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Multiphysics Modelling and Experimental Verification of Active and Passive Reduction of Structural Noise

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ABSTRACT

A fully-coupled multiphysics modelling is applied for the problem of simultaneous active and passive reduction of noise generated by a thin panel under forced vibration providing many relevant results of various type (noise and vibration levels, necessary voltage for control signals, efficiency of the approach). The active approach is validated experimentally. To this end, the plate of panel is excited in order to generate a noise consisting of significant lower and higher frequency contributions. Then, the low-frequency noisy modes are reduced by actuators in the form of piezoelectric patches glued with epoxy resin in locations chosen optimally thanks to the multiphysics analysis. The emission of higher frequency noise should be attenuated by well-chosen thin layers of porous materials. A fully-coupled finite element system relevant for the problem is derived; such multiphysics approach is accurate: advanced modelling of porous media can be used for porous layers, the piezoelectric patches are modelled according to the fully-coupled electro-mechanical theory of piezoelectricity, the layers of epoxy resin are considered, finally, the acoustic-structure interaction involves modelling of a surrounding sphere of air with the non-reflective boundary conditions applied in order to simulate the conditions found in anechoic chambers. The FE simulation is compared with some experimental results. The sound pressure levels computed in points at different distances from the panel agree excellently with the noise measured in these points. Similarly, the computed voltage amplitudes of controlling signal turn out to be good estimations.

INTRODUCTION

The paper presents a fully-coupled multiphysics approach to modelling of active-passive structural noise attenuators like acoustic panels, liners and composites. Such advanced modelling has been recently proposed [1, 2, 3] to deal with hybrid attenuators which involve an active approach as a remedy for the lack of performance at low frequency, and provide excellent passive absorption in medium and high frequency range. An additional effect based on the combination of both approaches is also important.

Here, a system for multiphysics modelling of such hybrid attenuators is briefly discussed and a generic two-dimensional example is presented. Then, the system is used to thoroughly model a problem of active vibroacoustic attenuation, and the numerical results are validated experimentally by tests carried out in anechoic chamber. The analysis is relevant for techniques of the vibration control and Active Structural Acoustic Control [4], and similar approach to such problems may be found, for example, in [5, 6, 7]. Some of the results briefly presented in this paper are extensively discussed in [8].

FULLY-COUPLED MODELLING OF PASSIVE AND ACTIVE NOISE REDUCTION

Accurate modelling of simultaneous active and passive noise attenuation by smart panels, liners or composites requires a multiphysics approach and the use of the finite element method to cope with complex geometries. A system for such an accurate modelling of active elasto-poroelastic noise attenuators has been developed and it combines the following theoretical models relevant to the problems involved:

- the Biot's theory of poroelasticity – to model the vibroacoustic transmission and passive dissipation of acoustic waves in porous layers,
- the linear acoustics – to model the propagation of acoustic waves in the surrounding air and in air-gaps,
- the linear elasticity – to model the vibrations of elastic faceplates and inclusions,
- the theory of piezoelectricity – to model the piezoelectric actuators and active control of low-frequency vibrations.

The system allows for the consideration of mutual interactions of all these various physical problems, since all the relevant couplings are implemented.

Figure 1 presents a simple generic example for two-dimensional analysis of an active-passive acoustic panel. A simple slat-shaped panel is composed of a single layer of highly-absorbing porous material fixed to an aluminium plate. The porus layer is made of a polyurethane foam modelled using the Biot's theory of poroelasticity. The plate is 1 mm-thick and the thickness of porous layer is 24 mm, so that the total thickness of panel is 25 mm. The width of panel is 120 mm, whereas its length is considered to be significantly bigger, so that such slat panel could be modelled using two-dimensional (plane strain) approach. The panel is put into a 120 mm-wide slit of a waveguide filled with air. The plate of panel is simply-supported at both ends (at the walls) and the skeleton of porous layer can freely slide along the waveguide walls. (Such boundary conditions can be easily realized in practice.) The panel should allow an active approach: thus, in the centre of the free face of aluminium plate a 0.3 mm-thick piezoelectric patch of PZT material is glued; its width is 20 mm. A plane harmonic acoustic wave propagates in air of the waveguide, onto the panel. The amplitude of source pressure is $p_0 = 1$ Pa. Depending on its frequency, the wave can be partially reflected and absorbed by the panel, or transmitted through it.

