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Smart panel with multiple decentralized units for the control of sound transmission. Part III: control system implementation

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Abstract

This is the third of three companion papers that summarize the theoretical and experimental work carried out to develop a prototype smart panel with 16 decentralized vibration control units for the reduction of sound transmission/radiation. In this paper the control effectiveness of the smart panel has been assessed experimentally by measuring, in a anechoic chamber, the reduction of its total sound power radiation when it is mounted on a Perspex box with very thick and rigid side walls. The panel has been excited either by the acoustic field produced by a loudspeaker placed in the Perspex box or directly by a point force generated with a shaker. The vibration averaged over the panel surface has also been measured with a laser vibrometer. Laser measurements have also been taken at some specific frequencies, in such a way as to highlight the spatial vibration of the panel with the 16 decentralized control units.

The various tests carried out have shown that the 16 control units can produce good reductions of the radiated sound power and averaged vibratory field over the panel surface, particularly when the panel is excited by the shaker. Indeed, when the panel is excited by the shaker, reductions of the sound radiation were measured in all the third octave bands in the frequency range 0–5 kHz and it was found that there were at least seven bands with reductions between 5 and 8 dB. Also, the averaged vibration field over the panel surface was found to be completely damped down in a frequency range between 0 and 1 kHz with reductions of about 10 dB in correspondence to all resonance frequencies.

1. Introduction

This is the third of three companion papers that summarize the theoretical and experimental work carried out to develop a prototype smart panel with 16 decentralized vibration control units for the reduction of sound radiation/transmission. The system studied consists of a thin aluminium

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panel of dimensions $414 \times 314\,\mathrm{mm}^2$ and thickness 1 mm with an embedded array of 4×4 square piezoceramic actuators. The sensing system consists equally of an array of 4×4 accelerometers that are arranged in such a way as to match the centre positions of the 16 piezoceramic patches. Each of the 16 sensor–actuator pairs is set to implement decentralized velocity feedback control, i.e., active damping. The theoretical study in Part I [1] has shown that, with this arrangement, both the kinetic energy of the panel and its transmitted/radiated sound power can be significantly reduced in the bandwidth up to 2 kHz provided an appropriate feedback gain is chosen. Also, in Part II [2], the design of 16 decentralized velocity feedback control units which implement active damping has been completed and has shown good reductions of the vibration at the sensor positions.

In this paper the control effectiveness of the smart panel is assessed experimentally by evaluating both the overall reduction of the vibration level of the panel and the reduction of the sound power radiation by the panel. The smart panel has been clamped to a rigid frame which is fixed on the top open side of a box made with 3 cm thick plates of Plexiglas. The panel is then excited either by the acoustic field generated in the Plexiglas box by a loudspeaker or directly by a shaker. With such thick side walls the sound transmission through the smart panel is about 10–20 dB higher than that through the side walls themselves and so the effective sound transmission/radiation by the smart panel is measured.

The following three sections describe the three fundamental elements that have been used in this experimental study: the smart panel, the testing facility and the multiple-channel decentralized feedback controller. Finally, in Sections 5 and 6 are presented the experimental results of a series of active control tests with reference to either the sound radiation or vibratory response by the panel when either a primary loudspeaker is exciting the acoustic cavity underneath the panel or a primary shaker is directly exciting the panel.

2. Smart panel

The prototype smart panel built for this study consists of an aluminium panel of thickness $h_s = 1\,\mathrm{mm}$, which has been clamped on a rigid frame so that the "vibrating area" is $l_x \times l_y = 414 \times 314\,\mathrm{mm}^2$. The panel is equipped with an array of 16 decentralized sensor–controller–actuator systems. As shown in Figs. 1 and 2a, a 4×4 array of piezoceramic actuators, of dimensions $a_x \times a_y \times h_p = 25 \times 25 \times 1\,\mathrm{mm}^3$, have been evenly bonded on one side of the panel so that the distances between the centres of two adjacent patches or between the centre of a side patch and the edge of the panel is $d_x = 82.8\,\mathrm{mm}$ and $d_y = 62.8\,\mathrm{mm}$.

As can be seen in Figs. 1 and 2b, on the other side of the panel a 4×4 array of high sensitivity ICP accelerometers (PCB Piezotronics, model 352C67) has been arranged in correspondence to the centres of the 16 piezoceramic patches. Each sensor–actuator pair is connected via a single channel analogue feedback controller. The controller is set to feedback a control signal proportional to the transverse velocity at the centre position of the actuator detected by the accelerometer.

