OPTIMISATION OF SANDWICH TRIM PANELS FOR REDUCING HELICOPTER INTERNAL NOISE

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Abstract

In the helicopter domain, the trim panels in cabin are generally provided with a core in Nomex honeycomb and external layers in composite fibres. This light assembly is not subjected to high static force and must just assure a sufficient stiffness not to be damaged during the helicopter life. Each material satisfies specific tests to be certified: behavior in high temperature, with humidity… .Nevertheless, to use these components can worsen the internal acoustic comfort.

Authors have developed analytic models to compute the acoustic Transmission Loss (TL) of sandwich panels, with a thick orthotropic core and multi-layered laminates. The TL represents the ratio between incident acoustic power, produced by a diffused acoustic field, and the acoustic power radiated by the panel.

Models consider elastic materials like homogeneous materials, composite fibres, visco-elastic materials, honeycombs or foams, described by their stiffness matrix. They can be applied to simulate helicopter "global" walls by the interaction of a structural panel (e.g. mechanical deck) and a trim panel separated by air gap or porous material (blanket).

In this paper is described the theoretical hypotheses assumed for models, then the acoustic behavior of current trim panels.

Sandwich panels with foam cores are suggested to replace current trim panels and tested by modal analysis to determine mechanical characteristics needed as input database in acoustic models.

The TLs measured in laboratory setup are compared with simulations to verify the validity models.

Finally, configurations with a "heavy" honeycomb and a foam are integrated numerically in a representative "global" wall.

Symbols

θ_1 and ϕ_1 : angles of incident waves

α_I, α_{II1} and α_{II2}: amplitudes of transverse displacement

u, *w*: displacements in x and z directions

PE, KE: potential and kinetic energies

ζ_{cx}: expansion term for core

Z: structural impedance

Ri: median axis of a layer *i*.

m_i: mass / unit area of a layer *i*.

u_{oi} and u_{oc}: membrane terms

W_I and W_T : incident and transmitted acoustic powers

p: acoustic pressure

ϕ_{ix} and ϕ_{cx}: shear terms

$\bar{E}_{ixx}, G_{ixz}, \bar{E}_{cxx}, \bar{E}_{czz}$ and G_{cxz} : elastic stiffnesses

$\varepsilon_{ixx}, \varepsilon_{izz}, \gamma_{ixz}, \varepsilon_{cxx}, \varepsilon_{czz}$ and γ_{cxz} : normal / shear strains

$\sigma_{ixx}, \tau_{ixz}, \sigma_{cxx}, \sigma_{czz}$ and τ_{cxz} : normal / shear stresses

t_i: thickness of a layer *i*

τ : transmission coefficient

TL: Transmission Loss

k_1: wave number in medium 1

Theoretical TL models

Let be plane acoustic pressure waves in medium 1 that excite a face of the panel I. The vibration of this one produces an acoustic radiation in medium 2 that excites the panel II. We are interested by the pressure radiated by this panel in medium 3 (Fig 1).

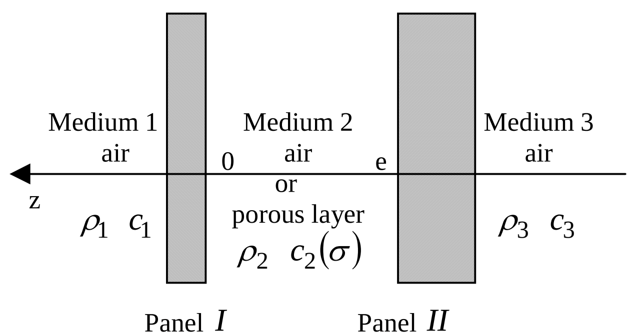

Fig 1 : Configuration of a "double-structure"

The direction of incident waves, with the wave number k_1, is defined by the angles θ_1 et ϕ_1 , as following (Fig 2):

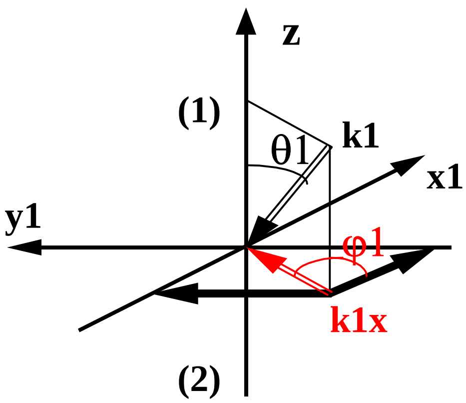

Fig 2 : Field and structural system of coordinates

k_{1x} is defined by: $k_{1x} = k_1 \cos(\theta_1)$

The system is led by:

Medium 1: $(\Delta + \frac{\omega^2}{c_1^2}) p_1(x,z) = 0$ (z>0)

$$\frac{\partial p_1(x,0)}{\partial z} = \omega^2 \rho_1 w_I(x)$$

Medium 2: $(\Delta + \frac{\omega^2}{c_2^2}) p_2(x,z) = 0$ (0<z<e)

$$\frac{\partial p_2(x,0)}{\partial z} = \omega^2 \rho_2 w_I(x)$$

$$\frac{\partial p_2(x,e)}{\partial z} = \omega^2 \rho_2 w_{II}(x)$$

Medium 3: $(\Delta + \frac{\omega^2}{c_3^2}) p_3(x,z) = 0$ (z<e)

$$\frac{\partial p_3(x,0)}{\partial z} = \omega^2 \rho_3 w_{III}(x)$$

Panel I: $Z_I . w_I(x) = p_2(x,0) - p_1(x,0)$

Panel II: $\begin{bmatrix} Z_{IIs} & Z_{IIas} \\ Z_{IIas} & Z_{IIa} \end{bmatrix} \begin{Bmatrix} w_s(x) \\ w_a(x) \end{Bmatrix} = \begin{Bmatrix} -p_s(x,e) \\ -p_a(x,e) \end{Bmatrix}$

With *p, w, Z* respectively pressure, displacement in direction z and structural impedance.

One can notice that the panel I has the same transverse displacement through the thickness (thin panel), while the panel II to take into account the possible expansion effect of a soft core (thick panel).

We assume that the medium 1 and 3 are of the same type: $\begin{cases} \rho_1 = \rho_3 \\ c_1 = c_3 \end{cases}$

The medium 2 can be composed of air or porous material defined like an equivalent fluid by a complex density ρ_2 and a flow resistivity σ.
The continuity of displacement *w* in the media 1 and 2, via the panel I is supposed:
$k_1 \sin(\theta_1) = k_2 \sin(\theta_2)$ (Snell-Descartes Law)

So:

$$w_I = \alpha_I \sin(k_1 \sin(\theta_1)x) e^{-j\omega t}$$

$$w_{II1} = \alpha_{II1} \sin(k_1 \sin(\theta_1)x) e^{-j\omega t}$$

$$w_{II2} = \alpha_{II2} \sin(k_1 \sin(\theta_1)x) e^{-j\omega t}$$

with α_I, α_{II1} and α_{II2} amplitudes of transverse displacement for panel I and external layers of panel II.

The pressures verify:

$$p_1(x,z) = \sin(k_1 \sin(\theta_1)x) \left(e^{jk_1 \cos(\theta_1)z} + B_1 e^{-jk_1 \cos(\theta_1)z} \right) e^{-j\omega t}$$

$$p_2(x,z) = \sin(k_1 \sin(\theta_1)x) \left(A_2 e^{jk_2 \cos(\theta_2)z} + B_2 e^{-jk_2 \cos(\theta_2)z} \right) e^{-j\omega t}$$

$$p_3(x,z) = \sin(k_1 \sin(\theta_1)x) \left(A_3 e^{jk_1 \cos(\theta_1)z} \right) e^{-j\omega t}$$

The continuity of wave numbers in plane (x1,y1) leads to: $k_2 \cos(\theta_2) = \left(k_2^2 - k_1^2 \sin^2(\theta_1) \right)^{\frac{1}{2}}$.

We consider the symmetric and antisymmetric motions for displacements and pressures at the interfaces of panel II (Ref 1):

$$w_s(x) = \frac{w_{II1}(x) - w_{II2}(x)}{2}, \quad w_a(x) = \frac{w_{II1}(x) + w_{II2}(x)}{2}$$

and

$$p_s(x) = \frac{p_2(x,e) + p_3(x,e)}{2}, \quad p_a(x) = \frac{p_2(x,e) - p_3(x,e)}{2}$$

$k_1, k_2, \rho_1, \rho_2, \theta_1$ being input data, as structural impedances, we have to solve a system with 7 equations for 7 unknown parameters, to determine the acoustic transmission.

With the Sommerfeld conditions, the acoustic transmission coefficient can be described by :

$\tau(\theta,\varphi)=\dfrac{W_T}{W_I}$ with W_I and W_T the incident and transmitted acoustic powers:

$$W_I=\frac{1}{2}Re\left\{\int_S p_I(x,0).\left(j\omega\, w_I(x)\right)^* dS\right\}=\frac{1}{2}\frac{\cos(\theta_1)S}{\rho_1 c_1}$$

and

$$W_T=\frac{1}{2}Re\left\{\int_S p_3(x,0).\left(j\omega w_{II2}(x)\right)^* dS\right\}$$

$$=\frac{\omega}{2}\operatorname{Im}\left\{A_3\,\alpha_{II2}{}^{*}e^{jk_1\cos(\theta_1)e}\right\}S=\frac{\omega^2}{2}\frac{\rho_1 c_1}{\cos(\theta_1)}\left|\alpha_{II_2}(\theta_1)\right|^2$$

The layers being anisotropic, the transmission coefficient depends on φ_1:

$$\tau(\theta_1,\varphi_1)=\omega^2\left(\frac{\rho_1 c_1}{\cos(\theta_1)}\right)^2\left|\alpha_{II_2}(\theta_1,\varphi_1)\right|^2$$

We define the Transmission Loss by:

$TL(\theta_1,\varphi_1)=10\log\left(\dfrac{1}{\tau(\theta_1,\varphi_1)}\right)$ en dB

In our case, we are interested in a diffuse field excitation. So, the transmission coefficient must be averaged over incidence orientation as follows, to obtain the diffuse field Transmission Loss :

$$TL_d=-10\log\left(\frac{\int_0^{2\pi}\int_0^{\theta_1 \lim}\tau(\theta_1,\varphi_1)\sin(\theta_1)\cos(\theta_1)\,d\theta_1\,d\varphi_1}{\int_0^{2\pi}\int_0^{\theta_1 \lim}\sin(\theta_1)\cos(\theta_1)\,d\theta_1\,d\varphi_1}\right)dB$$

with generally $\theta_{\lim}=78°$

To determine the global TL , it is necessary to determine the structural impedances of panel I and II.

