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**Improving sound transmission loss at ring frequency of a curved
panel using tunable 3D-printed small-scale resonators**

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1 **Abstract:** An important dip in the sound transmission loss of curved
2 panels occurs at the ring frequency. The relevance of using small-scale
3 resonators to solve this issue is experimentally demonstrated on an air-
4 craft sidewall panel. The effect of varying the spatial distribution of
5 single frequency resonators (including combination with a broadband
6 soundproofing treatment), as well as using multi-frequency resonators
7 with a fixed spatial distribution is studied. Large improvement of the
8 measured sound transmission loss under a diffuse acoustic field exci-
9 tation is obtained around the ring frequency with limited added mass
10 and very small alteration of the overall sound insulation performance.

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

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11

12 1. Introduction

13 The design of lightweight and efficient soundproofing packages is one of the top priorities
14 in the general transportation industry for noise, vibration and harshness objectives. A very
15 common solution to this weight-efficiency compromise is the use of light porous material
16 attachments on structures, the latter being usually simplified or modelled as flat whereas
17 they can have a variable curvature radius in reality (a ship hull, an aircraft cabin). In usual
18 sound transmission loss (STL) results for cylindrical shells under a diffuse acoustic field
19 (DAF) excitation¹, a first noticeable STL reduction occurs at the cylinder ring frequency
20 which is a function of the shell radius and longitudinal waves velocity. A second dip in STL
21 results happens at the coincidence frequency where structural and acoustic wavenumbers
22 coincide (common to both curved and plane structures). Classical porous treatments that
23 mainly provides broadband sound attenuation are not optimal for solving these reductions of
24 sound insulation performance in narrow frequency bandwidths. Moreover, for typical aircraft
25 sidewalls, the ring frequency-related dip occurs at low frequency (typically 400 to 800 Hz).

26 The design of ribbed or stiffened structures can be optimized to modify the frequency
27 bandwidth affected by the two previously cited specific frequencies². A large variety of
28 passive or active solutions have also been proposed. A typical example of an active approach
29 for controlling acoustic transmissibility of aircraft panels is decentralized active control with
30 distributed units^{3,4}. Concerning passive solutions and apart from the classical solution of
31 increasing the panel's structural damping, innovative approaches to tonal or narrow-band

32 problems have been largely inspired by recent research on locally resonant metamaterials and
33 often involve the use of periodic add-on units. The locally resonant bandgaps obtained using
34 either Helmholtz resonators^{5,6}, periodically distributed tuned mass dampers⁷ or surface-
35 mounted resonators^{8,9} can be used to improve the STL. For composites plates in which the
36 coincidence frequency zone is usually larger than for a single layer homogeneous plate, a
37 possible solution is then to use multi-modal resonators to broaden stop-bands¹⁰. Excepting
38 Ref.⁵, a commonality in previously cited references⁶⁻¹⁰ is that only plane structures and then
39 coincidence frequency are considered. To the best of the authors' knowledge, the reduction
40 of the negative impact of the ring mode on the STL of a curved panel by means of a
41 passive, tunable and multi-resonant approach has not yet been published. In this work,
42 a tunable resonator concept involving 3D-printed cantilever beams with variable tip end
43 magnets is experimentally studied on an aircraft sidewall panel including stringer and ring
44 frame attachments under a DAF excitation. Several spatial distributions of the resonators
45 including a multi-resonant configuration as well as possible addition of this narrow-band
46 solution to a broadband sound absorbing treatment are investigated.

47 **2. Proposed design of tunable 3D-printed cantilever small-scale resonators**

48 The proposed small-scale resonators are based on a 3D-printed core structure using polycar-
49 bonate polymer consisting in a beam of section $10 \times 4 \text{ mm}^2$ and length 23 mm supported by
50 a stiffener of section $4 \times 3.5 \text{ mm}^2$ (see various illustrations in Figs. 1(a-e)). The base part is a
51 cube of $10 \times 10 \times 8 \text{ mm}^3$ dimensions on which the beam and the stiffener are both connected.
52 The connection to the structure is made at the base using a sealant (see Fig. 1(d)). The

53 overall unit volume occupied by a resonator without a tuning mass is finally $33 \times 10 \times 8 \text{ mm}^3$,
 54 with a unit weight of 2.72 g.

