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SHAPE OPTIMIZATION OF INERTIA-AMPLIFIED MECHANICAL RESONATORS FOR VIBRATION DAMPING OF MOUNTED PLATE

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ABSTRACT

A well-known solution for suppressing undesired resonances in a vibrating structure is to place a tuned mass damper tuned to the frequency of the targeted mode. When a range of frequencies needs to be attenuated, vibroacoustic metamaterials are known as an effective solution. However, the periodic placement of resonators increases the added mass significantly, limiting its implementation in industrial applications. To address this, recent works have developed a methodology for optimizing the location and modal properties of a small number of resonators to achieve broadband attenuation in a desired frequency range. The aim of this study is to propose a practical implementation of this method for different mountings of a steel rectangular plate. Applying rotational inertia amplification for the local resonators offers a high dynamic to static mass ratio. The modal properties – mass and frequency - of the resonators are achieved by shape optimization to suit the target frequency range of a specific mounting case. This multi-objective optimization uses a practical data-driven generic design tool to avoid computationally heavy finite element models. The attenuation achieved by the rotational inertia amplifiers is then compared to the one achieved by mass-spring systems with equivalent modal properties.

Keywords: *vibration mitigation, shape optimization, inertia-amplified resonator.*

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1. INTRODUCTION

Periodic placement of resonators on a host structure to mitigate vibrations, a solution known as locally resonant metamaterials, results in the attenuation of not a single frequency, but a range of frequencies known as a bandgap. This solution poses limitations for industrial applications, as it involves adding a significant amount of mass to the system. For this reason, methods for locally treating the host structure have been investigated [1, 2]. In [3], Dedoncker et al. propose to apply two algorithms, gradient based and simulated annealing, in order to simultaneously optimize the placement and modal properties of a finite number of tuned mass dampers on a plate to achieve broadband attenuation. Based on a defined frequency range to be attenuated, the algorithm converges on a solution that maximizes the vibration attenuation of the host plate and returns the position and mass, frequency and damping of the desired number of tuned mass dampers (TMDs). However, this method is based on the strong assumption that TMDs are ideal mass-spring-damper systems and can exhibit any combination of modal properties. The design of realistic resonators, on the other hand, limits the practical application of the method. Machine learning approaches have been investigated to relate design parameters to physical parameters [4–6]. In [7], the modal properties of a spiral-shaped resonator are tuned by optimizing the spiral's design parameters. This approach provides a rapid way of designing resonators according to required modal properties, both frequency and mass, using a conditional variational auto-encoder (cVAE). In this work, we propose a practical implementation of these two methods to present an optimization routine for plate vibration treatment under different mounting scenarios. A gradient-based optimization routine calculates the optimal positions and modal properties of the resonators to be placed. These modal properties will be used by the cVAE-optimizer to design a series of complex-shaped mechanical resonators, a specific design which includes rotational inertia amplification. The vibration attenuation obtained by the set of rotational amplifier resonators, optimally placed on the plate, will be compared with the attenuation predicted by the gradient-based method using the set of ideal TMDs.

2. GRADIENT BASED METHOD FOR RESONATOR PLACEMENT AND MODAL PROPERTIES OPTIMIZATION

The following study considers a steel plate with a thickness of 3 mm and dimensions of 1,6 m (x-direction) by 1,2 m (y-direction). Two mounting conditions, shown in Figure 1, are investigated. In the first mounting scenario the plate is clamped on its left and right edges and excited by a force of 1 N (z-direction) at the center of the top edge. The second mounting scenario supposes that the plate is clamped on its bottom edge and excited by a force of 1 N (z-direction) on the top-right corner. The gradient based algorithm is used to optimize the placement and modal properties of 3 TMDs to maximize plate attenuation over the frequency range 150 Hz-200 Hz for these two mounting scenarios. The modal properties of the optimized TMDs are bounded by the distribution of the modal properties of the resonator dataset used to train the cVAE. Here, TMDs can take any frequency between 10 and 300 Hz, although their effective mass is constrained between 148 and 152 g. This high constraint on the effective mass is due to a narrow distribution of the modal properties of the resonator dataset considered. The modal properties of the 6 TMDs proposed by the algorithm are presented in Table 1. Figure 2 shows the average plate deformation with (blue) and without (black) TMDs. In both scenarios, we observe that 3 resonators are sufficient to achieve broadband attenuation over the frequency range specified: all resonance peaks are suppressed within this range.

