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Setting Up Plane And Thin Panels With Representative Simply Supported Boundary Conditions : Comparative Results And Applications In Three Laboratories

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ABSTRACT

Compared to numerous theoretical studies, the problem of experimentally realizing supported conditions on plane panels has received very little attention. A technique to setup a simply supported rectangular plane panel for laboratory vibroacoustic tests is described. Several application cases are reported, including three panels built in Groupe d'Acoustique de l'Université de Sherbrooke (GAUS, Sherbrooke, Canada), three panels built at Laboratoire de Mécanique des Structures et des Systèmes Couplés (LMSSC, Paris, France) and one at Laboratoire Vibrations Acoustique (LVA, Lyon, France). Covered subjects include vibration and sound radiation under deterministic and random excitations (diffuse acoustic field, turbulent boundary layer), similitude laws and passive control of vibrations using piezoelectric patches or viscoelastic treatments. All obtained results highlight the interest of having access to a laboratory tool that behaves closely to an exact mathematical model, and confirm that such a tool can be reproduced and generalized between laboratories.

Keywords: Rectangular plates, structural response, sound radiation

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1. INTRODUCTION

The book that Arthur Leissa published in 1969 [1] showed the already vast literature that concerned the free vibrations of rectangular plates with '164 pertinent references', even when excluding 'such complicating effects as orthotropy, in-plane forces, variable thickness, the effects of surrounding media, large deflections, shear deformation and rotary inertia, and non homogeneity' [2]. Considering all possible combinations of the three classical boundary conditions (*i.e.* free, clamped, or simply-supported), 21 distinct cases exist for rectangular plates. The specific case of finite simply supported plates is often used as a core problem or statement since such boundary conditions lead to a simple and exact analytical solution to the plate equation of motion.

In [3] was thoroughly described a technique that allows realizing representative simply supported boundary conditions on a panel and obtaining a functional experimental mean that can easily adapt to typical vibroacoustic laboratory measurements. Three different application cases were reported that concerned mechanical impedance measurement, sound transmission loss and active control applications, and several works from the same laboratory (GAUS) were also cited as possible applications [4–6].

This paper reports experimental results obtained using panels built following [3] in different laboratories. Details (especially theoretical ones) are voluntarily skipped, and the interested reader should pay attention to the indicated references. The three goals of this communication are

1. to briefly recall the methodology presented in [3],
2. to complement previously presented results with other tests conducted at GAUS laboratory since 2016,
3. to include results obtained in other laboratories with panels built following the methodology described in [3].

All the presented results highlight once more the interest of having access to such a laboratory tool. Finally, point 3 confirms that this tool can be effectively reproduced and generalized between laboratories which is a core result.

2. THEORETICAL FRAMEWORK

The problem under study considers a homogeneous and isotropic rectangular plate of length L_x and width L_y , and with a thickness h considered small compared to L_x and L_y (only the transverse deflection $w(x, y; \omega)$ is considered, thin panel hypothesis). For such condition, the free vibration equation for small harmonic displacements is

$$\tilde{D} \left(\frac{\partial^2}{\partial x^2} + \frac{\partial^2}{\partial y^2} \right)^2 w(x, y; \omega) - \rho h \omega^2 w(x, y; \omega) = 0, \quad (1)$$

where ρ is the mass density of the plate, ω the angular frequency and \tilde{D} the complex bending stiffness ($\tilde{D} = \tilde{E}h^3/12(1 - \nu^2)$, with \tilde{E} the complex Young's modulus including the structural damping factor η so that $\tilde{E} = E(1 + j\eta)$, and ν the Poisson's ratio). Achieving ideal simply supported conditions implies supporting a plane and thin structure, for which any edge can rotate freely while being restricted from out-of-plane displacements. Bending moments and transverse displacements are thus null on the plate boundaries. In this case, the eigenfrequencies $\tilde{\omega}_{mn}$ and eigenfunctions ϕ_{mn} , solutions of Eq.(1) take simple closed-form expressions :

$$\tilde{\omega}_{mn} = \left(\frac{\tilde{D}}{\rho h} \right)^{1/2} \left[\left(\frac{m\pi}{L_x} \right)^2 + \left(\frac{n\pi}{L_y} \right)^2 \right], \quad (2)$$

and

$$\phi_{mn}(x, y) = \sin\left(\frac{m\pi x}{L_x}\right) \sin\left(\frac{n\pi y}{L_y}\right), \quad (3)$$

with m, n non-zero, strictly positive integers.

