JOURNAL OF THEORETICAL AND APPLIED MECHANICS
49, 4, pp. 1135-1149, Warsaw 2011

PASSIVE-ACTIVE NOISE CONTROL OF AN ACOUSTIC DUCT

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In the recent years, noise control has become a significant factor in the design of automotive and aircraft industries. In general, methods of the passive noise control are practical and most effective at mid and high frequencies. On the other hand, active noise control techniques are more efficient at the low frequency range. Combined solutions seem to be the most appropriate key to cover the whole frequency range of frequencies. In the present study, a description of simulations for both feedback structural and acoustical structural control strategies are illustrated in order to maximize the damping within an acoustic duct by applying a passive foam layer bonded to an "active" surface.

Key words: acoustic structural control, damping noise control, feedback

1. Introduction

In the last years, there has been an increased interest for hybrid control strategies, to improve performance of active noise control in the low frequency range.

In general, passive sound absorbing materials are implemented for a large range of applications to attenuate the sound propagation. The dissipation energy occurs because the air molecules in the interstices oscillate with the frequency of the exciting sound wave. This oscillation results in friction losses. Changes in the flow direction, expansions and contractions of the flow throughout irregular pores result in a loss of momentum in the wave of propagation. These two phenomena account for the most of the energy losses in the high frequency range (Beranek, 1992). A porous layer can absorb a large amount of acoustic energy only if the thickness is comparable to the wavelength of the incident sound. This implies that such a passive technique is not very practical in the low frequency region.

On the other hand, active vibration control techniques are effective at low frequency ranges, without adding considerable weight to the vibrating structure. However, the main limitation of using active vibration control alone at higher frequencies stems from the fact that extensive computational requirements are needed for effective implementation of a proper controller. In addition, other drawbacks such as risk of instability due to control spillover and unreliability due to failure of sensors/actuators reduce the reliability of the active control systems.

Hybrid active-passive control techniques, which use both active and passive elements in either series or parallel, allow overcoming all of these limitations. The passive element usually carries the primary vibration attenuation characteristic, while the active component is used to enhance the performance of the passive system.

One of the first published works on hybrid techniques is that of Guickling and Lorenz (1984). The passive component was comprised of a porous plate located in an impedance tube, a small distance away from the open end of the tube which was terminated by a control speaker. One microphone placed in front of the porous plate was sent to the control speaker. The second microphone controlled the amplification factor such that the sound pressure at that location was minimized so as to produce a pressure release condition just behind the plate.

Gentry *et al.* (1997) developed a smart foam for passive active noise control. It consists of cylindrically curved sections of PVDF (Polyvinylfluoride film) embedded in a partially reticulated polyurethane acoustic foam. The performance of the active passive device is studied for controlling sound radiation from a vibrating piston.

Guigou and Fuller (1999) designed a foam-PVDF smart skin for aircraft interior noise control. Different reference signals were implemented for the feed forward controller and were compared in terms of interior noise attenuation achieved.

A hybrid absorption system comprised of a layer of sound absorbing material positioned at a distance from a movable wall acting as the active component was implemented by Smith *et al.* (1999) to achieve sound absorption over a broad frequency range.

Nabil (2004) investigated a new class of smart foam able to control simultaneously structural and acoustic cavity modes over a broad frequency range.

Thenail *et al.* (1994) presented an active system that included a fibreglass absorbing layer backed by an air cavity terminated with an active surface. Their work was intended to show that the pressure-release condition on the back surface of the fibreglass leads to improvement of absorption. They investigated two control approaches. In the first one, an error microphone was placed on the back surface of the fibreglass layer and the pressure at that location was minimized. The second approach was identical to the work by Guickling and Lorenz, where the porous plate was replaced with the fiber glass layer.

A hybrid passive-active system for damping augmentation, which is again based on the concept of mounting a layer of porous material on an elastic panel, is here proposed.

The system consists of a porous layer coupled with an active panel placed at one end of a duct, while the acoustic excitation is located at the other extremity, in order to achieve damping amplification.

In the initial approach, a structural feedback control strategy by using sensor accelerometers is considered. As actuators, a shaker and then a piezoelectric element have been taken into account. The signal sensed by the sensor applied on the plate is supplied back to the actuator in a classical co-located feedback configuration.