3. Testing facility

In order to assess the performance of the control system in terms of attenuation of the sound power transmission/radiation, the panel has been mounted on a Perspex box, which, as shown in

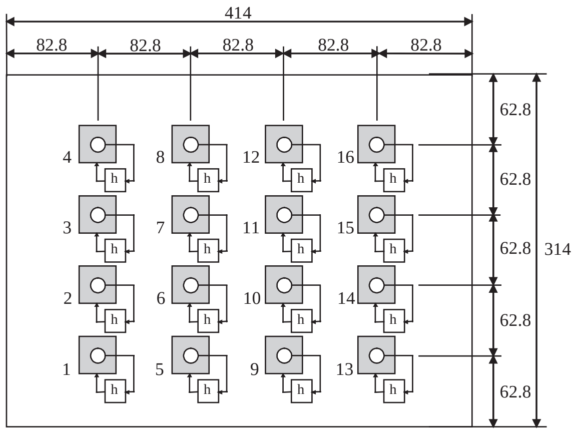

Fig. 1. Panel with 16 piezoelectric actuators, as shown by the squares, driven locally by the output of 16 velocity sensors, as shown by the circles, via individual control loops with a gain of h (dimensions are in mm).

Fig. 2. Panel with 16 piezoceramic actuators (a) and 16 collocated accelerometers (b). Each sensor actuator pair is driven by a decentralized analogue feedback controller. The panel is mounted on a Perspex box with a loudspeaker inside which generates the primary disturbance (b).

Fig. 2b, has a loudspeaker inside that generates the primary disturbance. As shown in Fig. 3, a pair of rigid aluminium frames has been used to clamp the smart panel on the open side of the box. Both frames have a width of w=30 mm, but they have different thicknesses: 25 mm for the bottom frame and 10 mm for the top one. The dimensions of the plate used to build the smart panel have been chosen to match the width and length of the clamping frame so that $l_{xs} \times l_{ys} \times h_s = 444 \times 344 \times 1\,\text{mm}^3$.

The box is made with relatively thick plates of Perspex so that the acoustic field generated by the loudspeaker inside it is essentially transmitted through the smart panel. The thickness of the Perspex plates have been chosen to be 30 mm so that, below 5 kHz the sound transmission through the smart panel is about 10–20 dB higher than the flanking component radiated by the

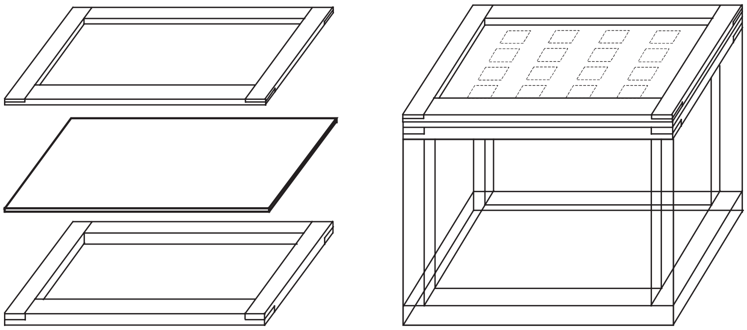

Fig. 3. Design of the rigid frames using for mounting the smart panel on the open side of the Perspex box.

Perspex walls. With this arrangement it is therefore possible to evaluate the reduction of sound transmission/radiation through the smart panel even when the control system is working and therefore attenuates the passive sound transmission by some 5–15 dB. As can be noticed in Fig. 2b, a set of electric plugs has been applied to one wall of the box to help the wiring between the 16 pair of sensor/actuator and the electrical equipment (power amplifier, phase-lag compensator and integrator).

It is important to emphasize that the choice of testing the smart panel when mounted on a cavity is for practical reasons. Indeed this configuration has enabled an easy estimate of the sound radiation/transmission by the panel in an anechoic chamber even though it has introduced some practical problems that have either affected the performance of the smart panel or have corrupted the measurement at some frequency bands as will be shown in Sections 5 and 6.

4. Multiple-channel decentralized controller

Using the theoretical and experimental work carried out to design one control unit, which is presented in Part II [2], sixteen independent single channel control units have been built for the smart panel. Each unit consists of an integrator circuit to transform the output of the accelerometer sensor to velocity, a power amplifier and a phase lag compensator circuit. The preliminary study presented in Part II [2] provided key information with regards to the electric characteristics of these three electric components, particularly for the design of the integrator and of the amplifier elements.