The displacement response of the plate at any (x, y) location is given by the following modal expansion

$$w(x, y; \omega) = \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} q_{mn}(\omega) \phi_{mn}(x, y), \quad (4)$$

where $q_{mn}(\omega)$ is the complex modal amplitude which for a given modal force F_{mn} (depending on the nature of the plate's excitation) can be written

$$q_{mn}(\omega) = \frac{1}{\rho h L_x L_y} \frac{F_{mn}}{\tilde{\omega}_{mn}^2 - \omega^2}. \quad (5)$$

The ratio of vibration velocity v to an arbitrary force F , $Y(\omega) = v/F$, is the mobility function of the structure. $Y(\omega)$ can be derived using Eqs.(2-5), with v calculated as the displacement w multiplied by $j\omega$, and finally

$$Y(\omega) = j\omega \frac{1}{\rho h L_x L_y} \sum_{m=1}^{\infty} \sum_{n=1}^{\infty} \frac{\phi_{mn}^2(x, y)}{\omega_{mn}^2 - \omega^2}. \quad (6)$$

3. APPROACHING ACTUAL BOUNDARY CONDITIONS OF A SIMPLY SUPPORTED PANEL

The accurate modeling of the boundary conditions applied along the edges of a structure as their practical implementation share common difficulties. Boundary conditions are generally modeled using a rotational spring of stiffness k_r and a linear spring of stiffness k_f . Unlike ideal simply supported boundary conditions that imply a null rotational stiffness and an infinite transversal stiffness (i.e. $k_r = 0$ and $k_f = +\infty$), realistic simply supported boundary conditions are usually achieved with a small but finite rotational stiffness, and a large but finite transverse stiffness (a small amount of stiffness restrains the edges rotation, and a small but non-zero movement is allowed in the z -axis). The simply supported boundary condition will be achieved here by rigidly attaching the plate's edges to glued thin vertical supports. The supports that will now be called 'blades' throughout the paper, should be flexible enough to provide a small but finite rotational stiffness, but also rigid enough to support the panel's weight and provide sufficient translational restraint. The structure under consideration is a plane panel with dimensions $(L_x \times L_y \times h)$ and Young's modulus E . The dimensions of the vertical blades are $(L_b \times l_b \times h_b)$, and Young's modulus E_b . At a given frequency, the rotation stiffness can be approximated by an equivalent rotational spring constant per unit length for the blades named k_r^b :

$$k_r^b = \frac{E_b h_b^3}{12 l_b}. \quad (7)$$

A coefficient k_r^p may also be defined for the panel to be supported :

$$k_r^p = \frac{E h^3}{12 L_x}, \quad (8)$$

A simple and necessary condition for approaching simply supported boundary conditions would be $k_r^b < k_r^p$. In order to better predict the error on the plate eigenfrequencies induced by imperfect boundary conditions regarding to perfect simply supported boundary conditions, a criterion based on the ratio of these two coefficients was

proposed in [3] and using a similar material for both panel and blades so that their Young's modulus are identical, that is

$$\frac{k_r^b}{k_r^p} = \frac{h_b^3 L_x}{h^3 l_b} \quad (9)$$

This ratio is used to relate frequency shift of eigenfrequencies when the height and thickness of the supporting blades are varied for fixed dimensions of the plate. Only the length L_x of the panel is considered for calculations since for a given value of the ratio k_r^b/k_r^p calculated with L_x and set as an upper bound, the criterion will be implicitly satisfied for L_y if $L_x > L_y$.

In [3], it was found that the values of the ratio k_r^b/k_r^p nearly vary as the percentage deviation on the frequency of mode (1, 1). Since the percentage change in terms of frequency tended to be smaller for higher modes compared to the fundamental one, the first vibration mode can be used to define a higher bound for frequency precision (that is then reached for all the other vibration modes). The ratio k_r^b/k_r^p can be linked to an objective upon frequency precision and be used as a design parameter to define adequate couples of height and thickness (l_b, h_b) values) for the blades supporting a given panel with fixed length, width and thickness. Such couples of values may be defined using a mapping as given in Fig.4 in [3]. If a maximum frequency deviation $\% \Delta f$ of approximately 5% compared to ideal simply supported conditions is targeted on the first natural frequency of a panel (panel A in this case, for which dimensions are given in Table 1), the following simple empirical condition was finally proposed in [3] : $\% \Delta f \leq 5\% \Leftrightarrow \frac{k_r^b}{k_r^p} \leq 0.1$.

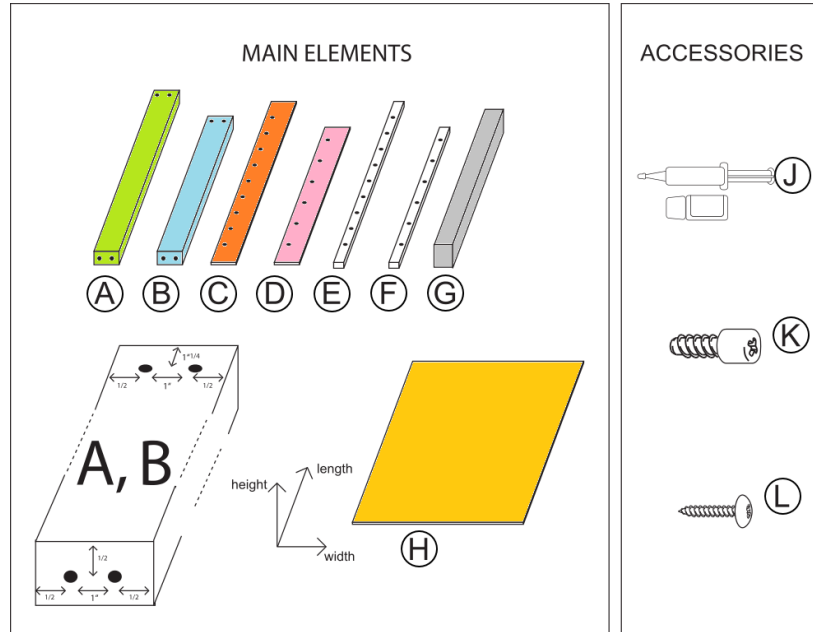


Figure 1. Simply supported panel schematic parts list.