The second approach considers a new scheme which combines acoustic sensors (two microphones) with a mechanical actuator (shaker). The pressure is captured in two different points close to each other and placed just in front of the actuator. The derivative of the pressure gradient has been supplied back to the shaker in order to increase damping capacity.

The effectiveness of this approach is here numerically demonstrated, while an experimental approach is verified by Tiseo *et al.* (2010).

Sound-absorbing materials coupled with an active layer can be used in the construction of aircraft, spacecraft and ships because of their low weight and effectiveness when used correctly. This trend is driven by demands for higher load capacity and reduced fuel consumption for cars, trucks and aerospace structures.

2. FE model description

A numerical simulation was performed within MSC/NASTRAN. The length of the duct is sufficiently large compared to its cross section dimensions, so that the acoustic waves travel along the axis of the duct with planar wave fronts. This assumption enables us to treat the duct as a one-dimensional system. An acoustic excitation, due to piston motion having frequency between 1 up to

4000 Hz and placed at the right extremity of the duct, has been considered, while an aluminium plate is located at the left side end of the duct. For sake of clarity, the lateral surfaces of the duct have not been represented. The duct is backed by a flexible plate with a bonded porous layer, Fig. 1.

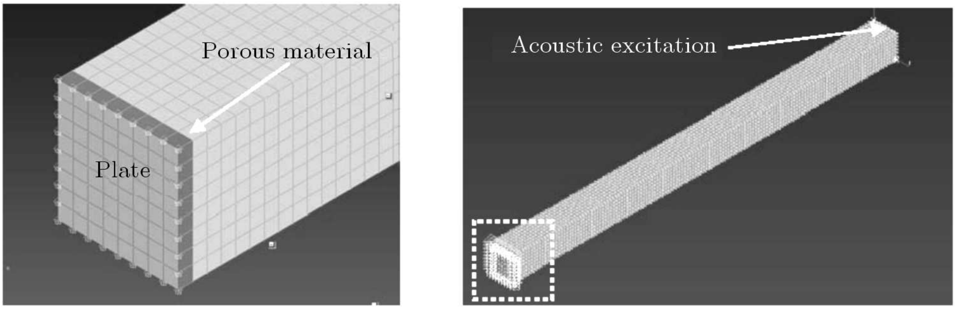

Fig. 1. Acoustic duct

The absorbing material is simulated by a mass attached to a spring and damper which are in parallel connection. The acoustic absorber is defined by the CHACAB (Connection Hexa Acoustic Absorber) elements constituting the interface between the structural elements (plate) and the fluid part (air).

The acoustic impedance of the porous material, in terms of resistance and reactance, Fig. 2, given as the input, allows Nastran to calculate the value of the mass, spring and damper used to simulate the behaviour of the absorbing material.

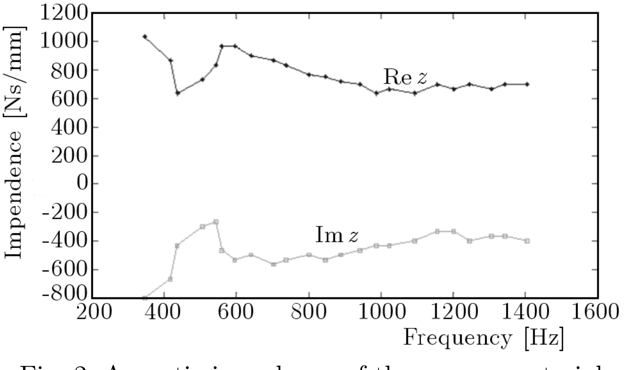

Fig. 2. Acoustic impedance of the porous material

This numerical approach, in which porous materials are represented through their global features, does not allow appreciating the dependence of material properties with variation of their thickness. This represents a limitation of the performed analysis.

3. Modal analysis: plate and acoustic cavity

Before the control strategies implementation, a quick verification of the model has been performed through a comparison with the results available in literature.

