A component study of the vibroacoustic damping of a non-baffled free plate partially covered with elastomer patches

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Introduction

The present study is part of an overall PhD work on noise reduction. This work aims at achieving the most appropriate passive control of any vibrating structure (cockpit, wagon) treated with localised damping material through a vibroacoustic modelling and a thorough understanding of the dissipation mechanisms.

As a starting point we focused on a suspended free thin aluminium plate partially covered with polymer patches which can be considered as a generic configuration for validating a numerical model.

The analysis of the patch location influence on the **vibration damping** was based on a 3D viscoelastic incompressible finite element formulation of the rubber together with a basic 3D formulation for the metal layers. Displacement and pressure variables were uncoupled by means of a perturbation technique. Based on a light fluid approximation this method finally yielded a fluid-structure coupling that is directly related to the **acoustic damping**.

In addition modal analysis with a non-contacting method was achieved by means of a loudspeaker and a laser-beam measurement apparatus.

Experimental set-up

As the numerical model clearly distinguishes the viscoelastic and the acoustic damping components it was attempted to reduce other damping sources for comparison purposes.

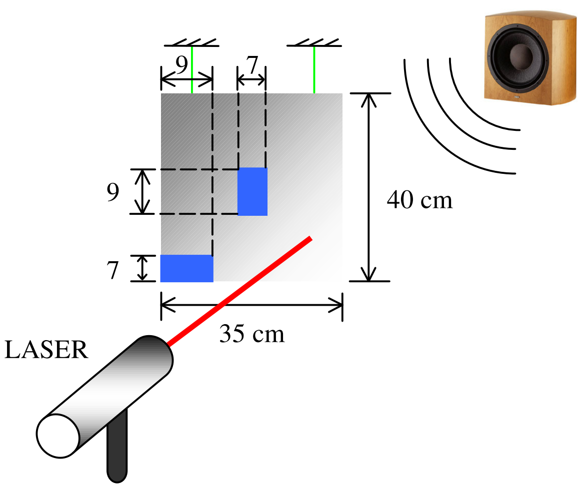


Figure 1: Modal analysis test bench

This is why a free aluminium 2mm-thick plate (35cm x 40cm) suspended by thin wires was studied for which the boundary condition damping is assumed to be relatively small.

The patches were double-layer 2mm-thick elastomer /aluminium laminae which were bonded at different locations. Four configurations were studied:

- The plate with no patch,
- A patch in the lower-left corner,
- A centred patch,
- Two patches in the same previous locations.

In a first experimental step structure modes were spotted thanks to a white noise covering the 40 – 1000 Hz frequency range.

Then the damping coefficient of each mode was measured through a generated sine noise whose frequency was tuned to the mode estimated frequency. After stopping the acoustic excitation it was noticeable (by using a logarithm scale) that the displacement signal at any point matched very well a straight line. From this line we deduced a decay coefficient which is related to a global modal damping.

Numerical aspect

Vibration Finite element model

A flexible 3D finite element code was developed to compute the structure in-vacuum displacement. Quadratic interpolation functions were used to correctly estimate the shear stresses of the polymer.

The element was constituted of 27 nodes, 27 gaussian integration points and had a specific incompressible formulation for the rubber layer with four discontinuous pressure points.

A solver capable of tackling more specifically sparse-complex matrix eigenproblems was used.

A solving iterative procedure

Isotropic material data was provided in the form of a complex Young modulus varying with the frequency. As the stiffness matrix depended on the frequency, an iterative technique was required.

Assuming the imaginary part of the modes eigenvalue was relatively small compared to its real part and to the spacing between the modes, the structure resonance modes were sought by inserting the material data at a fixed real frequency which we supposed to be close to the resonance mode's.

The computed eigenvalue real part was then used for another calculation. This strategy was repeatedly applied until good convergence was obtained. Typically three iterations were required to find the resonance's pulsation satisfying the equation (1).

$$K(Re(\omega_n)) - \omega_n^2 M = 0 \qquad (1)$$

Fluid-structure loading

The differential system of equations that governs both the displacement of the structure and the acoustic pressure was transformed into one single integro-differential equation of the displacement. This equation defines the frequency dependent eigenmodes $u_n(M)$ and the eigenfrequencies ω_n.

$$K(\omega)\,u_n(M) - \omega_n^2 M\,u_n(M) + 2\rho_f\,\omega_n^2 \int_\Sigma u_n(P)\,G_\omega(M,P)\,d\Sigma(P) = 0$$

The notation $K(\omega)u_n(M)$ expresses a scalar force value at point M. G_ω is the acoustic Green function of the spatial configuration (infinite medium with a free finite plate) when both M and P belong to the plate and was given by:

$$G_\omega(M,P) = g(M,P) - \mu_M(P)/2$$

Where g (M,P) is the free field Green function and μ_M (P) a double layer potential at point P generated by a source located in M.