The analogue control systems built for each decentralized control unit are composed by a cascade of six electrical elements: an integrator, the phase-lag compensator previously described, a low-power pre-amplifier for the velocity test point, a low-power amplifier for adjusting the gain level, a switching circuit driven by a manual switch to select the output (inverted, non-inverted, no output) and an ILP HY2001 power amplifier to drive the piezoelectric actuator [3]. Moreover, the controller includes an operational amplifier driven by a manual switch to enable/disable all the control signals at the same time. Such a controller provides a maximum voltage signal of 30 V peak to peak for an output power of 40 W RMS. All the 16 channels have been assembled in the single device shown in Fig. 4 below. The complete testing system is shown in Fig. 5 where on the

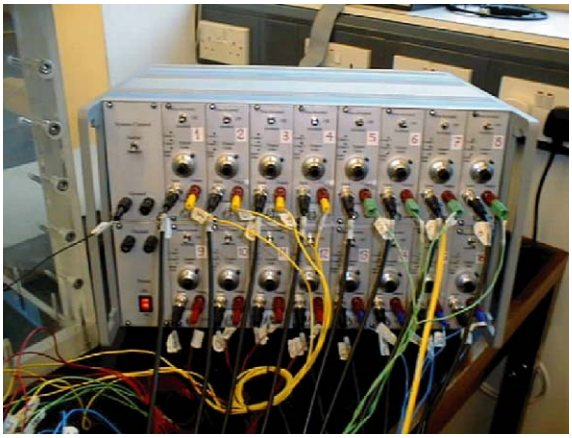

Fig. 4. The 16 channels decentralized feedback control system.

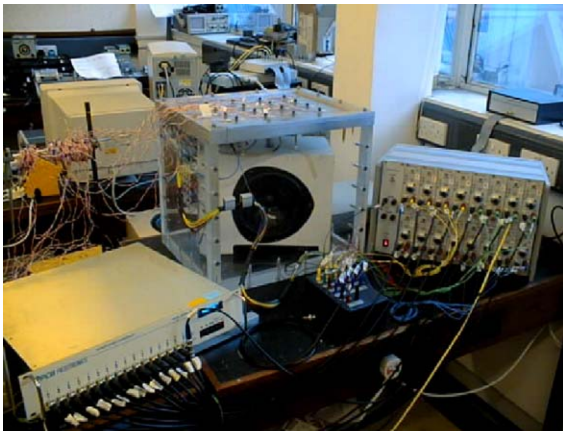

Fig. 5. Complete experimental set-up with the Perspex box with on top the smart panel and the control equipment: signal conditioner (left), controller (right).

centre can be noticed the Perspex box with the loudspeaker primary source inside and on top the smart panel with the 16 piezoactuators and accelerometer sensors. On the left side is visible the 16 channel PCB signal conditioner while on the right-hand side is shown the analogue 16 channels decentralized feedback control units system. The study carried out in Part II [2] has shown that the 16 decentralized control units are conditionally stable, nevertheless the control gain that guarantees the stability of a single working control unit could be used in all 16 control systems with guaranteed overall stability.

5. Active control of sound radiation tests in the anechoic chamber

This section presents the experiments carried out in an anechoic chamber to test the smart panel active control of sound radiation effectiveness. As shown in Fig. 6, the panel has been mounted in

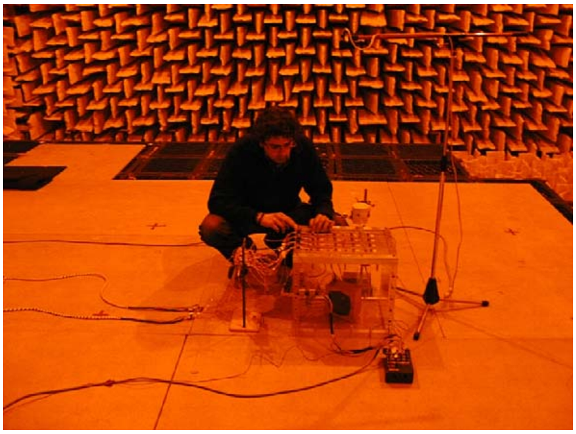

Fig. 6. Testing set-up in the anechoic chamber for the measurement of the sound power radiated by the smart panel when excited by the loudspeaker in the Plexiglas box or by the shaker mounted on the panel.

the testing facility described in Section 3 and has been excited either by an acoustic disturbance provided by the loudspeaker positioned inside the Perspex box or by a structural disturbance provided by a shaker mounted on the panel. Both are random, white and pink noise, excitations in a frequency band 0 to 5 kHz. The testing facility has been placed on a wooden floor made with a set of panels in order to get the sound radiation effect of a baffled panel. The 16 channels controller illustrated in Section 4 has been used with 16 equal feedback control gains that have been adjusted in such a way as to guarantee stability for the given primary disturbances.