Both for the plate and acoustic cavity, an evaluation of eigenvalues and eigenvectors has been executed, Fig. 3 and 4. The plate simply supported on all edges has the dimension of $30\,\mathrm{mm} \times 30\,\mathrm{mm} \times 0.15\,\mathrm{mm}$. A comparison between the numerical and analytical values has been performed. Results have been shown in Table 1.

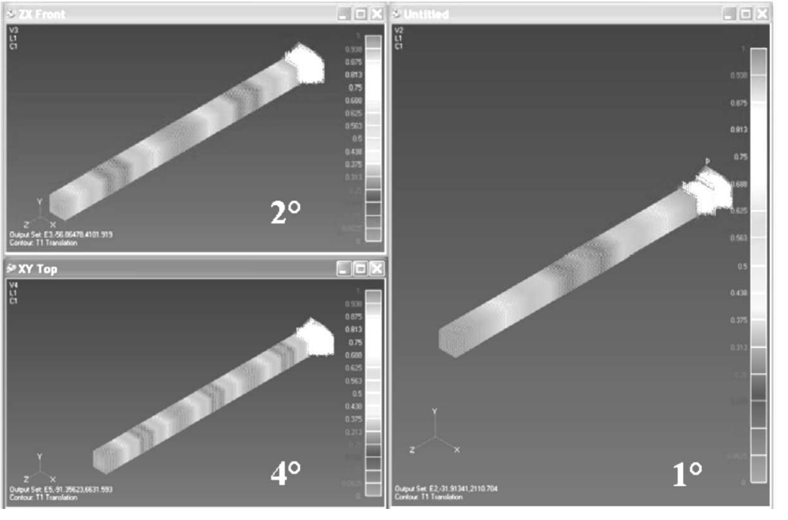

Fig. 3. Modal shapes (1-2-4 modes) for coupled structural acoustic system manly influenced by the acoustical part

Table 1. Natural frequencies. Numerical results vs. theoretical

Natural frequencies [Hz]					
Plate			Acoustic cavity		
FE analysis	Theory	$\Delta\%$	FE analysis	Theory	$\Delta\%$
803	819	1.9	343	345	0.5
1994	2048	2.6	686	690	0.5
3081	3277	5.9	1029	1035	0.5
3994	4096	2.5	1373	1380	0.5

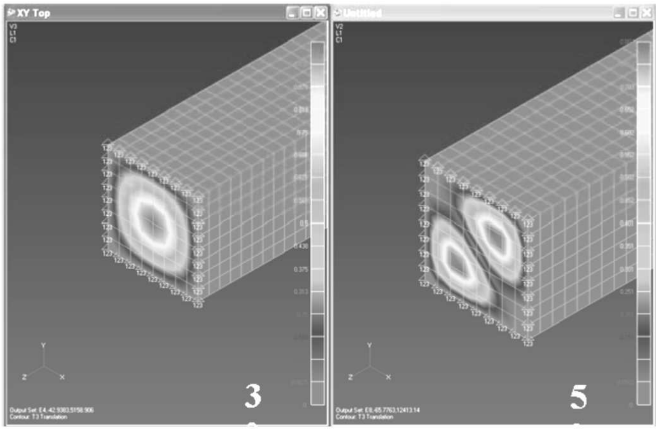

Fig. 4. Modal shapes (3-5 modes) for coupled structural acoustic system manly influenced by the structural part

4. Structural acoustic system: complex modal analysis

For the complex modal analysis, the direct solution defined by NASTRAN (SOL 107) has been used. The analysis has been conducted considering a 3% damping value for the fluid part.

As a consequence of application of the porous material, a slight variation of natural frequencies has been observed together with a significant increase of the damping value, especially for coupled modes which are influenced mainly from the acoustical part (shown in Table 2 written in bold).

Table 2. Coupled fluid – structure system-natural frequencies and damping coefficient

Numerical analysis: coupled fluid – structure system complex eigenvalue f and damping coefficient ζ			
without foam		with foam	
f [Hz]	ζ [%]	f [Hz]	ζ [%]
335	3.02	332	5.21
652	2.77	651	3.99
821	1.66	820	1.71
1055	2.75	1054	3.74
1975	1.05	1975	1.09

5. Structural control

The feedback control (FBC) strategy adopted consists in capturing the structural vibration through a velocity sensor applied to the plate, and feeding it back to an actuator also located on the plate, with the aim of modifying the system dynamic behaviour, Fig. 5.

