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Transmission loss prediction of double panels filled with porous materials and mechnical stiffners

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ABSTRACT

Sound transmission through hollow structures found its interest in several industrial domains such as building acoustics, automotive industry and aeronautics. However, in practice, hollow structures are often filled with porous materials to improve acoustic properties without adding an excessive mass. However the necessary mechanical links betweenn external panels are also transmitting noise and the TL can be strongly decreased compared to the ideal panel without mechanical links . This paper aims to predict vibroacoustic behaviour of such real double panels by using the patch mobility method recently published [1]. Equivalent fluid is a basic tool for the description of porous material and is used here. The mechanical links are constituted of shells stiffners modelled by Finite Element in order to take into account their complexity. After a global overview of the theoretical method, the case of a double panel filled with mineral whool and coupling stiffners is studied for highlighting the influence of mechanical links.

INTRODUCTION

Transmission of sound through double panels has been studied since a long time, and several particular aspects were identified such as critical frequency, double panel resonance, and spatial coincidence effects. The work of London on infinite double plates [2] showed for example that appreciable increases in transmission loss could be produced by double panels with very shallow air spaces over a single panel. Double panels, and more generally hollow structures, are therefore of great interest in several domains to enhance sound insulation without increasing the mass. Physically, the air gap between the panels permits to decouple the motions of the receiving panel from the excited one at medium-high frequency. This decoupling leads to the gain in sound insulation observed. However, at the double panel resonance frequency, the breathing phenomenon deteriorates the sound insulation properties of the double panel.

In order to improve the sound insulation properties of double panels without adding an excessive mass, poroelastic materials are advised. The gain in sound insulation obtained by these materials is due to the large viscous and thermal dissipations in the pores. In order to model the vibro-acoustic behaviour of such materials, the homogenized Biot's theory [3] is generally used. This theory enables to take into account the solid and the fluid phase and their elastic, inertial and visco-thermal interactions, however, due to the high computation cost required to implement this model, simplified numerical methods have been developed. Three classical simplifications of Biot's theory can hence be cited: the rigid frame formulation given by Zwikker and Kosten [4], the limp formulation given by Beranek [5] and studied more recently by Doutres et al [6]. When the solid phase elasticity is taking part in the response, but with negligible solid phase shear-stress, a fluid-fluid Biot's model has been derived by Chazot and Guyader [7]. In this paper the simpler model of rigid frame is used and the porous material is modelled by an equivalent fluid acoustic medium. A large number of approximation of equivalent fluid properties have been proposed in the following the equivalent fluid properties of references [8] and [9] are used.

In real double panels mechanical links exist and are limiting the transmission loss of the ideal double wall, this paper is focusing on this point and a model is derived to predict this effect.

The standard modelisation of panel transmission is based in plane wave excitation, however this approach is time consuming when diffuse field is considered and when complicated panels are of interest. This is the reason why the Patch Mobility Method was proposed in reference [1], two main advantages appeared: the substructuring allows the possibility of calculating separately substructures behaviour and the exciting pressure field can be calculated as the blocked cavity pressure instead of cumulating lot of uncorrelated plane wave's effects.

VIBRO-ACOUSTIC MODEL OF A DOUBLE PANEL FILLED WITH A POROUS MATERIAL

In this section, the Patch Mobility Method (P.M.M.) detailed in [1] is applied to calculate the vibro-acoustic response of a double panel containing a porous material. The double panel is coupled to a cavity and radiates in an open external space (see figure 2). Only the main lines of the method are reminded here.

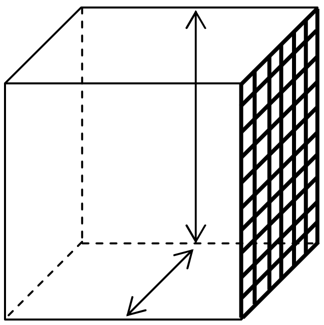

Figure 1. coupling surface of a cavity devided in patches

Let us consider a cavity that will be coupled through its surface and define the patch mobility as the ratio of averaged patch i velocity $\langle V \rangle_i$ to averaged patch j pressure force $\langle F \rangle_j$:

$$\langle\langle Y \rangle_i\rangle_j = \frac{\langle V \rangle_i}{\langle F \rangle_j} \tag{1}$$

Where $F = p.n_j$ and p is the pressure and n_j the outer normal to patch j area.