In order to estimate the total radiated sound power from the panel with and without control, the sound pressure level has been measured in nine positions around the box, according to the standard procedure described by the ISO 3744, for three different frequency ranges (0–1, 0–2 and 0–5 kHz). The measurements have been analyzed either in terms of narrow band frequency response functions between the averaged sound pressure measured by the nine microphones and the excitation of the primary source (loudspeaker or shaker) in the frequency ranges 0–1 and 0–2 kHz or in terms of the estimated sound power radiation plotted in third octave bands for a frequency range of 0–5 kHz (2 decades with centre band frequency from 63 to 4000 Hz).

5.1. Acoustic primary source in the cavity

Fig. 7 shows the frequency response function between the averaged sound pressure measured by the nine microphones and the excitation of the primary loudspeaker source in the two frequency ranges 0–1 (left-hand side plot) and 0–2 kHz (right-hand side plot). The solid line represents the averaged sound pressure when there is no control while the faint line gives the averaged sound pressure when the 16 control systems are turned on. The dashed line represents the averaged sound pressure due to the flanking sound radiation through the side walls of the Perspex box. This result has been obtained by replacing the smart panel with a very rigid block of metal, about 4 cm

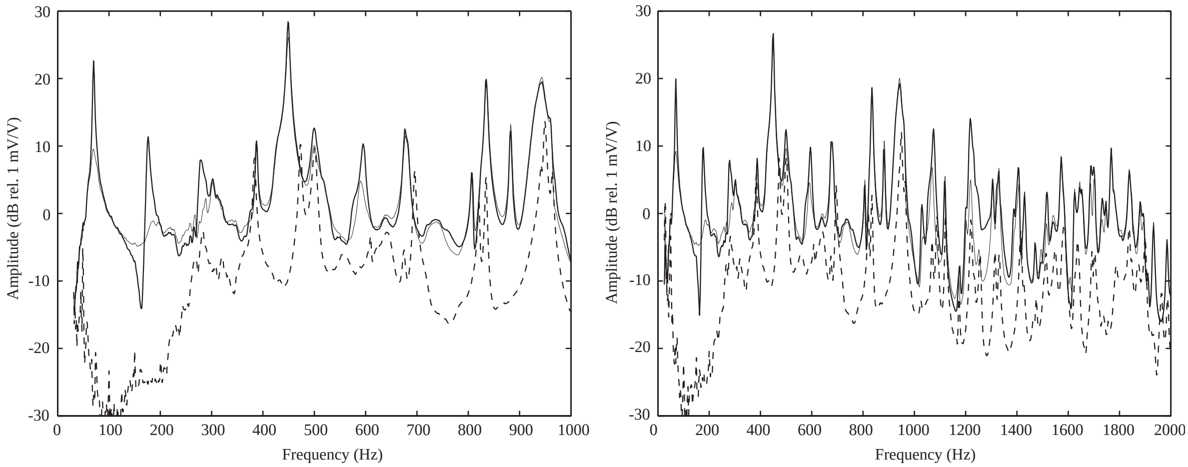

Fig. 7. Average of the frequency response functions between the sound pressure measured by the nine microphones and the excitation of the primary loudspeaker source in the frequency ranges 0–1 (left) and 0–2 kHz (right). Averaged sound pressure without (solid line) and with control (faint line). The averaged sound pressure due to the flanking sound radiation through the side walls of the Perspex box (dashed line).

thick, whose low frequency sound radiation is negligible. In general the flanking sound radiation is about 10–20 dB lower than that of the panel except at few resonance frequencies due to the cavity natural modes. This measurement validates the control results obtained at all frequencies except in correspondence to those resonance frequencies where the resonance effect in the rectangular cavity gives rise to an equal level of sound radiation through the smart panel and the side walls of the Perspex box in which case the reduction of sound transmission through the smart panel cannot be correctly estimated.

