One of the top priorities in the general transportation industry for noise, vibration and harshness objectives is the design of lightweight and efficient soundproofing packages. This paper focuses on a concept of small-scale distributed tuned mass dampers applied to the case of a realistic aircraft sidewall panel. The targeted bandwidth encompasses the curved fuselage’s ring mode, which is known to produce an important dip in the sound transmission loss of curved structures. A variety of experimental realizations are reported, including an investigation of the resonators distribution on the measured sound transmission loss of the panel under a diffuse acoustic field excitation.

Keywords: Vibration control; Curved panel; Sound Transmission Loss; Small-scale resonator; Ring frequency; Diffuse Acoustic Field

1. Introduction

Lightweight structures are ubiquitous in the field of transportation engineering. Typical examples are composite assemblies, sandwich, multi-layer configurations or stiffened panels, usually involving a large stiffness-to-weight ratio. This ratio theoretically goes against vibroacoustic performance, and usually requires additional soundproofing packages to ensure acoustic comfort. In order to enhance the acoustic insulation properties of such lightweight structures, two strategies can be considered. The first consists in exploiting sound absorption properties of porous materials to reduce the transmission/reflection of sound waves, which is generally efficient in the medium/high frequency range. The second approach relies on dynamical control techniques to reduce the vibration response, e.g. using viscoelastic layers and/or other damping materials. This latter approach is generally more effective in the low frequency range, where the acoustic wavelength is larger than the structure’s characteristic dimensions.

In both cases, a large number of "periodic" or "locally resonant" solutions, usually gathered under the term metamaterials, has been proposed to enhance the structural or acoustic insulation properties of
lightweight structures. A first example is the inclusion of Helmholtz resonators in porous soundproofing packages, these local resonances providing a narrow-band insulation capability in addition to the broad frequency coverage of porous-only soundproofing packages. Recent applications of this concept can be found in [1, 2]. Another examples concerns Tuned Mass Dampers (TMD), that were proposed in the seminal work of Ormondroyd and Den Hartog [3], and have been widely extended in the context of distributed (or multiple) TMDs (distributed TMD concepts are detailed in Fuller and Harne [4]). The integration of TMDs along periodic patterns to combine local resonances with periodic scattering effects (e.g. Bragg resonances) have contributed to revive the concept of distributed TMDs in the framework of locally resonant metamaterials. Some recent applications of locally resonant metamaterials to aircraft sidewall panels or curved shells can be found in [5, 6].

The present work focuses on the experimental validation of a multi-resonant TMD solution that aims at solving a specific issue met in the case of curved panels. Analogously to the coincidence dip at the critical frequency (that is common to plane and curved panels), the curvature of a panel leads to another dip in the panel sound transmission loss (STL) at the so-called ‘ring frequency’ of a full cylinder having the same radius as the panel [7, 8]. Droz et al. [6] proposed a proof-of-concept using 3D-printed tunable resonators for improving STL of an aircraft fuselage panel at its ring frequency. In this paper, the main results of [6] are reported, including the influence of resonators’ spatial distribution and frequency tuning on the STL improvement versus the overall added mass. Since the proposed concept involves tunable resonators, several multi-resonant solutions involving a combination of resonators tuned at different frequencies were investigated so as to enlarge the efficiency bandwidth of the locally resonant add-on. The possibility of coupling the TMDs with the existing sound packages is finally discussed.

2. Random fields and design of proposed 3D-printed resonators

When a structure is subjected to a single acoustic plane wave excitation, the structural-acoustic interaction effects (e.g. acoustic coincidence, ring modes, local resonances) produce narrow-band STL dips, which can typically be addressed by well tuned mass-spring systems. On the other hand, when the structure is subjected to Diffuse Acoustic Field (DAF) or Turbulent Boundary Layer (TBL) excitations, a broader STL dip is usually observed at critical frequencies. This phenomenon is due the enlarged complexity of the excitation (in terms of incidence angles, and magnitude), but also coupling with a structure that might not have a purely isotropic behavior. In the present work, where the structure is subjected to a DAF excitation, or in the case of a TBL excitation, this sensitivity of the STL to the directivity of the propagating waves requires a particular attention to the following design parameters:

- The distribution and spacing between the resonators,
- The directivity (or omni-directionality) of the bandgaps produced by the resonators,
- The damping, and more generally the bandwidth of the coupled host-resonator modes.

