JOURNAL OF THEORETICAL
AND APPLIED MECHANICS
49, 2, pp. 313-326, Warsaw 2011

EXPERIMENTAL CHARACTERIZATION OF HYBRID NOISE CONTROL SYSTEME ACTING IN ACOUSTIC DUCT

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In this investigation, a sound absorbing material is coupled with an active surface to reduce sound pressure within an acoustic duct. In this way, an enhancement of the passive foam performance can be observed because of further acceleration of the fluid particles within the foam pores.
The experimental activity implements noise control strategies to increase damping within the cavity through acoustic foam with a flexible backing plate, acted by piezoelectric actuators.

Key words: sound absorbing material, acoustic duct, foam, active surface, feedback control

1. Introduction

There are two fundamental approaches to control acoustic noise: passive and active. The traditional approach to acoustic noise control uses passive techniques to attenuate undesired noise. These passive treatments are valued for their high attenuation over a broad frequency range. However, they are relatively large, heavy, costly, and ineffective at low frequencies, making the passive approach to noise reduction often impractical. In an effort to overcome these problems, considerable interest has been shown in active noise control. The active noise control system contains an electroacoustic device that cancels the unwanted sound by generating an antinoise of equal amplitude and opposite phase. The successful application of active control is determined on the basis of its effectiveness compared with passive attenuation techniques. Active attenuation is an attractive way to achieve large amounts of noise reduction, particularly at low frequencies, where lower sampling rates are adequate, active control offers real advantages. Active noise control permits significant improvements in noise control, often with potential benefits in size, weight, volume, and cost of the system.

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 active control systems. Gentry *et al.* (1997) introduced an active passive noise control device, consisting of cylindrically curved sections of polyvinyldifluoride material (PVDF) embedded in a partially reticulated polyurethane acoustic foam. 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) investigated an active system that included a fibreglass absorbing layer backed by an air cavity terminated with an active surface.

In the study at hand, a sound absorbing material is coupled with an active surface to reduce sound pressure within an acoustic duct. In this way, an enhancement of the passive foam performance can be observed because of further acceleration of the fluid particles within the foam pores.

2. Experimental setup

The experimental mock up was a cylindrical duct, closed at both edges (Fig. 1). The length of the duct ($L = 1500$ mm) was sufficiently large compared to its cross section dimension ($D = 150$ mm) so that the acoustic waves travel along the axis of the duct with planar wave fronts. This assumption enabled us to treat the duct as a one-dimensional system.

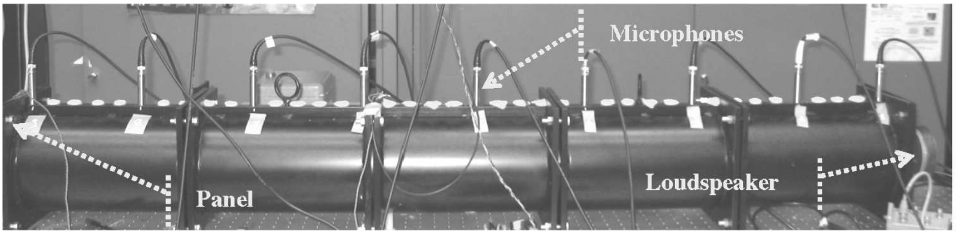

Fig. 1. Experimental mock up: acoustic duct

A loudspeaker was placed at the right end of the duct, while an exchangeable metallic panel was positioned at the left end. The duct, integrated a longitudinal narrow piece with 40 holes, to host microphones.

3. Acoustic cavity modal analysis

The experimental activity started with the evaluation of cavity normal modes in terms of natural frequencies and mode shapes. Initially, a steel rigid panel, 1 cm thick was pinned at the left side of the duct. A chirp signal (100-1500 Hz) was generated, amplified and given as input to the loudspeaker. As an acquisition and generation system, the LMS SCADAS System, integrated with the LMS Test.Lab software was used.

Frequency Response Functions (FRF) of microphone signals were evaluated in the frequency range of 100-1000 Hz. The experimental setup scheme was reported in the following figure (Fig. 2).