We consider 2 different models of plane and infinitely wide structures, inserted in PIAMCO (ONERA software):

- A "multi-layered" panel model for panel I
- A "dissymmetric" sandwich panel model.for panel II

<u>"Multi-layered" panel model</u>

This model concerns structures with P layers (Fig 3) whose orthotropy directions are different through the thickness. It is suited, for example, to composite fibers (kevlar, carbon or fiber glass) with resin, visco-elastic materials, or stiff or thin honeycombs in sandwich panels.

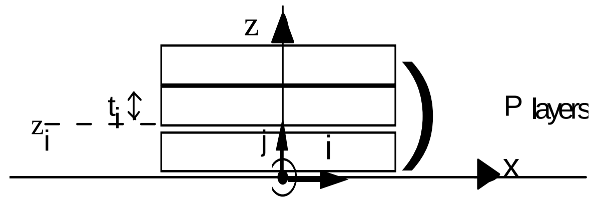

Fig 3: Geometry of "Multi-layered" panel

The study can be led in 2D (*xz*), where x is the direction of propagation across the panel surface and z the thickness direction.

The displacement field can be written, as below, for each layer *i* (Fig 4):

$$u_i(x,z)=u_{oi}(x)-(z-R_i)\left(\frac{\partial w_i(x,z)}{\partial x}+\phi_{ix}(x)\right)$$

$$w_i(x,z)=w(x)$$

with *u*, *w* displacements in x and z directions and *Ri* median axis of a layer *i*.

This expression includes respectively membrane bending and shear terms: $u_{oi}(x)$; $\dfrac{\partial w_i(x,z)}{\partial x}$, $\phi_{ix}(x)$.

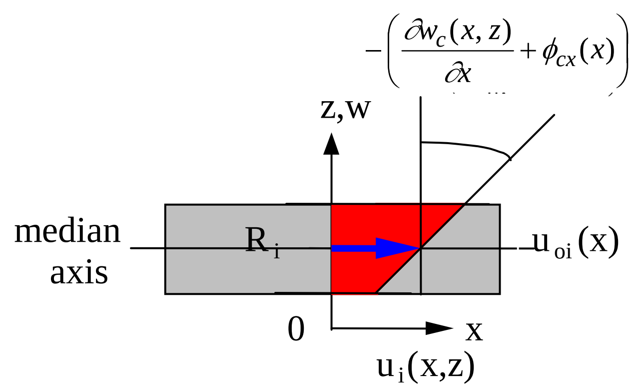

Fig 4 : Displacement field in a layer i

The normal and shear strains are the following:

$$\varepsilon_{ixx}(x,z)=\frac{\partial u_{oi}(x)}{\partial x}-(z-R_i)\left(\frac{\partial^2 w_i(x,z)}{\partial x^2}+\frac{\partial\phi_{ix}(x)}{\partial x}\right)$$

$$\varepsilon_{izz}(x,z)=0$$

$$\gamma_{ixz}(x,z)=-\phi_{ix}(x)$$

The stresses are related to the strains by the stiffness matrix as follows :

$$\begin{Bmatrix} \sigma_{ixx} \\ \tau_{ixz} \end{Bmatrix} = \begin{bmatrix} \overline{E}_{ixx} & 0 \\ 0 & G_{ixz} \end{bmatrix} \begin{Bmatrix} \varepsilon_{ixx} \\ \gamma_{ixz} \end{Bmatrix}$$

where $\overline{E}_{ixx}$ and G_{ixz} depend on elastic stiffnesses in the directions of orthotropy (*1,2,z*):

$$\overline{E_{ixx}} = \overline{E_{i11}}\cos^4(\varphi_1) + \overline{E_{i22}}\sin^4(\varphi_1) + 2(\overline{E_{i12}} + 2G_{i12})\cos^2(\varphi_1)\sin^2(\varphi_1)$$
$$G_{ixz} = G_{i1z}\cos^2(\varphi_1) + G_{i2z}\sin^2(\varphi_1)$$

These parameters are complex to take the damping loss factor η into account:

$$Mod_{complex} = Mod_{real}(1 - j\eta)$$

The hypothesis of plane strains in 2D (*1,2*) allows to determine the stiffnesses terms $\overline{E_{i11}}$ and $\overline{E_{i22}}$ according to the elastic moduli , E_{i1} and E_{i2} and the Poisson coefficient υ_{i12} :

$$\overline{E_{i11}} = \frac{E_{i1}}{1 - \frac{E_{i2}}{E_{i1}}\upsilon_{i12}^2} \quad \text{and} \quad \overline{E_{i22}} = \frac{E_{i2}}{1 - \frac{E_{i2}}{E_{i1}}\upsilon_{i12}^2}$$

