forum acusticum 2026
8-12 September · Graz, Austria

REDUCING VIBRATION TRANSMISSION THROUGH THIN PLATE JUNCTIONS USING RESONANT METAMATERIAL STRIPS

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Abstract

Reducing vibration transmission through plate junctions is a challenging problem for limiting flanking sound transmission. Conventional approaches rely on structural decoupling or addition of mass, often involving increased structural complexity, material use, and limited low frequency effectiveness. This work explores an alternative, lightweight solution using resonant metamaterials, implemented as a strip of subwavelength resonators along the junction line. Such resonators can introduce bandgaps that inhibit free traveling waves within targeted frequency ranges. To predict vibration transmission through plate junctions treated with a resonant metamaterial strip, a two-step prediction model is proposed. First, wave propagation through junctions between semi-infinite thin plates connected at arbitrary angles is described using a wave-based analytical model. Within this model, multiple wave types are considered, and the metamaterial treatment is included via homogenized effective properties. This allows the computation of frequency dependent reflection and transmission coefficients, accounting for wave conversion between different wave types. Second, these results are used to compute coupling loss factors and are incorporated in a statistical energy analysis model to predict vibrational energy exchange between the connected plates. The proposed framework is experimentally validated on a junction of two aluminum plates connected at different angles, both with and without resonant metamaterial strips. Different strip widths are considered. Predicted velocity level differences agree well with those obtained from the measurements, showing reduced vibration transmission near the local resonance frequency.

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Keywords: *wave-based analysis, statistical energy analysis, locally resonant metamaterials, effective medium theory, vibration transmission*

1. Introduction

Vibration transmission through plate junctions is an important mechanism governing structure-borne sound transmission in many engineering applications, including building acoustics. In buildings, junctions between walls and floors can form efficient flanking transmission paths that contribute significantly to the overall sound transmission between adjacent spaces. Conventional mitigation strategies generally rely on increasing structural mass or introducing resilient connections, but such solutions often increase structural complexity and may be difficult to implement in lightweight constructions or renovation projects.

Resonant metamaterials (RMMs) offer a lightweight alternative for vibration control. By attaching subwavelength resonators to a structure, frequency ranges can be created in which wave propagation is strongly attenuated [1, 2]. While resonant metamaterials have been successfully applied to airborne sound insulation [3, 4], their use for controlling vibration transmission through structural junctions remains largely unexplored.

The present work investigates the use of resonant metamaterial strips placed along plate junctions to reduce the transmission of vibrational energy between connected plates. A previously developed wave-based transmission model [5] is combined with a statistical energy analysis (SEA) framework [6, 7], enabling the prediction of velocity level differences for finite plate assemblies. The emphasis of the present contribution is on the experimental validation of the complete prediction framework rather than on the underlying analytical model itself. To this end, measurements are carried out on aluminum plate junctions equipped with resonator strips of different widths and for different junction geometries. Predicted and measured velocity level differences are subsequently compared to evaluate both the effectiveness of the resonant metamaterial treatment and the accuracy of the proposed prediction methodology.

The paper is organized as follows. Section 2 provides a concise overview of the two-step prediction framework, combining a previously developed wave-based transmission model [5] with a statistical energy analysis approach for predicting velocity level differences between connected plates. Section 3 presents the experimental validation and compares the predicted and measured velocity level differences for aluminum plate junctions equipped with a resonant metamaterial strip. Finally, Section 4 summarizes the main conclusions.

2. Prediction framework

The proposed prediction methodology combines a wave-based transmission model [8, 9] with a statistical energy analysis framework [6, 7]. First, transmission coefficients are determined for plate junctions incorporating a resonant metamaterial strip. Subsequently, these transmission coefficients are converted into coupling loss factors and used in a SEA model to predict velocity level differences between connected plates. A detailed description of the wave-based formulation can be found in [5] and is only briefly summarized here.

