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Noise Reduction in a Tractor Cab using resonators and active loudspeakers: FEM prediction and experimental verification

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ABSTRACT

In this study a newly developed 3D acoustic FEM model has been employed: it is capable of predicting the sound pressure level (Modulus & Phase) in each grid node inside the cavity, given the acoustic impedance of the boundary surfaces and the velocity shapes of the moving parts that radiate the noise.
Some noise reduction techniques have been tested by this FEM model. Two resulted very interesting: resonators accorded to the first natural frequency of the cab (obtained by vibrating panels and/or Helmoltz resonators), and an active loudspeaker system driven by a signal obtained from the acoustic input in the cab.
The accuracy of such noise reduction estimates has been confirmed through experiments conducted in a 1:1 model of the cab, built with steel and wood panels.
The sound reductions obtained (6 dB with resonators, and 8 dB with active loudspeakers) suggest that these two techniques can be a cost effective approach for reducing the low frequency noise in tractor cabs and in other similar enclosures.

0. Introduction

The noise in tractor cabs is usually higher than in other vehicle compartments, and its reduction is problematic due to the noise spectrum dominated by low frequency peaks. Furthermore, the numerical estimate of the acoustic behaviour of these enclosures is difficult, because the shape cannot usually be represented by a two dimensional FEM mesh.

Fig. 1, shows the pressure spectrum measured at the driver's ear in the cab of a tractor very similar to the one employed for this study. The harmonic structure of the noise is clearly visible, due to the fundamental ignition frequency and its harmonics. Some of them reach very high levels, because their frequency matches a cavity resonance. At least two peaks in fig. 1 show these effects, namely at 132 and 189 Hz.

Among many sound reductions techniques, only a few are suitable to this case: in fact, sound insulating panels give a poor insertion loss at low frequency, as they obey approximately the Mass Law. Also sound absorbing materials, obtained with soft tiles and foams, are ineffective at low frequency, and their capability to control the acoustic modes of the cavity is limited.

In this paper two control techniques have been tested: resonators (obtained both with vibrating panels and Helmoltz cavities) and Active Noise Control with loudspeakers. The latter is usually implemented through a very sophisticated system, built around a microprocessor controlled adaptive filter, with primary signal inputs near the noise source and reference microphones to evaluate the effect of the noise reduction system. Such an approach is very expensive, and is not actually realizable for mass-production. In this work a more simple system is suggested, based on low cost analog electronics, capable of controlling only one or two low frequency acoustic modes of the cab. It can be seen that the same job can be performed through passive resonators, properly tuned to those modes.

However, while the passive systems (resonators) are capable of reducing the amplitude of the acoustic resonances in the whole cavity, a properly tuned Active Cancellation system can move the nodal surface of the acoustic mode in space, positioning it over the ears of the driver, producing larger level reduction in this zone, but increasing the levels in other parts of the cab.

To obtain a quantitative evaluation of such effects, two testing techniques have been employed: a Finite Element computerized code (FEM), recently developed by the authors, and experimental tests conducted in a true-scale model of the cab. At the time of experimentation, a true tractor cab was not available, because the shape chosen for this study was that of a new tractor, and the first prototypes became available much later. Furthermore, this study shows that a reasonably good prediction of the acoustic behaviour of a vehicle compartment can be obtained at the first design stage, when shape or structure modifications to the project are still possible.