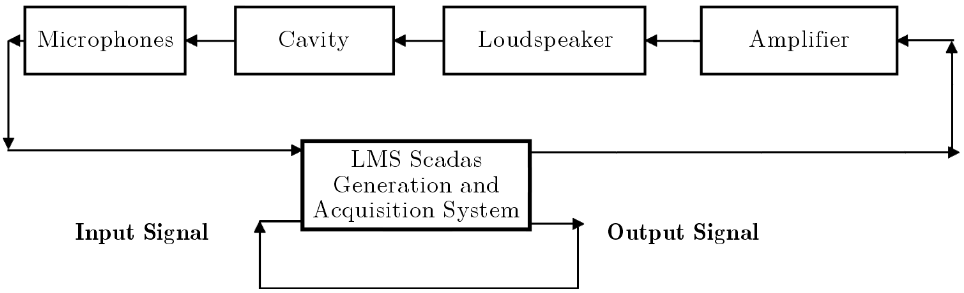

Fig. 2. Experimental setup scheme for normal mode analysis

By using LMS Test.Lab PolyMAX modal parameter identification algorithm, natural frequencies and mode shapes were evaluated (Fig. 3). Also the averaged pressure within the duct was calculated and plotted (Fig. 4).

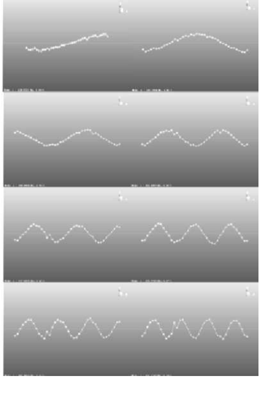

Fig. 3. Acoustic cavity: mode shapes

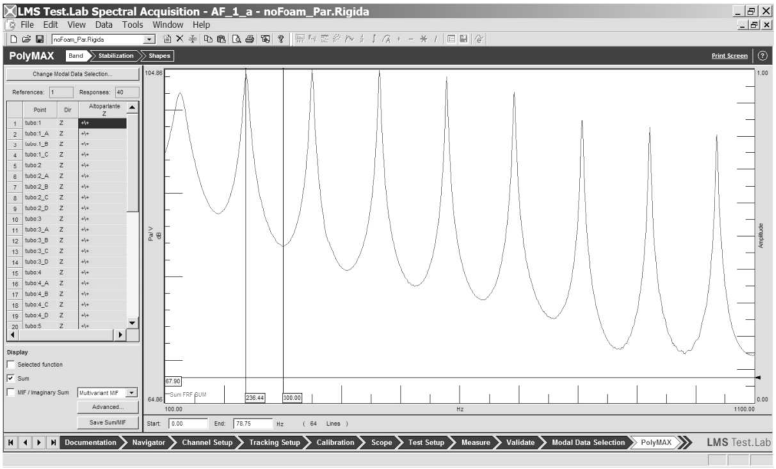

Fig. 4. Acoustic cavity averaged pressure versus frequency: naked panel

After that, a polyurethane foam was inserted and placed close to the rigid panel. The foamthickness was chosen to be 15 cm, being interested at the 0-500 Hz frequency range.

Also in this case, the microphone signals frequency response function, the averagedpressure within the cavity and the normal modes were evaluated.

The insertion of the passive foam implied a small shift of the natural frequencies (Table 1), no mode shapes modifications and a significant damping increase all over the frequency range of interest, even if this is more evident above the 300 Hz frequency (Fig. 5).

Table 1. Acoustic cavity natural frequencies: naked panel versus panel with foam

Natural frequencies [Hz]	
Naked panel	Panel with foam
125	125
236	237
369	329
462	423
576	515
690	639
805	756
919	872