The left-hand side plot in Fig. 7 shows that the first three resonance frequencies are well controlled with reductions of the sound level that goes from a minimum of about 8 dB for the third mode, to a maximum of about 13 dB for the other two modes. The 16 control units provide the control strength to damp the first three resonant modes of the panel which, as highlighted in the theoretical study in Part I [1], are strongly coupled to the volumetric excitation of the cavity underneath the panel. Between 350 and 1200 Hz there is little control effect except for the resonance at about 600 Hz. In particular, as anticipated with the theoretical study in Part I [1], the resonance frequency controlled by the first few cavity modes with natural frequencies at 445, 546, 608, 685 Hz cannot be controlled. Moreover, the flanking radiation of sound through the walls of the box affects the performance of the control system in correspondence to the three resonance frequencies at 385, 470 and 500 Hz. However, as shown in the right-hand side plot in Fig. 7, once more in agreement with the theoretical predictions of Part I [1], between 1200 and 1400 Hz reductions of about 10 dB of the measured sound level are registered. No further reductions of the radiated sound at higher frequencies are shown. Very few control spillover effects have been found and in most cases they correspond to antiresonance frequencies as one would expect with active damping.

Fig. 8 shows a bar plot with the total radiated sound power in third octave bands between 0 and 5 kHz. The white and grey columns represent, respectively, the sound power radiated with and

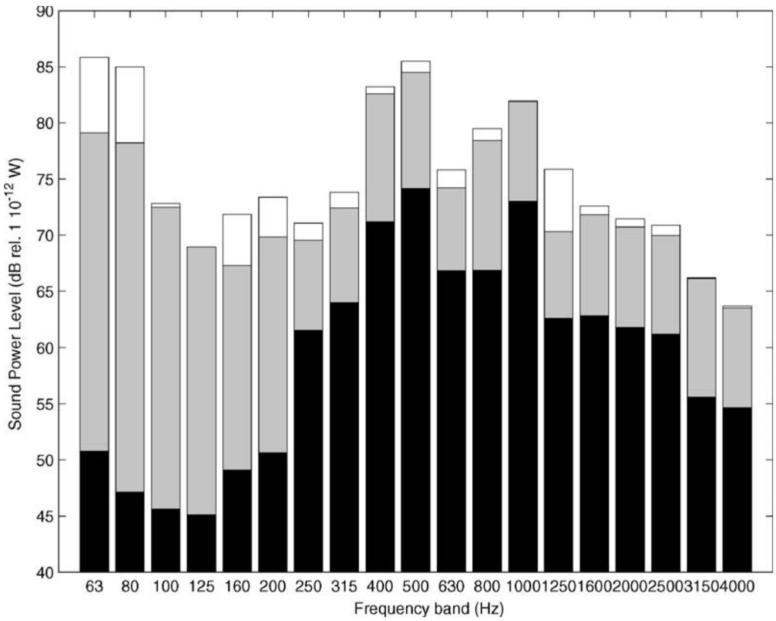

Fig. 8. Total radiated sound power in third octave bands between 0 and 5 kHz per unit primary excitation of the loudspeaker. Total sound power radiated without (white column) and with (grey column) control system. Total sound power radiated due to the flanking sound radiation through the side walls of the Perspex box (black column).

without control system turned on per unit loudspeaker excitation in the cavity. The black column represents the total radiated sound power through the side walls of the Perspex box per unit loudspeaker excitation in the cavity. Frequency averaged reductions of about 6 dB are found in correspondence to the third octave bands at 63, 80 and 1250 Hz. Smaller reductions are measured for the other third octave bands.

5.2. Structural primary source on the panel

Fig. 9 shows the frequency response function between the averaged sound pressure measured by the nine microphones and the excitation of the primary shaker source in the two frequency ranges 0–1 (left-hand side plot) and 0–2 kHz (right-hand side plot). The solid line represents the averaged sound pressure when there is no control while the faint line gives the averaged sound pressure when the 16 control systems are turned on.

Comparing the plots in Figs. 7 and 9 it can be noticed that the latter ones are characterized by a larger number of resonances at relatively low frequencies. This is because the shaker excites nearly all the structural modes of the panel while only the structural modes of the panel that are well coupled to the acoustic cavity underneath it are excited by the loudspeaker. As a result for example the sound radiation of the panel below 300 Hz is characterized either by three or five resonances depending whether the testing system is excited by the loudspeaker or by the shaker, respectively. In contrast at higher frequencies above 1 kHz it is the plot in Fig. 7 that is characterized by a relatively larger number of resonances. This is because above 400 Hz the

