Response of a non-baffled sandwich panel submitted to a reverberant chamber acoustic environment

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1- INTRODUCTION

It is well known, in the aerospace industry, that the acoustic pressure levels generated by the rocket launchers are extremely high. This strong acoustic pressure specificaly excites the payload in which a satelitte is located. Then, the satellites manufacturers have to submit their structural components to standardized tests in a reverberant chamber. We are then left with a situation where a panel is hung in a large reverberant room and immerged in a so-called diffuse field.. The excitation spectrum and levels are specified and measurements of acceleration levels induced in the structure are done. It leads to a vibroacoustical problem that presents many interesting challenges: (i) the panel is non-baffled (ii) the panel is excited on both sides by the acoustic field (iii) the coupling between the panel and the fluid has to be treated rigourously in order to obtain the pressure jump function across the panel (iv) the proposed formulation must take care of an extremely high modal density in the cavity, even at low frequencies.

In this paper, we are first presenting a new semi-analytical formulation to predict the behaviour of a non-baffled flexible plate in a rigid-walled rectangular cavity excited by an acoustical source. The second part of the paper presents some numerical results and discussions about the contribution of physical effects.

2- SEMI-ANALYTICAL FORMULATION

Consider a rigid-walled rectangular room (fig.1) of dimensions L_x, L_y, L_z, which contains a plate of surface $S_p=L_x \times L_y$ and enclosing a fluid of volume V. It is well known, using integral equation method, that the sound pressure P at any point point in V can be written

$$P(\vec{r}_0) = \iint_{S_S} dS\, G(\vec{r},\vec{r})\vec{\nabla}_{\vec{r}} P(\vec{r}) - \iint_{S_p} dS\, \overline{P}(\vec{r})\vec{\nabla}_{\vec{r}} G(\vec{r},\vec{r}_0)\cdot\vec{n}_p \qquad (1)$$

where S_S denotes the surface of the of the piston and S_p the surface of the panel. The quantity $\overline{P}$ represents the sound pressure jump across the surface S_p while $G(\vec{r},\vec{r}_0)$ is the Green's function for the empty room. The first term of (1) represents the contribution of the source while the second term is the contribution of the plate to the sound pressure. By coupling the equation for the motion of the plate and the boundary condition between the fluid and the plate at the interface S_p with equation (1), the problem is completely and rigourously defined.

A: Green's function
Instead of expanding the Green's function over modes of cavity as it is usualy done, the function is expanded over a set of two-dimensionnal functions. It reduces the number of summations by one, a great feature for computer calculations. Such an expasion is written in the following form

$$G(\vec{r},\vec{r}_0) = \sum_{tu} g_{tu}(z,z_0)\, \psi_{tu}(x,y)\psi_{tu}(x_0,y_0) \qquad (2)$$

where $g_{tu}(z,z_0)$ is a discontinous function at $z=z_0$. The basis functions ψ are just cosines along x and y and satisfy Neumann boundary conditions on the walls of the cavity.

B: Sound pressure jump
To compute the contribution of the plate to the pressure, we first expand the sound pressure jump function over the same basis functions used for the Green's function. One then write

$$\overline{P}(x,y) = \sum_{rs} \overline{P}_{rs}\, \psi_{rs}(x,y) \qquad (3)$$

where $\overline{P}_{rs}$ are unknown coefficients.

C: Fluid-structure coupling
The fluid-structure coupling is being taken into account in two parts. We first introduce the classical boundary condition at the interface S_p which states that the velocity of the fluid must be equal to the velocity of the plate on S_p in order to write

$$-\frac{1}{j\omega\rho}\frac{\partial P}{\partial z}\bigg|_{z=z_p} = -j\omega\, w(x,y) \quad \text{on } S_p\ , \qquad (4)$$

where w(x,y) is the deflection of the plate.

Since we have an acoustic excitation, we have a to consider the strong coupling limit in order to include *completly* the motion of the plate in the fluid. This motion is related to the sound pressure function, which is in fact the excitation force on the plate, by the following relation

$$\overline{P}(x,y) = Z(w(x,y)) \quad \forall\ (x,y) \in S_p \qquad (5)$$

where Z is an operator that represents the mechanical behaviour of the plate. If we expand the deflection of the plate over *in-vacuo* modes of the panel, introducing a set of unknowns b_{mn},

$$w(x,y) = \sum_{mn} b_{mn}\, \varphi_{mn}(x,y)\ , \qquad (6)$$

we can compute the operator Z and obtain

$$Z_{mnpq} = -\omega^2 M_{mnpq} + K_{mnpq}\ . \qquad (7)$$

The matrices **M** and **K** are respectively the mass and the stiffness matrices of the panel.

D: Linear system of equations
Inserting equations (1),(2),(3) and (6) in equations (4) and (5) and integrating over appropriate domains, we obtain a linear system of equations solvable with standard algorithm

$$\begin{bmatrix} SYS^{t}Z - \omega^{2} I \end{bmatrix}\begin{bmatrix} b \end{bmatrix} = \begin{bmatrix} -SW \end{bmatrix} \qquad (8)$$

where **S** is a change of basis matrix, **Y** a vector that contains the acoustic information of the problem and where **W** is the source vector.