Moreover, the displacement field parameters are assumed to be :

$$w(x) = \alpha \sin(k_{1x}x)$$
$$u_{oi}(x) = \beta_i \cos(k_{1x}x)$$
$$\phi_i(x) = \phi_i \cos(k_{1x}x)$$

The parameters of the layer *i* can be written in function of only the parameters of the layer *1*, α, β_1, ϕ_1 , with the continuity of displacements and shear stresses between each layer by an iterative procedure. Nevertheless, there is no continuity in normal stress and the shear stress is supposed independent of the thickness.

$$\phi_{(i+1)x} = \frac{G_{ixz}}{G_{(i+1)xz}}\phi_{ix}$$

and

$$-\frac{t_i}{2}\left(\frac{\partial w}{\partial x} + \phi_{ix}\right) + u_{oi} = \frac{t_{(i+1)}}{2}\left(\frac{\partial w}{\partial x} + \phi_{(i+1)x}\right) + u_{o(i+1)}$$

The potential and kinetic energy densities, *Ep_i* and *Ec_i*, can be expressed by:

$$2Ep_i = \overline{E_{ixx}}\varepsilon_{ixx}^2 + G_{ixz}\gamma_{ixz}^2$$
$$2Ec_i = \rho_i(\dot{u}_i^2 + \dot{v}_i^2 + \dot{w}_i^2)$$

The potential energy (PE) and the kinetic energy (KE) are calculated by integrating the different energy densities over a volume defined by the thickness of the panel (*z* direction), one wavelength in the *x* direction, and per a unit distance (*y* direction).

The Lagrange's equations are then used to obtain the parameters α, β_1, ϕ_1 :

$$\left(\frac{d}{dt}\right)\left(\frac{\partial KE}{\partial \dot{g}r}\right) - \frac{\partial KE}{\partial gr} + \frac{\partial PE}{\partial gr} = Qr$$

with *gr* corresponding to α, β_1, ϕ_1 (generalized displacements) and *Qr* the generalized forces coming from the pressures $p_1(x)$ et $p_2(x)$ acting respectively on the panel faces in media 1 and 2:

$$Q_r = \lambda \begin{Bmatrix} \frac{p_2 - p_1}{2} \\ 0 \\ 0 \end{Bmatrix}$$

with $P_1(x) = P_1\sin(k_x x)$ and $P_2(x) = P_2\sin(k_x x)$

With $w_I(x) = w(x)$, we can obtain the structural impedance Z_I and the acoustic coefficient transmission τ if the media 1 and 2 are identical:

$$\tau(\theta_1, \varphi_1) = \left(\frac{\omega\,\rho_1 c_1}{\cos(\theta_1)}\right)^2 \frac{4}{\left|Z_I - 2j\frac{\omega\,\rho_1 c_1}{\cos(\theta_1)}\right|^2}$$

<u>"Dissymmetric" sandwich panel model</u>

This model concerns dissymmetric structures with a thick orthotropic core and orthotropic multi-layered laminates (Fig 5).

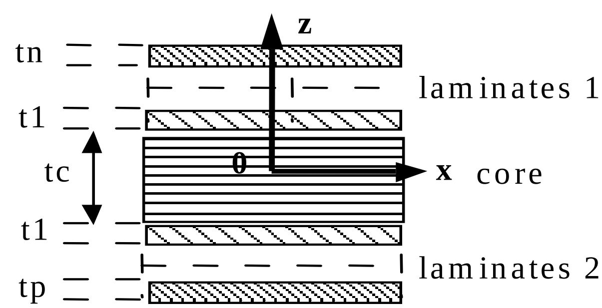

Fig 5: Geometry of a sandwich panel

This model can take into account more or less stiff cores as visco-elastic materials, honeycombs or foams.

As concerns the core, the displacement field satisfies :

$$u_c(x,z) = u_{oc}(x) - z\left(\frac{\partial w_c(x,z)}{\partial x} + \phi_{cx}(x)\right) + \zeta_{cx}(x)\cos\left(\frac{\pi z}{t_c}\right)$$

(Fig 6)

$$w_c(x,z) = \frac{w_{11}(x) + w_{21}(x)}{2} + \frac{w_{11}(x) - w_{21}(x)}{t_c} z$$

(Fig 7)

with (1,2) the layers 1 and 2 in contact with the core and $\zeta_{cx}(x)$ the expansion term. This formulation is similar to that employed by Ref 1 in the case of a single isotropic laminate on each side of the core.