2.1. Wave-based transmission model

The considered system consists of thin isotropic plates connected through a junction, and supports bending, longitudinal and shear waves. Vibration transmission is described by enforcing displacement compatibility and force equilibrium at the junction. For an incident wave, the model predicts the reflected and transmitted power per unit length for all angles of incidence ϕ with respect to the junction line, and the corresponding transmission coefficients between all wave-type combinations [8, 9]. The diffuse transmission coefficient for each frequency ω, independent of ϕ, for a wave of type s incident in plate i and a transmitted wave type t in plate j, can then be found as [8]:

$$\tau_{ij}^{st}(\omega) := \frac{1}{2}\int_0^{\pi} \tau_{ij}^{st}(\omega,\phi)\sin\phi\,\mathrm{d}\phi. \tag{1}$$

The resonator strip is incorporated by means of effective medium theory. Rather than modelling individual resonators, the periodically distributed resonators are represented by an equivalent medium with a frequency-dependent effective mass density [10]:

$$\rho_{\mathrm{eff},z}(\omega) = \rho\alpha_{\mathrm{st}} + \frac{1}{t}\sum_{k=1}^{n_m} \frac{m''_{z,k}(1+i\eta)}{(1+i\eta)-(\omega/\omega_k)^2}, \tag{2}$$

where ρ and t denote the mass density and thickness of the untreated plate, n_{m} the number of resonator modes, ω_k their fixed-base resonance frequencies, η the resonator damping loss factor, and $\gamma = m_{\mathrm{res}}/m_{\mathrm{plate}}$ the mass ratio. The quantity $m''_{\mathrm{z},k}$ represents the transverse modal effective mass of mode k, while α_{st} accounts for the dynamically inactive resonator mass [11].

2.2. Statistical energy analysis

The transmission coefficients obtained from the wave-based model are subsequently incorporated into a statistical energy analysis framework to predict velocity level differences between connected plates. The plate junction is decomposed into subsystems corresponding to all plate-wave type combinations, and the subsystem energies E_i and E_j are obtained from the power balance equation [6, 7]:

$$\sum_{(j,t)\neq(i,s)} E_j^t \eta_{ji}^{ts} + \frac{W_i^s}{\omega} = \left(\eta_{d,i}^s + \delta_{ir}^{st} + \sum_{(j,t)\neq(i,s)} \eta_{ij}^{st}\right) E_i^s, \tag{3}$$

where W_i^s is the input power, $\eta_{d,i}^s$ the damping loss factor, η_{ij}^{st} the coupling loss factor and δ_{ir}^{st} the energy dissipated in the resonators. The input power is obtained from the incident force and velocity at the excitation location [12], while the damping loss factor for bending waves for each frequency f is derived from the measured structural reverberation time T as [6]:

$$\eta_{\mathrm{d}i}^B = \frac{2.2}{fT}. \tag{4}$$

For longitudinal and shear waves, the corresponding damping loss factors may be taken as one third of the bending wave damping loss factor [7]. The coupling loss factors are calculated from the analytically predicted transmission coefficients as [6]:

$$\eta_{ij}^{st} = \frac{c_{g,i} L_{ij} \tau_{ij}^{st}}{2\pi^2 f S_i}, \tag{5}$$

where $c_{g,i}$ is the group velocity, L_{ij} the junction length, and S_i the source plate area [6]. In the presence of a resonator strip, part of the incident energy is dissipated within the resonators, which is accounted for through

$$\delta_{ir}^{st} = 1 - \tau_{ii}^{st} - \tau_{ij}^{st}, \tag{6}$$

with τ_{ii}^{st} and τ_{ij}^{st} representing the reflected and transmitted energy fractions, respectively.

After solving the SEA system, the velocity level difference between source and receiver plate is computed as [6]:

$$D_{\mathrm{v},ij}^{st} := 10\log\left(\frac{<v_i^{s\,2}>_{\mathrm{S}}}{<v_j^{t\,2}>_{\mathrm{S}}}\right) = 10\log\left(\frac{E_i^{\mathrm{s}} m_j}{E_j^{\mathrm{t}} m_i}\right), \tag{7}$$

where $<v_i^{s\,2}>_{\mathrm{S}}$ and $<v_j^{t\,2}>_{\mathrm{S}}$ denote the surface-averaged mean-square transverse velocities and m_i and m_j the corresponding plate masses.

3. Experimental validation

The proposed prediction framework is evaluated experimentally on two aluminum plate junctions with junction angles of $\theta_{12} = 180°$ and $\theta_{12} = 124°$. These two configurations were selected to assess the methodology for both a horizontal junction and a junction involving stronger wave conversion effects. For each geometry, four cases are considered: an untreated junction and junctions where one, two or three rows of resonators are positioned along the junction line. A schematic representation of a treated junction with three resonator rows is shown in Fig. 1.