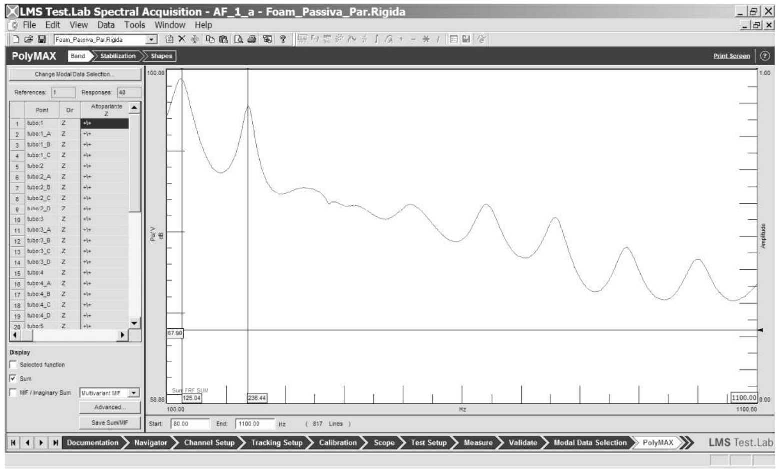

Fig. 5. Acoustic cavity averaged pressure versus frequency: panel with foam

Later, the rigid panel was replaced by a 1 mm thick steel flexible one. The same kinds of investigations performed earlier, were repeated.

In the case of flexible panel, the system response is superimposed to the one related to the rigid plate, except for the 300-400 Hz frequency range. Here, two peaks show the coupling of fluid and structure (Fig. 6). Furthermore, the system response is strictly dependent on panel-tube connection conditions, which are realised by clamping the panel between two rigid frames throughout four corner bolts (Fig. 7).

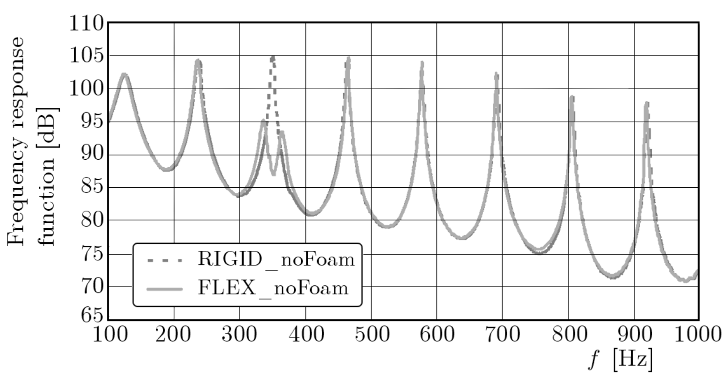

Fig. 6. Averaged FRF in the case of naked panel. Rigid versus flexible panel

The insertion of foam implies a considerable damping augmentation in the low frequency range. Very high damping values evidence a shift of natural frequencies in the middle and high frequency range (Fig. 8).

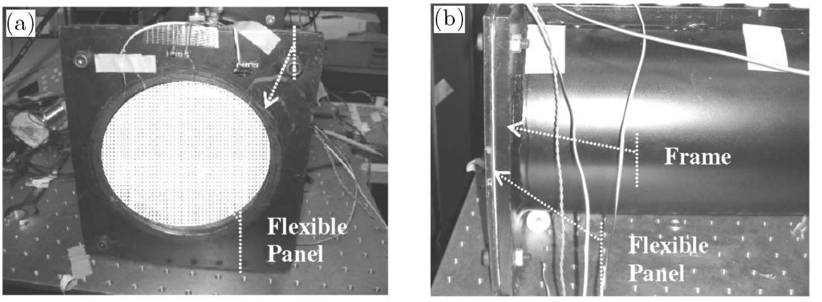

Fig. 7. Panel-duct connection: (a) frontal view, (b) lateral view

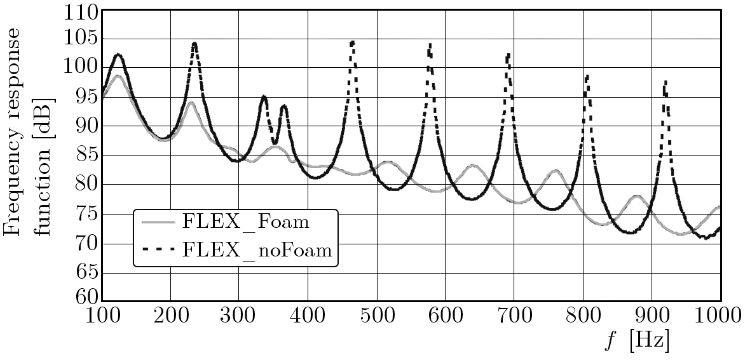

Fig. 8. Averaged FRF media in case of flexible panel: naked panel versus panel with foam

4. Collocated feedback structural control

The first control strategy consisted in improving passive foam performance, coupling it with the flexible panel which was "acted" by using a piezoelectric actuator.