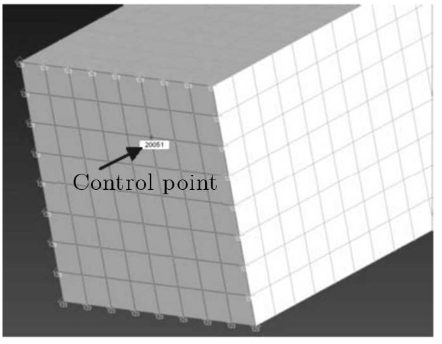

Fig. 5. Acoustic cavity. Structural control: excitation by shaker sensing by accelerometer

The system response variation is evaluated through change of natural frequencies and damping coefficients modification, estimated through complex modal analysis. The SISO control system (Single Input – Single Output) refers to co-located sensor-actuator that means both have been placed at the same physical point of the structure.

5.1. First configuration: shaker vs. accelerometer

In the first application, a shaker has been chosen as actuator and an accelerometer as sensor. They have been placed on a control node, which has been chosen away from the symmetry line of the plate, in order to excite as more as possible plate normal modes.

The signal measured from the accelerometer was amplified and supplied back to the shaker. The study has been conducted for different values of amplification factor (gain) looking for the optimal gain value, defined as the one able to dissipate the maximum energy.

Figure 7 shows the damping coefficient and natural frequencies (related to the first structural acoustical mode-335 Hz) with respect to the gain factor K.

While the control action is increasing (which means that k increases) the natural frequency raises because of the stiffness augmentation. In theory, $k \to \infty$, corresponds with a clamped condition of the control node. Figures 7

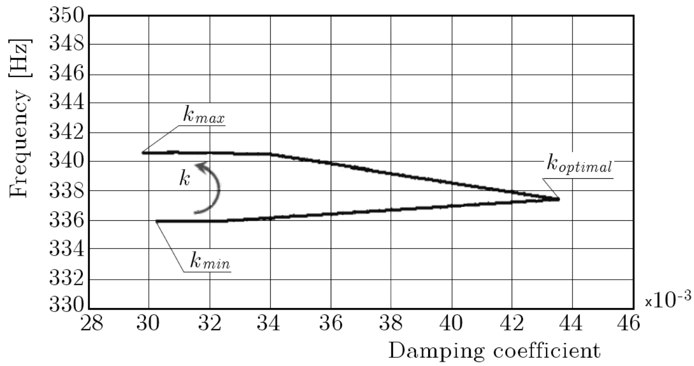

Fig. 6. Coupled system. Damping coefficient and natural frequencies vs. gain factor

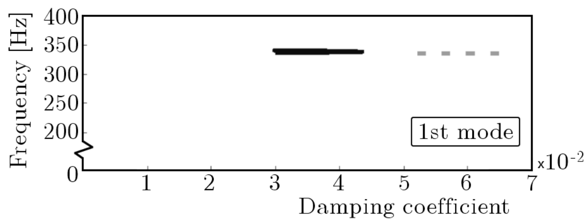

Fig. 7. FBC shaker vs. accelerometer, 1st mode. Damping coefficient and natural frequencies vs. gain factor; dashed curve – with foam, solid curve – without foam

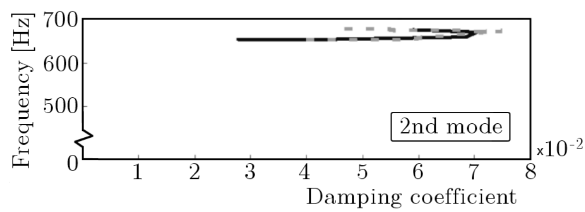

Fig. 8. FBC shaker vs. accelerometer, 2nd mode. Damping coefficient and natural frequencies vs. gain factor; dashed curve – with foam, solid curve – without foam

and 8 show the first and second complex modes of the coupled system. The insertion of the sound absorbing material implies a curve forward shift corresponding with an increase of the damping values. The insertion of the foam does not raise significantly the system damping of the third mode, which is mostly influenced by the structural part of the system. See Fig. 9 where the curves are superimposed. That is quite expected, since the porous material influences mainly the fluid damping than the structural one.

