

Dynamic Metamaterials for Structural Stopband Creation

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ABSTRACT

The NVH performance of conventional panels and structures is mainly driven by their mass. Silence often requires heavy constructions, which conflicts with the emerging trend towards lightweight design. To face the challenging and often conflicting task of merging NVH and lightweight requirements, novel low mass and compact volume NVH solutions are required. Vibro-acoustic metamaterials with stopband behavior come to the fore as possible novel NVH solutions combining lightweight requirements with superior noise and vibration insulation, be it at least in some targeted and tunable frequency ranges, referred to as stopbands. Metamaterials are artificial materials or structures engineered from conventional materials to exhibit some targeted performance that clearly exceeds that of conventional materials. They consist typically of (often periodic) assemblies of unit cells of non-homogeneous material composition and/or topology. Their superior performance may arise from the intrinsic properties of the individual unit cells and from the mutual interaction between the cells. If the transfer path of vibrations is known, metamaterials targeting different frequency zones can be applied to achieve wider stop bands to block vibrations traveling along the given transfer path. This paper investigates the efficiency of different metamaterial solutions for blocking vibrations along a certain transmission path: sequential metamaterial lay-out versus a mixed lay-out. The efficiencies of the different strategies for vibration attenuation are numerically and experimentally evaluated and compared.

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INTRODUCTION

The vibration harshness of products is gaining importance due to both more restrictive legal requirements and consumer expectations. Ecological trends and the associated run for efficiency, however, increase the necessity of lightweight design. The combination of reduced weight and increased stiffness in lightweight applications leads to a deterioration of the vibration harshness, resulting in a challenge to meet both lightweight and vibration harshness requirements.

In previous publications, the authors showed the potential of vibro-acoustic metamaterials as possible candidates for lightweight material systems with superior noise and vibration insulation, be it at least in some targeted and tunable frequency ranges, referred to as stop bands [1]. A novel metamaterial concept was introduced based on the addition of resonant structures to a host structure. In this paper the potential of metamaterial as vibration insulation along known transmission paths is investigated. Contrary to acoustic transmission problems, mostly characterised by two-dimensional acoustic loading, in vibration problems, the transmission path is often one-dimensional, such as in ducts. This opens up the potential to combine metamaterials of different lay-outs along the propagation direction of these waveguides. In this paper the efficiency of a metamaterial with two stop bands is compared to the efficiency of a layout with two separate metamaterials, each with one stop band, in a sequential layout.

After the introduction, the second and third sections of this paper introduce the application case and the metamaterial design. Next the vibrational stop band behavior is predicted based on unit cell models. These numerical predictions are compared to experimental results: the fifth section describes the test set-up and section six compares the experimental results with the predicted stop bands. The paper ends with the conclusions.

APPLICATION CASE

A common case for vibration propagation in one direction, are ducts. In view of computational effort and simplification of the wave propagation involved, a rectangular non-fluid loaded duct is selected as application case. On the one hand, within fluid loaded ducts, energy can propagate through both the fluid as the structure of the duct, meaning that the efficiency of the stop band effect can be reduced. Also from an experimental point of view, fluid loaded ducts would increase complexity of the test procedure. On the other hand, the choice for a rectangular duct is mainly driven by the available stop band prediction tools: cheap and fast two dimensional unit cell models can then be applied without having to include curvature in the models, which would be a non-trivial extension of the current available tools.

A rectangular PVC duct of outer dimension 100mm x 100mm and 2mm thickness is chosen. In the experiments, a finite duct with a length of 1m is used. The density is retrieved through a weight measurement of the test section. The Young's modulus and Poisson's ratio of the PVC can be obtained based on the correlation between the modeshapes retrieved with a Finite Element (FE) model and the experimentally measured modeshapes. For the FE model 4000 shell elements were used. For the experimental test the duct was freely suspended and roving hammer excitation was used in combination with 4 reference lightweight accelerometers. The same equipment and measurement procedure was used as explained in the Section ‘experimental characterisation method’ below.

Table 1 shows the material parameters which lead to a maximum error of 2.28 % on the first 30 experimental modes of the tube.

Table 1. Material parameters of the PVC ducts used in the tests.