A piezoelectric patch and a physically collocated accelerometer sensor identify a typical structural collocated feedback control strategy.

The panel vibration was measured by the accelerometer whose signal was derived, amplified and fed back to the actuator. The loudspeaker driven by a signal generator was used to excite the system at resonance frequencies. By using a control card, the optimal gain value K (defined as the one which implied the higher damping coefficient and consequently the maximum attenuation of peaks) was evaluated, observing the accelerometer signal on a oscilloscope.

Having identified the "optimal K", the experimental activity continued with a broadband acoustic excitation and the microphone signals acquisition. Two cases were considered: single and dual channels control, implying one and two actuator-sensor pairs, Fig. 9-13.

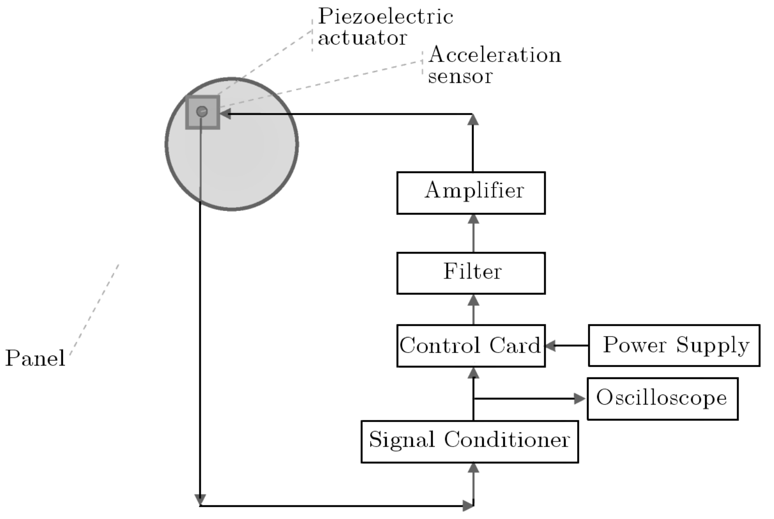

Fig. 9. Single channel collocated feedback structural control: scheme

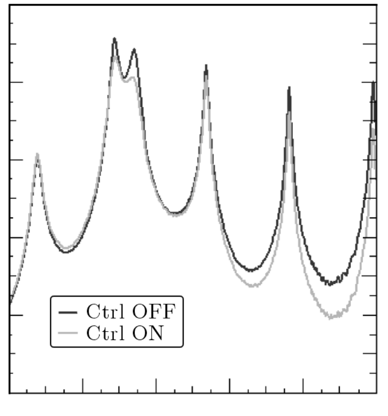

Fig. 10. Single channel collocated feedback structural control – acceleration signal naked panel – Ctrl OFF vs ON

The single channel feedback control strategy affected the system response only around the mode mainly influenced by the structural part of the system (Figs. 10-12).

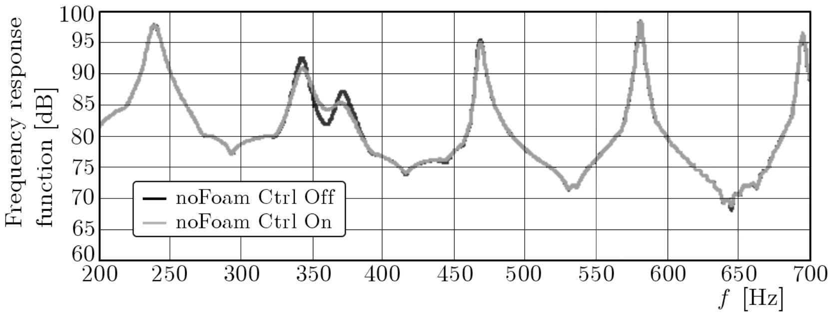

Fig. 11. Single channel collocated feedback structural control – averaged FRF – naked panel