55 A 1 mm-thick base magnet is glued at the tip end and the resonators are then tuned
 56 using different neodymium magnets of known masses. The tuning masses and corresponding
 57 resonance frequencies for the resonators shown in Figs. 1(a-c) are 2.24 g and 670 Hz (F1),
 58 1.11 g and 820 Hz (F2), and 0.74 g and 980 Hz (F3), respectively. These average resonance
 59 frequencies were estimated on a few numbers of resonators when installed on the tested panel
 60 using a laser Doppler vibrometer and a shaker excitation.

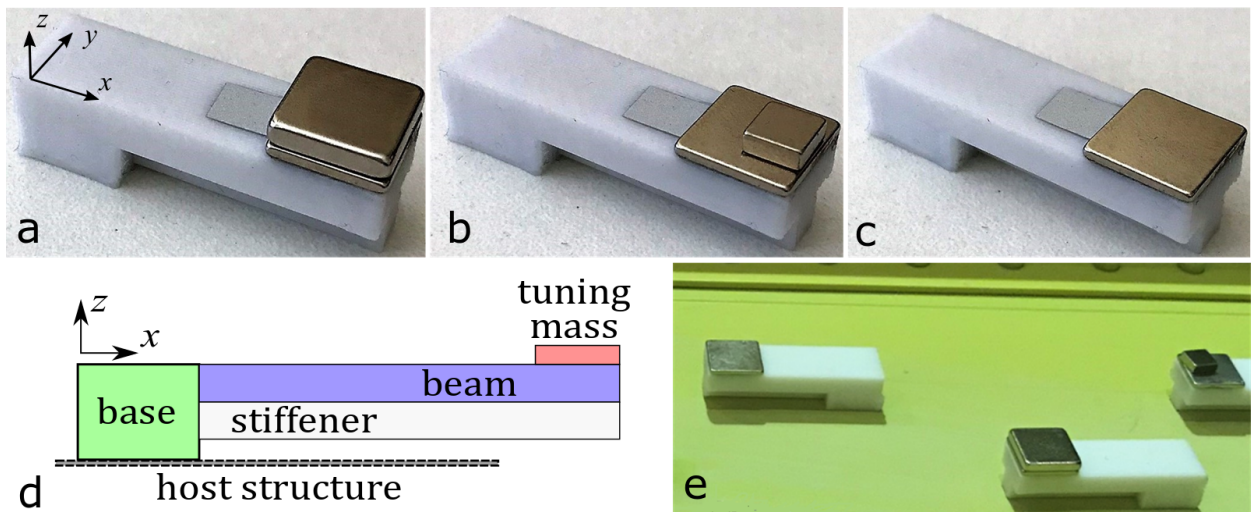


Fig. 1. (Color online) Pictures and schematic of the small-scale resonators. (a) Resonator F1. (b) Resonator F2. (c) Resonator F3. (d) Schematic describing the resonator parts. (e) Close-up view of the three resonators types mounted on a panel.

