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THE INFLUENCE OF INTERNAL STRUCTURES ON THE SOUND RADIATED FROM A MACHINERY HOOD

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Abstract

An analytical model of shell is used to study its sound radiation under the influence of internal structure of high modal density. The numericals results show a smoothing of cylinder's velocity and radiation spectra that can be associated to an increased damping. The paper present the theoritical derivation of the method, in the case of a cylinder coupled to internal system described by point mobilities.

1. INTRODUCTION

A machinery is often composed of a hood radiating sound to the exterior and internal substructures (I.S.). Ignoring this substructures when calculating the sound radiated by the hood can lead to bad results, when compared to experiment. However taking into account the substructure in a determinist sense, is generally unrealistic due to their complexity and their hight modal density.

The goal of the paper is to describe the effect of I.S. in specified way based on their input mobilities, and present the basic effects on the sound radiation. In particular conditions, that will be described, the internal substructures can increase considerably the apparent damping of the hood, and then reduce the radiated sound.

2. RADIATION OF CYLINDER

Let us consider a finite cylindrical shell,terminated by infinite cylindrical rigid baffles and immersed in a fluid. The shell is excited by harmonic point force of pulsation ω.
The shell equation of motion is:

$$[L]\begin{pmatrix} u(Q) \\ v(Q) \\ w(Q) \end{pmatrix} + \omega^2 m \begin{pmatrix} u(Q) \\ v(Q) \\ w(Q) \end{pmatrix} = -\begin{pmatrix} 0 \\ 0 \\ F_w \end{pmatrix} + \begin{pmatrix} 0 \\ 0 \\ p(Q) \end{pmatrix} \quad (1)$$

where

- m is the shell mass per unit area

- u(Q), v(Q) and w(Q) are the longitudinal, tangential and radial displacement at point Q of the shell mid-surface.

- F_w is the radial component of driving force.

- p(Q) is the shell boundary pressure.

- L is the Donnell's operator.

The displacement for a shell with simply supported boundaries is sought by introducing the following expression

$$\begin{pmatrix} u \\ v \\ w \end{pmatrix} = \sum_{\alpha=0}^{1} \sum_{n=0}^{\infty} \sum_{m=1}^{\infty} \sum_{j=1}^{3} A_{nmj}^{\alpha} \begin{bmatrix} D_{nmj} sin(n\phi + \frac{\alpha\pi}{2}) cos(\frac{m\pi x}{l}) \\ E_{nmj} cos(n\phi + \frac{\alpha\pi}{2}) sin(\frac{m\pi x}{l}) \\ sin(n\phi + \frac{\alpha\pi}{2}) sin(\frac{m\pi x}{l}) \end{bmatrix} \tag{2}$$

where

- A_{nmj}^{α} is the modal amplitude of a shell mode of circonferentiel order n, longitudinal order m, type j and symmetry character α.

- l is the shell length.

- $(D_{nmj}, E_{nmj}, 1)$ are components of the eigenvector.

In the fluid, the acoustic pressure is governed by the Helmholtz's equation:

$$\nabla^2 p(M) + (\frac{\omega^2}{c^2}) p(M) = 0 \tag{3}$$

where ω is the angular frequency and c the speed of sound. At the interface between the baffled shell and the fluid medium of density ρ_0, the continuity of normal velocities requires

$$\begin{cases} \frac{\partial p(Q)}{\partial r} = \rho_o \omega^2 w(Q) & \text{on the shell} \\ \frac{\partial p(Q)}{\partial r} = 0 & \text{on the cylindrical baffle} \end{cases} \tag{4}$$

The pressure field solution can be expressed with the integral Eq.(5) (see LAULAGNET).

$$p(Q) = -\rho_0 \omega^2 \int_s \sum_{\alpha=0}^{1} \sum_{n=0}^{\infty} \sum_{m=1}^{\infty} \sum_{j=1}^{3} A_{nmj}^{\alpha} G(Q, Q_0) sin(n\phi + \frac{\alpha\pi}{2}) sin(\frac{m\pi x}{l}) ds \tag{5}$$

where $G(Q, Q_0)$ is the Green's function satisfying the Neumann boundary condition on the cylinder and the baffle.