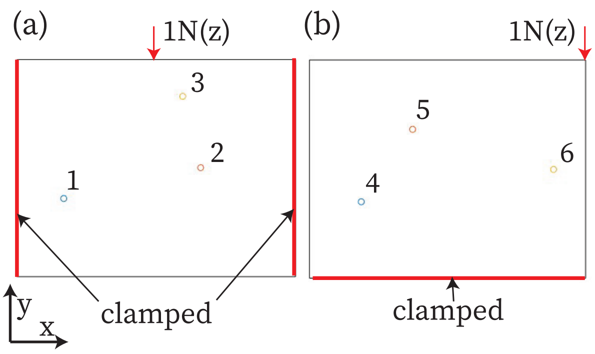

Figure 1. Mounting scenarios 1 (a) and 2 (b) and optimized positioning of the TMDs 1 to 6.

Table 1. Modal properties of the TMDs required by the gradient based optimization for the mounting scenario 1 (TMDs 1,2 and 3) and the mounting scenario 2 (TMDs 4,5 and 6).

	Frequency (Hz)	Effective mass (kg)
TMD 1	176	0.1506
TMD 2	165	0.1506
TMD 3	160	0.1505
TMD 4	177	0.1506
TMD 5	154	0.1504
TMD 6	163	0.1506

3. SHAPE OPTIMIZATION OF ROTATIONAL AMPLIFIERS USING CONDITIONAL VARIATIONAL AUTOENCODERS

The geometry of the resonator is chosen as follows: an hourglass shaped soft core is connected to two identical heavy disks at the bottom and top. The disks are connected by tilted struts with spherical joints as in [8]. The tilted struts allow the translation of the disks to be coupled to their rotation thereby inducing inertia amplification to the system as described in [9, 10]. As a consequence, the rotational amplifier achieves a high static-to-dynamic mass ratio, further minimizing the mass added to the host structure. The torsional and axial stiffnesses connecting the disks together are defined by the design of the hourglass-shaped core, which therefore influences the amplifier's modal properties. Realistic resonators are designed using the cVAE developed in [7] to achieve the modal properties given in Table 1. The geometry of the ro-