61 **3. Experimental methods and tested configurations**

62 According to standards^{11,12}, STL is determined in coupled reverberant-anechoic rooms using
63 measurements of the spatially averaged sound pressure level in the source room L_p and of
64 the spatially averaged average sound intensity level L_i over a scanning surface S_m on the
65 receiving side (both in dB), $STL = L_p - L_i - 6 - 10 \log_{10}(S_m/S)$ (with S the effective panel
66 area, considered equal to the scanning area S_m considered equal so that the last term was
67 neglected). All measurements were conducted in the coupled reverberant-anechoic rooms at
68 groupe d'acoustique de l'Université de Sherbrooke. A DAF excitation was generated in the
69 reverberant room ($7.2 \times 6.5 \times 3$ m³ dimensions) using a loudspeaker with a white noise input
70 in the 50 – 5000 Hz frequency range. The average sound pressure level in the reverberant
71 room L_p was obtained by rotating a half-inch PCB microphone with more than a complete
72 rotation of its supporting arm during the signal acquisition time (of 120 seconds for each
73 test). The average radiated sound intensity level L_i was measured in the anechoic room
74 ($6.8 \times 6.5 \times 3$ m³ dimensions) using a Bruel & Kjaer sound intensity probe composed of two
75 half-inch microphones and a 12 mm spacer. Manual scanning was performed at a distance of
76 10 cm from the panel surface following recommended scan patterns^{11,12}. Standards^{11,12} also
77 recommend not exceeding a pressure-intensity indicator value of 10 dB for a sound-reflecting
78 test specimen which was verified in each measurement. Finally, all the results presented
79 hereinafter are provided in 12th octave bands between 150 and 5000 Hz.

80 The structure under test is a curved rectangular fuselage panel equipped with axial
81 and circumferential stiffeners (see Figure 2(a)). The panel is made of 1.27 mm-thick aero-
82 nautic grade aluminum with a 21.4 kg total weight. Its length is 1.7 m with outer and inner

83 circumferences of 1.45 m and 1.3 m, respectively. Its curvature radius is approximately 1.6
 84 m. The equivalent mass per unit area of the fuselage equipped with attachments is approx-
 85 imately $8.68 \text{ kg}\cdot\text{m}^{-2}$ while the bare aluminum shell has a theoretical $3.43 \text{ kg}\cdot\text{m}^{-2}$ mass per
 86 unit area. The panel was mounted in the test window using a frame made of plywood with
 87 acoustic sealing made of neoprene adhesive and silicone. The frame and surrounding surfaces
 88 were covered with a sound insulating barrier including an open-cell sound absorbing material
 89 and a PVC heavy layer.