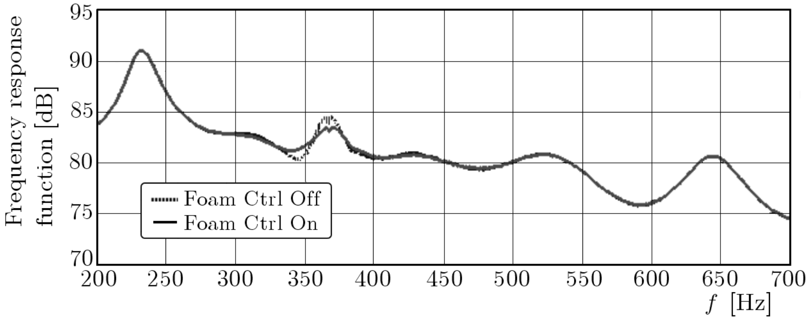

Fig. 12. Single channel collocated feedback structural control – averaged FRF – panel with foam

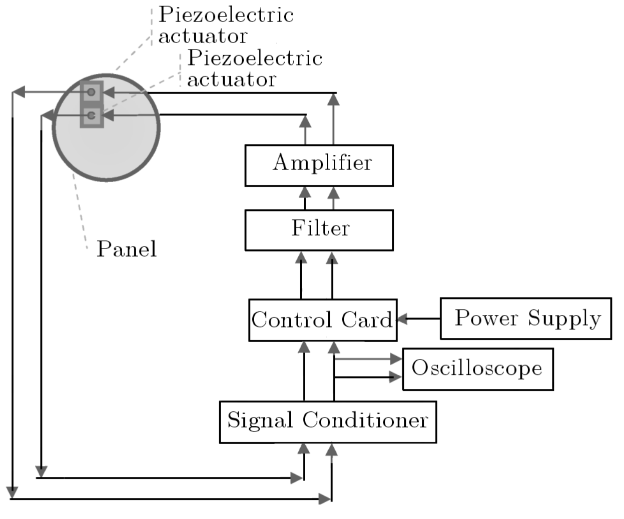

Fig. 13. Dual channel collocated feedback structural control: scheme

In the second configuration characterized by two pairs of collocated actuators and sensors, the piezoelectric actuators were located so that the maximum numbers of structural modes within the frequency range of our interest were excited.

An attenuation of 4 dB on the accelerometers signals within the whole frequency range was observed, at the optimal gain value (Figs. 14 and 15).

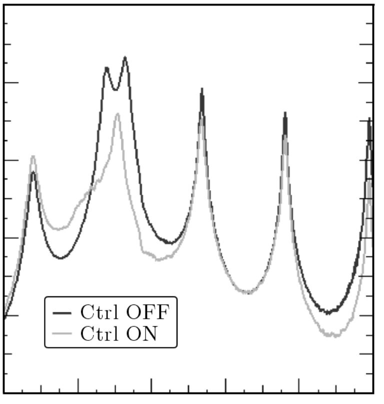

Fig. 14. Dual channel collocated feedback structural control – acceleration signal 1: naked panel – Ctrl OFF vs ON

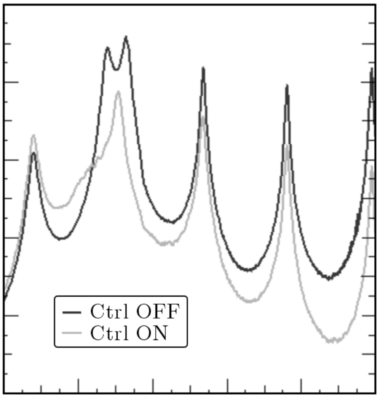

Fig. 15. Dual channel collocated feedback structural control – accelerometer signal 2: naked panel – Ctrl OFF vs ON

The adopted control strategy, in the case of naked panel, also affected the system acoustic response, above 300 Hz frequency (Fig. 16). When the panel was covered by foam, the influence of control was evident only around 369 Hz mode (Fig. 17).