4. PANEL CONSTRUCTION

This section briefly describes the general method used to construct all the panels that have been used in this paper. Figure 1 schematically describes the main elements and accessories needed. The choices for the panel setup and its constitutive components were guided by simplicity, with only eight main components (parts A to H) and simple building steps (see Figure 2).

Precise descriptions and additional technical details for all needed parts (A to L) are provided in [3]. The frame is made of parts A, B, E and F, all made of steel. The panel (part H) and supporting blades (parts C and D)

are made of aluminium. It is precised that steel is chosen for the frame partially for reasons of cost, but mainly to ensure a high '*panel-and-edges to frame*' mass ratio so that the frame nearly behaves as a rigid and massive foundation from the panel's point of view. A mass ratio of approximately 0.09 is finally obtained for panel A, described in section 5.1 (with a weight of 20.7 kg for the frame, and 1.8 kg for the panel and edges).

Figure 2 then gives visual instructions for assembly, which are to follow from left to right and top to bottom. It is precised that a main advantage of this setup is that the '*panel-and-blades*' part can be easily disassembled and reassembled from the frame.

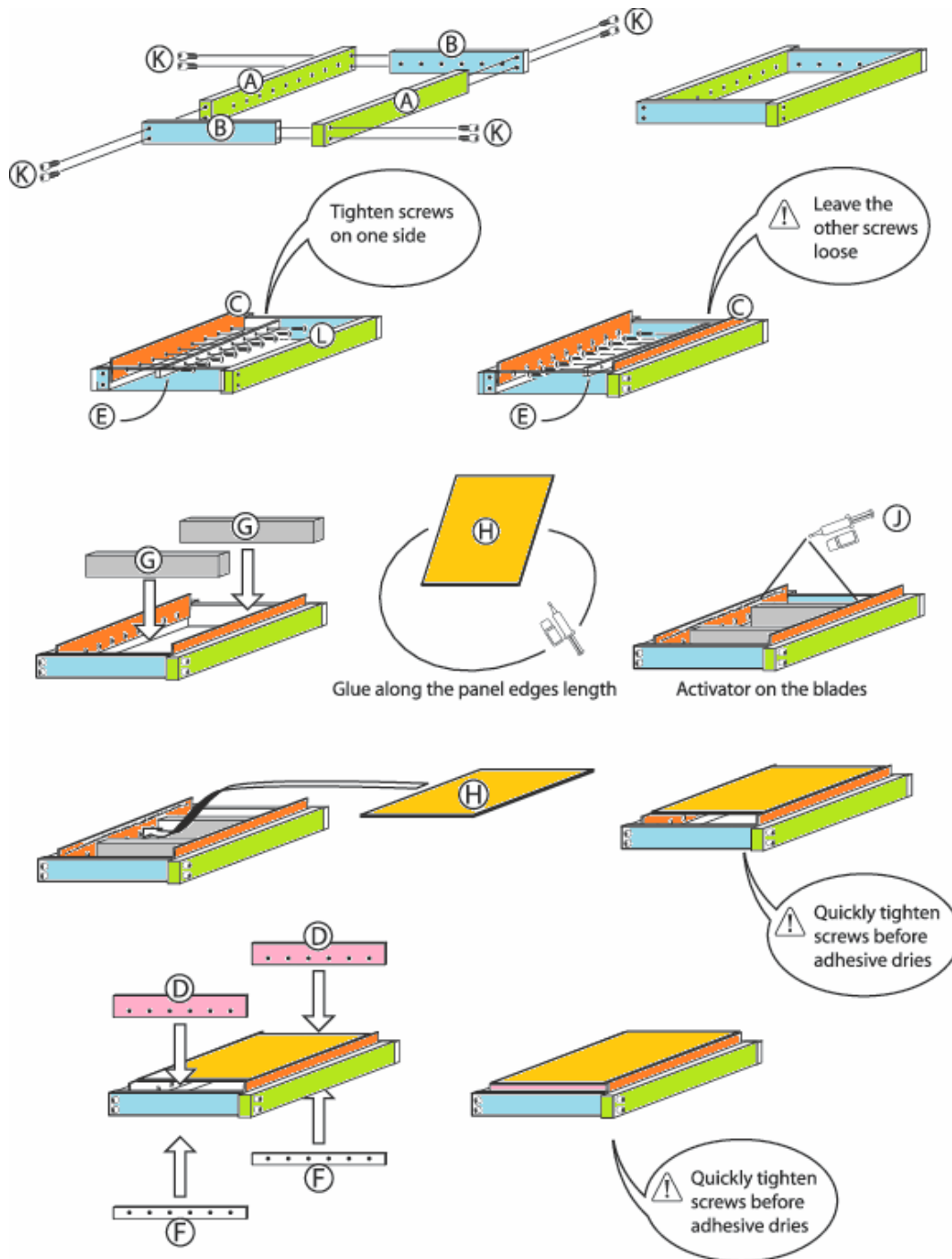


Figure 2. Assembly plan (to be read from left to right and top to bottom).