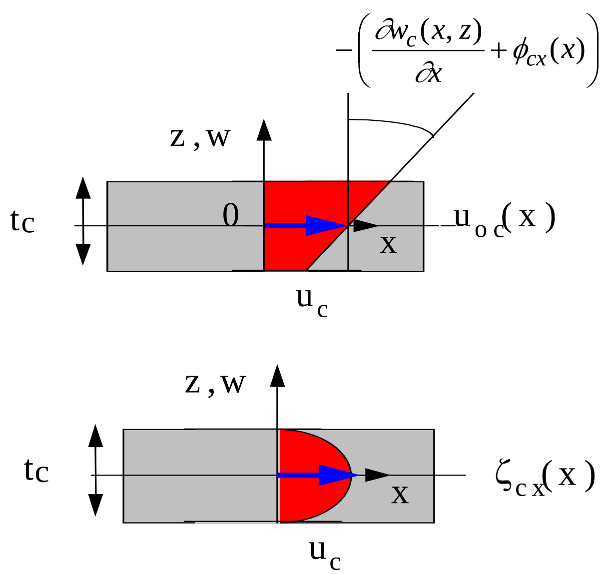

Fig 6: Core displacement in x direction

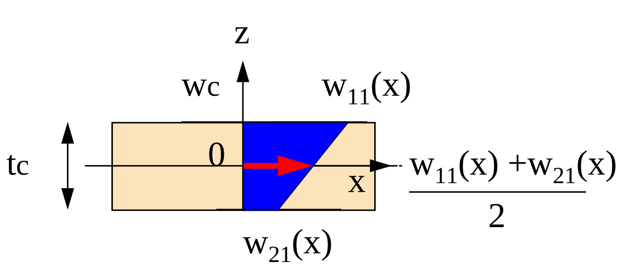

Fig 7: Core displacement in z direction

The normal and shear strains are the following:

$$\varepsilon_{cxx}(x,z) = \frac{\partial u_{oc}(x)}{\partial x} - z\left(\frac{\partial^2 w_c(x,z)}{\partial x^2} + \frac{\partial \phi_{cx}(x)}{\partial x}\right) + \frac{\partial \zeta_{cx}(x)}{\partial x}\cos\left(\frac{\pi z}{t_c}\right)$$

$$\varepsilon_{czz}(x,z) = \frac{\partial w_c(x,z)}{\partial z}$$

$$\gamma_{cxz}(x,z) = -\phi_{cx}(x) - \frac{\pi}{tc}\zeta_{cx}(x)\sin\left(\frac{\pi z}{t_c}\right)$$

The stresses are related to the strains by the stiffness matrix as follows :

$$\begin{Bmatrix} \sigma_{cxx} \\ \sigma_{czz} \\ \tau_{cxz} \end{Bmatrix} = \begin{bmatrix} \bar{E}_{cxx} & \bar{E}_{cxz} & 0 \\ \bar{E}_{cxz} & \bar{E}_{czz} & 0 \\ 0 & 0 & G_{cxz} \end{bmatrix} \begin{Bmatrix} \varepsilon_{cxx} \\ \varepsilon_{czz} \\ \gamma_{cxz} \end{Bmatrix}$$

with

$$\bar{E}_{cxx} = \bar{E}_{c11}\cos^4(\varphi_1) + \bar{E}_{c22}\sin^4(\varphi_1) + 2(\bar{E}_{c12} + 2G_{c12})\cos^2(\varphi)\sin^2(\varphi_1)$$

$$\bar{E}_{cxz} = \bar{E}_{c1z}\cos^2(\varphi_1) + \bar{E}_{c2z}\sin^2(\varphi_1)$$

$$G_{cxz} = G_{c1z}\cos^2(\varphi_1) + G_{c2z}\sin^2(\varphi_1)$$

The stiffness terms comply with the hypothesis of plane strain (3D).

Writting the continuity of displacements and shear stresses between the core and the laminates, the remained unknown parameters are α_1, β_{11} (laminate 1), α_2, β_{21} (laminate 2) and ξ_c.

As concerns the laminates 1 and 2, the displacement and stress fields follow the approach of the "multi-layered" model.

It is interesting, for the following, to replace α_1, α_2, β_{11}, β_{21} by symmetric (*s*) and antisymmetric (*a*) terms :

$$w_s(x) = \alpha_s \sin(k_{1x}x), \quad w_a(x) = \alpha_a \sin(k_{1x}x)$$

$$u_s(x) = \beta_s \cos(k_{1x}x), \quad u_a(x) = \beta_a \cos(k_{1x}x)$$

$$\zeta_{xc}(x) = \zeta_{xc}\cos(k_{1x}x)$$

with

$$\alpha_s = \frac{\alpha_{11} - \alpha_{21}}{2} \text{ and } \alpha_a = \frac{\alpha_{11} + \alpha_{21}}{2}$$

$$\beta_s = \frac{\beta_{11} + \beta_{21}}{2} \text{ and } \beta_a = \frac{\beta_{11} - \beta_{21}}{2}$$

As for the "multi-layered" model, the Lagrange's equations are then expressed :

$$\left(\frac{d}{dt}\right)\left(\frac{\partial KE}{\partial \dot{g}r}\right) - \frac{\partial KE}{\partial gr} + \frac{\partial PE}{\partial gr} = Qr$$

with *gr* corresponding to α_s, β_s, and ξ_c (symmetric generalized displacements) and α_a, β_a (antisymmetric generalized displacements).

Qr are the generalized forces coming from the pressure $P_s \sin(k_{1x} x)$ (symmetric pressure) and $P_a \sin(k_{1x} x)$ (antisymmetric pressure).