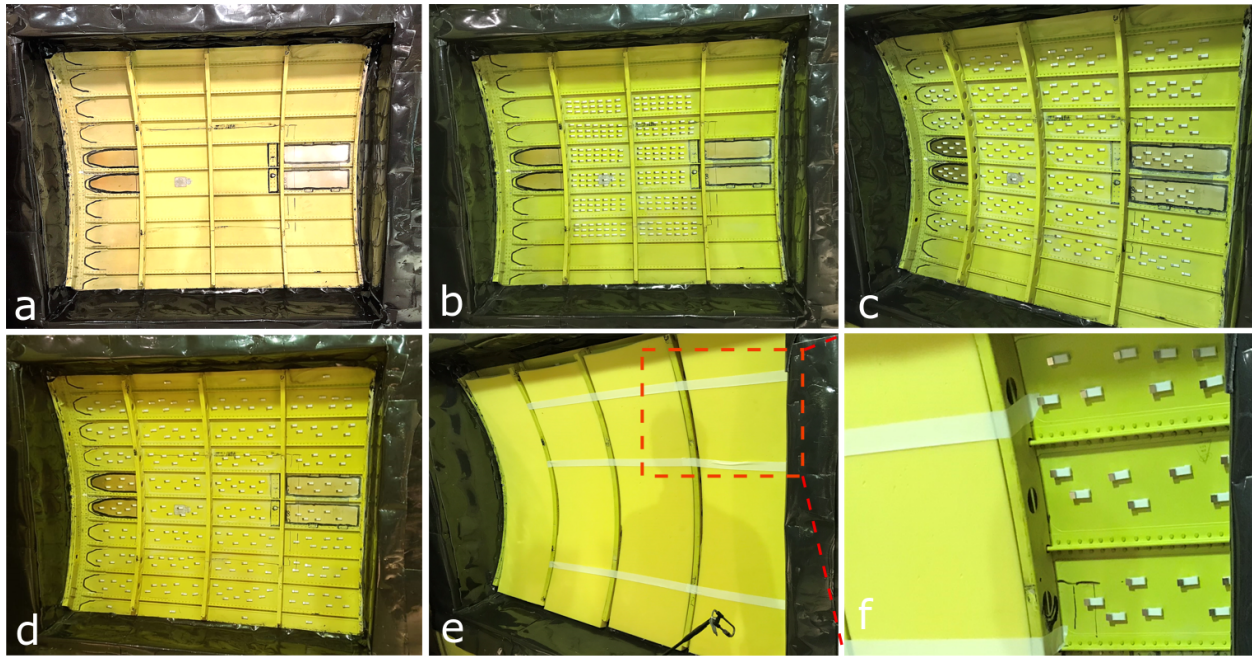


Fig. 2. (Color online) Pictures of the tested configurations. (a) Bare panel. (b) D1 distribution of resonators. (c) D2 distribution of resonators. (d) D3 distribution of resonators. (e) Panel fully covered by melamine foam. (f) Close-up view of the panel equipped with D3 distribution of resonators prior mounting of the lateral melamine foam piece.

90 The tested configurations are visually displayed and summarized in Figs. 2(a-f).
91 Fig. 2(a) corresponds to the reference configuration, *i.e.* the bare panel, with measure-
92 ments conducted both at the beginning and at the end of the series of experiments in order
93 to verify repeatability and data consistency. The difference between those two tests was be-
94 low 0.5 dB per 12th octave band. For all tests, the resonators were bonded on the panel skin
95 with wax typically used to mount accelerometers. Fig. 2(b) illustrates the D1 distribution of
96 resonators. In this case, 246 resonators are placed on the central portion of the panel (21 or
97 18 resonators per bay on 12 bays). The D2 distribution of resonators is depicted in Fig. 2(c).
98 The available 246 resonators are now distributed on the panel at the exception of upper and
99 lower parts of the panel, with 7 to 10 resonators per bay on 28 bays. Fig. 2(d) illustrates the
100 D3 distribution of resonators that now fully cover the panel area. The number of resonators
101 per bay varies between 5 and 9 for a total number of 240 resonators on 32 bays (including
102 one additional resonator on each of the 8 upper and lower edges' bays). The choice of these
103 configurations is mainly empirical for this first proof-of-concept.

104 The positioning of a sound absorbing treatment over the panel area is finally shown
105 in Figure 2(e) (including possible combination with the D3 distribution of resonators). A
106 melamine foam of 50.8 mm thickness was chosen to study its effect on measured STL when
107 used alone or combined with resonators. This material has a mass density of 6.1 kg/m³,
108 a static air flow resistivity of 7920 Ns/m⁴, a tortuosity of 1, a porosity of 0.98 and viscous
109 and thermal lengths of 132 μm and 149 μm , respectively. Four foam layers of dimensions
110 1.34 m \times 0.37 m are inserted between the circumferential stiffeners, supported by stringers

111 and kept in place with tape (see Figure 2(e)). Since the resonators have a smaller height
 112 than the stringers, the resonators are not in contact with the foam layer. The total added
 113 mass for the foam is 0.6 kg. Figure 2(f) presents a close-up view of the panel equipped with
 114 resonators prior complete installation of the melamine foam.

115 All the tested configurations are summarized in Table 1 with corresponding percent
 116 added mass and figure result number. Using only F1 resonators, the D1, D2 and D3 dis-
 117 tributions are first tested (see Figs. 2(b,c,d), respectively). Based on the D2 distribution
 118 (Fig. 2(c)), combinations of F1, F2 and F3 resonators are then used to produce a multi-
 119 modal locally resonant add-on covering the anti-resonance region. Note that although the
 120 number of resonators remains unchanged for the three configurations, the added mass is
 121 slightly decreased in the multi-modal configuration, as F2 and F3 resonators are lighter than
 122 F1 resonators (see Table 1). The effect of the sound absorbing treatment is finally tested
 123 alone and in combination with the D3 distribution of resonators in single mode (F1).