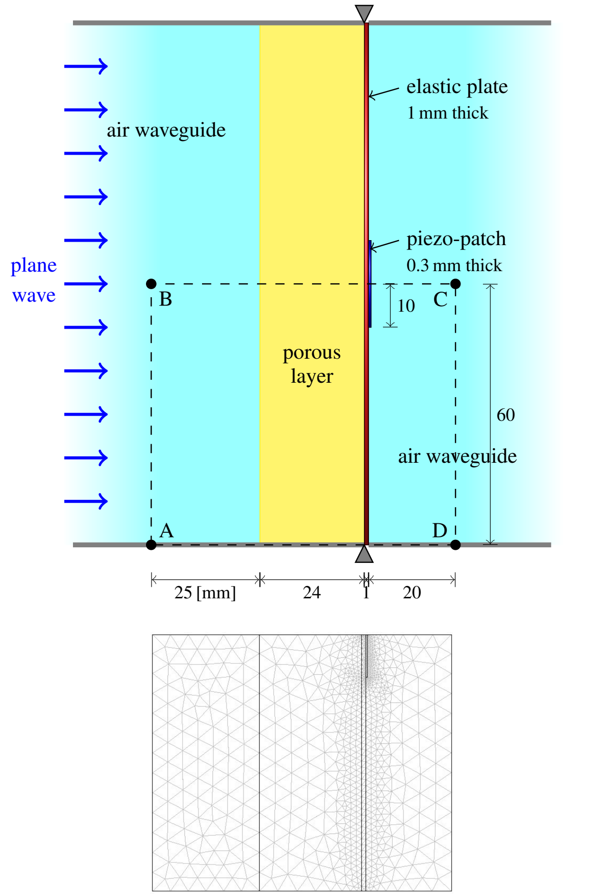


Figure 1: A generic two-dimensional example of an active-passive acoustic panel, the representative modelled area ABCD, and the finite element mesh

By taking advantage of the symmetry, the problem can be modelled using the rectangular domain ABCD shown in Fig. 1, where the finite-element mesh of the modelled domain is also presented. Appropriate boundary conditions must be applied. The rigid-wall or sliding conditions are set on relevant parts of boundaries AD and BC (in the latter case, these conditions are valid because of the symmetry plane). On the boundary CD, the free radiation condition is set, or alternatively, the impedance boundary condition (using the characteristic impedance of air) can be set. (It was checked that both approaches give very similar results.) Such conditions simulate for a finite domain (modelled with finite elements) the fact that there is no reflection at the relevant boundary and the wave can freely propagate outside the domain. Finally, a free radiation condition with a plane incident pressure wave is set on the boundary AB. Another approach, where simply $p = p_0$ is set on the boundary would result in the appearance of some additional cavity resonances.

Firstly, the behaviour of panel in passive state was analysed. The passive behaviour means that the piezoelectric patch was simply shunted (no voltage signal was sent to its electrodes), and the only excitation provided to the system was by the plane harmonic wave. The passive results are shown in Fig. 2 in the form of sound pressure level (SPL) curves calculated at point C for a wide range of frequencies. Remembering that the SPL of source is app. 94 dB (for the pressure amplitude $p_0 = 1$ Pa), these curves illustrate the expected reduction of noise. Two curves are presented for the passive state, namely, the SPL curve obtained for the poro-elastic panel is compared with the SPL curve obtained for the plate in the absence of the porous layer (in the latter case, the subdomain of porous layer is modelled as air using the same mesh of elements). From