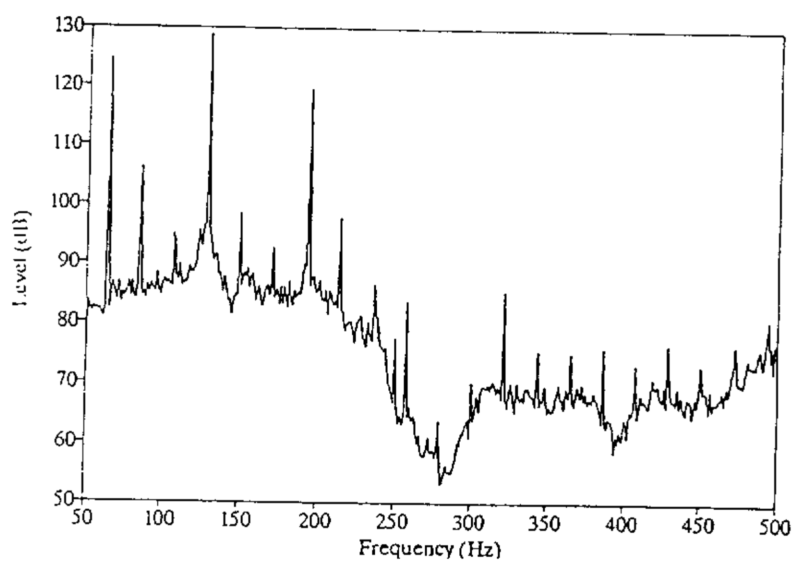

Fig. 1 - Noise spectrum in a tractor cab

1. Finite Element Mathematical Model

The formulation of the finite element method for time harmonic acoustic wave propagation is well documented [1,2]. The method produces a matrix equation of the form:

$$\left[[K] - \left(\frac{\omega}{c}\right)^2 \cdot [M] + j\omega \cdot [D] \right] \cdot \{p\} = \{f\} \tag{1}$$

where [M] is the acoustic "mass" matrix, [D] is the acoustic "damping" matrix, [K] is the acoustic "stiffness" matrix, ω is the angular frequency, c is the sound speed, {p} is the vector of acoustic pressure at the model grid points and {f} is the vector modelling the acoustic excitation mechanism (in this case boundary velocities).
The assembly of matrix M and K reflects some common assumptions: firstly it is possible to build the local matrices for each element in the grid, based on the shape function vector N:

$$[M]_e = \int_V \{N\} \cdot \{N\}^T \, dV \tag{2}$$

$$[K]_e = \int_V [\nabla N] \cdot [\nabla N]^T \, dV \tag{3}$$

in which $[\nabla N]$ can be espressed as

$$[\nabla N] = \begin{bmatrix} \frac{\partial N_1}{\partial x} & \frac{\partial N_1}{\partial y} & \frac{\partial N_1}{\partial z} \\ \frac{\partial N_2}{\partial x} & \frac{\partial N_2}{\partial y} & \frac{\partial N_2}{\partial z} \\ \frac{\partial N_3}{\partial x} & \frac{\partial N_3}{\partial y} & \frac{\partial N_3}{\partial z} \end{bmatrix} \tag{4}$$

The eqs. 2 and 3 are obtained from volume integrals taken over the volume V of each element. Furthermore, over the boundary surfaces, one can compute the terms of D and f from the surface shape vectors N_s:

$$[D]_e = \rho_o \cdot \int_S \frac{\{N_s\} \cdot \{N_s\}^T}{Z} \, dS \tag{5}$$

$$\{f\}_e = -j\omega\rho_o \cdot \int_S u \cdot \{N_s\} \, dS \tag{6}$$

where Z is the normal specific acoustic impedance of the boundary (absorbing) surfaces, and u is the normal velocities of the surfaces that are moving (radiating sound into the cavity).
Choosing a very simple element shape and interpolating function makes the evaluation of the former integrals easy. The numerical code used in this work is based on tethraedrical elements with only four nodes, and linear interpolation through both the element volume and the boundary surfaces. In this way the following expressions were found, for the generic element

contributions to the global matrices:

$$[K_{i,j}] = \frac{N_1(i)\cdot N_1(j)+N_2(i)\cdot N_2(j)+N_3(i)\cdot N_3(j)}{36\cdot V} \tag{7}$$

$$[M_{i,j}] = \begin{cases} \frac{V}{20} & (i \neq j) \\ \frac{V}{10} & (i=j) \end{cases} \tag{8}$$

$$[D_{i,j}] = \begin{cases} 0 & \text{(nodes i or j not on the boundary)} \\ \frac{\rho_o}{Z}\cdot\frac{A}{12} & \text{(nodes i and j on the boundary, } i\neq j) \\ \frac{\rho_o}{Z}\cdot\frac{A}{6} & \text{(nodes i and j on the boundary, } i=j) \end{cases} \tag{9}$$

$$\{f_i\} = -j\omega\rho_o\cdot u\cdot\frac{A}{3} \qquad (i^{th} \text{ node on the surface}) \tag{10}$$

where V and A are respectively the volume and the boundary surface area (if any) of the element, the indices i and j run from 1 to 4 (the vertices of the tethraedron), and the interpolating area functions N_1, N_2 and N_3 are expressed as follows:

$$\begin{aligned} N_1(i) &= \left(x_{i+1}\cdot y_{i-1}-x_{i-1}\cdot y_{i+1}+x_{i+2}\cdot y_{i+1}-x_{i+1}\cdot y_{i+2}+x_{i-1}\cdot y_{i+2}+x_{i+2}\cdot y_{i-1}\right)\cdot(-1)^{i+1} \\ N_2(i) &= \left(y_{i+1}\cdot z_{i-1}-y_{i-1}\cdot z_{i+1}+y_{i+2}\cdot z_{i+1}-y_{i+1}\cdot z_{i+2}+y_{i-1}\cdot z_{i+2}+y_{i+2}\cdot z_{i-1}\right)\cdot(-1)^{i+1} \\ N_3(i) &= \left(z_{i+1}\cdot x_{i-1}-z_{i-1}\cdot x_{i+1}+z_{i+2}\cdot x_{i+1}-z_{i+1}\cdot x_{i+2}+z_{i-1}\cdot x_{i+2}+z_{i+2}\cdot x_{i-1}\right)\cdot(-1)^{i+1} \end{aligned} \tag{11}$$

The global matrices can then be assembled by adding the contributions of the single elements together: this process is consistent with the enforcing rules of compatibility and continuity over the whole model volume.
The matrix equation (1) can be solved for a given excitation angular frequency ω and a prescribed boundary velocity vector {u}. In general, all the matrix variables are complex: so complex velocities can be input, meaning that parts of the boundary can move with different relative phase. The boundary surface impedance Z is also usually complex. Consequently, also the unknown nodal pressures {p}, obtained from the solution of the linear equations system (1), are complex quantities, which show a variable local phase shift in respect to the excitation.
The prediction of the acoustic response of the cavity is possible repeating the solution for different excitation frequencies, constructing the response spectrum with the desired frequency resolution.
Usually, once the matrices have been assembled, a preliminar eigenvalue problem is solved, obtained by removing the forcing vector in the right part of eq. (1), and regarding the angular frequency ω as the variable. In this way the eigenvalues $\omega_1,\omega_2,..\omega_n$ are found, and the corresponding eigenvectors {p} can be computed, showing the modal shape of the resonance in the cavity. In this case the absolute value of the pressure is meaningless, and is usually adjusted to fit the range ±1. If the matrix [D] is non-zero, the eigenvalues and eigenvectors become

complex, making more involved the numerical solution of the problem [3]: because usually the presence of absorbing material is not capable of modifying substantially the resonance frequencies and the modal shapes, in this work the matrix [D] has been neglected in the eigenvalue problem solutions, making it possible to use a standard library subprogram. Instead the "Damping" matrix was considered in the subsequent analysis, concerning the cavity response to forcing actions, because in this case the absolute value of the acoustic pressure is important, and it depends strongly on the boundary absorption.

2. Numerical and Physical Models of the cavity.

The tractor cab that is the object of this work has been numerically modelled starting from the design drawings, showing the main sections of the structure. A three dimensional surface model of the cavity was first created using AUTOCAD: fig. 2 shows such a model, both as a wireframe view and as an opaque one.

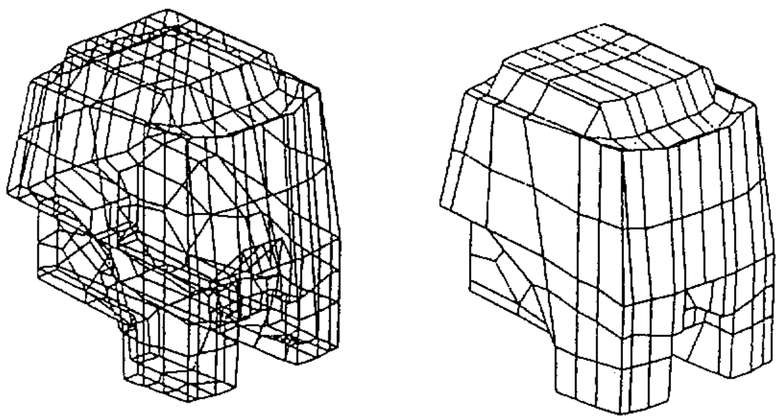

Fig. 2 - Autocad (TM) model of the cab surfaces.

The internal volume was then subdivided in esaedra. Each esaedron can be broken down into 5 tethraedra, as demonstrated in fig. 3. Such a breakdown is automatically performed from the computer code, which reads the Autocad DXF file directly.