In the PMM, the averaged velocity on patch i of the subsystem after coupling, , is calculated by summing the averaged velocity before coupling $\langle \tilde{V} \rangle_i$ due to internal subsystem-sources and the averaged velocity responses due to coupling patch pressure forces with other subsystems, this relies on linear behaviour.

$$\langle V \rangle_i = \langle \tilde{V} \rangle_i + \sum_{j=1}^{N_c} \langle\langle Y \rangle_i\rangle_j . \langle F \rangle_j \tag{2}$$

In the case of a double panel containing a porous material, the system is first divided into five sub-systems as depicted in figure 2: the emission chamber A, the first panel B, the cavity C filled with a porous material, the second panel D, and the semi infinite medium E. Existing coupling surfaces between subsystems i and j are defined:

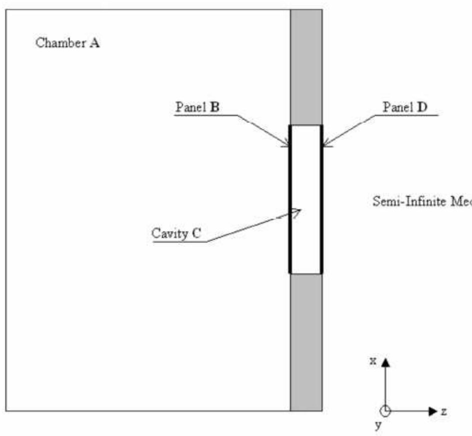

Figure 2. Double panel coupled with a source room and a semi-infinite medium

The behaviour of the first panel is described with equation:

$$\langle V_B \rangle_i = \sum_{j=1}^{N_{AB}} \langle\langle Y_B \rangle_i\rangle_j . \langle p_{bl} \rangle_j + \sum_{j=1}^{N_{BC}} \langle\langle Y_B \rangle_i\rangle_j . \langle F_{C\to B} \rangle_j \ ,$$

where $\langle p_{bl} \rangle_j$ is the pressure in the emmiting room acting on the panel, it characterizes the excitation and is approximated by the pressure existing when the panel is blocked. This assumption is generally acceptable since the radiated pressure is very small compared to the bloked pressure

The cavity of porous material is modelled through an equivalent acoustic medium; its behaviour is described with equations:

$$\langle V_{C1} \rangle_i = \sum_{j=1}^{N_{BC}} \langle\langle Y_{C1} \rangle_i\rangle_j . \langle F_{B\to C} \rangle_j + \sum_{k=1}^{N_{CD}} \langle\langle Y_{C1} \rangle_i\rangle_k . \langle F_{D\to C} \rangle_k$$

$$\langle V_{C2} \rangle_i = \sum_{j=1}^{N_{CD}} \langle\langle Y_{C2} \rangle_i\rangle_k . \langle F_{D\to C} \rangle_k + \sum_{k=1}^{N_{BC}} \langle\langle Y_{C2} \rangle_i\rangle_j . \langle F_{B\to C} \rangle_j$$

The second panel behaviour is described with equation:

$$\langle V_D \rangle_i = \sum_{k=1}^{N_{CD}} \langle\langle Y_D \rangle_i\rangle_k . \langle F_{C\to D} \rangle_k + \sum_{k=1}^{N_{DE}} \langle\langle Y_D \rangle_i\rangle_k . \langle F_{E\to D} \rangle_k$$

Finally the receiving medium is described by its plate boundary velocity:

$$\langle V_E \rangle_i = \sum_{k=1}^{N_{DE}} \langle\langle Y_E \rangle_i\rangle_k . \langle F_{D\to E} \rangle_k$$

To solve the problem one has to satisfy two continuity conditions at interfaces: the patch pressure forces coupling must be balanced and the patch coupling velocities must be equal. For subsystems X and Y interface, it writes:

$$\langle F_{X\to Y} \rangle_k + \langle F_{Y\to X} \rangle_k = 0$$

$$\langle V_X \rangle_i = \langle V_Y \rangle_i$$

The complete set of equations given in reference [1] that describes all the physical interactions taking part in the vibroacoustic response of the double panel can be lightly simplified because the radiated field, coming from the plate induced vibrations, is quite negligible compared to the direct excitation of the source room. The coupling of the first panel with the source room is therefore omitted, and the double panel excitation is introduced with blocked patch pressure existing in the emiting room surface.

PANEL PATCH MOBILITY

The panel patch mobilities can be calculated from the Love-Kirchhoff equation of motion (flexural vibration of thin plates) using a modal expansion of the plate transverse displacement with simply supported boundary conditions.