5. APPLICATION CASES

5.1 LMSSC laboratory

In this section, the pursued application is the study of vibration damping, either provided by viscoelastic patches or by piezoelectric patches (see Figures 3(b-c), respectively). Those experiments follows previous works on (1) inverse characterization of viscoelastic materials using vibration measurements on structures [7] and (2) vibration control using piezoelectric coupling with a passive electrical network composed of 42 elemental cells [8]. Three aluminium panels of dimensions $420 \times 360 \times 3 \text{ mm}^3$ were built at LMSSC laboratory following the indications provided in [3], and are used to evaluate damping performances of different strategies, and perform a cross-validation between simulation and experimental results.

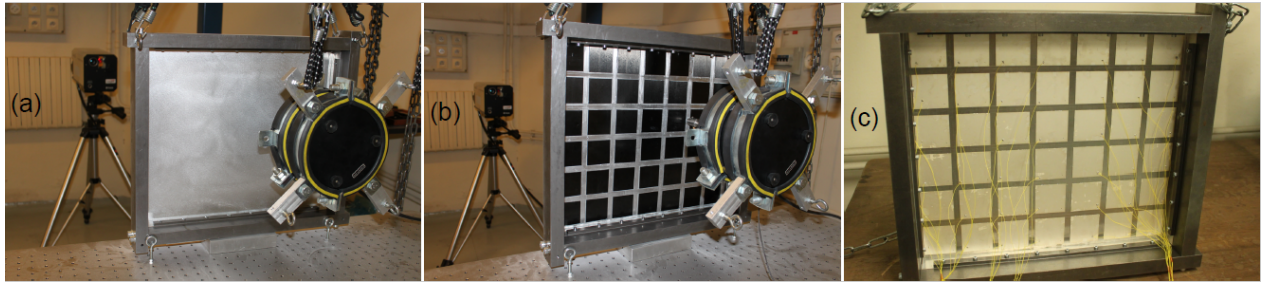


Figure 3: Measurement at LMSSC using vibration shaker and scanning laser vibrometer - (a) Bare panel - (b) Viscoelastic patches - (c) Piezoelectric patches.

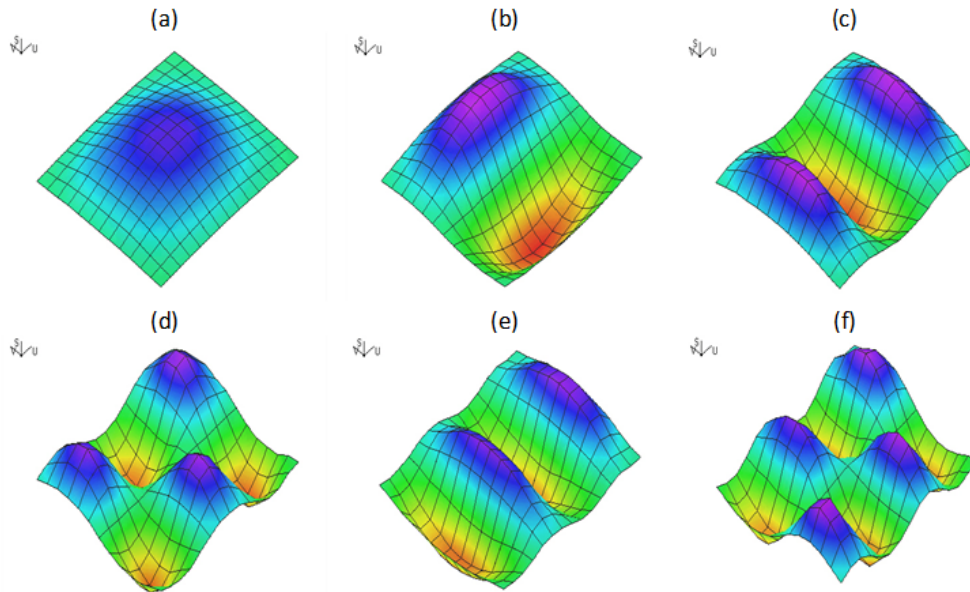


Figure 4: Operational deflection shapes measured at LMSSC laboratory - (a) mode (1,1); (b) mode (1,2); (c) mode (3,1); (d) mode (3,2); (e) mode (4,1); (f) mode (4,2).