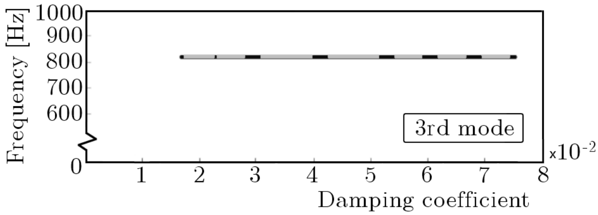

Fig. 9. FBC shaker vs. accelerometer, 3rd mode. Damping coefficient and natural frequencies vs. gain factor; dashed curve – with foam, solid curve – without foam

5.2. Second configuration: piezo vs. accelerometer

The control strategy consists in capturing the plate vibration (in terms of displacement or velocity) on the selected node and feeding it back to the piezoelectric element, which has been placed at the same accelerometer location, Fig. 10. This scheme can still be considered as a collocated feedback control within the frequency range in which the half wavelength of the considered mode is of the same order of magnitude of piezo dimensions.

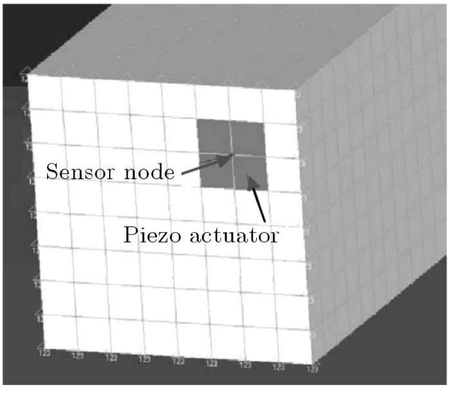

Fig. 10. Feedback structural control: excitation by piezo sensing by accelerometer

In the FE simulation, the piezo action has been represented through transmitted forces to the structure.

Comparing this control strategy with the previous one (shaker/accelerometer), the following can be observed:

- In the case of piezo actuator, the damping increment is lower than the one evaluated in the case of the shaker actuator. This is true for the naked and porous material covered plate. The first and second mode has been illustrated in Figs. 11 and 12.

- The insertion of the porous material implies damping augmentation for all modes, Figs. 13-15. This is even more accentuated for modes mainly influenced by the acoustical part (1, 2, 4 modes).

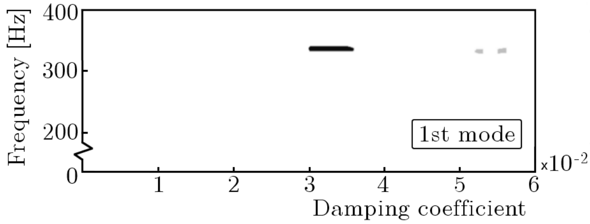

Fig. 11. FBC piezo/accelerometer, 1st mode. Damping coefficient and natural frequencies vs. gain factor; dashed curve – with foam, solid curve – without foam

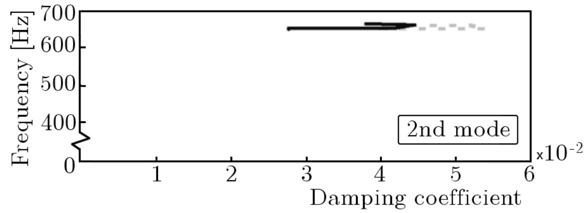

Fig. 12. FBC piezo/accelerometer, 2nd mode. Damping coefficient and natural frequencies vs. gain factor; dashed curve – with foam, solid curve – without foam

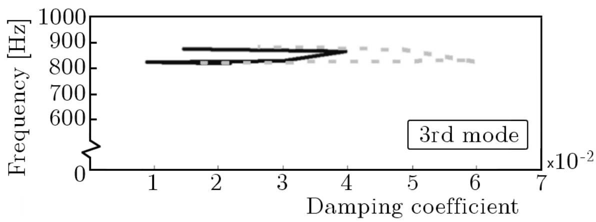

Fig. 13. FBC piezo/accelerometer, 3rd mode. Damping coefficient and natural frequencies vs. gain factor; dashed curve – with foam, solid curve – without foam