Young’s modules	Poisson’s ratio	Density
2700 Mpa	0.25	1242 kg/m ³

METAMATERIAL SOLUTION

To obtain metamaterials with stop band behaviour two conditions need to be met; (1) resonant cells have to be added to a host structure on a scale smaller than the structural wavelengths to be influenced [2] and (2) the net sum of the forces on the hosting structure contributed by a resonator should be non-zero [5]. Stop band behaviour can thus be achieved through the introduction of any system that introduces local resonant behaviour: in literature examples can be found of grids of mass-spring systems that act as resonant system [2,6,7].

In previous work stop bands were created by inclusion of resonant structures within the cavities of a sandwich structures with a hollow core [1]. In this paper, stop bands will be achieved by adding resonant structures on the outside of the duct.

As resonant structures, different designs are possible. In this paper the choice is made to design resonant structure as a cantilever beam with an end point mass, with a support to connect the resonant structure to the duct. Figure 1 shows the design of the resonant structures used in this paper. Depending on the mass at the end point, the resonance frequency of the resonant structure can be changed.

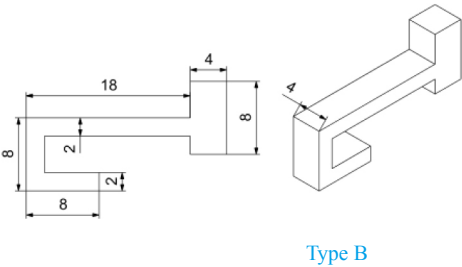
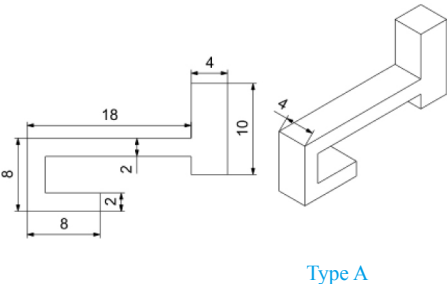


Figure 1. Design of the resonant structures used in this paper. The dimensions are given in mm. Top: design A. Bottom: Design B.

The resonant structures are made by lasercutting the samples from a 4 mm thick PMMA (Polymethylmethacrylat) plastic plate. Table 2 gives an overview of the material parameters of the used plastic. These parameters were again retrieved through correlation of the modeshapes of a numerical models and the measured modeshapes of a test sample. In this case the test sample was a flat PMMA plate with dimensions 300×200×4 mm.

Table 2. Material characteristics of the PMMA plastic for the resonant structures.

Young’s modules	Poisson’s ratio	Density
3380 Mpa	0.245	1167 kg/m ³

The designs of Figure 1 in combination with the material parameters of Table 2 leads to a predicted resonance frequency of 493 Hz for resonator A and 550 Hz for resonator B. For these predictions, a FE model of the resonant structure was used made of solid elements and the base of the resonant structures was modelled as clamped. A convergence study showed that the models were converged to an accuracy of less than 1%, which suffices in view of the trend prediction is this paper.

Five different configuration will be considered:

- Original configuration: A duct without resonant structures.
- Configuration A: A duct with 640 resonant structures of type A, resulting in a 27.4% mass addition.
- Configuration B: A duct with 640 resonant structures of type B, resulting in a 26% mass addition.
- Sequential configuration: A duct with 320 resonant structures of type A on the first half of the duct and 320 resonant structures of type B on the second half, resulting in a 26,7% mass addition.
- Mixed configuration: A duct with 320 resonant structures of type A and 320 resonant structures of type B mixed as shown on the left of Figure 2, resulting in a 26,7% mass addition.
- Configuration half A: A duct with 320 resonant structures of type A as shown on the right of Figure 2, resulting in a 18.2% mass addition.

Type B	Type A	Type B	Type A
Type A	Type B	Type A	Type B

	Type A		Type A
Type A		Type A	

Figure 2. Visualisation of the resonant structure lay-out for the mixed configuration (left) and the configuration half A (right).

To achieve stop band behaviour, the resonant structure will be added on a grid of 26 mm in perimetral direction and 25 mm in length direction.

In this way 16 resonant structure can be added across the perimeter of one cross section and 40 resonant structures are added in the length of the tube, leading to a total of 640 resonant structures per tube.