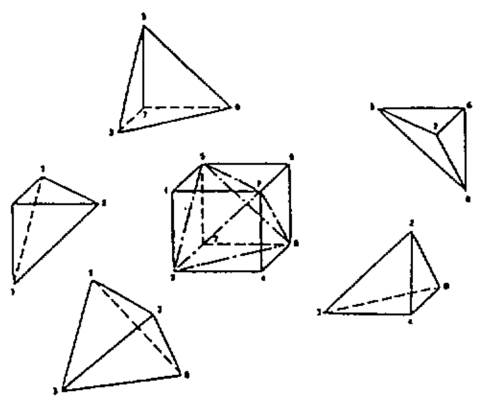

Fig. 3 - Breaking down an esaedron into 5 tethraedra.

The resulting grid was constituted by 252 esaedra, with 437 nodes. The average node-to-node distance is 167 mm: because this distance must be at least 8 times shorter than the wavelength, the maximum frequency is limited to about 250 Hz.
The walls of the cab are modelled with a frequency-independent impedance of 30000 rayls (steel and glass parts). An attempt was made, covering all the steel surfaces with an absorbing coat, the impedance of which was measured in an impedance tube: the Sound Pressure Levels were reduced by less than 0.3 dB, so the constant impedance boundary condition was maintained for all subsequent tests.
After obtaining the eigenvalue solution for the undamped, unforced problem, and having found the

resonance frequencies of the modes that probably cause problems, two sound reduction techniques were modeled with the aim of reducing the amplitude of the third resonance, occurring at 152 Hz: a vibrating panel, and active loudspeakers.

The vibrating panel, placed on the roof of the cab, is constituted of a rigid sheet having a surface density σ of 2 kg/m^2, backed with a soft poliurethane layer, having a unit surface compliance k_w of about 456000 m^3/N. A certain amount of damping can be expected from the poliurethane, so at the resonance frequency the surface impedance of the panels does not fall to zero, but tends to a constant value that has been assumed to be 715 rayls. At other frequencies, the imaginary part of the impedance can be computed as:

$$IM[Z] = j\omega\sigma + \frac{k_w}{j\omega} \tag{12}$$

while obviously the real part remains constant at 715 rayls.

The active loudspeaker system was simulated by changing the boundary condition over a part of the roof from finite impedance to prescribed velocity. The moving part was fed with a velocity input having an amplitude and phase varying with frequency, following the typical behaviour of a narrow-band filter, tuned to the proper resonance frequency. The overall gain and phase were adjiusted by trial and error to the optimum values: 0.22 mm/s for the peak velocity and 15° for the phase shift (the primary source was fed with 10 mm/s and no phase shift).

In parallel to such numerical evaluations, a true-scale model of the cab was built, as shown in fig.4. The structure is made in steel profiles and plates, and resembles as closely as possible the designed structure of the real-world cab. The glass parts were substituted by conglomerate wood panels, having sufficient thickness to give a flexural stiffness comparable to that of glass panels. Obviously the dynamic response of such a structure is far from that of the true cab, but for this experiment, involving only cavity effects, it is enough that the enclosure shape is reproduced.