124 4. Experimental results and discussions

125 Results obtained are reported in Figs. 3(a-d). The reference STL curve for the bare panel
 126 is systematically included in Figs. 3(a-c) in order to ease comparison between test cases
 127 described in Table 1. In Figure 3(d), the effect of four key configurations is also plotted in
 128 terms of insertion loss so as to highlight STL improvement brought by each setup.

129 A theoretical ring frequency f_r of approximately 530 Hz is calculated using the sim-
 130 plified relation for a non stiffened thin shell¹ $f_r = \sqrt{E/\rho(1 - \nu^2)}/(2\pi r)$, with E the Young's
 131 modulus (= 70 GPa), ρ the mass density (= 2700 kg/m³), ν the Poisson's ratio (= 0.3) and

Table 1. Summary of tested configurations.

Tested configuration [results figure number]	Number of resonators			Total and tip end mass (both in kg)	Foam	Total added mass (%)
	F1	F2	F3			
D1, D2 and D3 distributions, single mode (F1) [Fig.3(a-b-c)]	246	0	0	1.22 , 0.55	No	5.7
D2 distribution, Two modes (F1-F2) [Fig.3(b)]	142	104	0	1.10 , 0.43	No	5.15
D2 distribution, Three modes (F1-F2-F3) [Fig.3(b)]	116	87	43	1.05 , 0.39	No	4.9
D3 distribution, Single mode (F1) [Fig.3(c)]	246	0	0	1.82 , 0.55	Yes	8.5
Foam alone [Fig.3(c)]	0	0	0	0.6 , -	Yes	2.8

132 r the curvature radius. This approximated f_r is indicated by a vertical arrow in Figure 3(a),
133 and the largest decrease of measured STL is obtained above this frequency at the 630 and
134 670 Hz twelfth octave bands. This bias might be explained by the fact that the mass density
135 of the panel skin alone is considered in the previous calculation and that stiffeners and their
136 effect are not accounted for. Nevertheless, the F1 frequency of the resonators is tuned for
137 enlarging the STL in the frequency band where it is found to be the lowest.