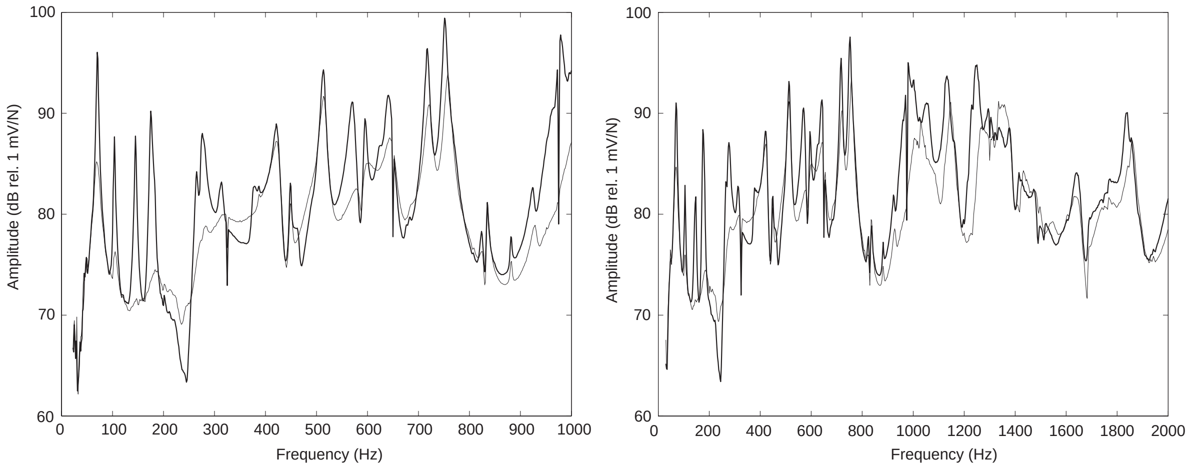

Fig. 9. Average of the frequency response functions between the sound pressure measured by the nine microphones and the excitation of the primary shaker source in the frequency ranges 0–1 (left) and 0–2 kHz (right). The averaged sound pressure without (solid line) and with 16 decentralized control units (faint line).

response of the cavity is characterized by modes whose number grows with the square of frequency. Thus, when the panel is excited by the acoustic field in the cavity, a large number of modes is found in the sound radiation which are indeed due to the cavity modes. Alternatively, when the panel is excited directly by a point force very little effects of the cavity resonances are seen in the radiated sound power.

The plot in Fig. 9 shows that the first five resonance frequencies are well controlled with reductions of the radiated sound power that, in this case, goes from a minimum of about 12 dB for the first and fifth modes to a maximum of about 18 dB for the other three modes. Significant control effects are also found in the frequency bands between 350 and 1300 Hz and between 1600 and 1800 Hz with reductions of the averaged sound pressure of the order of 3–8 dB. In this case the acoustic cavity underneath the panel produces only a passive loading effect which is not characterized by strong coupling effects between the cavity and panel natural modes and thus smaller control strength is necessary to produce the wanted active damping on the panel.

Also in these tests relatively few control spillover effects have been found. In most cases they occur at antiresonance frequencies except in the frequency band between 1300 and 1350 Hz where a relatively high enhancement of the averaged sound pressure, about 3 dB, has been measured.

Fig. 10 shows the bar plot with the total radiated sound power in third octave bands between 0 and 5 kHz. The white and grey columns represent the sound power radiated with and without control system turned on per unit loudspeaker excitation in the cavity. Comparing this plot with that in Fig. 8 it is evident that much larger reductions of the radiated sound power are measured for all frequency bands. In this case reductions of the sound radiation are measured in correspondence of all third octave bands and there are at least seven bands, at 63, 80, 125, 160, 250, 315, 400, 1000 Hz, with reductions of about 5–8 dB.