The introduction of Eqs (2) and (5). into Eq. (1) yields after use of shell modes orthogonality properties:

$$M_{pqk}[\omega_{pqk}^2(1 - i\eta_c) - \omega^2] A_{pqk}^{\alpha} - i\omega \sum_{r=1}^{\infty} \sum_{j=1}^{3} Z_{pqr} A_{prj}^{\alpha} = F_{pqk}^{\alpha} \tag{6}$$

where

- M_{pqk} is the generalized mass of the shell mode.

- ω_{pqk} is the eigen angular frequency of the cylinder.

- η_c is the shell loss factor.

- Z_{pqr} is the acoustical radiation impedance of the cylinder modes, involving circonferentiel order p and logitudinal orders (q,m).
- $\tilde{F}^{\alpha}_{pqk}$ is the generalized force due to the mechanical driving force.

The determination of the modal amplitudes allows us to calculate the quadratic velocity $< V^2 >$ and the radiated power W of the shell with Eqs.(7) and (8).

$$< V^2 >= \frac{\omega^2}{4} \sum_{\alpha=0}^{1} \sum_{p=0}^{\infty} \sum_{q=1}^{\infty} \sum_{j=1}^{3} \sum_{k=1}^{3} \frac{A^{\alpha}_{pqj} A^{*\alpha}_{pqk}}{\epsilon_p} \tag{7}$$

$$W = \frac{\omega^2}{2} \Re(\sum_{\alpha=0}^{1} \sum_{p=0}^{\infty} \sum_{m=1}^{\infty} \sum_{r=1}^{\infty} \sum_{j=1}^{3} \sum_{k=1}^{3} Z_{pmr} A^{\alpha}_{pmj} A^{*\alpha}_{prk}) \tag{8}$$

where $\Re$ indicate the real part.

3. INTERNAL SUBSTRUCTURES

The Internal Substructures (I.S.) will be described in a simplifed way, using model of input mobility. To every I.S. is associated an input mobility of a beam like structure in longitudinal motion (Eq. (9)).

$$M_{internal} = \frac{-i}{S\sqrt{\rho E} \tan(kL)} \tag{9}$$

where S, E, ρ, L represent respectively the section, the Young's modulus, the density and the length of the beam. k, being the mechanical wave number, is expressed by:

$$k = \frac{\omega}{c} = \omega \sqrt{\frac{\rho}{E}}$$

We present in Fig. 1 the input mobility of I.S. in the case of High Modal Density (I.S.H.M.D.) (0.4 mode/Hz) and a damping loss factor $\eta = 10^{-2}$, and the shell input mobility. In this case, the two mobilities have approximately equal geometric mean, that significate impedance matching.

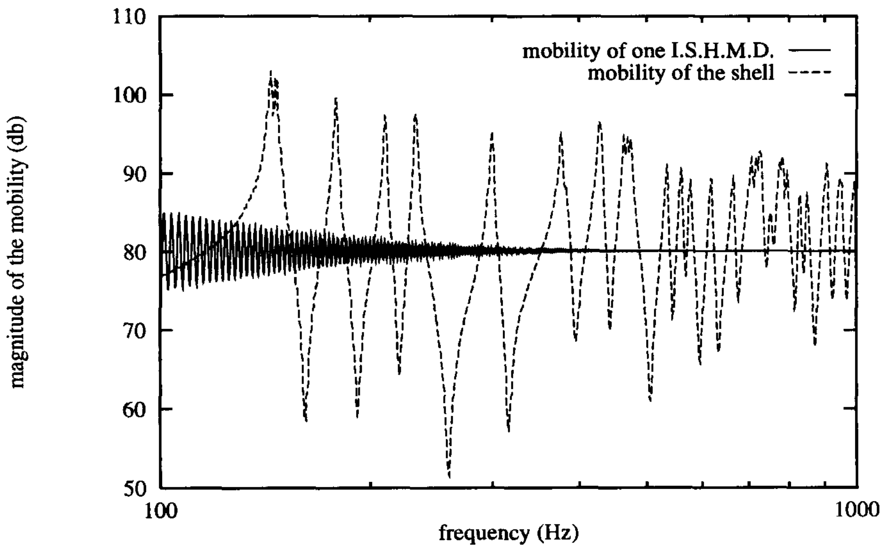

FIG. 1 - *mobility modulus of the shell and of one I.S.H.M.D.*

In a machinery, the I.S. are generaly not directely coupled to the shell, but are attached through a spring. The input mobility of the internal system is then given by Eq. (10):

$$M_{internal} = -i\omega \frac{ESk\tan(kl) + K}{ESk\tan(kl) \times K} \quad (10)$$

where K is the stiffness of the coupling spring.