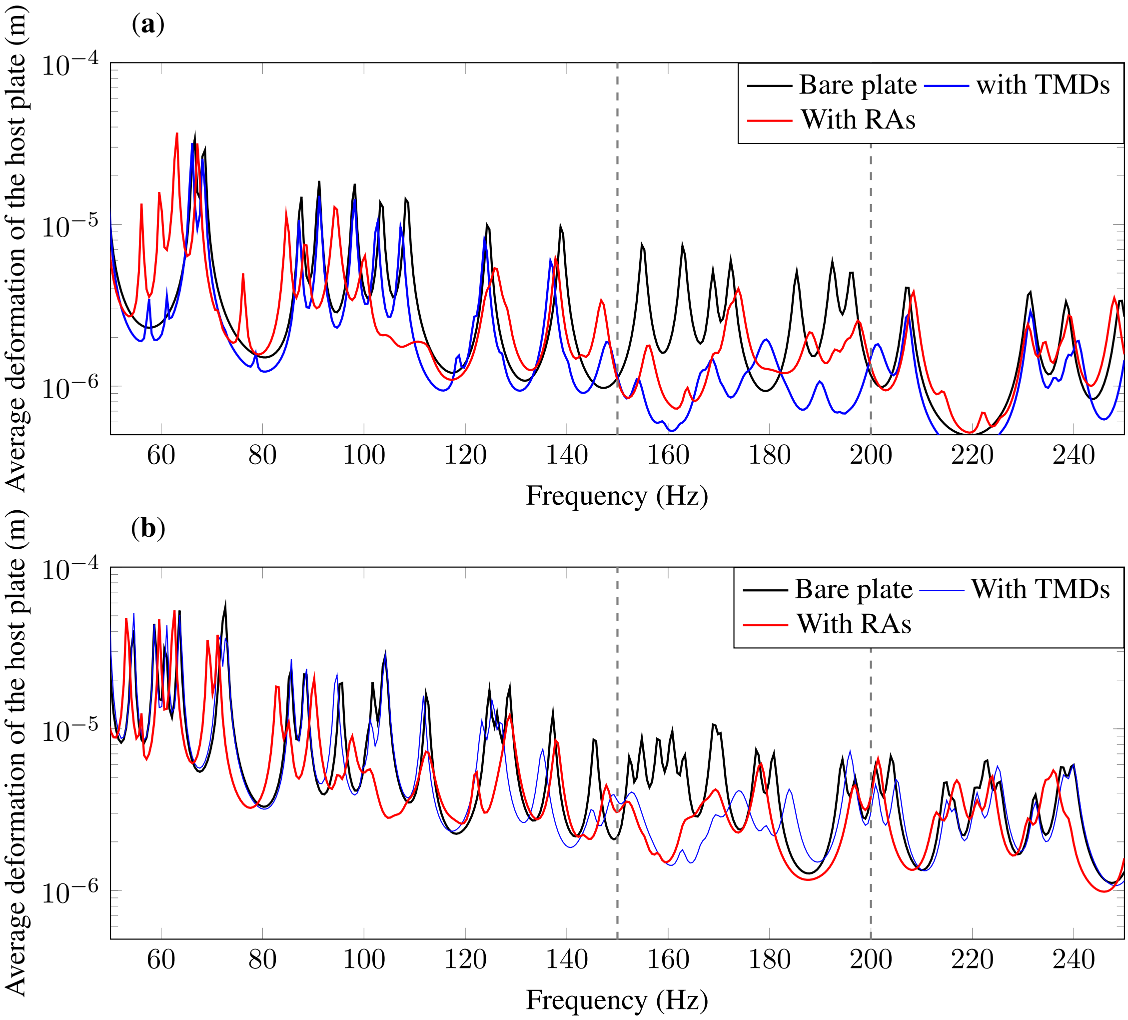

Figure 2. Average deformation of the mounted plate under mounting scenarios 1 (a) and 2 (b): bare plate in black, with ideal TMDs in blue and with RAs in red.

tational amplifier (RA) is parameterized according to the dimensions d1, d2, d3, d4 and angles a1 and a2 shown in Figure 3(a). 1100 designs are generated by Latin Hypercube sampling (LHS) of the resonator's geometrical parameters. The frequency and effective mass of the predominant axial vibration mode of each design, shown in Figure 3(b), are calculated performing a modal analysis in AnsysR2024V2. The conditional variational auto-encoder is then trained on the dataset created and finds the best designs to reach desired modal properties. The modal properties predicted for the best design given by the cVAE are compared with those required by the gradient-based algorithm. These latter modal properties are then compared to those observed on a FEM model of the suggested design. The geometric parameters of the 6 RAs best fitting the modal properties of the 6 TMDs required by the gradient based optimization are shown in Table 2. For the 6 ideal TMDs, Table 3 shows an error of less than 5% for all frequencies and effective masses.

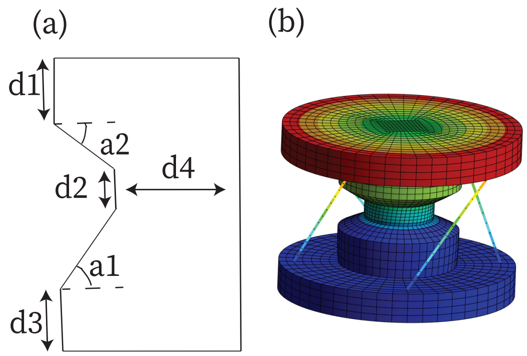

Figure 3. (a) Parametrized shape of the soft core. (b) First torsional mode of the rotational amplifier.