2- NUMERICAL RESULTS

To ensure convergence of our method,we have established a simple geometrical criterion : *the smallest wavelength of the acoustic basis functions* , λ^{ac}_{min}, *must be less or equal to the smallest wavelength of the in-vacuo modes,* λ^{struc}_{min} . Figure 2 shows the case of $\lambda^{ac}_{min} < \lambda^{struc}_{min}$ (sub-critical) and $\lambda^{ac}_{min} \geq \lambda^{struc}_{min}$ (critical and super-critical).
Results for the sound pressure jump and velocity of the plate are shown in fig.3 for a 0.7×0.5 m^2 panel in a 2.6×2.0×3.0 m^3 cavity. It shows a stronger response of the plate at empty room modes frequencies comparatively to the response at *in-vacuo* eigenfrequencies.
The influence of the mechanical properties of the plate on the sound pressure jump is shown on fig 4. One can see the sound pressure jump remains unchanged for some values of the mass per unit area and flexural rigidity. It suggests the fact that the panel acts primarily as a if it is rigid so that the governing phenomena is the diffraction.

3- CONCLUSION

We have developed a novel semi-analytical approach that is able to to take care of many interesting features (i) acoustic excitation (ii) complete fluid-structure coupling (iii) non-baffled panel (iv) light and fast computer code. Preliminary results show good agreements with expected results and show the possibility to simulate aerospace applications such as the dynamic response of a composite panel with attached electronic equipments.

4- ACKNOWLEDGMENTS

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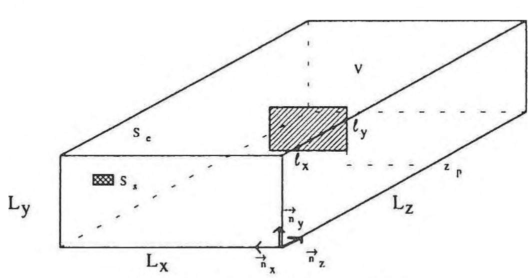

fig1 :Geometry of the problem

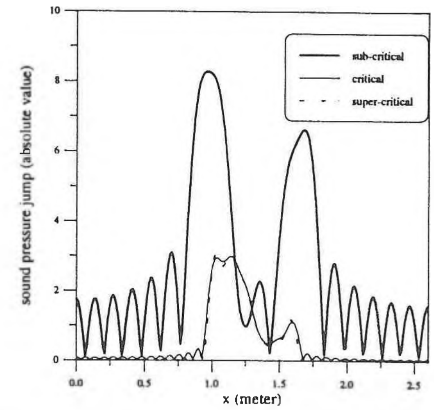

fig.2 : Sound pressure jump as a function of x (y being fixed) for three cases of λ^{ac}_{min}.

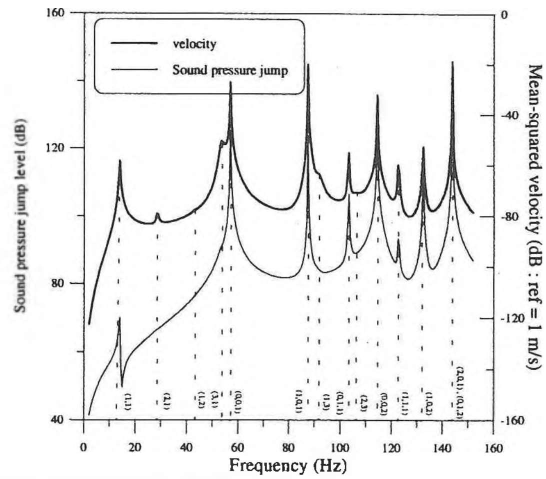

fig.3: Mean-squared sound pressure and velocity levels as a function of the frequency for a 0.7×0.5 m^2 panel. (ijk) denotes empty cavity modes and (mn) denotes *in-vacuo* modes

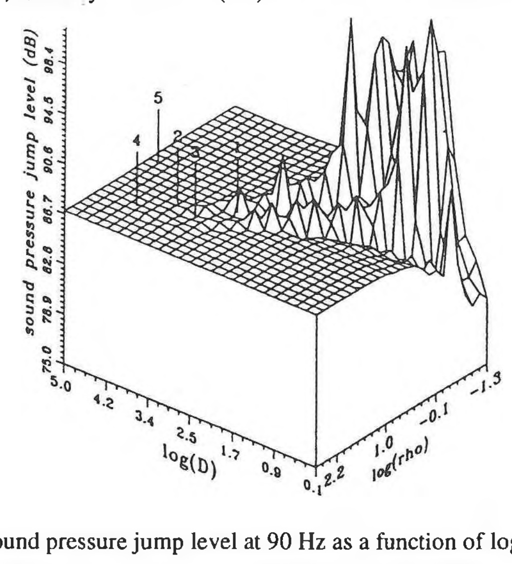

fig 4: Sound pressure jump level at 90 Hz as a function of log(ρ_S) and log(D) where ρ_S is the mass per unit area and D is the flexural rigidity. #1-#2 : 5 mm. and 1 cm. of aluminium, #3-#4 : 5 mm. and 1 cm. of steel, #5 : typical sandwich panel.