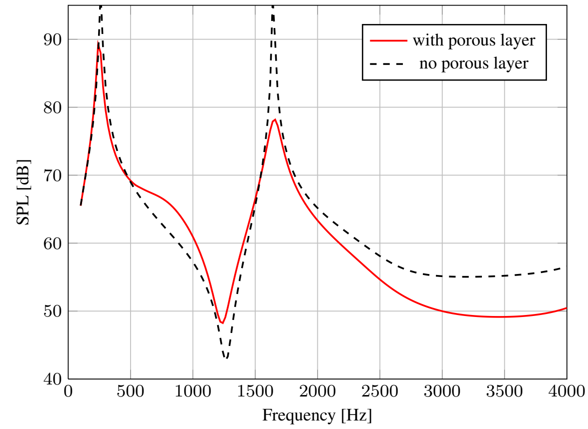


Figure 2: SPL at point C in the whole frequency range for the passive panel

the comparison it is clearly visible that for frequencies above 1.5 kHz the performance of the panel is better that that of the plate, and above 3 kHz this improvement reaches and exceeds 6 dB. The absorbing effect of rather thin porous layer allows for significant reduction of vibro-acoustic transmission in higher frequencies. An important observation concerns two peaks that appear at app. 240 Hz and 1700 Hz. They are the vibro-acoustic noise resulting from the first and second eigenmode vibrations of the plate, caused by the acoustic wave excitation of relevant frequency. It can be observed that the porous layer damps slightly the effect of noise emitted at plate resonances; it is not so in the absence of porous layer (the inherent plate damping is comparatively very small and it was considered in the model of elastic plate). Nevertheless, an active approach is necessary in order to reduce the vibro-acoustic transmission not only at these eigenfrequencies, but for lower frequencies in general.

Figure 3 shows some results of active analyses carried out for the panel excited by the plane acoustic wave with harmonic frequency of 240 Hz (which is approximately the first eigenfrequency of the plate) and a voltage signal of the same frequency applied to the electrodes of the piezoelectric patch in order to reduce the noisy vibrations. The generated/transmitted low-frequency noise was observed at point C for different amplitudes of the voltage signal in the range from 0 to 1 V. (It is worth to mention that a similar noise would be observable even farther from the plate since for the first eigenmode shape there is no auto-cancelation of waves generated by the neighbouring parts of the plate and the total emitted wave quickly becomes plane in the waveguide.) The sound pressure level computed at point C for different voltage amplitudes is shown in Fig. 3: from this SPL curve it can be assessed that the necessary amplitude of the voltage signal is slightly more than 0.9 V. One should remember, however, that in practice, the noisy vibrations are often induced

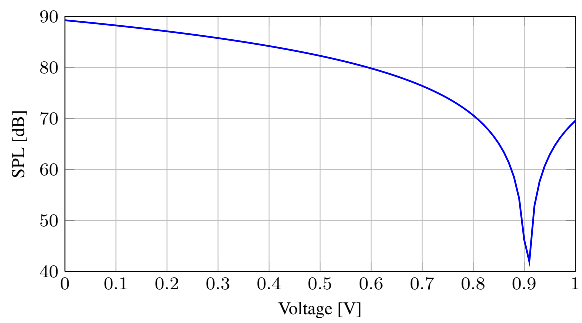
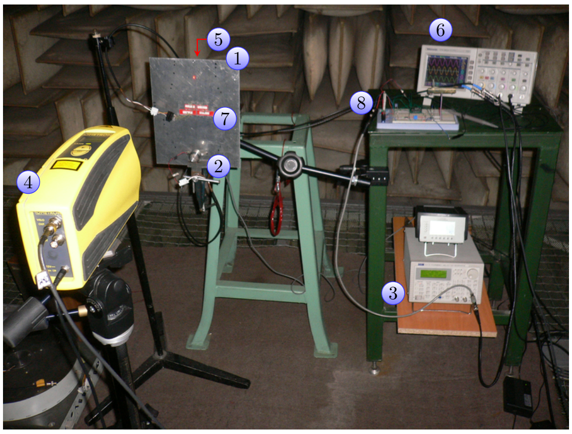


Figure 3: SPL at point C at 240 Hz for different values of the voltage amplitude of the active signal

by causes stronger than impinging acoustic waves, and would generally require higher voltages for active control, as will be illustrated in the next section.

MODELLING AND EXPERIMENTAL VALIDATION OF ACTIVE VIBROACOUSTIC ATTENUATION

Figure 4 shows the experimental stand which was used for tests carried out in an anechoic chamber in order to validate a fully-coupled modelling of active vibroacoustic reduction. The main part of the stand is a simple generic panel, or rather an elastic faceplate of panel to which some active elements are fixed. The plate is made up of aluminium and the active elements are: a few piezoelectric patches of PZT glued to its surface by the epoxy resin, and the amplified piezoelectric exciter fixed close to the clamped area. The exciter is aimed to induce low-frequency vibrations of the plate and so it is connected to a signal generator. The vibrations of plate can be measured by a laser vibrometer which registers the velocities of deflection at a chosen point. The vibrating plate generates an acoustic wave which propagates in the surrounding air and can be measured by a microphone placed at some distance (typically several centimeters) behind the plate. All the generated and produced-at-measurements signals are observed on an oscilloscope. Eventually, an electric signal is sent to the electrodes of the piezo-patch actuators so they expand and contract thus affecting the vibrations of plate by locally-induced bending. The electric signal is produced by the signal generator: in fact, it is supposed to be the same harmonic signal sent to the exciter but transformed by a signal changer.