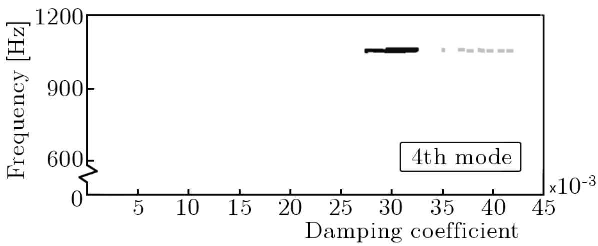

Fig. 14. FBC piezo/accelerometer, 4th mode. Damping coefficient and natural frequencies vs. gain factor; dashed curve – with foam, solid curve – without foam

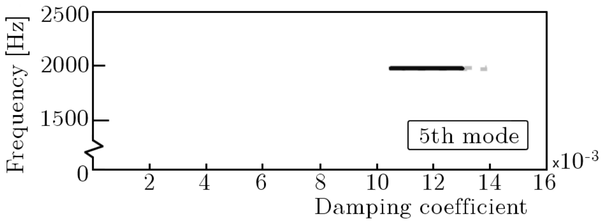

Fig. 15. FBC piezo/accelerometer, 5th mode. Damping coefficient and natural frequencies vs. gain factor; dashed curve – with foam, solid curve – without foam

6. Structural acoustic control

The pressure is captured in two different points close to each other. The first point is coincident with the place of application of the force exerted by the shaker, Fig. 16.

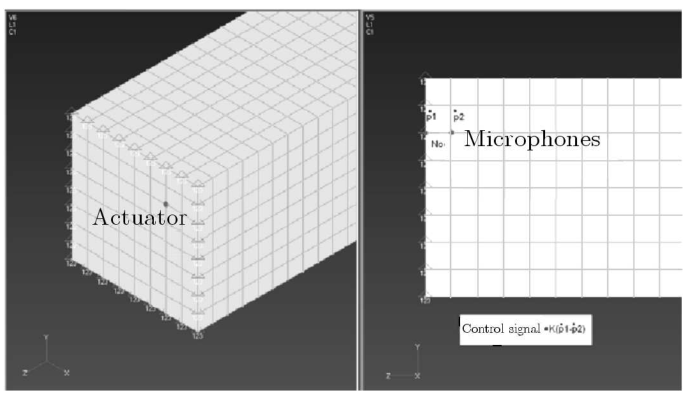

Fig. 16. Feedback structural acoustic control: excitation by shaker sensing by microphones

The derivative of the pressure gradient has been feedback to the shaker. The reason of this choice, originates from the coupled plate and fluid system boundary conditions

$$\frac{\varDelta p}{\varDelta n} = -\rho \ddot{w} \tag{6.1}$$

This means that in order to increase system damping capacity (strictly related to velocity), the feedback signal has to be proportional to the time variation of the pressure gradient.

The point control application has been chosen to be the same as in the previous cases, in order to compare the obtained results.

The first and the second complex mode seem to be significantly influenced by the application of the sound absorbing material, Figs. 17 and 18. The third mode, which is mainly effected by the structural part of the coupled system, shows a considerable decrease of the frequency while the damping coefficient rises. The third mode, which is mainly influenced by the structural part of the system, is quite altered from the adopted control strategy: the natural frequency drops down while the damping coefficient ascends, Fig. 19. The first trend is due to the fact that the control strategy aims to reduce the pressure within the cavity; hence, the plate accomplishes the pressure fluctuation introducing a structural stiffness decrease. Such kind of behaviour could cause structural instability, which implicates the inapplicability of the adopted control. Anyway, this would occur for extremely high values of the control gain. The combination of structural control executed by using acoustic measures, increases both the damping capacity of the structural system and the acoustical one, synchronizing the structural movement and acoustic wave.