Details of these calculations are given in reference [1]. The panel patch mobility hence obtained writes:

$$\left\langle \left\langle Y_B \right\rangle_i \right\rangle_j = \sum_{k=1}^{N} \sum_{l=1}^{M} H_{kl} \left\langle \varphi_{kl}\left(x, y\right)\right\rangle_i \left\langle \varphi_{kl}\left(x_0, y_0\right)\right\rangle_j$$

Where: $\varphi_{kl}\left(x, y\right) = \sin\left(\frac{k\pi}{l_x} x\right) \sin\left(\frac{l\pi}{l_y} y\right)$ is the mode shape of a simply supported plate of dimensions $\left(l_x, l_y\right)$ and $H_{kl} = \frac{1}{M_{kl}\left(\omega_{kl}^2 - \omega^2 + 2j\varepsilon_{kl}\omega_{kl}\right)}$ is the mode impedance.

POROUS MATERIAL CAVITY PATCH MOBILITY

The cavity patch mobility is otained indirectly by inversion of patch mobility impedance matrix. Let us consider a cavity of poro-elastic material modelled by an equivalent acoustic medium of complex wave velocity c_{eq} and complex equivalent mass density; these quantities are obtained with different models depending on parametes that must be measured as: porosity, tortuosity, characteristic lengths... In this paper the coefficients of reference [] are used.

The averaged pressure created on patch k by a vibration of the patch j having unit velocity amplitude is given by the following expression

$$Z_{kj} = \left\langle p\left(x, y, z\right)\right\rangle_k = \sum_{p,q,r} A_{pqr} \left\langle \psi_{pqr}\left(x, y, z\right)\right\rangle_k \left\langle \psi_{pqr}\left(x_0, y_0, z_0\right)\right\rangle_j$$

Where:

$$A_{pqr} = \frac{j\omega\rho_{eq}}{\left(\left(\omega / c_{eq}\right)^2 - k_{pqr}^2\right) N_{pqr}},$$

$$N_{pqr}\left(x, y, z\right) = \int_V \psi_{pqr}^2\left(x, y, z\right) dV ,$$

$$\psi_{pqr}\left(x, y, z\right) = \cos\left(\frac{p\pi}{l_x}\right) \cos\left(\frac{q\pi}{l_y}\right) \cos\left(\frac{r\pi}{l_z}\right),$$

$$k_{pqr} = \sqrt{\left(\frac{p\pi}{l_x}\right)^2 + \left(\frac{q\pi}{l_y}\right)^2 + \left(\frac{r\pi}{l_z}\right)^2} .$$

To obtain patch mobilities the matrix of patch mobility impedance must be inverted:

$$\left(Y_{kj}\right) = \left(Z_{kj}\right)^{-1}$$

Let us remark that it is possible to obtain directly the patch mobilities by using Dirichlet mode shapes instead of rigid cavity wall mode shapes.

SHELL STIFFNERS PATCH MOBILITY

Due to the complexity of the stiffner structure finite element model is used for calculating mode shapes of the isolated stiffner, 300 modes were calculated and served to calculate input and transfer point mobility, the general expression of the stiffner mobilities has the same general expression given for the plate :

$$\left\langle \left\langle Y_B \right\rangle_i \right\rangle_j = \sum_{k=1}^{N} \sum_{l=1}^{M} H_{kl} \left\langle \phi_{kl}\left(M\right)\right\rangle_i \left\langle \phi_{kl}\left(M_0\right)\right\rangle_j ,$$

However the mode shapes $\phi_{kl}\left(M\right)$ are obtained numerically through a finite element code. In addition, the links between stiffners and plates are spot welding points and point mobilities are only calculated by FEM at coupling point M_i . The patch mobilities used in the method are calculated by approaching

$$\left\langle \phi_{kl}\left(M_0\right)\right\rangle_j = \phi_{kl}\left(M_j\right) S_j ,$$

Thus, the patch mobilities for the stiffners are calculated by:

$$\left\langle \left\langle Y_B \right\rangle_i \right\rangle_j = \sum_{k=1}^{N} \sum_{l=1}^{M} H_{kl} \phi_{kl}\left(M_i\right) \phi_{kl}\left(M_j\right) S_j S_i$$

PLATE EXCITATION IN THE SOURCE ROOM

In the Patch Mobility Method the excitation of the structure is modelled by the blocked patch pressure in the source room. It can be measured or calculated. Of course the blocked patch pressure is not easy to calculate because modal decomposition must be used, however the calculation is standard as the room as rigid cavity wall and for a rectangular room, analytical mode shapes are available that simplify calculation. The enourmous advantage of the PMM compared to standard calculation with excitation by uncorrelated plane waves, is that when the blocked patch pressures are known, just one resolution of the vibroacoustic problem is necessary contrary to the standard approach that necessitate as many resolutions than the number of waves. That means a lot of calculation to insure convergence.