$p_s(x)$ and $p_a(x)$ come from the following equations:

$$p_s(x) = \frac{p_2(x,e) + p_3(x,e)}{2}$$

$$p_a(x) = \frac{p_2(x,e) - p_3(x,e)}{2}$$

With $w_{IIs}(x) = w_s(x)$ et $w_{IIa}(x) = w_a(x)$, we can obtain the impedance matrix that includes a coupling term between the symmetric and antisymmetric behaviours (Z_{IIas}).

The transmission coefficient can be described by :

$$\tau(\theta_1,\varphi_1) = \left| \frac{j\omega \frac{\rho_1 c_1}{\cos(\theta_1)}\left(Z_{IIs} - Z_{IIa}\right)}{\left(Z_{IIs} - j\omega \frac{\rho_1 c_1}{\cos(\theta_1)}\right)\left(Z_{IIa} - j\omega \frac{\rho_1 c_1}{\cos(\theta_1)}\right) - Z_{IIas}^2} \right|^2$$

Simulations applied to current trim panels

To optimize the assembly of materials, we have simulated (Fig 8) the sound Transmission Loss (TL) of a panel defined from a fractional plan using a database, composed of several Nomex honeycomb (with variable thickness), fiber glass, kevlar, carbone and viscoelastic materials. The optimal configuration has the maximum global TL in the frequency range 500-5000 Hz (Ref 2) and verifies initial requirements, like mass and thickness below 6 kg/m^2 and 20 mm, and presence of a viscoelastic layer on both sides of the core. The panel is so 6 kg/m^2 and 8.2 mm, in mass and thickness, with a core of 5 mm thick.
In the mentioned frequency band, the TL is similar to this produced by a steel panel of equal weight. The coincidence frequency and the double wall resonance appear beyond the band (12 and 18.4 kHz) (Ref 3). So, the TL follows only the mass law. Moreover, the high damping provided by the viscoelastic layer (about 20 %) is not efficient outside the coincidence frequency.

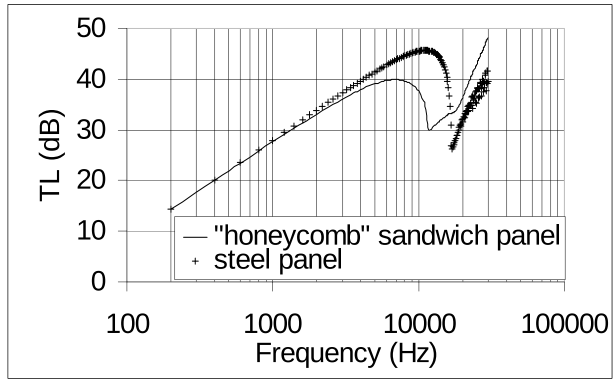

Fig 8: Simulated TL of optimal "honeycomb" sandwich and steel panels

ONERA proposal

To take advantage of the high TL that exists after the double wall resonance frequency, it can be interesting to use foam, less stiffness than honeycomb, and so to place this frequency below the interesting frequency range. With this aim, simulations have been realized to define optimal configurations with standard foams, whose mechanical characteristics are estimated.
Two square panels (0.90 x 0.90 m^2) composed of a foam core with closed or open cells placed between two orthotropic fiber glass layers (Table 1) have been made to be tested in laboratory.

Characteristics	"closed cell" panel	"open cell" panel
Total surfacic mass (kg/m^2)	5.6	5.3
Foam surfacic mass (kg/m^2)	0.47	0.15
Total thickness (mm)	17.7	10.5
Foam thickness (mm)	14.6	7.4

Table 1: Mass and thickness of "foam" panels

The "closed cell" foam is a strong, resilient and low density material, with cells whose mean diameter is about 5 mm. It is mainly used for vibration damping but also for sound absorption. The "open cell" foam is a soft material with cells of 1 mm in mean diameter. It is generally pasted on a face of a structural panel, for the sound absorption. The stiffness matrix of a fiber glass layer is completely given by the aeronautic manufacturer (Young and Shear moduli in all directions). On the other hand, the tests of high static compression deformation (10%, 25% and/or 50 %), led on foams by manufacturers, does not allow to determine the mechanical characteristics valid for an acoustic excitation.

Modal analysis

An experimental modal analysis is carried out on each type of sandwich panel clamped on the boundaries (surface: 0.84 x 0.84 m^2).
As concerns the “closed cell” panel, 7 modes have been identified between 54 and 220 Hz with a damping of about 6 %. In spite of the square section, the resonant frequencies of (m,n) and (n,m) modes (m or n anti-nodes in a given direction) are very different because of, mainly, the orthotropic behavior of the foam. An analytical model is used to compute the resonant frequencies of a clamped multi-layered panel (Ref 4) and so, by comparison with experimental results, to fit mechanical characteristics of foams. The mode shapes follow the Warburton formulations defined for clamped plates with bending displacements (Ref 5). We can show that taking into account membrane and shear terms in displacement field is compatible with the formulations (negligible errors). Some adjustments have allowed to determine precisely the values of the foam shear moduli G_{c2z} and G_{c1z} (Fig 9-a). It turns out that a slight variation of these values produces important variations of resonant frequencies (e.g. G_{c2z}: Fig 9 b). On the other hand, the accuracy of $\overline{E}_{c11}$ and $\overline{E}_{c22}$ is not determining.
For the “open cell” panel, the experimental resonant frequencies and damping are clearly lower: that is 8 modes between 18 and 69 Hz and a mean damping of 1.5 %. The foam behaves like an isotropic material (Fig 10-a) with a very low shear modulus. As for the previous foam, only this type of parameter must be defined accurately (Fig 10-b).