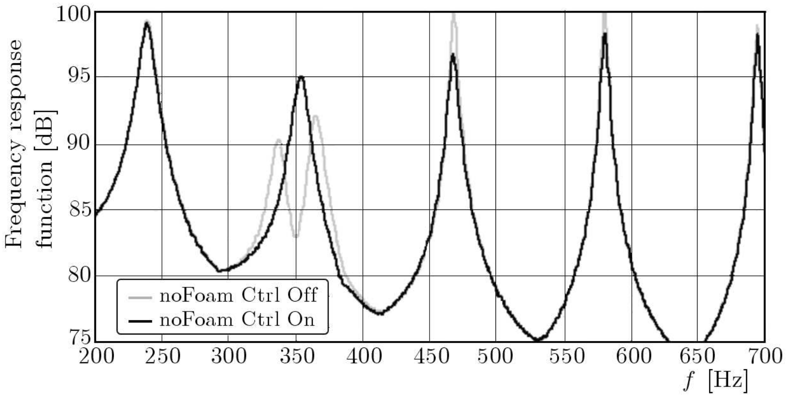

Fig. 16. Dual channel collocated feedback structural control – averaged FRF – naked panel

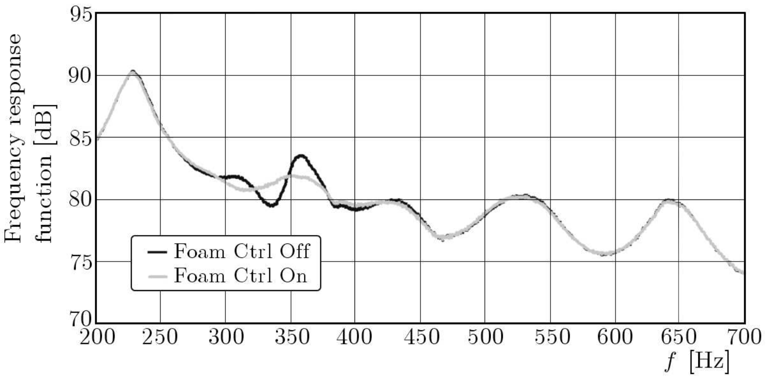

Fig. 17. Dual channel collocated feedback structural control – averaged FRF – panel with foam

5. PVDF control

In this application, polyvinyldifluoride (PVDF) material was used as the actuator because of its specific features.

PVDFs are light weight, very thin, very flexible and readily shaped in complex shapes to cover irregular structures. They can also be easily integrated with existing structures, where classical point actuators like shakers or loudspeakers can not be used.

The main disadvantage is that its piezoelectric constants are two orders of magnitude lower than those of PZT. Additionally, high driving voltage and limited temperature range have to be mentioned as well.

An active foam was realised by incorporating a 0.28 μm thick PVDF film within a foam. The film was opportunely curved to couple plane deformation due to the piezoelectric effect with vertical motion that is needed to accelerate fluid particles and, hence, to increase damping capacity.

A PVDF film can be driven by a maximum input of 500 V. Two uncorrelated signals were given as the input to the PVDF film and loudspeaker. Two opposite trends occurred: high input voltage, probably increased damping capacity of the foam (with noise reduction, as a consequence). At the same time, the PVDF worked as an additional noise source which increased the noise within the cavity, cancelling the first tendency.

To avoid it, a high frequency excitation outside the range of hearing (> 20 KHz) was given as the input to the piezo film. It resulted in small vibration amplitudes, incapable to accelerate adequately the fluid particles within the pores, so that to increase the dissipation energy.

6. Feedback structural acoustic control

The control strategy consisted in capturing the pressure within the tube by using two adjacent microphones and located close to the piezoelectric patches, Fig. 18.