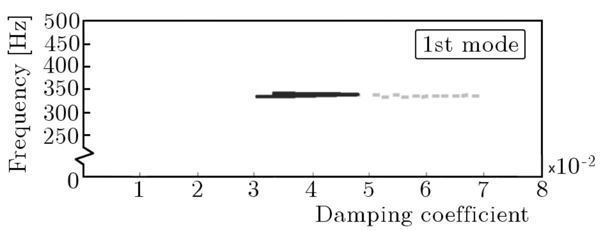

Fig. 17. FBC shaker/microphones, 1st mode. Damping coefficient and natural frequencies vs. gain factor; dashed curve – with foam, solid curve – without foam

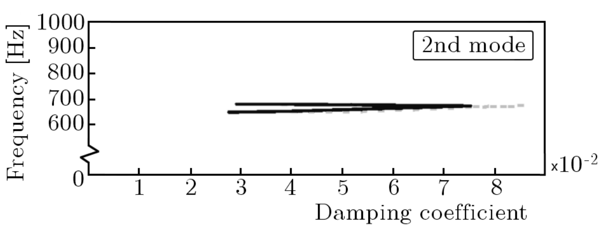

Fig. 18. FBC shaker/microphones, 2nd mode. Damping coefficient and natural frequencies vs. gain factor; dashed curve – with foam, solid curve – without foam

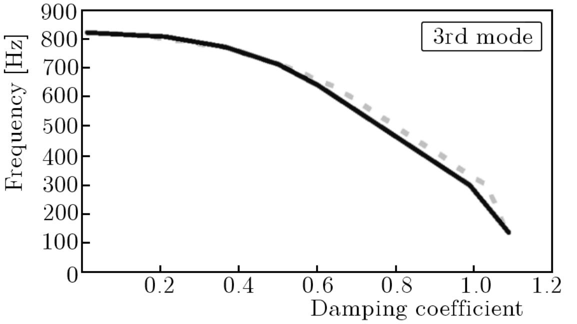

Fig. 19. FBC shaker/microphones, 3rd mode. Damping coefficient and natural frequencies vs. gain factor; dashed curve – with foam, solid curve – without foam

7. Conclusions

Feedback control strategies have been implemented to increase damping capacity of a structural-acoustical system represented by an acoustic duct. The application was particularly aimed to investigate the possibility of using a numerical model which contemplates the use of an active system together with insertion of a porous material. The investigation was performed using solutions already existing in the MSC/Nastran reference code.

The reference system is an acoustic duct, squared in the cross section. The length of the duct is sufficiently large compared to its cross section dimensions, so that the acoustic waves travel along the axis of the duct with planar wave fronts. This assumption enables us to treat the duct as a one-dimensional system.

Performances of three different kinds of the control system have been estimated: shaker accelerometer, piezo-accelerometer and shaker-microphones. Two configurations were referred to structural systems while the last one combined a structural actuator with a pair of microphones. All configurations envisaged a porous material coupled with a vibrating panel to explore active-passive combined solutions.

The results obtained show a large increase of the damping coefficient. The addition of the porous material implies a forward shift of the curves along the x axis.

In this paper, only control strategies using velocity, acceleration and pressure gradient as the input signals were investigated. Sensitivity of the feedback loop to phase changes linked to delays or structural and environmental modifications should be also considered in view of a real application. This step is being considered in the next stage of the study aimed at implementing the suggested control architecture onto a DSP system for early experimentation on a real lab test article.

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Pasywno-aktywne metody kontroli hałasu w przewodzie akustycznym

Streszczenie

W ostatnich latach problem sterowania poziomem hałasu stał się szczególnie znaczący w projektowaniu na potrzeby przemysłu samochodowego i lotniczego. Jako ogólną zasadę można przyjąć, że pasywne metody redukcji hałasu są efektywne dla średnich i wysokich częstości dźwięku, podczas gdy metody aktywne dobrze się

sprawdzają dla częstości niskich. Rozwiązanie oparte na kombinacji tych metod stanowi naturalny i najbardziej skuteczny sposób ograniczania hałasu w całym przedziale częstości. W pracy przedstawiono opis strategii sterowania poziomem hałasu bazującym na strukturalnym i akustycznym sprzężeniu zwrotnym, celującym w maksymalizację tłumienia wewnątrz przewodu akustycznego, na którego „aktywną" powierzchnię naniesiono warstwę pasywnej pianki.

Manuscript received July 1, 2010; accepted for print Feruary 2, 2011