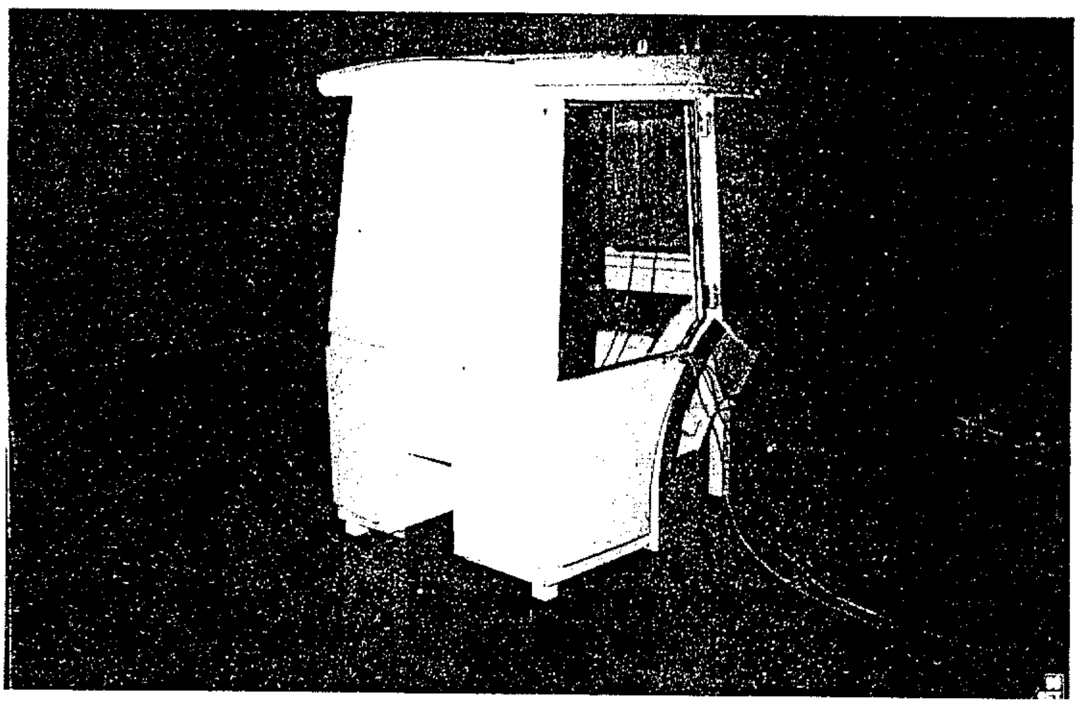

Fig. 4 - True-scale physical model of the tractor cab.

The same investigations have been performed on the physical and numerical model. First the acoustical modes of the cavity were found, exciting the volume with a small loudspeaker, and sampling the sound pressure response in a grid of points. The sampled points lie on five straight lines, intersecting in two points corresponding to the centre of the driver's head in standard driving position and in reversed position (the latter is possible when working with tools attached to the back of the tractor). One line is longitudinal, two are vertical and two

are transversal.
Afterwards, the acoustic frequency responses were measured at the driver's ear in both standard and reverse position, exciting the cab with a sound source placed roughly in the same place where the numerical excitation section was chosen.
The effects of two sound reduction techniques were detected: Helmoltz resonators accorded to the third resonance (158 Hz) at the driver's ear in reverse position, and active noise cancellation by loudspeakers of the fourth resonance (189 Hz), with the driver in standard position.
Helmoltz resonators were built inserting two flexible pipes in the cavedium existing in the roof structure: these pipes ended with a neck, whose length was experimentally accorded to give a resonance frequency of 158 Hz. Two pipes were placed, with their openings located at the front upper corners of the cab.
The active noise control system was constructed with very poor equipment: a microphone to sample the signal near the source, a narrow band filter tuned to the resonance frequency, and a small power amplifier connected to two car-audio woofers. No delay or phase adjustment was possible with this system. Nevertheless, it was possible to tune the cancellation region simply by moving the speakers inside the cab, making the path length right to obtain the wanted 180° phase shift between primary and cancellation signals at the driver's ears. As the 4th resonance is symmetric, and has two non intersecting nodal planes, the primary excitation region is out-of-phase with the position were the A.N.C. speakers were placed (roughly the same as the resonator mouths). For this reason, it was very difficult to produce acoustic feedback in the cab, and the system worked with good stability even in the absence of dedicated control systems.

3. Results

The numerical and experimental results are only partially comparable, because the excitation points and sound reduction techniques are slightly different. However, the comparison between experimental and numerical resonance frequencies show a very good agreement up to 250 Hz:

Resonance n.	Numerical freq. (Hz)	Experimental freq. (Hz)
1	117	116
2	150	146
3	152	158
4	190	189
5	212	not found
6	241	243

Also modal shapes are perfectly corresponding: for example, fig. 5 shows the comparison between the modal shape at 3rd resonance obtained experimentally and the numerical one. It can be seen that this is a symmetrical mode, with a nodal plane coming from the upper-rear corner of the cab and going down to the front-lower corner. The maximum Sound Pressure Levels are expected in the region occupied by the driver's head in reverse position.