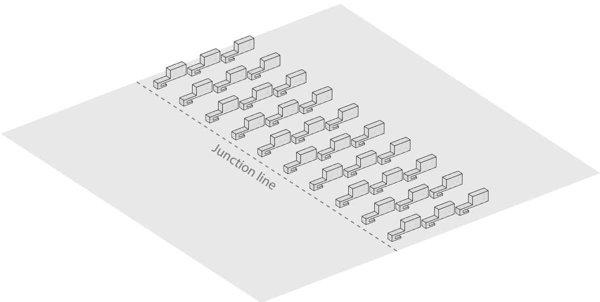
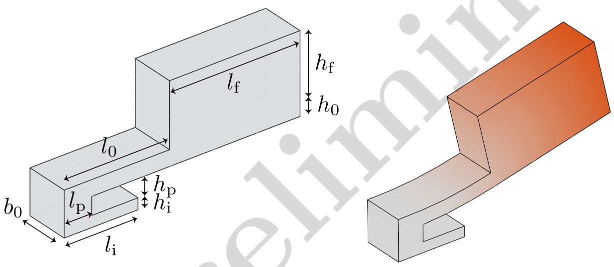


Figure 1: Schematic representation of a plate junction with three rows of resonators along the junction line.

3.1. Resonator design and material characterization

The resonators were laser-cut from an 8 mm PMMA plate and designed according to a cantilever-type configuration as proposed in [13]. The geometry was optimized using a finite-element model of the fixed-base resonator such that the first transverse resonance occurred close to the target frequency of 250 Hz while maintaining a lightweight design with a mass ratio below 20%. Figure 2 shows the resonator geometry together with its first transverse vibration mode.



(a) **(b)**

Figure 2: Geometry of the PMMA resonator (a) and corresponding first transverse vibration mode (b).

The designed and measured dimensions of the resonator are summarized in Table 1. Small geometric deviations were introduced during the laser-cutting process and were subsequently accounted for in the numerical model through the measured dimensions.
Experimental characterization of the resonance frequency of ten fabricated resonators resulted in an average resonance frequency of 260.8 Hz and a damping loss factor of $\eta = 0.055$. The slight shift between the targeted and measured resonance frequency can be attributed to the geometric deviations introduced during manufacturing. These experimentally identified resonance characteristics were subsequently used in the effective-medium representation of the resonator strip.

Table 1: Designed and measured dimensions of the resonator.

Parameter	Design [mm]	Measured [mm]
l_i	13.10	13.10
h_i	2.00	1.95
l_p	5.00	4.89
h_p	3.00	3.11
l_0	18.70	18.70
h_0	2.70	2.73
l_f	23.30	23.17
h_f	11.30	11.21
b_0	8.00	7.92

The experimental validation of the prediction model was performed on a junction of two aluminum plates with dimensions $0.909 \times 1.002 \times 0.003$ m and $1.03 \times 1.002 \times 0.003$ m, respectively. To ensure physically representative model predictions, the material properties of both the aluminum plates and the PMMA resonators were identified experimentally. The densities were determined from the measured mass and geometry, while the elastic properties were obtained through modal testing and model updating [14]. The resulting material properties are summarized in Table 2 and were subsequently used as input for the prediction model.

Table 2: Material properties used in the prediction model.

Property	Aluminum	PMMA
Density ρ [kg/m^3]	2676	1166
Young's modulus E [GPa]	63.5	4.42
Poisson's ratio ν [-]	0.33	0.31

The damping loss factors required by the SEA model were obtained from structural reverberation time measurements [15]. Since freely suspended aluminum plates exhibit very low total damping values that are not representative of practical engineering structures, damping patches were applied to increase the total damping. This resulted in frequency-dependent damping loss factors ranging from approximately 0.01 to 0.02 over the frequency range of interest.

3.2. Experimental setup

To validate the proposed prediction framework, vibration transmission measurements were performed on the two plate junction configurations ($\theta_{12} = 180°$ and $\theta_{12} = 124°$) considered in this study. The objective was to compare the predicted and measured velocity level differences for the untreated junction and for junctions equipped with one, two or three rows of resonators. Each row consisted of 20 resonators distributed uniformly along the junction length.
The junctions were suspended with three elastic cords

Figure 3: Experimental setup for the measurement of the vibration field of the aluminum plate junction. (a) Polytec PSV-Qtec scanning laser Doppler vibrometer pointed towards the measurement surface. (b) Experimental setup for the plate junction of $\theta_{12} = 180^\circ$ with one resonator row. (c) Experimental setup for the plate junction of $\theta_{12} = 124^\circ$ with three resonator rows. (d) Detailed view of the three resonator rows.

to approximate free-free boundary conditions. The plate with dimensions $0.909 \times 1.002 \times 0.003$ m was used as the source plate, whereas the $1.03 \times 1.002 \times 0.003$ m plate acted as the receiver plate. Excitation was provided in the transverse direction using an electrodynamic shaker (Brüel & Kjær Type 4809), while the resulting out-of-plane vibration field was measured with a Polytec PSV-Qtec scanning laser Doppler vibrometer.