We present in Fig. 2 the mobility modulus of one internal structure coupled with a spring between $10^4 kg.s^{-2}$ to $10^8 kg.s^{-2}$. When the spring stiffness decreases, the mobility modulus increases. So one can say that the shell motion will be free at the coupling points.

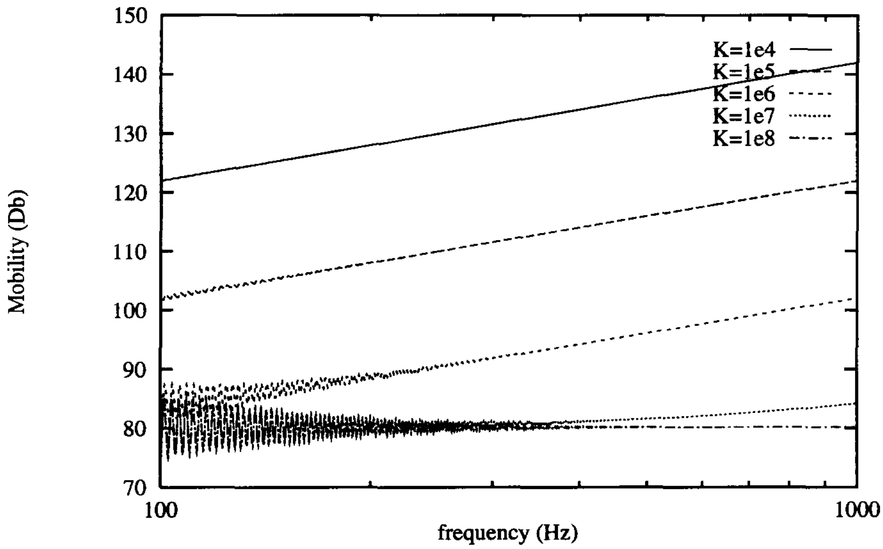

FIG. 2 - *Influence of the value of the spring on the mobility modulus of one I.S.H.M.D.*

4. COUPLING BY MOBILITY

The coupling of the shell and the internal structures is carried out using the mobility method, which consists of calculating the coupling forces, solving the linear system (11):

$$[\bar{\bar{M}}_{shell}(I,J) + \bar{\bar{M}}_{internal}(I,J)]\{\vec{F}(I)\} = \{\vec{V}_{internal}(I) - \vec{V}_{shell}(I)\} \quad (11)$$

where

- $F(I)$ is the coupling force at point I.
- $V_{internal}(I)$ is the uncoupled internal structure velocity at the coupling point I due to the excitation's force in the internal structure.
- $V_{shell}(I)$ is the uncoupled shell velocity at the coupling point I due to the excitation force applied to the shell.
- $M_{internal}(I,J)$ is the transfer internal structure's mobility between points I and J.
- $M_{shell}(I,J)$ is the transfer shell's mobility between points I and J. The expression of the mobility's matrix of the shell is the ratio between the radial velocity $V_{x_i\phi_i}$ and the driving force, at each coupling's point:

$$M_{shell}(I,J) = \frac{V_{x_i\phi_i}}{F_{x_j\phi_j}} \quad (12)$$

and is calculated solving the linear system (6), one per losation of a I.S.H.M.D..

The determination of $F(I)$ allows us to take into account the effects of the I.S. on the shell calculating the response to a driving force equal to directly applied force and coupling forces.

4.1 Position of the driving force

The driving force can be located either on the shell or in the internal system. In the mobility method, the driving force is taken into account by the determination of the uncoupled velocity of every structure before coupling. This velocities correspond to the velocities of each structure at the coupling's points, for a position of the driving force given. If the driving force is on the shell, the uncoupled internal structure velocity is zero, and vice versa.