Transmission Loss

Transmission Loss measurements are obtained according the procedure described in Ref 6. Simulations are led with PIAMCO (Ref 7), supplied in input by the mechanical characteristics fitted after modal analysis. Only the transverse Young modulus of foam $\overline{E}_{czz}$ remains unknown. The figure 11 a-b bring to the fore its influence on the TL and, in particular, on the value of the double wall resonance frequency. The figure 12 compares the theoretical and experimental TL of the two panels with the optimal values of $\overline{E}_{czz}$, G_{c2z} and G_{c1z} . These ones appear very different between materials (factor 20 to 40) (Table 2). One can notice that the theoretical behavior of the “closed cell” panel is in accordance with the reality, with a double wall resonance frequency around 2800 Hz. In the case of the “open cell” panel, this frequency around 550 Hz leads to a TL about 60 dB at 10 kHz. The high values of TL obtained at high frequency range, hard to assure and to measure precisely in laboratory, can explain in part the difference between theory and experimentation.

Mechanical characteristics	“closed cell” foam	“open cell” foam
$\overline{E}_{czz}\ (Mpa)$	6	0.15
$G_{c2z}\ (Mpa)$	1.1	0.08
$G_{c1z}\ (Mpa)$	1.75	0.08

Table 2: Optimal mechanical characteristics of foams

Finally, configurations with a "heavy" honeycomb and a "light" foam panels (respective masses: 8.3 and 5.4 kg/m^2) (Table 3) are simulated as trim panels preceded by a representative helicopter structural panel (sandwich panel with nomex honeycomb and carbon layers) and an air gap (Fig 13). The honeycomb panel appears less interesting than the foam panel from 800 Hz, with a difference about 30 dB at high frequencies, in spite of an higher mass and the presence of a visco-elastic layer.

Panel	Structural panel	"Honeycomb" trim panel	"Foam" trim panel
Surfacic mass (kg/m^2)	2.8	8.3	5.4
Thickness (mm)	16.5	11	18.1

Table 3: Surfacic masses and thicknesses of panels integrated in the helicopter "global" wall

Conclusion

This paper is relative to the theoretical models developed by authors to represent the acoustic behavior of structural and trim helicopter panels. The aim is to improve the acoustic Transmission Loss of helicopter structures while satisfying industrial requirements (mass, thickness…). These models are besides integrated into the software PIAMCO supplied to Eurocopter.

Simulations of TL and a validation in laboratory have shown the efficiency of a sandwich panel with open cell foam for helicopter applications.
Future experiments in flight or in realistic set-up. must nevertheless be achieved to confirm this result.

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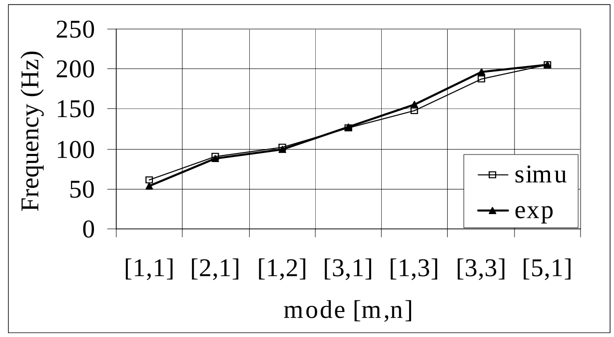
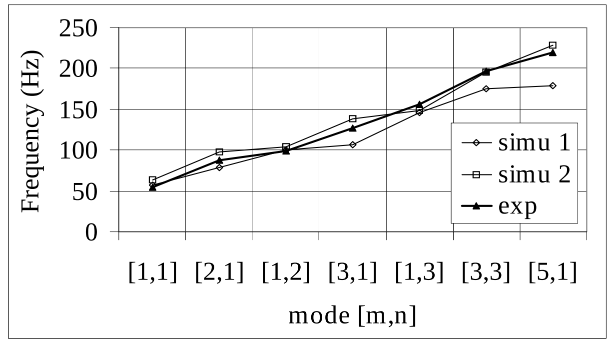
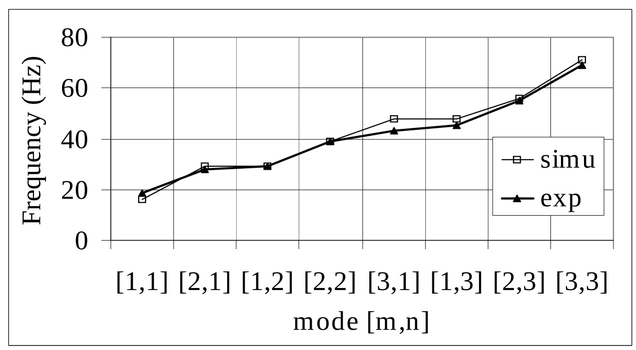