The vibration response was acquired on a spatial grid covering both plates for three excitation locations on the source plate. For each configuration, the H1 estimator of the velocity frequency response functions was determined from the measurements [16]. The spatially averaged mean-square velocities of the source and receiver plates were then obtained as:

$$v_{\mathrm{ms}}(\omega) = \frac{1}{2}\Big\langle\, |H_1(\omega)|^2 \Big\rangle_S. \tag{8}$$

These quantities were subsequently used to compute the velocity level differences according to Eq. (7), allowing a direct comparison with the velocity level differences predicted by the proposed two-step prediction framework.

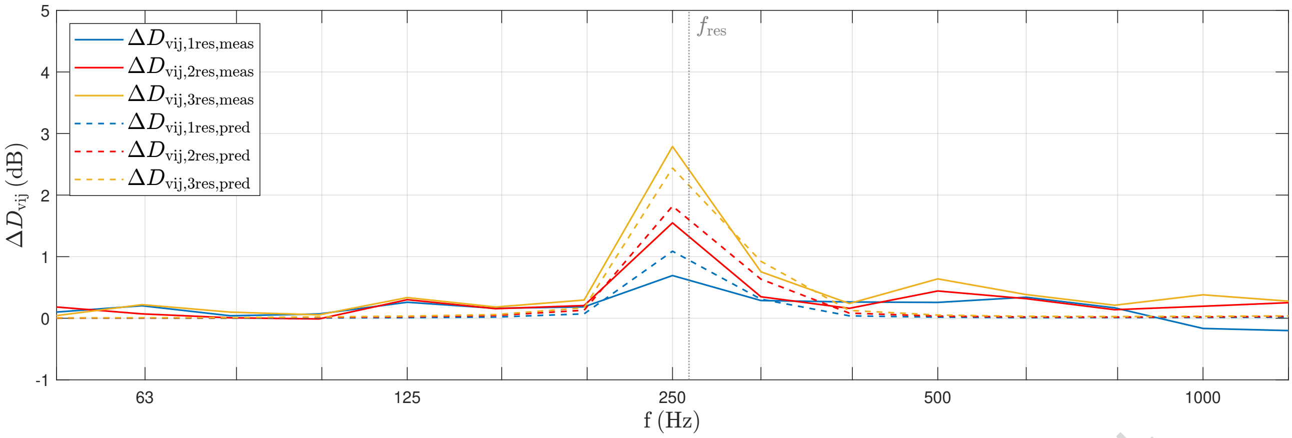
3.3. Results and discussion

The experimentally identified material and resonator properties of Sect. 3.1 were implemented in the wave-based model described in Sect. 2.1. The corresponding transmission coefficients were subsequently used to determine the velocity level differences through the SEA model described in Sect. 2.2. For the measurements, the velocity level differences were calculated from the spatially averaged mean-square velocities obtained from the measured vibration field as discussed in Sect. 3.2.