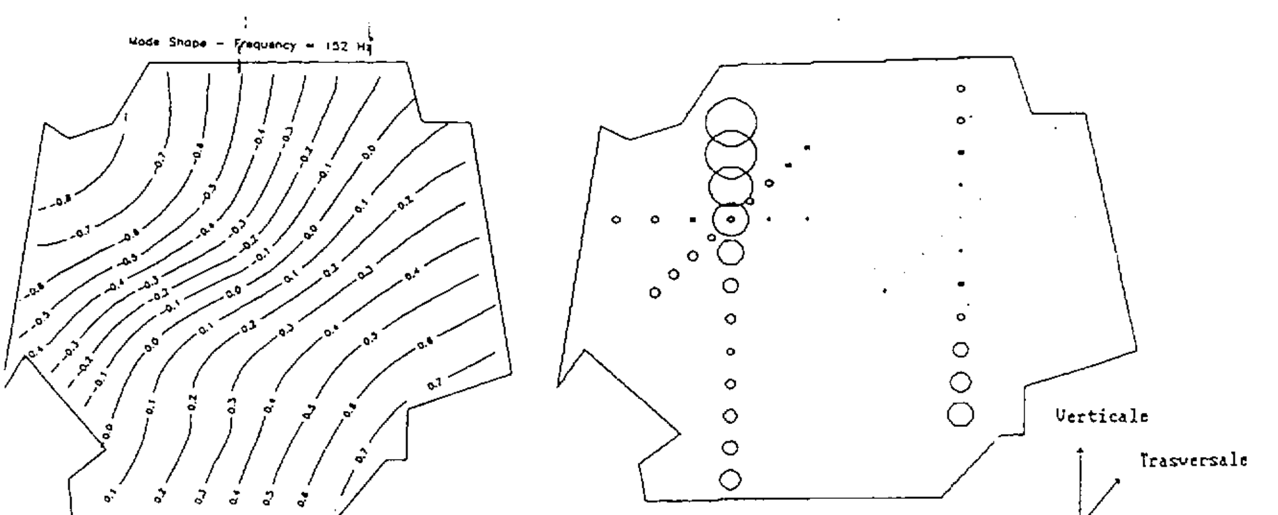

Fig. 5 - Modal shapes of 3rd resonance: 152 Hz (numerical), 158 Hz (experimental).

Also the frequency response of the cab without sound reductions shows a good agreement between numerical predictions and experiment, as reported in fig. 6; these responses were found in a point located near the vibrating floor.

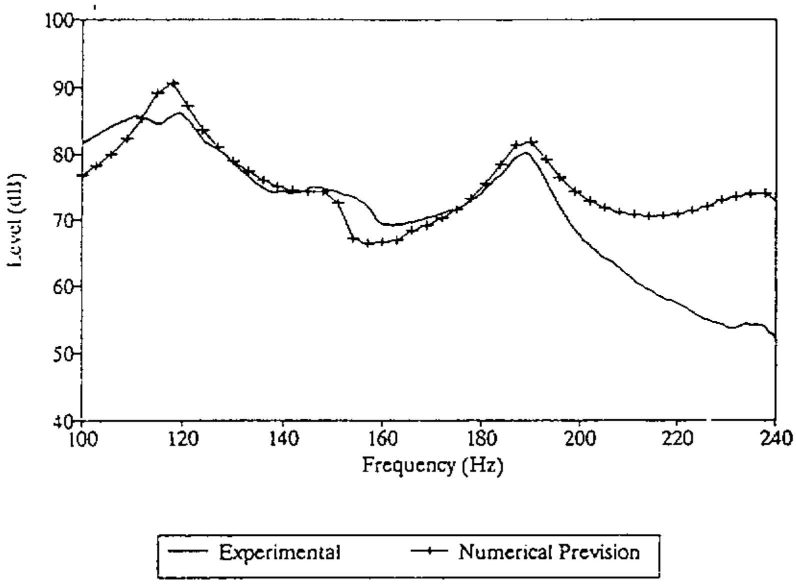

Fig. 6 - Comparison between frequency responses obtained by numerical and physical models.

Fig. 7 reports numerical simulations comparing the response of the untreated cab with those obtained with the vibrating panel and the active loudspeakers, at the dirver's left ear in reverse position. Although in the frequency domain these results appear similar (the A.N.C. simply gives a greater attenuation), the spatial effects of the two systems are different, as reported in fig. 8, which compares the SPL contour maps, at the resonance frequency. While the vibrating panel reduces the level everywhere, the A.N.C. increases it in the lower part of the cab.

