_l$ supposed time invariant)
$W(z)$ = adaptative filter (order N, coefs. w_i)
$x'_j(n) = \sum_{k=0}^{Q-1} x(n-k) \cdot \hat{s}_j(k)$ = filtered reference for microphones. #j
$e_j(n)$= error (output of microphones. #j)

Practically, a partial update algorithm is used to obtain an acceptable computational cost.

2. APPLICATION

A. Configuration

Active control processes are led in the test bench (developed in project COMBE) which is a cylindrical duct (diameter=17.6 cm), supplied with air by upstream fan. It is dedicated for development of active control concepts in multi-modal configuration[7]. It can be equipped with different sections for generation of "acoustic waves" or measurement systems (e.g. in figure 2). In our case (figure 3), we use a "pair" of loudspeakers (1) to generate acoustic waves (primary source). Error sensors (1 or 2 microphones) coupled with an other pair of loudspeakers (7) (secondary source) are used to reduce locally acoustic pressure or intensity. The reduction of pressure is then verified downstream thanks to 8 microphones mounted according to two sections.
The cut-off-frequency of the first azimuthal mode is at 1144 Hz without flow.
The reflection coefficient of the duct outlet (figure 4) shows a decrease according frequency, with a level between 0.3 and 0.5 in the frequency range 800-1200 Hz. This implies presence of backward waves that can justify the use of an active control based on intensity cost function.

Figure 2: Example of configuration for test bench

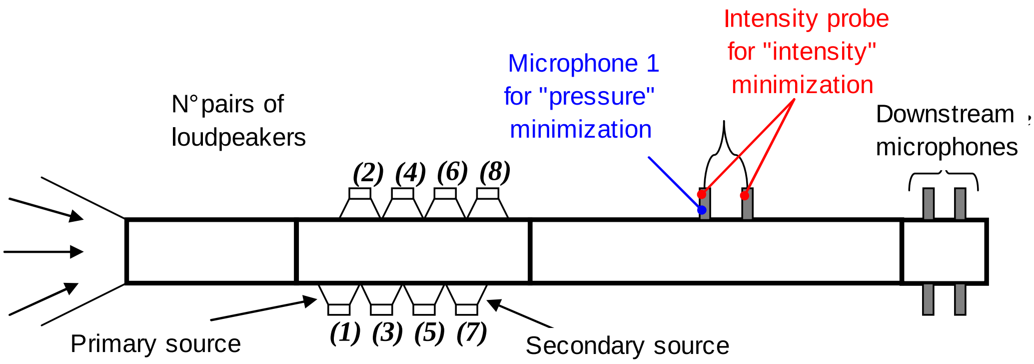

Figure 3: Diagram of Test bench used for active control

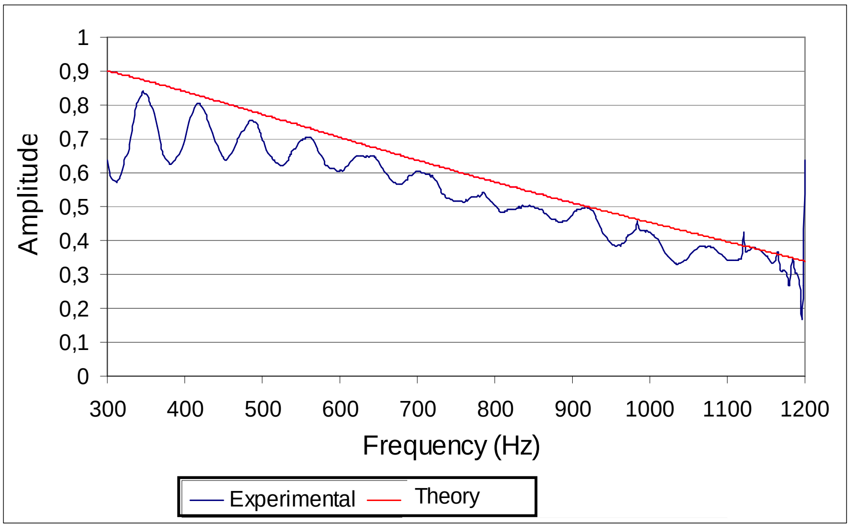

Figure 4: Reflection coefficient of outlet duct without flow – experimental and theory[8]

The purpose is to achieve a «virtual» adaptive control (figure 5) from data acquired in the test bench in "real" condition:

- transfer functions between sources (primary / secondary) and sensors ("error" microphones / downstream microphones)
- "aerodynamic" noise (turbulence fluctuations).

The process of "virtual" control is a useful step to evaluate mainly the performance of algorithm to reduce intensity and downstream pressures in real test bench, without effects of active control hardware and electronic noise.