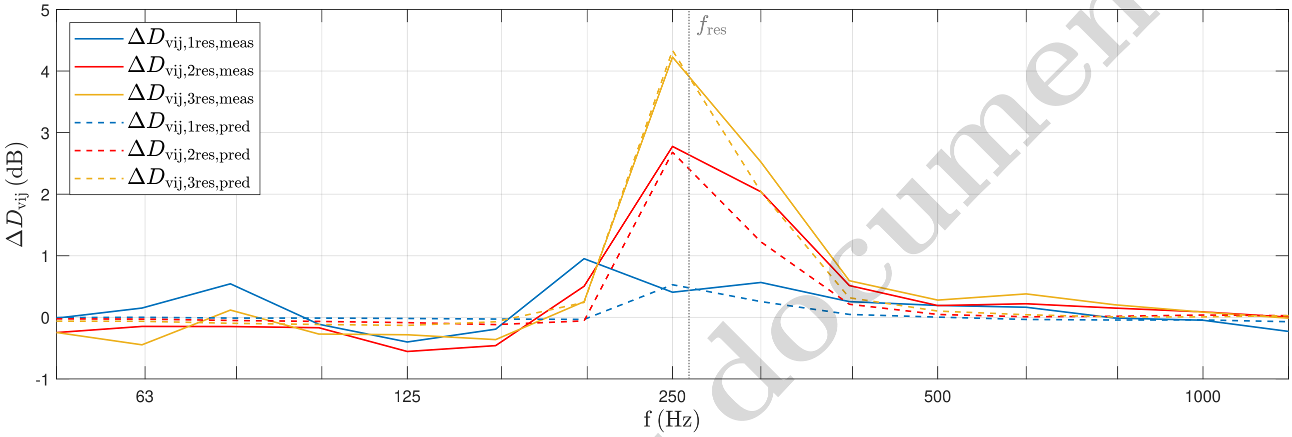
Figure 4 compares the predicted and measured changes in velocity level difference relative to the bare junction for both plate configurations. Presenting the results relative to the untreated junction allows the influence of the resonator treatment to be isolated, thereby reducing the influence of the modal behaviour of the finite plate junctions on the comparison between measurements and predictions. Only the results for bending waves are shown, as the resonator strip was specifically designed to attenuate out-of-plane vibration through its transverse resonance mode. Furthermore, the out-of-plane vibration field was directly accessible from the laser Doppler vibrometer measurements.

For both junction geometries, the resonator treatment leads to an increase in velocity level difference around the target resonance frequency, indicating a reduction in vibration transmission through the junction. Furthermore, the attenuation increases systematically as additional resonator rows are introduced.

A good correspondence is observed between the predicted and measured results in the frequency range of interest. The proposed framework accurately predicts both the frequency at which the attenuation occurs and the trend of increasing attenuation with increasing strip width. The agreement is particularly



(a)



(b)

Figure 4: Comparison between the predicted and measured change in velocity level difference $\Delta D_{\mathrm{v},ij}$ for bending waves for plate junctions with $\theta_{12} = 180°$ (a) and $\theta_{12} = 124°$ (b), with one, two or three rows of resonators. Results are presented in 1/3-octave bands.

encouraging given that the resonator strip is represented through homogenized effective properties and subsequently integrated into a SEA framework.

Overall, the measurements confirm that resonant metamaterial strips provide an effective lightweight solution for suppressing vibration transmission through thin plate junctions. At the same time, the results demonstrate that the proposed prediction framework can reliably predict the resulting changes in velocity level difference.

4. Conclusion

A two-step prediction framework was presented for assessing vibration transmission through plate junctions treated with resonant metamaterial strips. The proposed approach combines a wave-based analytical model with a statistical energy analysis framework, allowing wave transmission characteristics at the junction to be translated into velocity level differences of the coupled plate system. The resonator strip was represented through an effective-medium formulation, resulting in an efficient prediction methodology.

The framework was validated experimentally on aluminum plate junctions with junction angles of 180° and 124°, equipped with one, two, or three rows of PMMA resonators. For both configurations, the resonator treatment produced a clear increase in velocity level difference around the resonator resonance frequency, demonstrating a reduction in vibration transmission through the junction. Increasing the strip width further enhanced the attenuation performance. Good agreement was observed between the predicted and measured changes in velocity level difference. The model accurately captured both the frequency range in which attenuation occurs and the increase in attenuation obtained with additional resonator rows. The results therefore demonstrate the effectiveness of lightweight RMM strips for reducing vibration transmission through thin plate junctions, while confirming the capability of the proposed two-step prediction framework to predict their vibro-acoustic performance. Future work will focus on extending the framework to more complex junction configurations and broadband resonator designs, enabling its application to realistic building assemblies and flanking transmission problems.

5. Acknowledgments

The presented research has been performed within the framework of the doctoral fellowship “Resonant metamaterials for broadband suppression of flanking sound transmission in buildings” (Grant 1S38725N), funded by the Research Foundation Flanders (FWO), Belgium. Stijn Moons gratefully acknowledges the financial support of Flanders Innovation & Entrepreneurship (VLAIO) through the Baekeland fellowship HBC.2022.0712. Daniele Giannini gratefully acknowledges the financial support of the Research Foundation Flanders (FWO) through the postdoctoral fellowship 12A3Q24N. Furthermore, this work was supported by KU Leuven internal funds under project C2E/24/031. The authors thank Zander Quatacker for his valuable support in preparing the experimental setups.

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