For the bare panel (i.e. without any damping device), operational deflection shapes were obtained using a scanning laser vibrometer (see figure 3(a)). The corresponding results for six vibrations modes [(1,1), (1,2), (3,1), (3,2), (4,1), (4,2)] are reported in Fig. 4, with an excellent agreement with theoretical mode shapes (see Eq. 3). Also, in Table 1, the theoretical and measured eigenfrequencies are reported and confirm the very good

agreement between analytical and experimental results. As for the GAUS panel (see next section), the largest deviation is seen for the first vibration mode. For the ten first vibration modes, the mean percentage deviation is 1%.

The theoretical mobility function (vibration velocity upon force) is calculated at the excitation point by using Eq. (6) for a unit force input. The experimental mobility function at the excitation point was extracted for the two firstly built panels. Figure 5 shows the comparison of the two experimental and the single theoretical mobility functions. The agreement between analytical and experimental results is very satisfactory on the considered frequency range. The similarity of response between the two built panels is especially impressive.

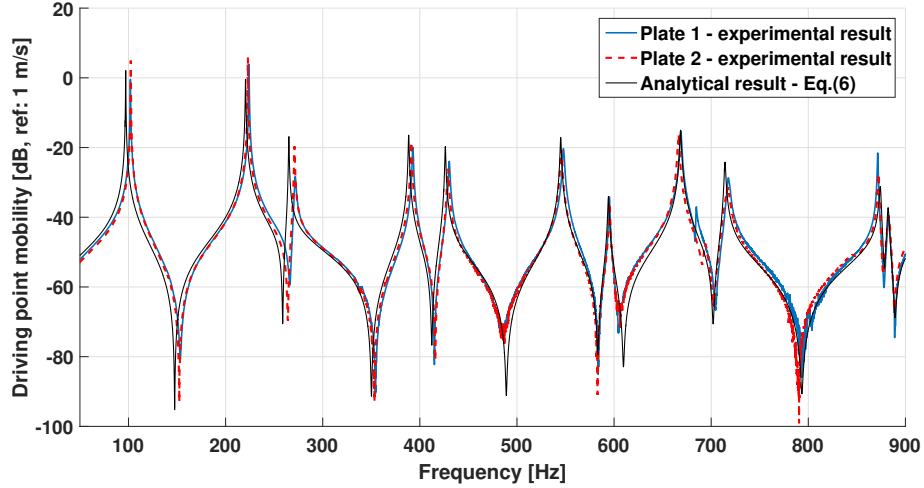


Figure 5: LMSSC laboratory - Measured driving point mobility for two plates of same geometry successively built, compared with an analytical result obtained with Eq.(6).

Fig.6 shows preliminary results comparing damping performances of passive viscoelastic and piezoelectric treatments (the case of the bare panel is compared with those two treatments). To achieve damping, the viscoelastic solution consists in a constrained layer damping, while the array of piezoelectric patches is coupled to a multi-resonant passive electrical network. Over the frequency range of interest (0-900 Hz), the amplitude of the first ten resonance peaks is reduced by more than 20 dB. This validates the efficiency of both solutions and allows comparisons with numerical models.

5.2 GAUS laboratory

This part of the paper concerns similitudes for the vibration of plates under a random excitation (like the turbulent boundary layer), and both analytical and experimental analyses were conducted in [9, 10]. Additional details can be found in these two references, and results concerning similitude for the dynamic vibration response and radiated acoustic power of flexural plates under a point mechanical excitation are provided in [10]. For the case of a turbulent boundary layer excitation, it was shown that under the assumption of proportional sides, perfect similitude in terms of vibration response scaling can theoretically be achieved between plates of variable thicknesses. A scaling procedure was derived using a modal approach and including scaling coefficients for both structure and excitation. Indeed, when the case of a panel under a random excitation is considered, the excitation and the plate vibration response must be both described in terms of auto-spectral and cross-spectral density functions. A modal approach can be considered to calculate the panel response to this excitation, and the auto spectral density function of the displacement at point $A(x_A, y_A)$ due to an assigned random distributed

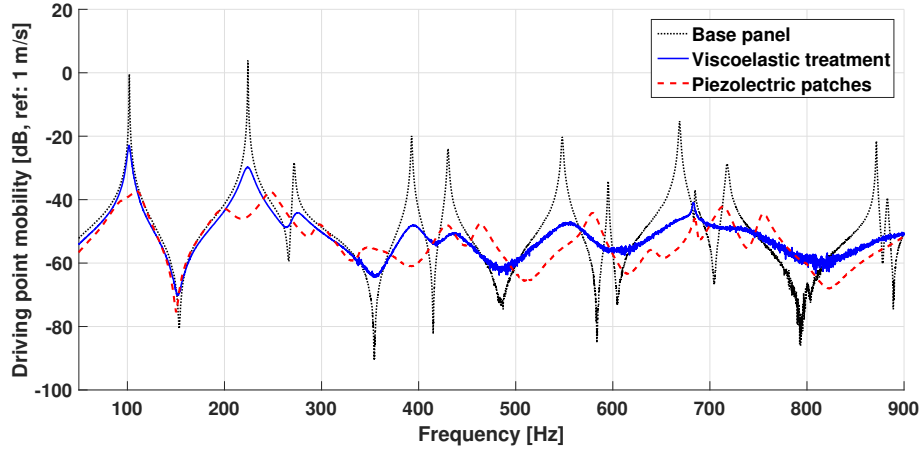


Figure 6: LMSSC laboratory - Measured mobility for bare panel, panel with viscoelastic treatment and piezoelectric patches.