Table 2. Geometric parameters of the best design suggested by the cVAE fitting the modal properties of the 6 TMDs required.

	d1 (m)	d2 (m)	d3 (m)	d4 (m)	a1 (rad)	a2 (rad)
RA1	0.011	0.008	0.010	0.015	0.99	0.89
RA2	0.011	0.012	0.010	0.014	0.34	0.94
RA3	0.012	0.008	0.009	0.012	0.72	0.57
RA4	0.009	0.011	0.010	0.016	0.81	1.00
RA5	0.009	0.007	0.009	0.013	0.90	0.89
RA6	0.008	0.008	0.009	0.014	0.88	0.10

Table 3. Relative errors between the required (req), designed (cVAE) and observed (FEM model) modal properties (frequency f and modal mass m).

	f_{req} to f_{cVAE}	m_{req} to m_{cVAE}	f_{req} to f_{FEM}	m_{req} to m_{FEM}
TMD 1	0.3%	0.1%	0.9%	1.0%
TMD 2	0.3%	0.2%	0.8%	1.1%
TMD 3	0.0%	0.3%	4.4%	1.3%
TMD 4	0.1%	0.1%	1.5%	1.5%
TMD 5	0.1%	0.2%	2.4%	4.0%
TMD 6	0.3%	0.0%	2.6%	2.8%

4. PRACTICAL IMPLEMENTATION OF THE DESIGNED ROTATIONAL AMPLIFIERS

An explicit FEM model created in ANSYS2024R2 couples the resonators designed with the cVAE to a plate with dimensions defined above at their optimized position as shown in Figure 4. The rotational amplifiers for the two considered mounting scenarios attenuate the targeted resonance peaks between 150 and 200 Hz. As an interesting side effect, the chosen resonators also attenuate the plate over unforeseen frequency ranges. A clear vibration reduction is noticeable near 100 Hz for both configurations: the resonators undergo a bending mode in this frequency range, and attenuate the motion of the plate. The approach presented in [3] allows the accurate placement of realistic resonators. However, the attenuation level achieved by the realistic resonators is lower than that achieved by ideally tuned mass dampers, as observed at 170 Hz for both mounting scenarios. This difference can be explained by two assumptions made by the gradient based optimization method. The optimized TMDs feature a single vibration mode, allowing only the predominant axial mode of the resonator to be modeled, and thus ignoring the more complex resonator dynamics. Moreover, this method assumes a point-wise contact between the resonator and the host plate, thus neglecting the local stiffening that can be observed at attachment points.

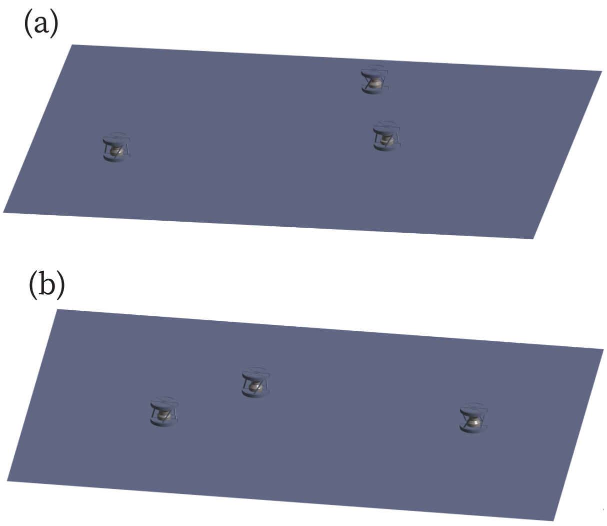

Figure 4. FEM model of the mounting scenarios 1 (a) and 2 (b) and optimized positioning of rotational amplifiers designed.

5. CONCLUSION

In this study, two optimization methods are combined to enable a fast routine for positioning and designing a small number of resonators to attenuate a mounted plate. This method allows the resonator design to be adapted to the different mountings and excitation points of the plate. Coupling the two methods results in a clear attenuation over the desired frequency range containing multiple resonance peaks. However, the fast optimization methods use simplifying assumptions, such as ideal, point-like mass-spring systems and the omission of multiple resonances of the resonators. The more complex dynamics of realistic resonators should be considered explicitly, by full harmonic simulations of the final setup.

6. ACKNOWLEDGMENTS

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