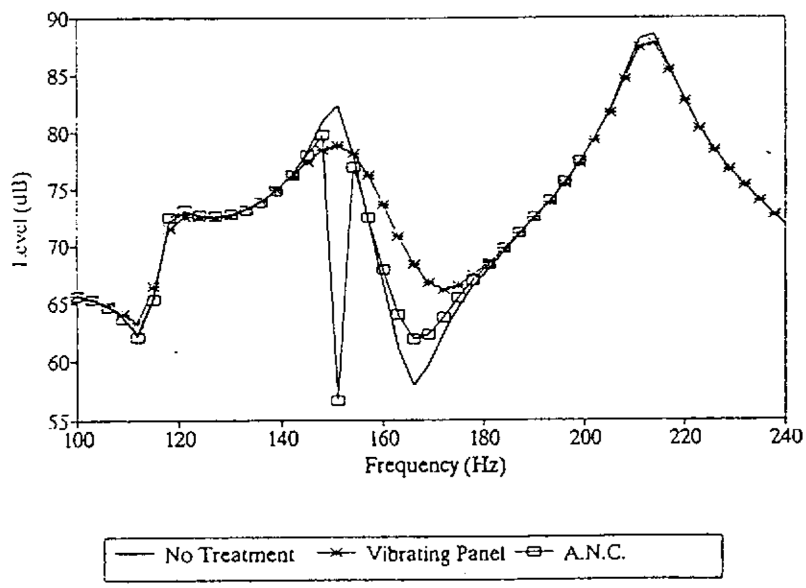

Fig. 7 - Numerical prevision of attenuation obtainable with a vibrating panel and with A.N.C.

Finally, figs 9 and 10 show the experimental noise reductions obtained with Helmoltz resonators and Active Noise control. In fig. 9 it can be seen that the resonators are capable of a good reduction at the tuned frequency (about 8 dB), but their response is too narrow, and the original peak split in two side-band peaks, whose amplitude is only 4 dB less than the previous one. However, there is no noise increment at any frequency.

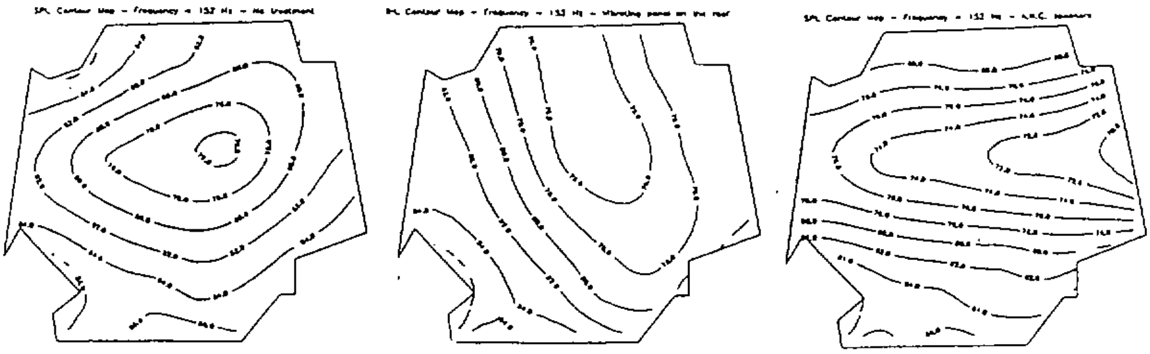

Fig. 8 - Comparison between S.P.L. Contour Maps - Numerical prevision at 152 Hz.

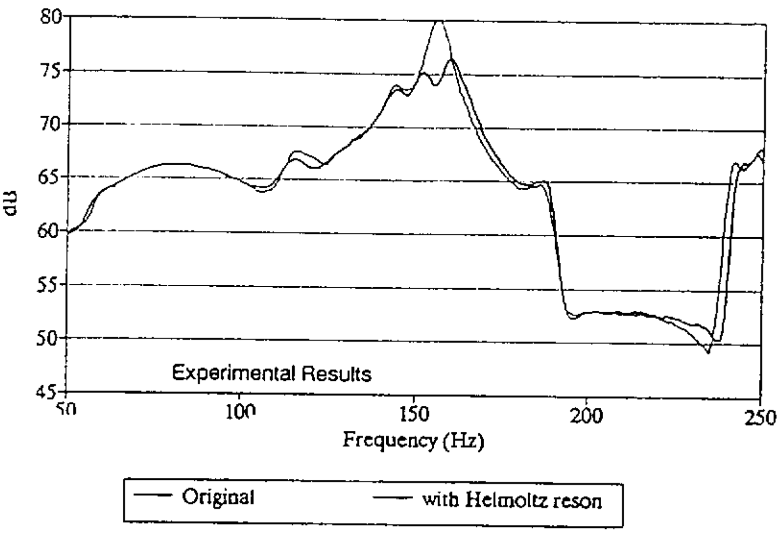

Fig. 9 - Experimental frequency response of the cab:
effect of Helmoltz resonators at reversed driver's right ear.