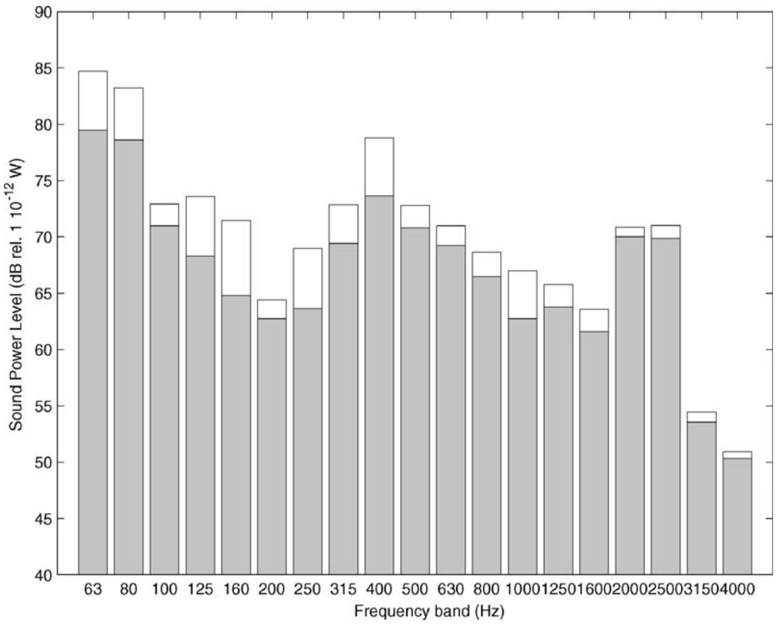

Fig. 10. Total radiated sound power in third octave bands between 0 and 5 kHz per unit primary excitation of the shaker. Total sound power radiated without (white column) and with (grey column) control system. Total sound power radiated through the side walls of the Perspex box (black column).

6. Active control of vibration tests using a laser vibrometer

This section presents the experiments carried out using a laser vibrometer in order to evaluate how much and in which way the vibration over the surface of the smart panel varies when the sixteen decentralized control units are turned on. As for the experiment in the anechoic chamber, the panel has been mounted on the testing facility described in Section 3 and has been excited either by an acoustic disturbance provided by the loudspeaker positioned inside the Perspex box or by a structural disturbance provided by a shaker mounted on the panel. In this case the excitation point of the shaker is moved quite close to one corner of the panel and the shaker is oriented at an angle with respect to the normal to the panel in order to enable the vibration measurement over the whole surface with the vibrometer. Both the loudspeaker and shaker excitations have been set to be white noise in a frequency band between 0 and 1 kHz. The laser vibrometer is equipped with a scanning system which has enabled the measurement of the transverse vibration of the panel over a grid of 133 points evenly distributed over the panel surface as shown in Fig. 11. In this way 133 frequency response functions have been measured which gives the transverse velocity of the panel per unit excitation of the shaker with and without the 16 control units turned on. The vibration averaged over the panel surface has therefore been plotted between 0 and 1 kHz. Also, the vibratory field over the panel surface has been plotted in correspondence to resonance frequencies where the control system produced an interesting variation of the vibratory field over the panel surface. The 16 channels controller illustrated in Section 4 has been used with 16 equal feedback control gains that have been adjusted in such a way as to guarantee stability for the given primary disturbances.

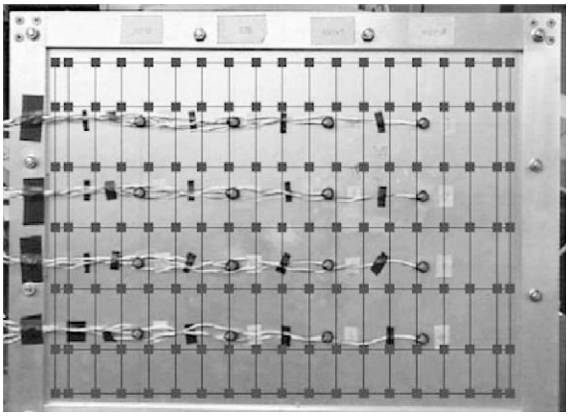

Fig. 11. Grid of 133 measurement points scanned with the laser vibrometer.