NUMERICALLY PREDICTED STOP BAND BEHAVIOUR

Stop bands can be predicted for periodic structures through unit cell modelling and the application of the Bloch-Floquet boundary conditions. Based on an undamped FE model of the unit cell and the application of periodicity boundary conditions, dispersion diagrams for freely propagating waves in an infinite periodic structure can be derived. Frequency zones for which no solutions are found, correspond to frequency zones without free wave propagation and thus a stop band region. In previous work, the authors discussed the derivation and interpretation of such dispersion diagram [1,2]. For the sake of brevity, this discussion is not repeated here.

To predict the stop band behaviour, two approaches can be followed. On the one hand stop bands can be predicted based on a model of two-dimensional infinite PVC plates. This means that the shape of the duct itself is not taken into account for stop band calculation. On the other hand, one-dimensional periodic structure models can be used and thus the entire cross section of the duct has to be modelled to predict stop bands. In this paper the first approach is chosen, since this allows for fast prediction based on small numerical models. But both approaches will be compared in future work.

From literature it is known that wave propagation through infinite periodic structures can be investigated through unit cell modelling [3, 4]. Based on an undamped FE model of the unit cell and the application of periodicity boundary conditions, dispersion curves for freely propagating waves in an infinite periodic structure can be derived. Frequency zones for which no solutions are found, correspond to frequency zones without free wave propagation and thus a stop band region. For this approach a in house code is available which has been explained in previous papers by the authors [2]. For this study a FE model of the unit cell was used which is comprised of 169 linear shell elements for the PVC part and 68 linear solid elements for the resonant structure. Convergence of the model has been verified.

For each configuration the dispersion contours are given in Figure 3. In these graphs, only the dispersion lines corresponding to out-of-plane bending motion are shown. Based on these dispersion diagrams, the predict stop band widths are given in Table 3. In the case of the sequential configuration, the stop band is predicted based on the dispersion diagram of configuration A and B.

As expected, in the mixed configuration and in configuration half A, the stop band width is reduced with respect to configuration A. This can be explained by inspecting the mass ratio of the resonant structure with respect to the mass of the remainder of the structure: according to previous research, this mass ratio correlates to the stop band width [2]. In configuration half A, the remainder of the structure is twice the mass of the case of configuration A. In the mixed configuration the remainder of the mass is twice the mass of configuration A, and additionally the mass of resonant structure B. As can be seen in Table 3, resonant structure A creates the largest stop band in configuration A, and the smallest stop band in the mixed configuration.

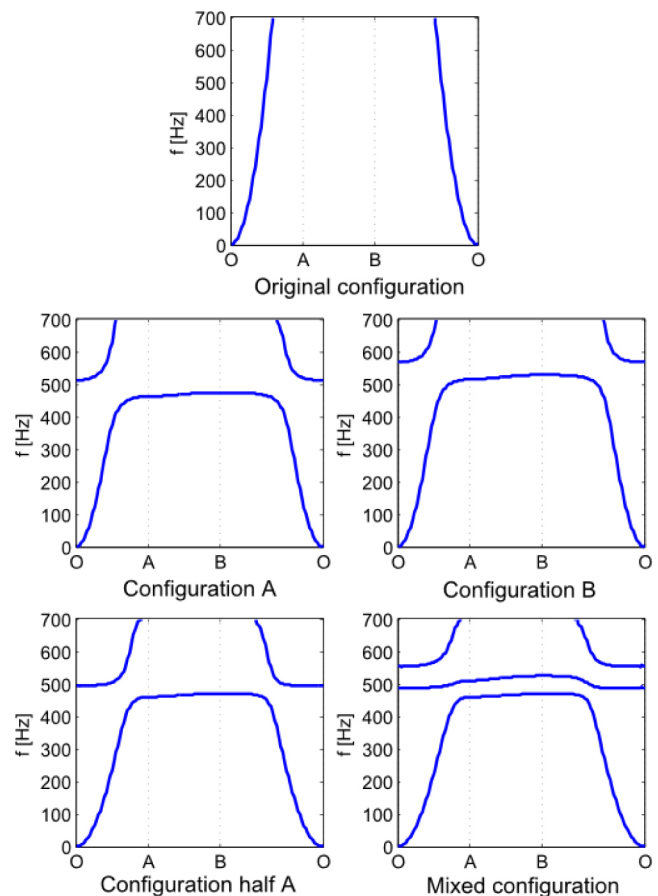


Figure 3. The dispersion contours for the different configurations. Figure 4 gives the interpretation of the horizontal axis of the Brillouin contour.