In this paper only the case of excitation located in the internal substructures will be considered and we will firstly study the effect of distance of the driving force position to the coupling point. The driving force appear in the right-hand side of the equation (11), through the uncoupled volocity of every structure before coupling :$\tilde{V}_i^s$ et $\tilde{V}_i^c$. Taking again our assumption of I.S.H.M.D., of beam like structures vibrating longitudinally, one can obtain the expression (13)

$$\tilde{V}_i^S(0) = \frac{-iF_{exc.}}{s\sqrt{\rho E}} \times [\sin(kx_0) + \frac{\cos(kx_0)}{\tan(kL)}] \tag{13}$$

where the driving force of modulus $F_{exc.}$ is located at point x_0 and the coupling point is zero.

In Fig. 3, we show the uncoupled internal structure velocity modulus for three different positions of the driving force at the coupling point, in the midlle and at the opposite end of the beam.

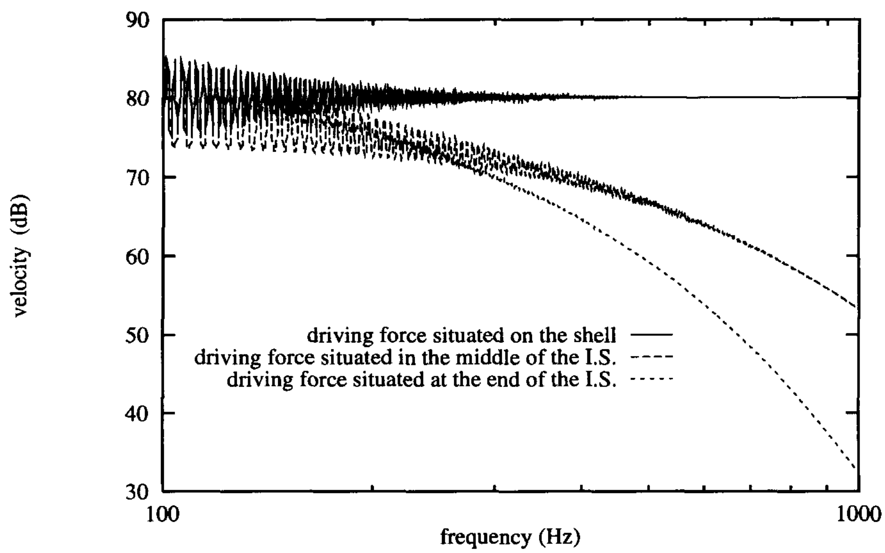

FIG. 3 - *Influence of the position of the driving force on the uncoupled velocity modulus of one I.S.H.M.D.*

Let us remark that the velocity is the same when the I.S.H.M.D. is terminated by a spring, and thus in this case only the mobility will be changed.

5. NUMERICAL SIMULATION

5.1 Direct coupling

In this case the I.S.H.M.D. is directly connected to the shell, or through a spring having a minimal value of $10^8 kg.s^{-2}$. Let us take the example of a baffled cylindrical shell, coupled with eight I.S.H.M.D. ditributed in different points of the shell. The internal structures are defined by the mobility given in Fig.1.

In Figs. 4 is presented the calculation of the quadratic velocity of the only cylindrical shell and of the shell coupled with I.S.H.M.D. for the three positions of the driving force given in Fig. 3.