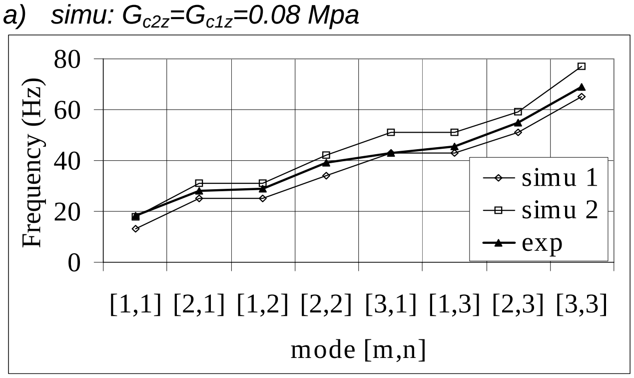
a) simu: G_{c2z}=1.1 Mpa / G_{c1z}=1.75 Mpa



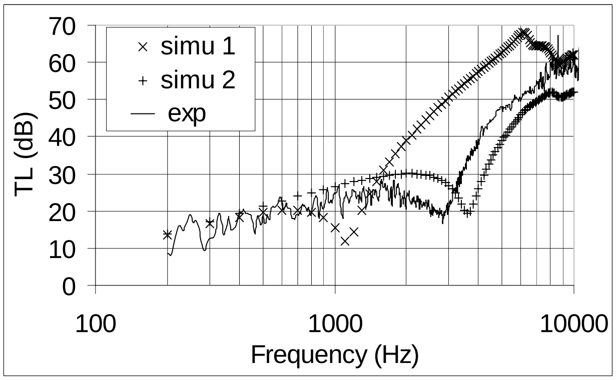
b) simu 1: G_{c2z}=0.7 Mpa - simu 2: G_{c2z}=1.4 Mpa
Fig 9: Simulated / experimental resonant frequencies of “closed cell” panel



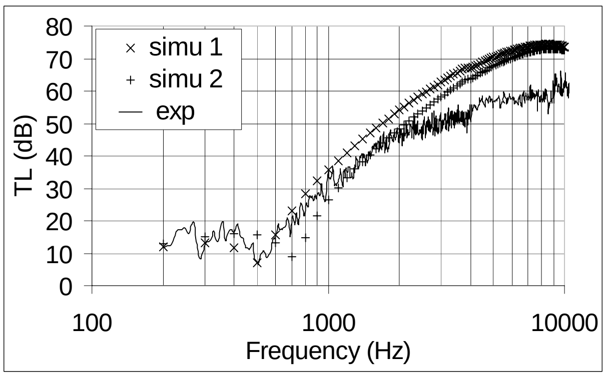
a) simu: G_{c2z}=G_{c1z}=0.08 Mpa



b) simu 1: G_{c2z}= G_{c1z}=0.065 Mpa - simu 2: G_{c2z}= G_{c1z}=0.12 Mpa
Fig 10: Simulated / experimental resonant frequencies of “open cell” panel



a) "closed cell" panel - simu 1: E_{c1z}=1 Mpa - simu 2: E_{c1z}=10 Mpa



b) "open cell" panel - simu 1: E_{c1z}=0.1 Mpa - simu 2: E_{c1z}=0.2 Mpa

Fig 11: Simulated / experimental TL

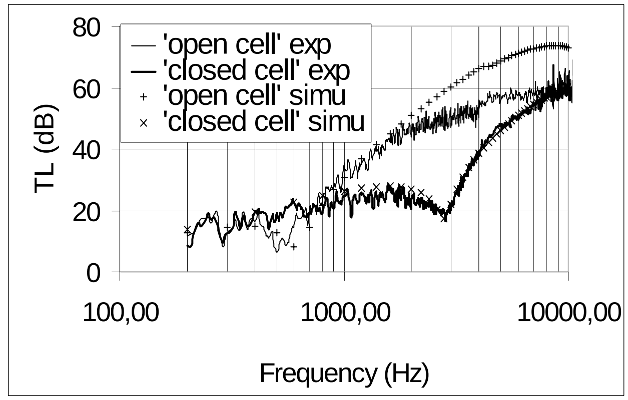


Fig 12: Optimal simu. / exp. TL of "closed and open cell" panels

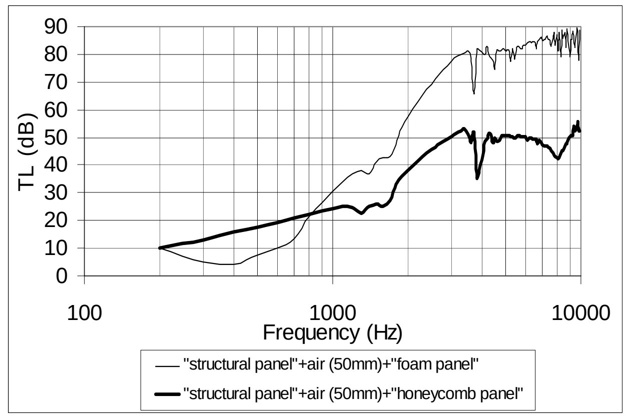


Fig 13: Simulated acoustic Transmission Loss of global wall with honeycomb or foam trim panel