Table 1: Theoretical and experimental eigenfrequencies for the ten first modes, and corresponding percentage deviation.

(m,n) indices of considered mode	(1,1)	(2,1)	(1,2)	(2,2)	(3,1)	(1,3)	(3,2)	(2,3)	(4,1)	(3,3)
GAUS laboratory - panel dimensions : 480 × 420 × 3.2 mm³										
Theoretical frequency (Real part, Eq. (2)) (Hz)	77	177.3	207.9	304.1	344.3	426.1	475.2	526.3	578.1	693.3
Measured frequency at response peak (Hz)	82.1	181.3	212.5	307.8	348.4	431.3	478.1	530.6	584.4	696.9
Percentage frequency deviation %	6.6	2.3	2.2	1.2	1.2	1.2	0.6	0.8	1.1	0.5
LMSSC laboratory - panel dimensions : 420 × 360 × 3 mm³										
Theoretical frequency (Real part, Eq. (2)) (Hz)	97.5	221.4	266.1	390	427.9	547.2	596.5	671.1	717	877.6
Measured frequency at response peak (Hz)	102.3	225.4	269.8	391.9	430	547.7	595.2	670.2	718.3	883.1
Percentage frequency deviation %	4.9	1.8	1.4	0.5	0.5	0.1	0.2	0.1	0.2	0.6

excitation can be found with the following modal expansion:

$$G_{ww}(x_A, y_A; \omega) = \sum_{i=1}^N \sum_{k=1}^M \left[\frac{\phi_i(x_A, y_A) \phi_k(x_A, y_A)}{Z_i^*(\omega) Z_k(\omega)} \right] \left[\frac{S_p(\omega) (L_x L_y)^2}{\gamma_i \gamma_k} \right] A_{Q_i Q_k}(\omega), \quad (10)$$

where for the plate i -th structural mode (i and k indices corresponding to couples of (m, n) values), Z_i is the plate modal impedance, $\gamma_i = \int_0^a \int_0^b \phi_i^2(x, y) dy dx$, and the terms $A_{Q_i Q_k}(\omega)$ are usually known as the acceptances: joint acceptance for $i = k$, or cross acceptance for $i \neq k$. Finally, $S_p(\omega)$ is the single point spectral density of the forcing pressure field. Theoretical details concerning the calculations for reference panels and derived similitude laws can be found in [9, 10]

The interest of setting up a scaled family of simply supported panels was to provide an experimental validation of the theoretical scaling procedure. Three plates are considered and their dimensions are reported in Table 2. In table 1 are also reported the theoretical and measured eigenfrequencies for the panel A, as well as the corresponding percentage frequency deviation (with the largest deviation for the first vibration mode which is a known result, see [3]). For the ten first vibration modes, the mean percentage deviation is 1.8%.

Table 2. Plate dimensions (mm) and corresponding scaling coefficients

Plate	L_x	L_y	h	r_x	r_y	r_h
A (Reference)	480	420	3.2	-	-	-
B (Proportional Sides, same thickness)	600	525	3.2	1.25	1.25	1.00
C (Proportional Sides, \neq thickness)	600	525	2.4	1.25	1.25	0.75

The three panels were all built following [3]. The material properties are those of a typical aluminum alloy, with a Young's modulus value $E = 70$ GPa, a mass density $\rho = 2700$ kg.m⁻³, a Poisson's coefficient $\nu = 0.3$ and a structural damping $\eta = 0.02$ (considered constant as a function of frequency). The derived similitude ratios (or scaling coefficients) are also reported in Table 2 (denoted r_x , r_y and r_h for length, width and thickness, respectively). The plate A is considered as the reference plate, so that measurement results for panels B and C are scaled to reobtain panel A response.

Measurements were conducted in the anechoic flow facility at Université de Sherbrooke. Each plate was flush-mounted in a wood baffle placed at the convergent nozzle exit, and mechanical decoupling was ensured and verified (the panel was individually supported by its own stand). Seven accelerometers were glued on the panel in order to estimate the vibration autospectrum and cross-spectrum under turbulent boundary layer excitation. The experimental measurements were carried at scaled flow speeds, with speeds of 35, 28 and 21 m/s (for panels A, B and C, respectively). Acceleration autospectral density measured at several discrete points on the three parent panels at different flow speeds are finally scaled following the theoretical procedure defined in [9, 10]. The unscaled and scaled experimental results are presented in figure 7. The agreement between the three structural responses is very satisfactory up to a frequency of 1.5 kHz and confirms the validity of the proposed scaling approach.