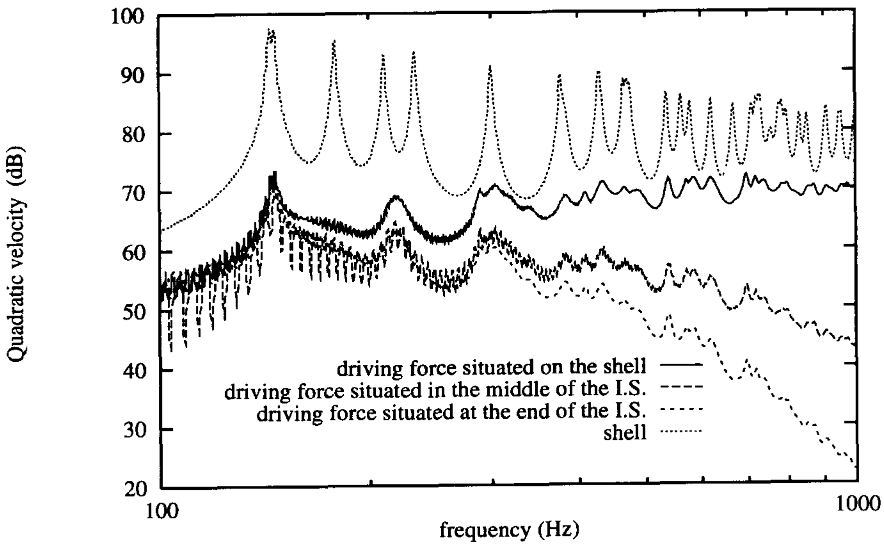

FIG. 4 - *Influence of the position of the driving force on the quadratic velocity*

One can see a smoothing of results, in every cases, between the empty shell and these coupled one. This effect is associated to the dissipation in the internal structures. Even if the loss factor of I.S. is weak, because of the great number of modes, the total power dissipated is big.

In addition, the influence of the position of the driving force in the I.S.H.M.D. is only rendered by a decrease of the level of the quadratic velocity versus the frequency, without changing its behavior. This decrease is as much higher than the driving force is far from the shell.

We show too, in Fig. 5, the acoustical behavior by presenting the power radiated, for the same cases as precedently.

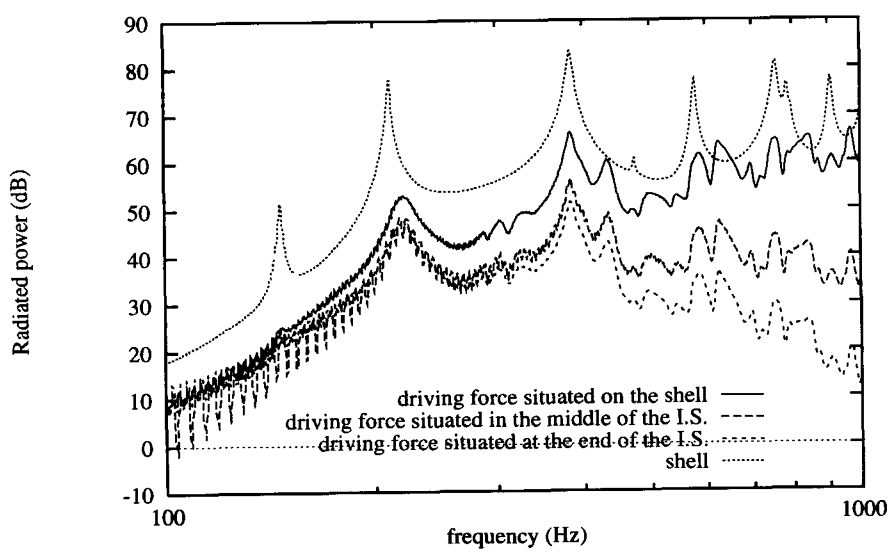

FIG. 5 - *Influence of the position of the driving force on the radiated power*

We can do the same observations as precedently about the effect of the I.S.H.M.D., that is a smoothing and a decrease with the distance of the driving force, excepted from 400 Hz where a dynamical appear. So the shell coupled radiate more than this empty one.

5.1 Non-direct coupling

In this case, the I.S.H.M.D. are not directely coupled to the shell, but are attached through a spring of stiffness $10^4 kg.s^{-2}$. The input mobility of the I.S.H.M.D. is higher than this one of the shell, hence we don't respect the condition of matching impedances of the preceeding paragraph, that is to say, point mobility of shell and I.S.H.M.D. having close geometrical average.

Fig. 6 shows the quadratic velocity of the only cylindrical shell and of the shell coupled with I.S.H.M.D. for the three positions of the driving force given in Fig. 3.