A free software has been developed, it can be downloaded on the web at http://www.utc.fr/~chazotje/ (request password to the authors). It can generate boundary patch pressure for rectangular rooms and also contains standard room pre-calculated results. Thus, for a given test room facility, the exciting blocked patch pressure is a stored data usable for any plate.

A second advantage is the possibility of using measured excitation boundary pressure in the source room. Two examples of measured boundary pressure are presented in Figs 3 and 4. It can be clearly seen that the pressure is not uniform as supposed in diffuse field assumption even at frequencies higher than the cutt of frequency.

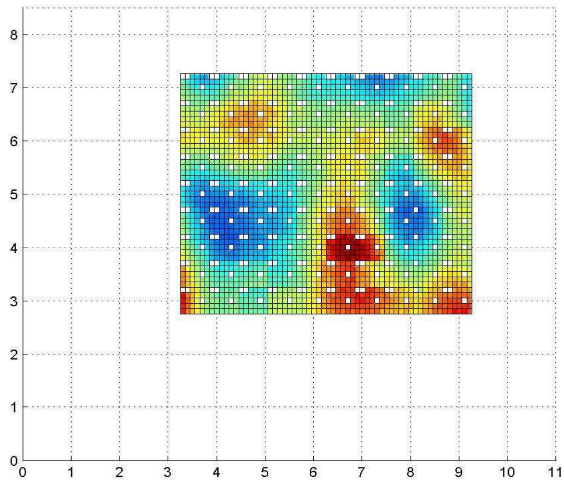

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Figure 3. Measured blocked patch pressure at 80 Hz in a room of dimensions: lx=11.5 m, ly=8.69 m, lz=4.03 m - Patch size: X=0.04 m, Z=0.037 m - Source position in the room : Xs =2 m, Ys=2 m, Zs=1 m . Diffuse field cutoff frequency: 187 Hz.

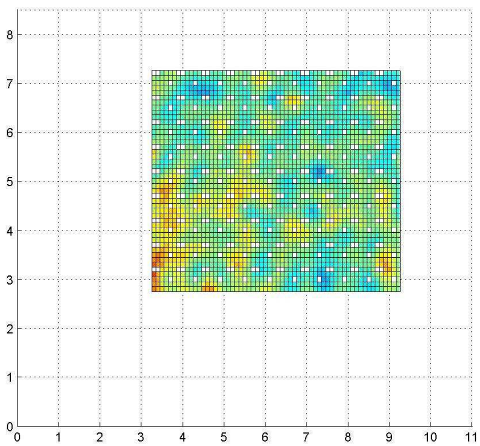

Figure 4. Measured blocked patch pressure at 250 Hz in a room of dimensions: lx=11.5 m, ly=8.69 m, lz=4.03 m - Patch size: X=0.04 m, Z=0.037 m - Source position in the room : Xs =2 m, Ys=2 m, Zs=1 m . Diffuse field cutoff frequency: 187 Hz.

CALCULATION OF TRANSMISSION LOSS

The basic expression of the transmission loss is depending of incident to transmitted powers ratio:

$$TL = 10 Log\left(\frac{\pi_{ray}}{\pi_{inc}}\right),$$

In experiments the incident power cannot be directly measured and it is estimated from reverberant pressure assuming that diffuse field exists in the emmiting room.

$$\pi_{inc} = \left\langle p_{rev}^2 \right\rangle_V \frac{S}{4\rho_0 c}$$

In our case the blocked pressure acting the plate in the emmiting room is considered, it is related to the reverberent pressure by the following expression, if perfect diffuse field is existing in the room.

$$\left\langle p_{rev}^2 \right\rangle_V = \frac{1}{2}\left\langle p_{blocked}^2 \right\rangle_S$$

In the following the TL is calculated with the incident power expression:

$$\pi_{inc} = \left\langle p_{blocked}^2 \right\rangle_S \frac{S}{8\rho_0 c}$$

It can be observed that the pressure field diffusivity can be estimated by the ratio between volume quadratic averaged pressure in the room and surface quadratic blocked pressure :

$$\beta = \frac{\left\langle p_{blocked}^2 \right\rangle_S}{\left\langle p_{rev}^2 \right\rangle_V},$$

For perfect diffuse field; $\beta = 2$, if $\beta < 2$ some waves are missing in the exciting pressure and if $\beta < 2$ correlated waves are present in the acoustic field.