Configurations are as following:

- Primary excitation with monochromatic noise from 800 to 1300 Hz (ΔF=100 Hz)
- Without and with flow ($M_0 = 0.15$)
- Active control with cost function $J = e_1(n)^2$ or $J = i(n)^2$ (pressure or instantaneous intensity)
- Hypothesis: "aerodynamic" and "loudspeaker" noises are uncorrelated.

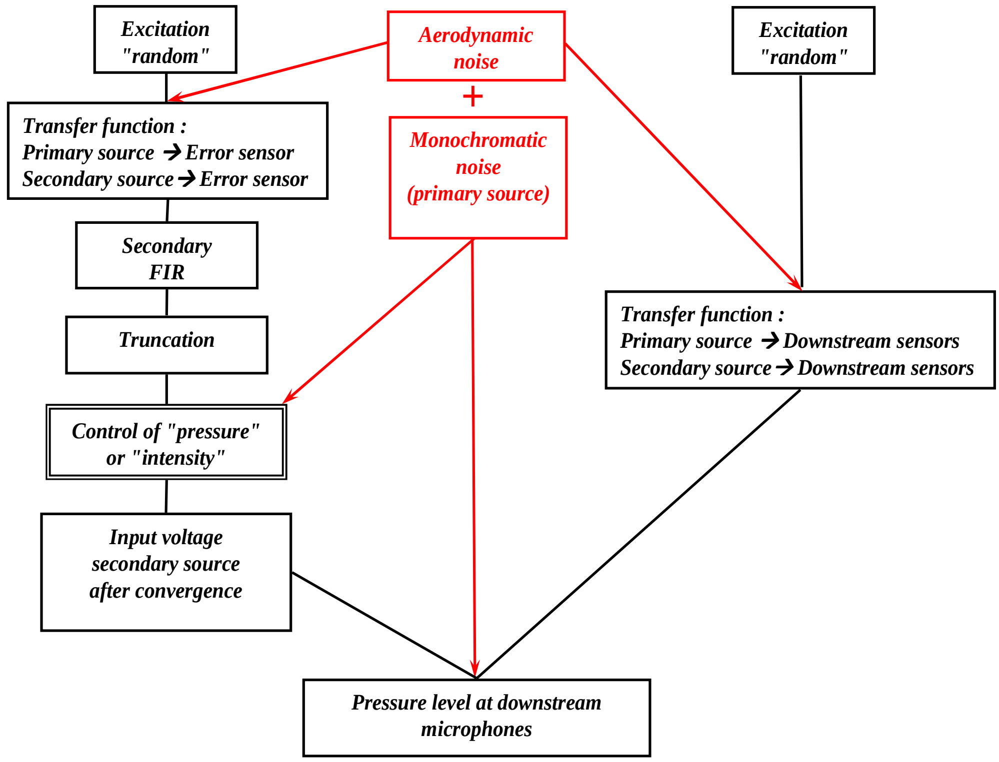

Figure 5: Process of "virtual"active control

B. Results

Main parameters of virtual control (representative of real values) are respectively for a "pressure" and an "intensity" minimization (table 2).

Table 2: *Parameters for* $J = e_1(n)^2$ *or* $J = i(n)^2$

Excitation frequency (Hz)	Sampling frequency (Hz)	N (filter coefficients)	Q (secondary path)
800-1300	4000 or 6000	20 ($J = e_1(n)^2$) or 5 ($J = i(n)^2$)	120 or 250

The number of filter coefficients is chosen lower for "intensity" minimization to reduce the computational cost.
One can notice that the time convergence needed to obtain a stable minimization is longer for this type of control and increases with frequency (figures 6 and 7).
The average pressure level measured at downstream microphones, with and without control, is shown respectively without flow (figure 8) and with flow at $M_0 = 0.15$ (figure 9).
From 800 to 1100 Hz, only plane waves are propagated. Above, appears contribution of azimuthal mode with significant level differences between downstream microphones mounted on a same section.
Without flow, at 800 Hz, the local instantaneous intensity control can't reach the low level of active intensity naturally obtained with local pressure control. Consequently, the acoustic power remains higher, as average downstream pressure.

Above, to control local intensity instead of local pressure generates a pressure reduction higher donwstream (from 1 to 16 dB).
The difference between the two types of control is nevertheless low when the first transverse mode can be propagated.