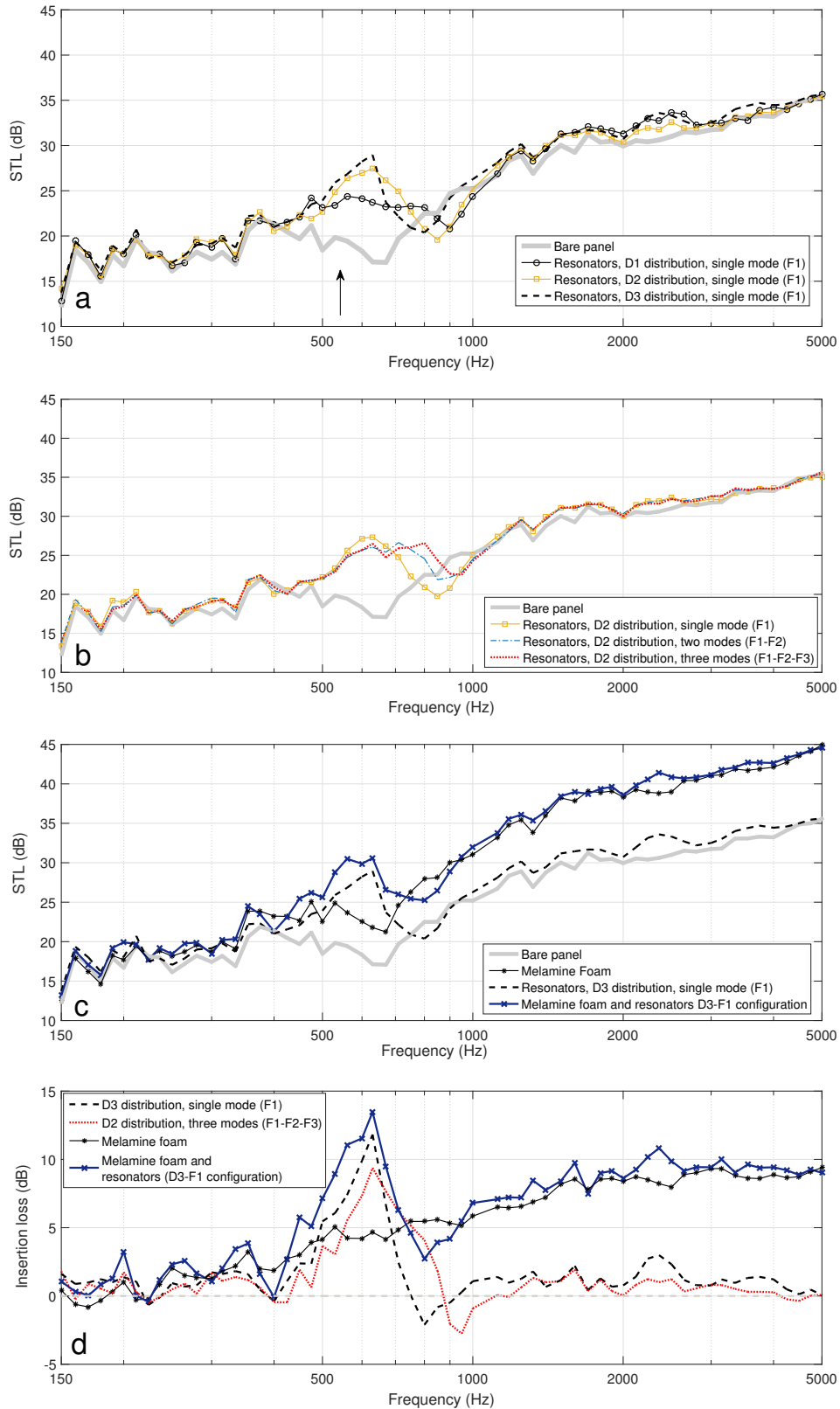


Fig. 3. (Color online) Summary of absolute and relative STL results obtained. (a) Effect of resonators distribution with single mode tuning (the ring-mode frequency is illustrated by a vertical arrow). (b) Effect of single to three-modes tuning with fixed distribution. (c) Separated and cumulative effect of melamine foam and resonators D3-F1 distribution. (d) Insertion loss (difference between considered solution and bare panel) for four key configurations.

138 The STL of the fuselage is measured before and after placing the resonators tuned on
139 a single mode (F1, 670 Hz) with distributions D1, D2 and D3 (see Table 1). Corresponding
140 results are provided in Figure 3(a). For the three distributions, the largest STL enhancements
141 are not obtained in the 670 Hz twelfth octave band but in the preceding one, centered on
142 a frequency of 630 Hz. Compared with the bare panel, the obtained STL gains for D1, D2
143 and D3 distributions are 6.6, 10.3 and 11.8 dB, respectively. The more concentrated D1
144 distribution produces a lower and smoother STL pic than when the resonators are more
145 uniformly spread (*i.e.* D2 and D3 distributions). This result might indicate that local sound
146 radiation is predominant around the ring mode's frequency bandwidth and that a spread
147 spatial distribution of the resonators improves the sound insulation properties in the tuned
148 frequency's bandwidth. This could also be a consequence of the spatial averaging done
149 when measuring radiated sound intensity, that includes untreated bays in the case of the D1
150 distribution. The STL reduction observed around between 800 and 900 Hz in Fig. 3(a) is a
151 known consequence of the tuned mass dampers anti-resonance. The dip nevertheless appears
152 at different center frequencies depending on the considered D1, D2 or D3 distribution, with
153 reduction of 5.1, 2.9 and 2.1 dB at twelfth octave bands frequencies of 900 Hz, 850 Hz and
154 800 Hz, respectively (with the bare panel result as a reference). Note that at the exception
155 of this STL reduction, no other alteration of the sound insulation occurs in the 100-5000 Hz
156 frequency range (see the result given in terms of insertion loss in Figure 3(d)).