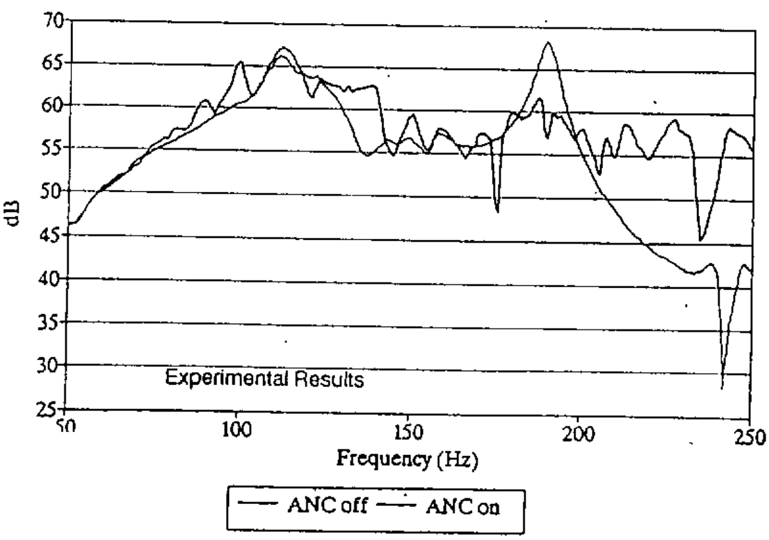

Fig. 10 - Experimental frequency response of the cab:
effect of Active Noise Control at forward driver's right ear.

Fig. 10 shows that at the tuned frequency the A.N.C. is capable of reducing the noise by more than 8 dB, without spurious side-band peaks. But at higher frequency, a consistent level increase was observed, up to 15 dB at the 6th resonance (243 Hz). This effect was caused by the insufficient filter rejection of high frequencies, accompanied by uneven speaker response; furthermore, the high power required to match the primary source sound level caused distortion both over the amplifier and the speakers.

4. Discussion and conclusion

Two reduction techniques have been tested on a tractor cab: a resonator and an active noise control system. Both the techniques have been investigated by a numerical FEM model and by a 1:1 physical model of the tractor cab. These two sound reductions techniques have been implemented with very cheap components, so they can actually be used for mass-production.
The resonators are obviously the cheapest: the only problem is their volume occupation in the roof structure, where many important components have to be installed. Their acoustic effect is safe, reliable, and can be accurately adjusted to match the cab response. Furthermore, there is no risk of level increase.
A pure Helmoltz resonator has been shown to be too selective, giving absorption only in a narrow frequency band; probably a partial filling with mineral wool can broaden the response, reducing however the attenuation.
The vibrating panel has shown a better frequency response, and probably wastes less space than the previous one. The problem is that this system is not easily tuned, and slight changes in the properties of the soft layer backing the panel can shift the resonance frequency at values where it gives a very poor contribution to the overall A-weighted level. This system requires further experimental investigation before real-world applicability.
Finally, the Active Noise Control system presented here shows performances very far from the better digital-processing units, developed in the last few years. However, if properly tuned, this low cost system is capable of giving significant (audible) reductions, and can be incorporated in a sound diffusion system. The only problem is with the signal pick-up. In this work a microphone was used, but on a tractor, it is better to sample a sync signal directly from the engine which excites the structure of the cab. Regarding this point, only experiments conducted on real-word tractors should show what is better.
However the prediction techniques employed for this work showed that it is possible to accurately predict the acoustic response of cavities by numerical codes, and that a 1:1 scale model can be build in the very early stage of vehicle development. For these reasons, the combined numerical-experimental technique should be largely employed, in years to come, for the acoustic optimization of both cavity shape and noise reductions techniques.

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