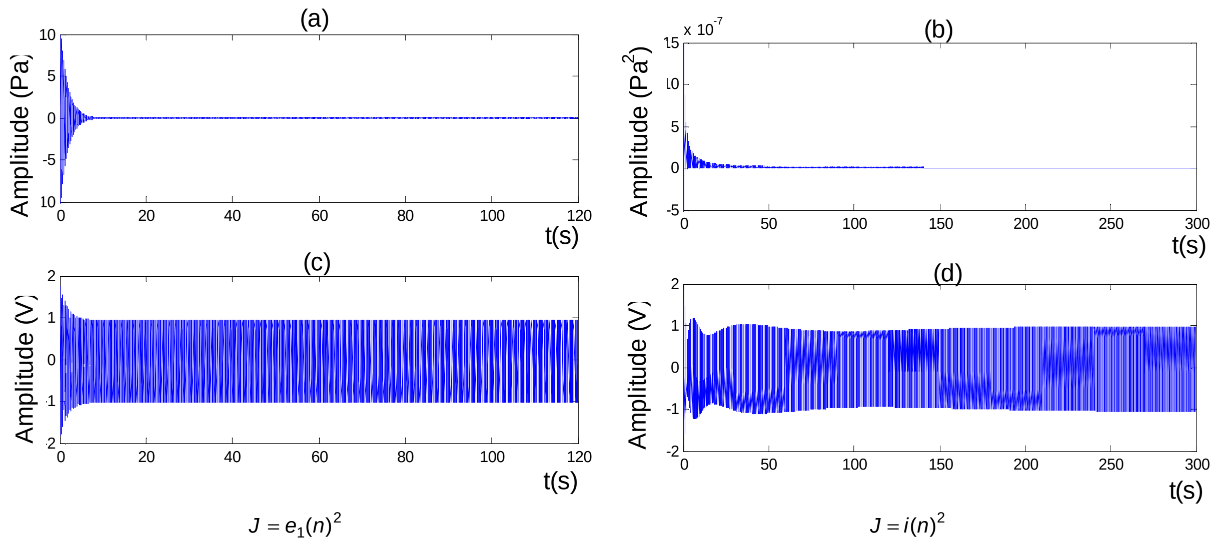

Figure 6: Time signals $e_1(n)$ and $i(n)$ (respectively (a) and (b)) / time signal $y(n)$ (respectively (c) and (d)) with "pressure" or "intensity" minimization at 800 Hz without flow ($M_0 = 0$)

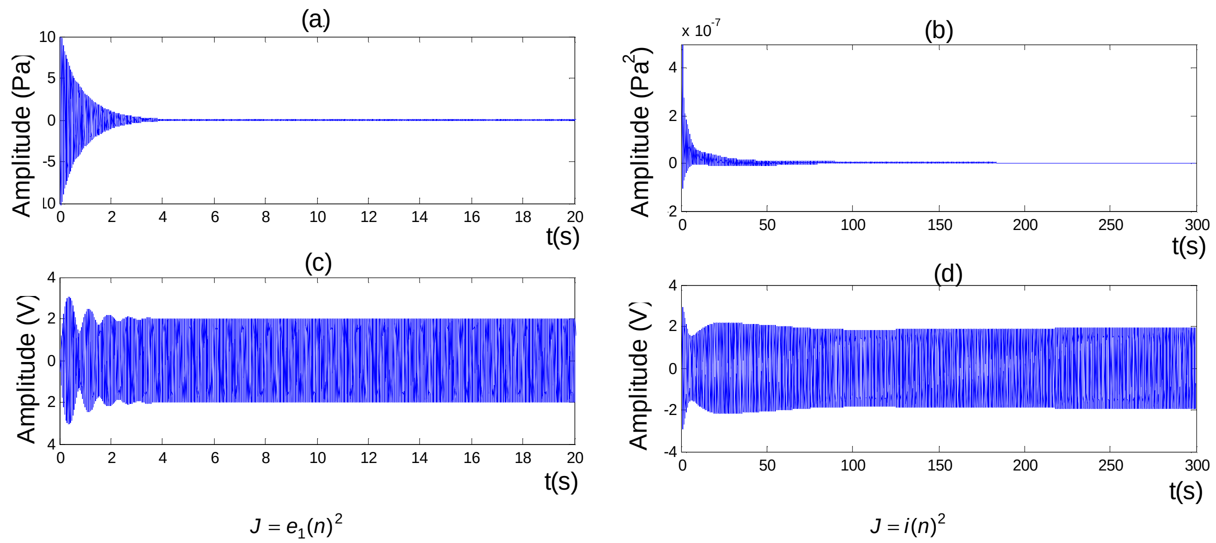

Figure 7: Time signals $e_1(n)$ and $i(n)$ (respectively (a) and (b)) / time signal $y(n)$ (respectively (c) and (d)) with "pressure" or "intensity" minimization at 1300 Hz without flow ($M_0 = 0$)