By using the small parameter $\varepsilon = 2\rho_f / h\rho_s$ (ratio of the fluid to the metal density) and setting $\omega = \varpi_n$ the resonance modes v_n were given by :

$$K(\varpi_n)\,v_n(M) - \rho_p\varpi_n^2\left(\left(\frac{M}{\rho_p}\right)v_n(M) - \varepsilon\int_\Sigma v_n(P)\,G_{\varpi_n}(M,P)\,d\Sigma(P)\right) = 0$$

Assuming that $\varepsilon \ll 1$ (light fluid loading) each resonance mode was sought as a perturbation series

$$v_n = v_n^0 + \varepsilon\, v_n^1,\ \varpi_n = \varpi_n^0 + \varepsilon\,\varpi_n^1$$

This procedure yielded two sets of equations. The first one characterises the in-vacuum vibration of the plate and was resolved thanks to the FEM code.

Expanding the term v_n^1 as a series of in-vacuum resonance modes, the second set of equations led to the expression of the frequency shift $(\varpi_n^1)^2 = (\varpi_n^0)^2\,(1 + \varepsilon\,\beta_{\varpi n}(v_n^0, v_n^0))$ which depends on the modal radiation impedance $\beta_{\varpi n}$.

$$\beta_{\varpi n}(v_n^0, v_n^0) = \iint\iint v_n^0(x,y)\,G_{\varpi n}\,v_n^0(x',y')\,dx\,dy\,dx'\,dy$$

This term defines a fluid/structure coupling which is directly related to the acoustic damping.

The numerical model eventually gave a viscoelastic damping component, an acoustic damping term and the mode shifted frequency.

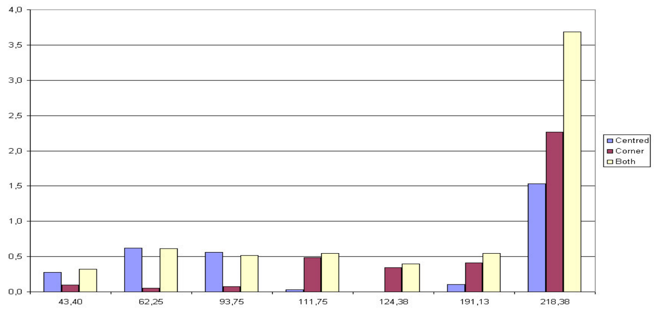
Experimental/numerical comparison

Resonance frequencies

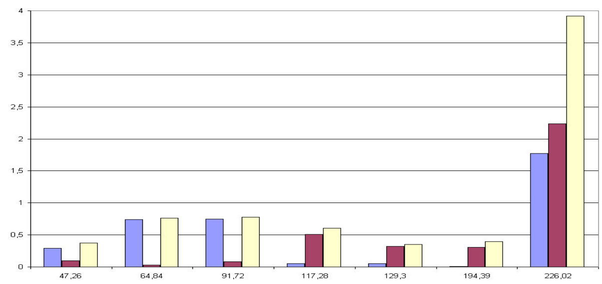
Numerical and experimental frequencies match fairly well. Nevertheless numerical modes are overestimated. This is probably due to the fact that $\beta_{\varpi n}$ is being computed via an infinite medium Green function at the moment (quadruple integration of the μ-term is not achieved yet): it is still not able to sufficiently lower the pulsation real part.

Damping

For the time being acoustic damping results are not satisfactory. It is still possible yet to compare an estimated experimental viscoelastic damping to a numerical one by substituting the plate's damping (without patch) to the damping of the other configurations (figure 2).



Experiment



FEM

Figure 2: Viscoelastic damping (left column: centred patch, middle one: corner patch, right one: both).

We notice that an excellent estimation of the viscoelastic damping is obtained.

Future work

Modelling of the fluid/structure coupling needs to be terminated (the present baffled estimation is unsatisfactory). Another case study with boundary-related damping is forecasted as well as an optimization work of the patch location for noise reduction purposes.

References

[1] *The Finite ElementMethod*, Thomas J.R. Hughes

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