5.3 LVA laboratory

As a complement to the previous section, another method for calculating the vibration response of a panel to a random excitation considers coupling so-called sensitivity functions and a cross-spectral density function describing the excitation, now both defined in the wave number domain. The one-sided frequency auto spectral density function of the velocity $G_{vv}(\mathbf{x}, f)$ at point $\mathbf{x}(x, y)$ can be approximated by a discrete summation [11]:

$$G_{vv}(x, y; \omega) \approx \frac{1}{4\pi^2} \sum_{\mathbf{k} \in \Omega_{\mathbf{k}}} |H_v(x, y; \mathbf{k}; \omega)|^2 G_{pp}(\mathbf{k}, \omega) \delta \mathbf{k}. \quad (11)$$

with $\mathbf{k} = (k_x, k_y)$ is the wave vector defined in the plane (x, y) , $\delta \mathbf{k}$ represents the wave number resolution and $\Omega_{\mathbf{k}}$ is the finite wave number domain over which the discrete integration is performed. The function $G_{pp}(\mathbf{k}, \omega)$ corresponds to the cross-spectral density function of the wall-pressure field (for instance a diffuse acoustic field or a turbulent boundary layer excitation, that can be described using models). The term $H_v(x, y, \mathbf{k}, \omega)$ is called the sensitivity function and characterize the vibration behavior of the panel.

A first goal in this work was to show that the sensitivity functions (see Eq. (11)) could be determined using the reciprocity principle, which makes it easier to be determined from an experimental point of view (and is a key point into the methods proposed in [11, 12]). Theoretically, these functions are defined as the panel's response to wall-pressure plane waves and characterize the panel's vibroacoustic behavior. Practically and thanks to the used

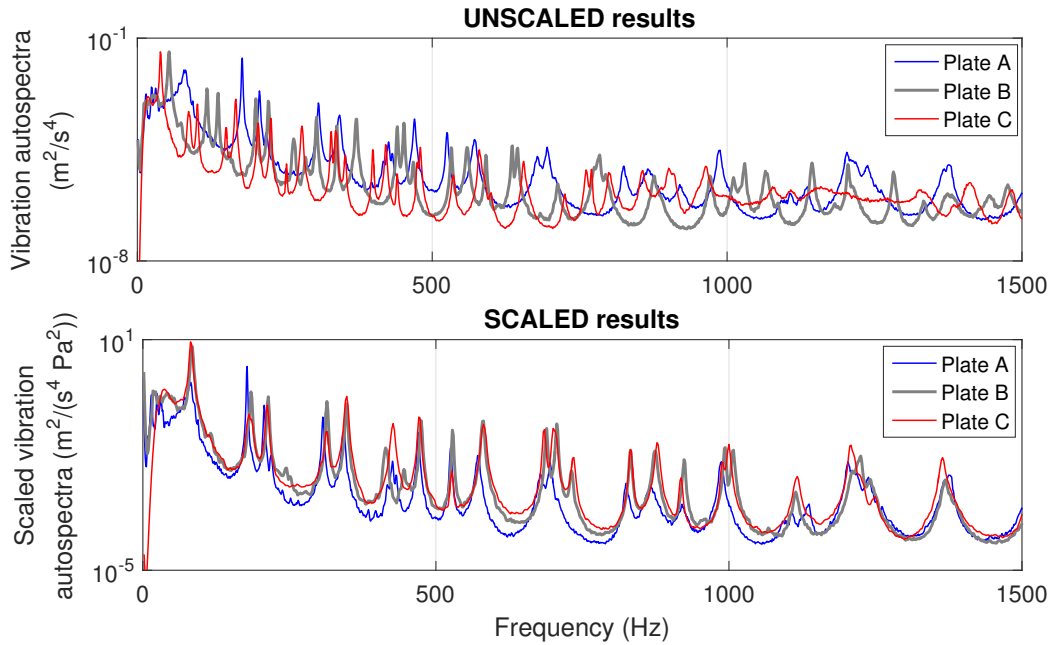


Figure 7: Measurement at GAUS laboratory - Unscaled (upper part) and scaled (lower part) mean vibration autospectra - flow speeds of 35, 28 and 21 m/s are considered for panels A, B and C, respectively (adapted from [9]).