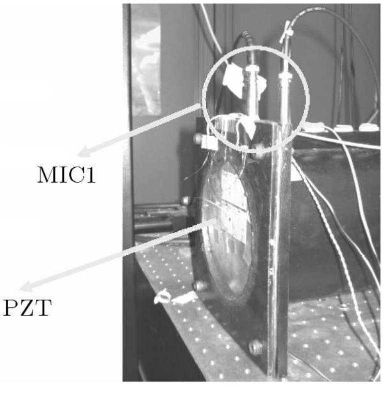

Fig. 18. Structural acoustic feedback control

The pressure signals were subtracted, derived and fedback to the piezo actuators Fig. 19. Strictly speaking, in that case, it was improper to speak of a "collocated" feedback control, since the two microphones were positioned at the plate, but not to the same height of the piezoelectric.

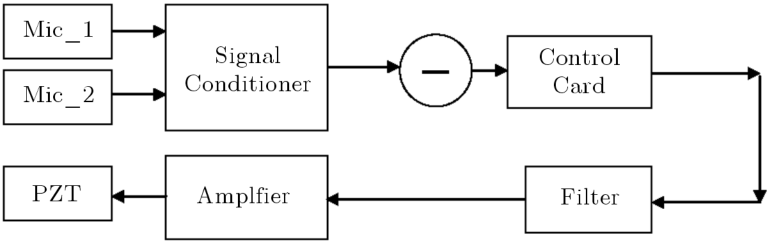

Fig. 19. Structural acoustic feedback control: scheme

The results obtained show an attenuation of about 3 dB, within the range of 0-500 Hz, in the case of naked panel, Fig. 20. In the case of foam, the adopted control strategy effected only the 369 Hz mode, Fig. 21.

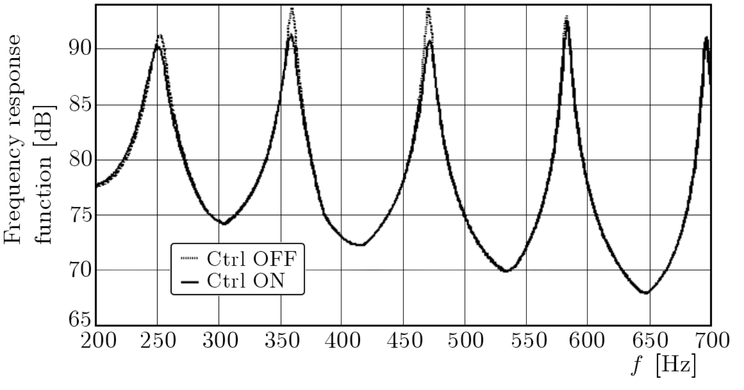

Fig. 20. Structural acoustic feedback control – averaged FRF – naked panel

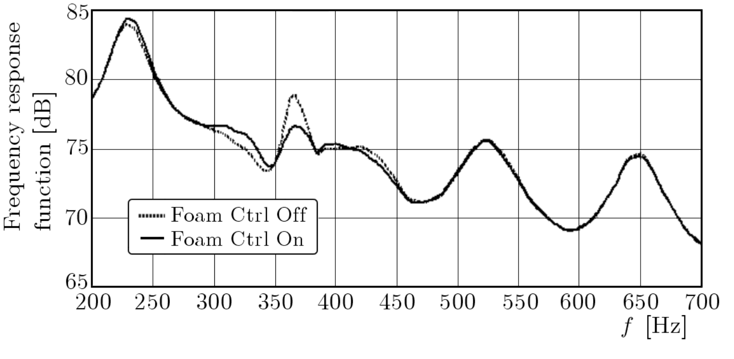

Fig. 21. Structural acoustic feedback control – averaged FRF – panel with foam

7. Collocated feedback structural acoustic control

The last experimental configuration, a collocated structural-acoustic control, was implemented. A collocated actuator sensor pair was used. It was realised by inserting a microphone within the foam Figs. 22-24. Compared to the nocollocated case, no changes were observed.