The first parameter affects the wave propagation, either by altering or introducing orthotropy in the structure, or by producing Bragg or other periodicity-related scattering effects. Interested reader can refer to recent work on the design and modelling of similar resonators [9]. The sensitivity to the resonators’ distribution and spacing increases with frequency. The second parameter has similar consequences but is solely dependent on the geometry of the resonators, even when the wavelength is large compared with the periodic unit-cell’s dimensions. The last parameter is the most influential in the case of a random aerodynamic load. As mentioned above, the orthotropy of the curved fuselage results in broader bandwidth of the STL dips at critical frequencies. Thus, the use of higher damping values is a solution to
enlarge the bandgap produced by the locally resonant add-on. The Wave Finite Element Method [10][11] is used to compute the dispersion diagrams and identify the bandgaps of the resonators. The classical WFEM exploits periodic structure theory to derive the propagation constants $\Lambda$ and eigenvectors $\Psi$ from the direct Bloch formulation, which writes in the 1D case:

$$[\mathbb{D}_{RL}(\omega)\Lambda^{-1} + (\mathbb{D}_{RR}(\omega) + \mathbb{D}_{LL}(\omega)) + \mathbb{D}_{LR}(\omega)\Lambda]\Psi = 0$$

(1)

where the dynamic stiffness sub-matrices are denoted $\mathbb{D}_{ab}(\omega) = K_j + j\omega C - \omega^2 M$ with $a$ and $b$ the left and right sides of the unit-cell while we denote $K$, $C$ and $M$ the finite element stiffness, damping and mass matrices of the periodic cell. The influence of material damping on the wave reflection, transmission and absorption coefficients of the considered locally resonant add-on was investigated in [12] using 3D FEM and a diffusion-based modelling technique. This technique allows a de-coupling between the computation of the Bloch waves propagating in the host structure and the locally resonant add-on, represented as a coupling waveguide’s singularity. In order to reduce the computational effort a reduced state vector technique [13][14] is used with the Wave Finite Element method [10][11].

At the end of this procedure, the proposed design for small-scale resonators is based on a 3D-printed core structure using polycarbonate polymer, see Figure 1. It consists in a beam of section $10 \times 4 \text{mm}^2$ and length 23 mm supported by a stiffener. The base part is a rectangular prism of dimensions $10 \times 10 \times 8 \text{mm}^3$ on which the beam and the stiffener are both connected. The connection to the host structure is made at the base using an adhesive. The overall unit volume occupied by a resonator without a tuning mass is finally $33 \times 10 \times 8 \text{mm}^3$, with a unit weight of 2.72 g.

A 1 mm-thick base magnet is glued at the tip end and the resonators are then tuned using different neodymium magnets of known masses. The corresponding resonance frequencies for the resonators are 2.24 g and 670 Hz, 1.11 g and 820 Hz, and 0.74 g and 980 Hz, respectively. These average resonance frequencies were estimated from a few resonators once installed on the tested panel using a laser Doppler vibrometer and a shaker excitation.

![Figure 1](image1.png)

Figure 1: a) Schematic describing the different parts of the resonator (b) Close-up picture of the resonator tuned at 980 Hz.

### 2.1 Experimental methods

The panel tested is a curved rectangular fuselage panel equipped with circumferential and axial stiffeners, all riveted to the skin of the panel (see Figure 2). The panel’s length is 1.7 m with respective inner and outer circumferences of 1.45 m and 1.3 m, with an approximate curvature radius $r = 1.34$ m. The panel is made of 1.27 mm-thick aerospace grade aluminum with a 21.4 kg total weight. The experimental set-up is also detailed in Droz et al. [6] and is shown in Fig.2. The approximated theoretical ring frequency $f_r = 634$ Hz is determined using the simplified relation for a non stiffened thin shell [7].
\[ f_r = (2\pi r)^{-1} \sqrt{E/\rho(1-\nu^2)}. \]
In this relation, \( E \) is the Young’s modulus (\( = 70 \) GPa), \( \rho \) the mass density (\( = 2700 \) kg/m\(^3\)) and \( \nu \) the Poisson ratio (\( = 0.3 \)).