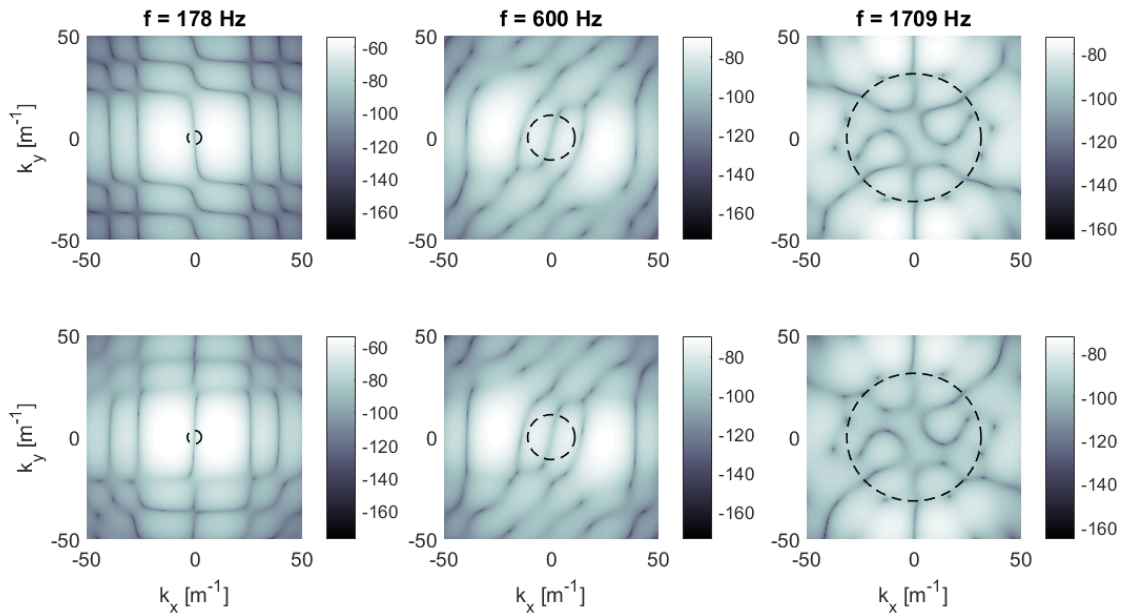


Figure 8: LVA laboratory - Calculated (upper part) and measured (lower part) sensitivity functions at three different frequencies using vibration shaker and scanning laser vibrometer - The indicated circle has a radius k_0 , the acoustic wavenumber - Adapted from [11].

simply-supported panel, it has been shown that these functions can be determined from the panel velocity field in the wavenumber domain when the system is excited by a source of unit amplitude at a point of interest. The panel used in this section is similar to panel A in section 5.1, with the difference that the *panel and edges* part had to be rebuilt at LVA after being damaged during transport between GAUS and LVA laboratories. Compared to original

panel A from GAUS, it includes possible variability on construction. The cousin panel at LVA was used in [11] while the original panel A (see Table 1) was used at GAUS in [12]. One great advantage in this study was to be able to conduct distant and parallel tests (between Canada and France) with two panels that behaved closely with differences lower than 1 %.

In figure 8 are compared the measured sensitivity functions (using the reciprocity principle) to theoretical ones. The result at a frequency of 178 Hz corresponds to the (2,1) vibration mode, whereas the results at 600 and 1709 Hz are off-resonance cases. The numerical and experimental results are generally in very good agreement, especially within the circle of radius k_0 , the acoustic wave number.

Once the reciprocity approach for determining the sensitivity was validated, the vibration response of the panel to a diffuse acoustic field excitation could be calculated (Eq.(11)). Figure 9 compares the obtained numerical and experimental results. A good agreement is observed between the two results, which allows validating the proposed methodology for the case of a diffuse acoustic field excitation (prediction of the sound transmission loss was also shown to be feasible [11]). The methodology was also applied and validated for the case of a turbulent boundary layer excitation [12], still using panel A for testing (now in a wind-tunnel) and by performing comparisons with analytical results (obtained for a simply-supported panel).

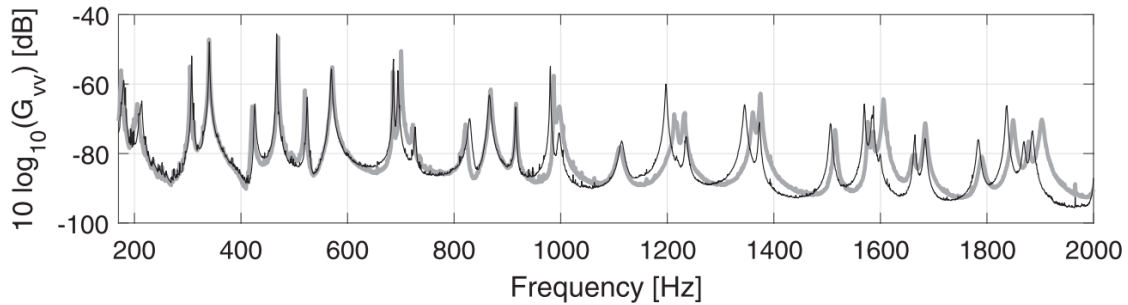


Figure 9: LVA laboratory - Velocity ASD functions G_{vv} (dB, ref. $1 \text{ m}^2 \text{ s}^2$) obtained using Eq.(8) with theoretical and measured sensitivity functions (a theoretical model for the diffuse acoustic field is used) - Numerical result (bold gray line) vs experimental result (light black line) - Adapted from [11].

6. CONCLUSION

This paper described a technique for realizing representative simply supported boundary conditions on a panel and obtaining a functional experimental mean that can easily adapt to typical vibroacoustic laboratory measurements. In order to ease the spread of this laboratory tool, precise technical guidelines were given with an assembly plan in [3] hoping that it could be acquired, used and improved by the research community. All the obtained experimental results in the three different laboratories and applications cases shown good agreement with analytical solutions and numerical predictions. This confirms the interest and the robustness of the proposed methodology for building panels with such boundary conditions. The results obtained in LMSSC and LVA laboratories with panels built following the instructions provided in [3] show that the tool can be effectively reproduced between laboratories.

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