1. aluminium plate
2. low-frequency exciter (fixed to the plate)
3. signal generator
4. laser vibrometer
5. ~~~
6. microphone (behind the plate)
7. oscilloscope
8. piezo-patch actuators (glued on the plate)
9. signal changer

Figure 4: Experimental stand for validating the fully-coupled modelling of active vibroacoustic reduction

A multiphysics approach was applied to model the problem. The thin elastic plate (with thickness 0.93 mm and dimensions 280 mm$\times$224 mm) is modelled using the finite element method based on the displacement formulation of linear elasticity. The elastodynamic behaviour of the plate can be affected by the presence of piezoelectric exciter and patches, as well as by the surrounding air, (see Fig. 5). The plate is clamped at one end and the piezoelectric patches are fixed to its upper surface so by stretching they bend the plate. To grasp this effect directly, the thin plate is modelled as three-dimensional structure. A spherical region of air surrounding the panel is modelled using the classical pressure formulation of linear acoustics. The acoustic wave propagating in this region results from the vibrations of

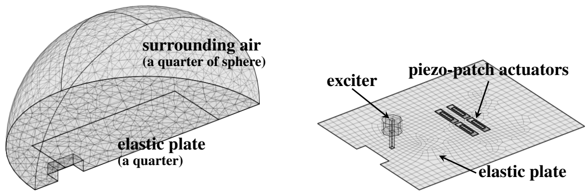


Figure 5: Some finite-element meshes used for the problem

aluminium plate. Eventually, only a quarter of the sphere can be modelled (see Fig. 5, left) in case of harmonic oscillations, because the most relevant cases of "noisy" vibrations are linked with symmetrical mode shapes. The region of surrounding air is modelled with finite elements: some distance from the plate, non-reflective boundary condition is applied in the form of the acoustic impedance of air. On other boundaries the relevant boundary conditions are specified: the so-called no penetration condition (by setting the normal displacements of fluid particle to zero) on the rigid walls of the vice clamping the plate, and on the (vertical) plane of symmetry; the zero-pressure condition on the (horizontal) plane of antisymmetry.

In order to simulate an unwanted behaviour of a panel transmitting a low-frequency noise, the generic faceplate of panel is excited at low frequency inducing a "noisy" symmetric mode. To this end, the vibration exciter is used, which was deliberately located closely to the clamped area. Such a localization of the exciter justifies a simplified approach to its modelling, in particular, its effect may be simply modelled by a concentrated force, but even in the case of a more advanced modelling any model errors are less influential. It has been found experimentally (by sweeping through the frequency of harmonic signal sent to the exciter) that for the plate the lowest "noisy" mode exists at app. 113 Hz. This eigenfrequency and mode has been confirmed by a numerical analysis and the corresponding mode shape is shown in Fig. 6 (the black marks on the plate visible on the photograph form the neutral curves of deflection, that is, the points where the deflection velocities have zero amplitude). This is the 5th eigenmode of the plate and a very noisy mode because of its symmetrical bulging shape (like a "loud-speaker"), see Fig. 6. For further investigations the frequency of 115 Hz is chosen, as the one which produces stable vibrations of the shape presented in Fig. 6, generates a distinct low-frequency noise, and is very well represented by the modelling.

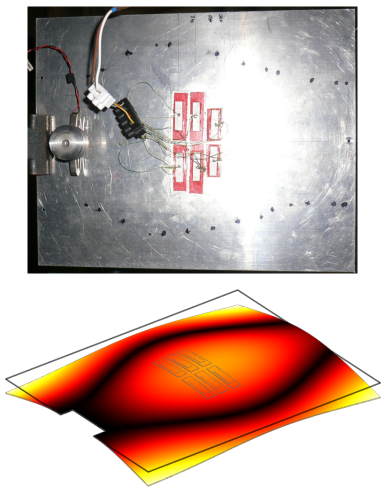


Figure 6: Aluminium plate and a "noisy" low-frequency mode

Figure 7 presents a scheme of the experiment carried out in the anechoic chamber shown on the photograph in Fig. 4. The scheme can be summarized as follows:

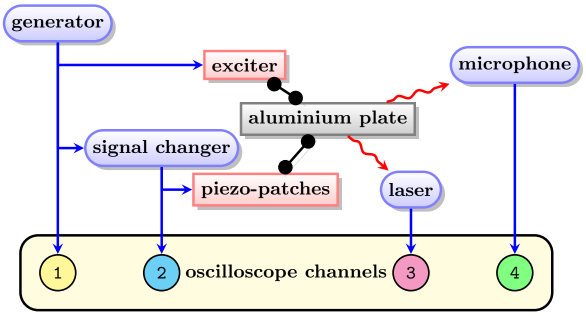


Figure 7: Scheme of the experiment

1. a harmonic signal of 115 Hz (which is the chosen "low and noisy" frequency) is sent by the generator to the exciter fixed on the elastic plate (and to the oscilloscope, where it can be observed);
2. the exciter induces significant vibrations of the plate, and the amplitude of velocity of the normal deflection of plate are measured in some points by the laser vibrometer and observed on the oscilloscope;
3. the vibrating plate generates a low-frequency noise which propagates in the air from both surfaces of the plate in the form of harmonic acoustic waves;
4. the noise, or more specifically, the amplitude of acoustic pressure, is measured by the microphone placed at some chosen distance from the plate (the measured signal is sent to the oscilloscope);
5. in case of the active approach, the same harmonic signal which is being sent to the exciter is now also sent to a signal changer where it can be adjusted;
6. from the signal changer, the adjusted signal is sent to the electrodes of the piezo-patch actuators which begin to expand and contract harmonically, and that behaviour locally affects the bending of the plate since the piezo-patch actuators are glued to its surface;
7. the action of piezo-patch actuators is aimed to reduce the "noisy" modes of vibration of the plate in order to reduce the generation of noise – the reductions are observed on the oscilloscope (thanks to the laser continuously measuring the vibrations, and the microphone continuously measuring the noise) and the observations can be used to better adjust the signal sent to the actuators.

First, the modelling has been validated in the passive case where the noise and vibrations are not reduced actively (the electrodes of piezoelectric patches are simply shunted). Some of the passive results are shown in Fig. 8 in the form of the sound pressure level (SPL) at different distances from the plate. It is observed that SPL reaches 90 dB at 2 cm from the plate surface, and at 12 cm it does not exceed 78 dB. These results are validated by the measurements taken by the microphone (and, independently, with the usage of a Bruel&Kjaer Hand-held Sound Analyzer) in different points behind the plate: particularly, at some points on the line defined by the intersection of planes $x = 15$ cm and $y = 0$ cm (assuming that the x-axis is along the length of plate – it the middle and in the direction of its free edge, the y-axis is along the clamped edge, and the z-axis is perpendicular to the plate). The computed and measured results (see Fig. 8) are very much alike.

Several numerical tests of active reduction of structure-borne noise were carried out before the piezo-patch actuators were actually glued to the plate. The tests allowed to choose an efficient placement of piezo-patch actuators and check the feasibility of the active approach. Moreover, the voltage amplitudes for the active signal were correctly estimated during the numerical tests. The numerical tests involved one, two, and three pairs of