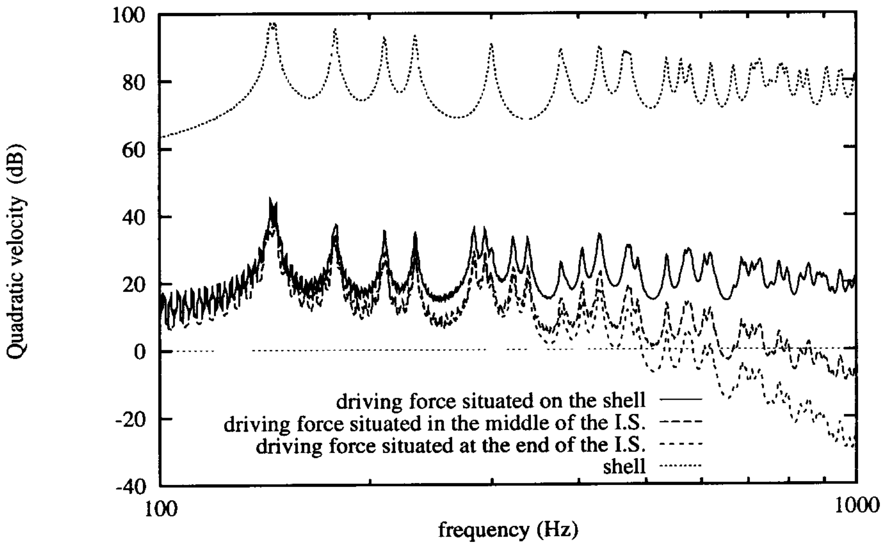

FIG. 6 - *Influence of the position of the driving force on the quadratic velocity*

As the condition of matching impedance is not respected, the effect is not an increased damping, but a little increase of the modal density. The driving force being in one internal structure, the presence of the spring leads to a big decrease in the quadratic velocity.

The same remarks can be done on the acoustical radiated power showing in Fig. 7.

In a machinery, when the I.S.H.M.D. are not directely coupled to the shell, but are attached through a spring, the phenomenom of smoothing can't appear by this way that is, a dissipation due to a transfert of the energie from the shell to the I.S.H.M.D., because of non respect of the matching impedance.

To put a spring between the shell and the I.S.H.M.D. is only interesting if the driving forces are in the I.S.H.M.D., what is generally the case in a machinery hood.

6. CONCLUSION

The smoothing of vibration and radiation behavior of structures containing I.S. has been evidenced theoretically and experimentally. It is necessary to have both matching impedance and high modal density, of I.S. compared to the structure.

These two conditions are not often encountered in structures and the phenomenom will be in general not observed.

When the internal structures are decoupled from the shell through a spring, then we lost the phenomenom of smoothing because the condition of matching impedance is not respected, moreover the level is very much lower.

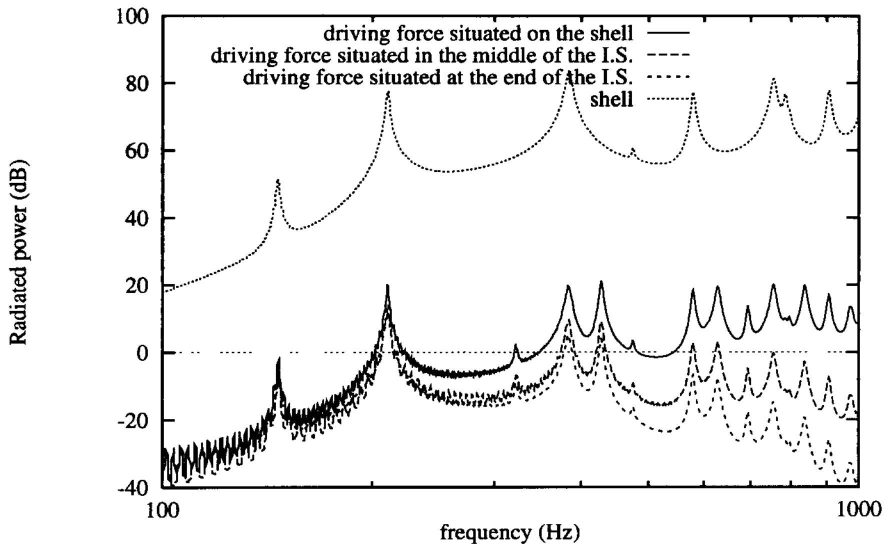

FIG. 7 - *Influence of the position of the driving force on the radiated power*

The position of the driving force influence only the level at high frequency but not the dynamical behavior.

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