DOUBLE PLATE WITH STIFFNER

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The double plate used as example in the following is made of two sandwich steel-polymer-steel plates, connected by a stiffner in different points, the double plate can be or not filled by mineral whool.

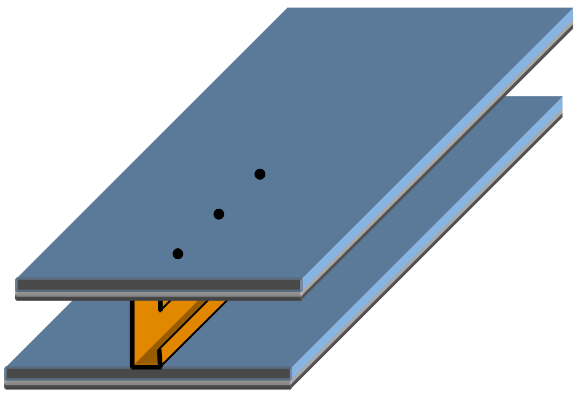

Figure 3. Double panel with stiffner point coupled to plates

Plates dimensions: 1.2 m x 2.4 m Sandwitch Plates 1 mm Steel- 0.05 mm PVB-1 mm Steel, 4 Screws on face 1, 3 Screws on face 2

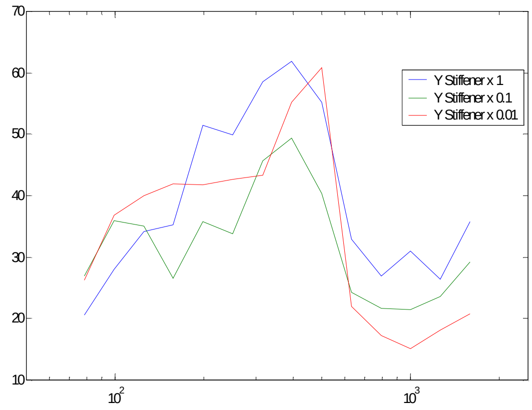

Figure 4. Transmission loss of the double panel with stiffner and glass whool. Influence of stiffner.

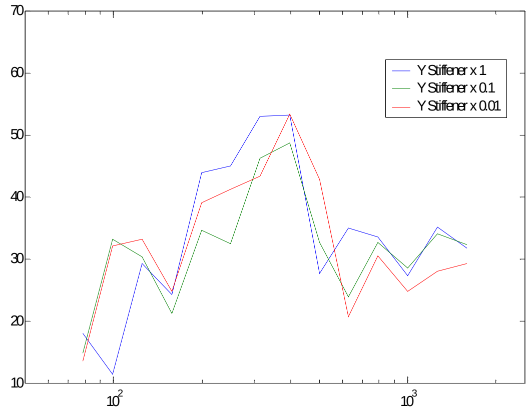

Figure 4. Transmission loss of the double panel with stiffner and without glass whool.

The blue curve in Fig.4 is the basic result obtained with the stiffner mobilities, the red (resp. black) curve corresponds to a reduced stiffner input and transfer point mobility of 0.1 (resp.0.01) of the original one. As a general trend the decrease of stiffner mobilitbies increase the sound transmitted by the panel compared to the basic case, this is the expected trend as the two panels are more and more mechanically coupled, however this effect is more important for the double panel filled with mineral whool than the one with air cavity.

CONCLUSION

A model of sound transmission through double panels filled with porous material and having mechanical links has been presented. It is based on the Patch Mobilty Method. Two advantages appear with this approach; first the possibility if substructuring allows us to use different methods for calculation of patch mobilities of separate structural components (modal decomposition, FEM, analytical modes), secondly the excitation is modelled by the blocked pressure acting on the panel in the emmiting room. This permits the TL calculation with a single resolution of the vibroacoustic problem, instead of the classical decomposition of the exciting field with decorelated waves that necessitate a huge number of vibroacoustic problem resolution.

The influence of mechanical links on the double plate TL is presented , the increase of transmission compared to the ideal casewith no mechanical link is predicted, the effect is more important when the gap between panel is filled of porous material.

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