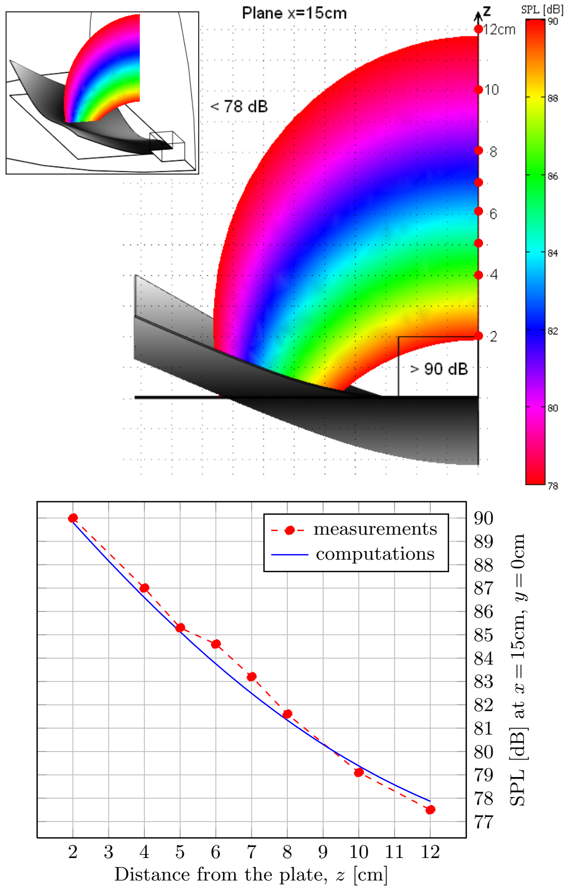


Figure 8: Sound pressure level (SPL) at different distances from the plate (passive case)

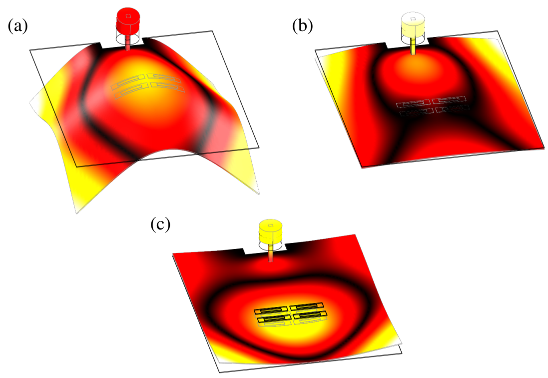


Figure 9: Numerical analyses for the plate: (a) passive behaviour, (b) active approach, (c) overloading

symmetrically placed actuators. Figure 9 shows final numerical analyses for the plate with three pairs of piezo-patches. A "noisy behaviour" of panel is shown in Fig. 9(a); the electrodes of piezoelectric patches are shorted (i.e., the voltage is set to 0 V), and so the actuators are passive. The plate driven by the exciter freely vibrates and emits a significant low-frequency noise. The amplitude of acoustic pressure (registered at the point of measurement by the microphone) is 0.22 Pa which corresponds well with the computed value. Figure 9(b) shows an experimental verification of the active reduction of structure-borne noise. Now, the harmonic signal of the peak-to-peak amplitude of approximately 23 V is applied to the electrodes of piezo-patch actuators. Such amplitude is chosen in order to damp the "noisy" mode of vibration – it is slightly smaller than 20 V estimated numerically. Now, the amplitude of plate deflections is less than 5% of the original (undamped) value, and the noise is no longer audible. The shape of vibrations is significantly different than the undamped one, and as a source of sound it is not so efficient. Moreover, the amplitudes of deflection are now much smaller, especially, at the center of the plate (notice that the plots in Fig. 9 have the same scale of deformation). The reduction of noise – assessed on the basis of the amplitudes of acoustic pressure registered by the microphone – is very big indeed and reaches 97% (on the basis of a logarithmic scale such as SPL – it means 36% reduction). Finally, Fig. 9(c) shows a case when an overload signal is sent to the electrodes of piezoelectric patches: the peak-to-peak amplitude of 27 V is too big. Since the signal is too strong, the counter-vibrations caused by the piezo-patch actuators are significant (the amplitude of deflection is nearly 17% of the amplitude of undamped vibrations). Thus, a noise is audible again, although it is more quiet than the noise of the passive phase. The microphone measurements permit to estimate it for nearly 20% of the undamped noise, so with such moderate overloading one can still experience a very significant reduction of noise by 80% (it means 17% reduction of SPL). These relative quantitative results of experimental tests are shown in Fig. 10.

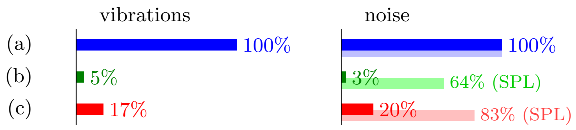


Figure 10: Relative results of the tests: (a) passive behaviour, (b) active approach, (c) overloading

FINAL REMARKS

The accurate estimations obtained from the proposed fully-coupled modelling were confirmed by the experimental tests. It was shown that the chosen setup of piezo-patch actuators allows for an almost complete reduction of a low-frequency noise generated by a "noisy" mode of vibrations. However, an efficient control system is necessary to prevent overloading and work in reality. In order to validate experimentally models involving porous materials the identification of some crucial material parameters of such media (required by the Biot's theory of poroelasticity) is indispensable.

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