157 In summary, results presented in Figure 3(a) indicate that the largest STL enlarge-
158 ments near the ring frequency (and the lowest alteration above) are reached with the D3

159 distribution. Nevertheless, the D2 distribution provides a good compromise in terms of over-
160 all STL improvement in the 500-800 Hz frequency range and is therefore chosen for studying
161 the effect of different resonators frequencies. Combinations of F1, F2 and F3 resonators
162 are used with D2 distribution and the obtained results are provided in Figure 3(b). Re-
163 sults show that the 2-modes and 3-modes configurations help limit the negative effect of the
164 anti-resonance and enlarge the STL in the 800-950 Hz bandwidth. Compared with a single
165 frequency configuration, the three modes configuration leads to an average STL improve-
166 ment of 3.3 dB per 12th-octave with a maximum of 5.7 dB at 800 Hz. This multi-resonant
167 solution even reduces the add-on's weight from 5.7% to 4.9% (see Table 1). A multi-modal
168 configuration of the resonators clearly increases the upper limit of the covered bandwidth
169 from 750 Hz to 900 Hz. Interestingly, changing the frequency distribution within the fixed
170 number of resonators does not modify the STL results in the rest of the frequency spectrum
171 (see the result given in terms of insertion loss in Figure 3(d)).

172 In figure 3(c), the results for the D3-F1 configuration is recalled (see Figure 3(a)), and
173 compared to the obtained results with the melamine foam alone and combined with the D3-
174 F1 configuration. The addition of the foam treatment has for consequence an enlargement of
175 the measured sound insulation starting at 250 Hz but nevertheless fails to fully compensate
176 the STL loss resulting from the panel's ring mode as a result of its broadband behaviour
177 (limited to a masking effect). When combined with the D3-F1 resonators configuration,
178 the broadband effect of the melamine is almost additive to the narrow-band effect of the
179 resonators. Compared with the foam alone case, the obtained STL result is enlarged at all

180 frequencies (at the exception of two STL reduction that can be found at 400 and 800 Hz
181 frequencies). It is noteworthy that the additive effect is only limited at the peak enhancement
182 brought by the resonators (see the result given in terms of insertion loss in Figure 3(d)). The
183 maximum STL increase is found at the 630 Hz frequency band, with values of 11.8 and 13.5
184 dB for the foam alone and combined with the resonators, respectively.

185 5. Concluding remarks

186 The performance of a locally resonant solution for enlarging sound transmission loss of
187 curved panels at ring frequency is demonstrated on an aircraft sidewall panel under a diffuse
188 acoustic field excitation. The STL reduction occurring at and near the ring mode frequency
189 is entirely suppressed with up to 10 dB gain with resonators tuned at a single frequency. The
190 use of a multi-resonant configuration proves to be a solution for both enlarging the efficiency
191 bandwidth and limiting the anti-resonance negative effect. It is finally demonstrated that
192 the proposed concept can be combined with a broadband sound insulation treatment, the
193 resonators' narrow-band and foam's broadband effects being nearly always additive in terms
194 of STL benefits. Based on this first proof-of-concept, the on-going steps of this work are
195 (1) a numerical model guiding an optimization of both the frequency tuning and number
196 of resonators so as to be able to draw a 'cost function-like' map to propose reachable dB
197 enhancements *vs* a target added mass, and (2) the design of a solution for plane panels so as
198 to solve ring and coincidence frequency STL reductions. Finally, this compact and low-cost
199 solution of 3D-printed small-scale cantilever beams could be also tested on arbitrarily shaped
200 complex industrial-scale structures for solving other vibro-acoustic issues.

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