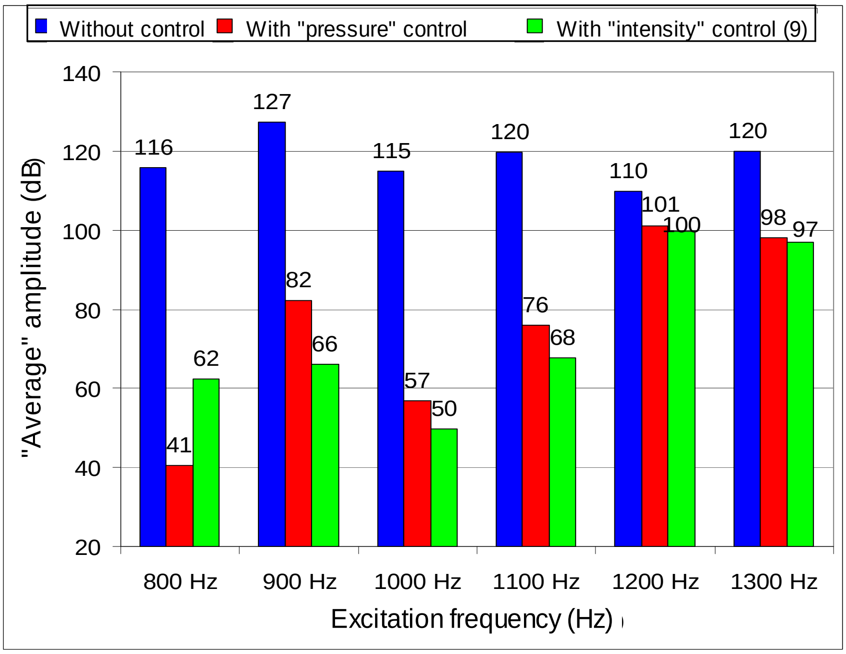

Figure 8: Average pressure level (dB) without and with control, without flow ($M_0 = 0$)

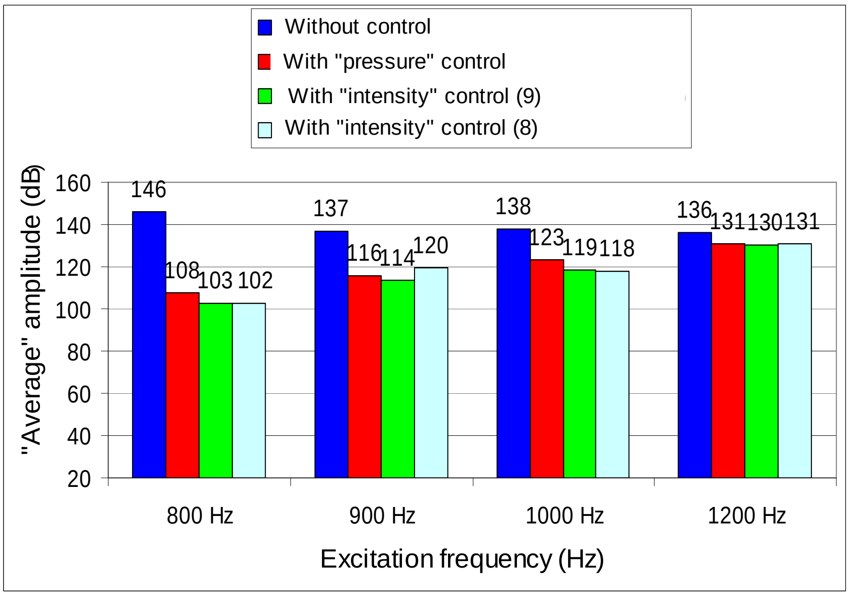

Figure 9: Average pressure level (dB) without and with control, with flow ($M_0 = 0.15$)

With flow ($M_0 = 0.15$), are tested two types of cost function relative to (8) and (9), to consider or not contribution of flow in intensity formulation.

Presence of turbulent flow is responsible for an "inaccurate" identification of transfer functions before control, a low ratio "signal/noise" and so on, a reduced effect of active control process (in some cases, for instance, at 1100 and 1300 Hz, active control is not applicable).

One can notice (figure 9) that contribution of flow in (8) is not preponderant for this low Mach number and can even generate a lower reduction because of interaction of all terms (because of measurement errors or computational errors).

Nevertheless, when active control can be applied, intensity cost function gives better results than pressure cost function (from 1 to 5 dB), with an effective reduction of active intensity.

4. CONCLUSIONS

A new FXLMS algorithm suited to control "instantaneous" acoustic intensity in presence of flow has been presented. Some experiments of virtual control have been led from data acquired in an aero-acoustic test bench dedicated to control process. The control process has generated a minimization of time average intensity (active intensity) representative to power flow. We have generally noticed a positive effect of this type of concept in comparison with local reduction of pressure. Unfortunately, interest of flow contribution in used formulation has not really been proved because of a low Mach number (0.15). It will be necessary to achieve new simulations from existing data or to realize new experiments for higher Mach number to settle.

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