Figure 2: Picture of the inner (left) and outer (right) parts of the aircraft fuselage, placed in the coupled reverberant-anechoic rooms with embedded resonators.

The panel was mounted in the test window between coupled reverberant-anechoic rooms using a frame made of plywood with acoustic sealant and subjected to a Diffuse Acoustic Field excitation (see [15] for description of tests conducted with a similar set-up on this panel). Following standards [16, 17], STL is determined using measurements of the spatially averaged sound pressure level in the source room \( L_p \) and of the spatially averaged average sound intensity level \( L_i \) over a scanning surface \( S_m \) on the receiving side (both in dB), \( \text{STL} = L_p - L_i - 6 - 10 \log_{10}(S_m/S) \) (with \( S \) the effective panel area, considered equal to the scanning area \( S_m \) so that the last term was neglected). \( L_p \) was obtained using a rotating half-inch PCB microphone on the reverberant room side, while the average radiated sound intensity level \( L_i \) was measured in the anechoic room using a Bruel & Kjaer sound intensity probe composed of two half-inch microphones and a 12 mm spacer. Manual scanning was performed at a distance of 10 cm from the panel surface following recommended scan patterns [16, 17]. All results are provided in 1/12th-octave bands in the 150-5000 Hz frequency range.

3. Experimental results

The first comparison concerns the influence of the resonators’ distribution on the STL performances of the panel. The tested configurations are presented in Table 1 and results are given in Figure 3. 246 single-frequency resonators are distributed over various numbers of bays (12, 28 and 32) with a constant overall added mass. The second comparison investigates the performances of a multi-resonant configuration, involving one-, two- and three-frequency resonators combinations. The tested configurations are presented in Table 2 and results are given in Figure 4. The last discussion focuses on the addition of the resonators below an existing soundproofing package, involving a 50.8 mm-thick layer of melamine foam.
covering the inner surface of the fuselage. The tested configurations are presented in Table 3 and results are given in Figure 5. In each tested configuration, the STL result for the bare panel is indicated as a reference.

![Table 1: Tested configurations with varying spatial distribution.](image)

![Figure 3: Influence of the resonators’ distribution on measured STL.](image)

![Table 2: Tested configurations with various tuning frequencies.](image)
Figure 4: Influence of a multi-resonant configuration on the bandwidth of the STL improvement.

Table 3: Tested configurations with combination with foam layer.

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Resonators at 670 Hz</th>
<th>Distribution</th>
<th>Foam layer</th>
<th>Added mass (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>#3</td>
<td>246</td>
<td>32 bays</td>
<td>no</td>
<td>5.7</td>
</tr>
<tr>
<td>#6</td>
<td>0</td>
<td>-</td>
<td>yes</td>
<td>2.8</td>
</tr>
<tr>
<td>#7</td>
<td>246</td>
<td>32 bays</td>
<td>yes</td>
<td>8.5</td>
</tr>
</tbody>
</table>

Figure 5: Result with foam-alone, TMDs configuration # 3 alone, and combination of both treatments.
4. Discussions / conclusions

The following conclusions can be drawn from the results shown in Figs.3, 4 and 5:

- Increasing the spatial spreading of the resonators produces a larger and narrower STL improvement around the tuning frequency (see Fig.3) while a concentration of resonators on a reduced number of bays results in a smoother STL pic (with a similar overall added mass). A possible explanation lies in the localization of the acoustic radiation in the bays, which is then spatially averaged during the measurement of the transmitted sound pressure level. Alternatively, one can note that some coupling effects may be encountered between the ring mode and the local modes of the plates.

- A multi-resonant solution proves to be an effective way for increasing the bandwidth of STL gains, by increasing the STL improvement within the usual anti-resonance region (see Fig.4). This extension of the efficiency of the proposed solution also results in a reduction of the overall added mass, with very small decrease of the performances at the initial ring frequency.

- The beneficial effect of the resonators appears to be nearly additive with the one provided by a foam layer according to experimental results (see Fig.5). This indicates that the broadband (and generally high-frequency) STL benefits of the foam can be complemented with the narrow-band effects of the resonators at targeted critical frequencies.

REFERENCES