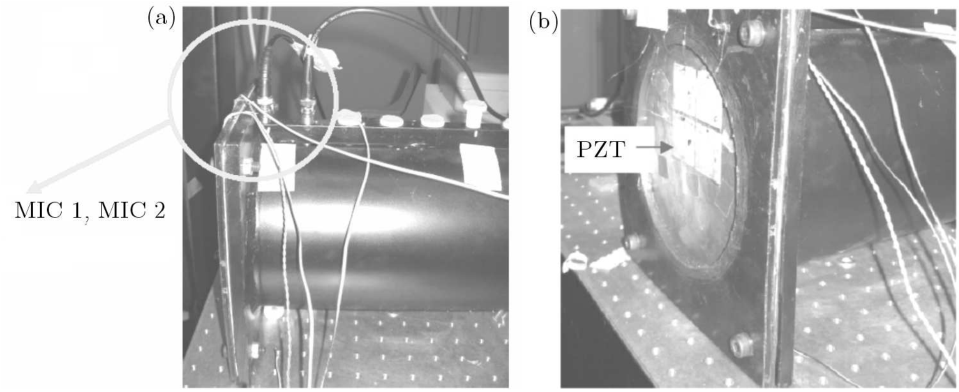

Fig. 22. Collocated feedback control: (a) microphones, (b) piezoelectric actuators

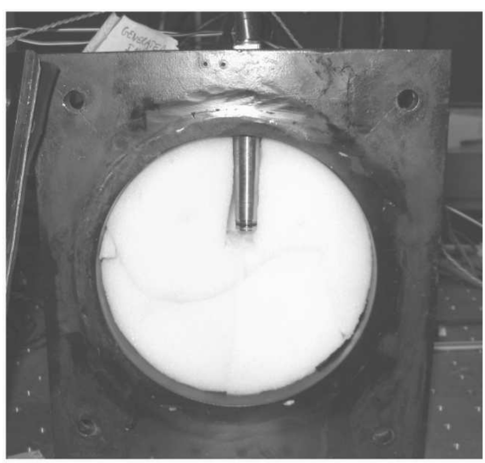

Fig. 23. Collocated feedback control: microphones within foam

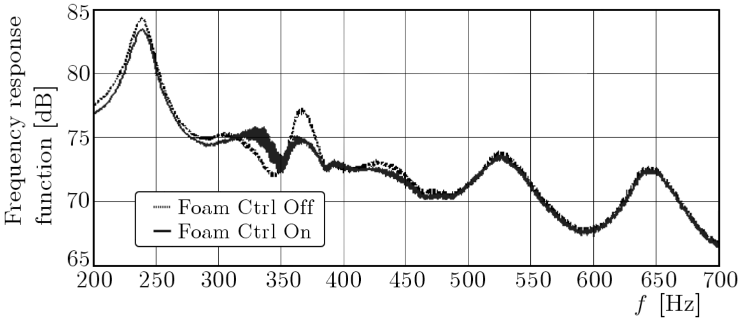

Fig. 24. Structural acoustic feedback control – averaged FRF – panel with foam

References

1. Gentry C.A., Guigou C., Fuller C.R., 1997, Smart foam for applications in passive active noise radiation control, *Journal of Acoustical Society of America*, **101**, 4
2. Nabil W., 2004, *Smart Foam for Active Vibration and Noise Control*, dissertation Thesis, Department of Mechanical Engineering, University of Maryland
3. Thenail D., Galland M., Sunyach M., Sunhack M., 1994, Active enhancement of the absorbent properties of a porous material, *Smart Mater. Struct.*, **3**, 18-25

Eksperymentalna charakterystyka hybrydowego układu redukcji hałasu działającego w akustycznym dukcie

Streszczenie

W pracy zbadano skuteczność zastosowania materiału absorbującego dźwięk współpracującego z aktywną powierzchnią przeznaczoną do redukcji ciśnienia akustycznego w dukcie. Zaobserwowano, że takie skojarzenie znacznie poprawia pasywne właściwości samej dźwiękochłonnej pianki wskutek zwiększenia przyspieszeń cząstek płynu zawartego w porach pianki. W wyniku badań doświadczalnych uzyskano strategie kontroli hałasu zwiększające tłumienie poprzez użycie akustycznej pianki na elastycznej płycie domykającej dukt i sterowanej elementami piezoelektrycznymi.

Manuscript received July 30, 2010; accepted for print September 22, 2010