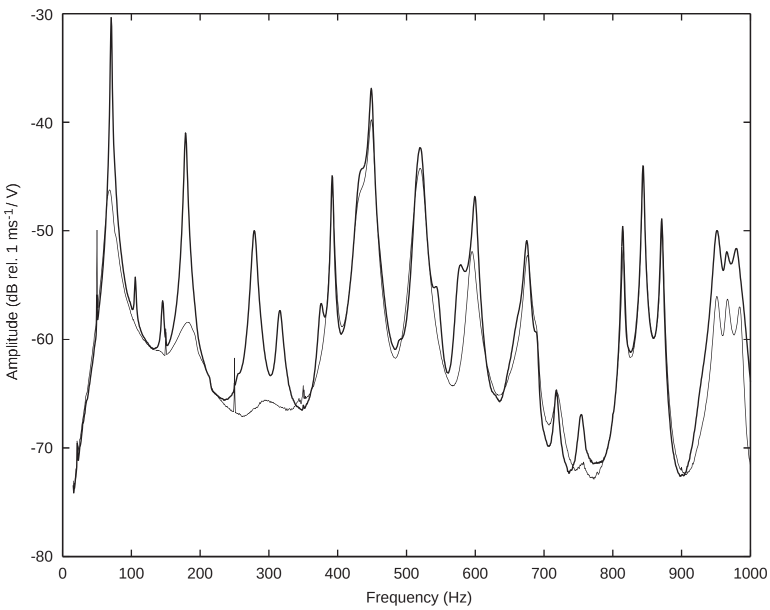

Fig. 12. Average frequency response function of the panel velocity (0–1 kHz) per unit excitation of the loudspeaker calculated as the average of all the 133 frequency response functions measured at the grid points: no control system (solid line) and with decentralized feedback control (faint line).

6.1. Acoustic primary source in the cavity

The average spectrum of the vibration level of the plate in the frequency range between 0 and 1 kHz when the smart panel is excited by the acoustic field in the cavity generated by the loudspeaker is shown in Fig. 12. The two curves in this plot have been calculated as the average of all the 133 frequency response functions measured by the laser vibrometer in correspondence of the points of the grid, considering again the response of the system before (solid line) and after (faint line) the control effort. It can be seen that the average attenuation factors of the first five resonances are, respectively, 16 dB (at 70 Hz), 4 dB (at 102 Hz), 18 dB (at 178 Hz), 15 dB (at 280 Hz) and 10 dB (at 326 Hz). Moreover, there are further attenuations of about 5 dB in the narrow bands around 600 and 950 Hz. The overall attenuation factor in the frequency range of 0 to 1 kHz is 3.6 dB. No reductions of the vibration of the panel are found corresponding to the resonances controlled by the cavity response and in correspondence of resonances where the response of the panel cannot be well controlled as will be discussed below for the resonance frequency at 448 Hz.

Fig. 13 shows the vibratory field of the panel without control (left-hand side pictures) and with control (right-hand side pictures) when it is excited by the acoustic field generated by the loudspeaker in the cavity underneath of the panel at the frequencies of 70 Hz (top pictures), 178 Hz (centre pictures) and 448 Hz (bottom pictures). The two top and centre pictures show the vibration of the panel in correspondence to the resonance frequencies associated, respectively, to the (1,1) and (1,3) modes of the smart panel. The two pictures on the left show that as the 16 control gains are turned up the amplitude of the vibratory fields at the resonance frequencies of 70 and 178 Hz still are characterized by the (1,1) and (1,3) natural modes of the panel but their amplitudes are much lower. In general the nearfield sound radiation of a panel is directly associated to the panel vibration itself since there are no cancelling effects as can be seen in the farfield radiated sound [4]. Therefore, it is expected that, at resonance, the 16 control units produce both a reduction of the nearfield and farfield sound radiation as was highlighted in Fig. 7. The two bottom pictures show the vibration of the panel in correspondence to the resonance frequency of 448 Hz associated to the (1,4) mode of the smart panel. In this case as the control gain is turned up there is just a very little variation of the vibratory field with nearly no reduction of the vibration amplitude. As a consequence at this resonance frequency there is no reduction of the nearfield sound radiation and, as shown in Fig. 7 also the farfield sound radiation is not reduced. The two bottom pictures of Fig. 13 suggest that the 16 control units are not producing any control effect primarily because they are lying along the nodal lines of the (1,4) natural mode of the smart panel. Therefore in order to control the vibration associated to this mode, the 16 control units should be arranged with a different geometry over the panel surface. Probably a less regular arrangement of the 16 control units would allow the control of a larger number of modes provided larger control gains could be generated by each control unit in order to make up for the lower number of effective control units for each mode of the panel.

6.2. Structural primary source on the panel

The average spectrum of the vibration level of the plate excited by the shaker in the frequency range between 0 and 1 kHz is reported in Fig. 14 for both the case of uncontrolled (solid line) or controlled (faint line) system. It can be seen that the average attenuation factors of the first five resonances are, respectively, 6 dB (at 70 Hz), 13 dB (at 102 Hz), 10 dB (at 144 Hz), 13 dB (at 185 Hz) and 12 dB (at 250 Hz). Moreover, there are further attenuations of about 5–10 dB in correspondence of the higher frequency resonances. The overall attenuation factor in the frequency range of 0–1 kHz is 9.3 dB. This is a very good result that highlights the potential control effectiveness of the nearfield sound radiation of this type of smart panel.

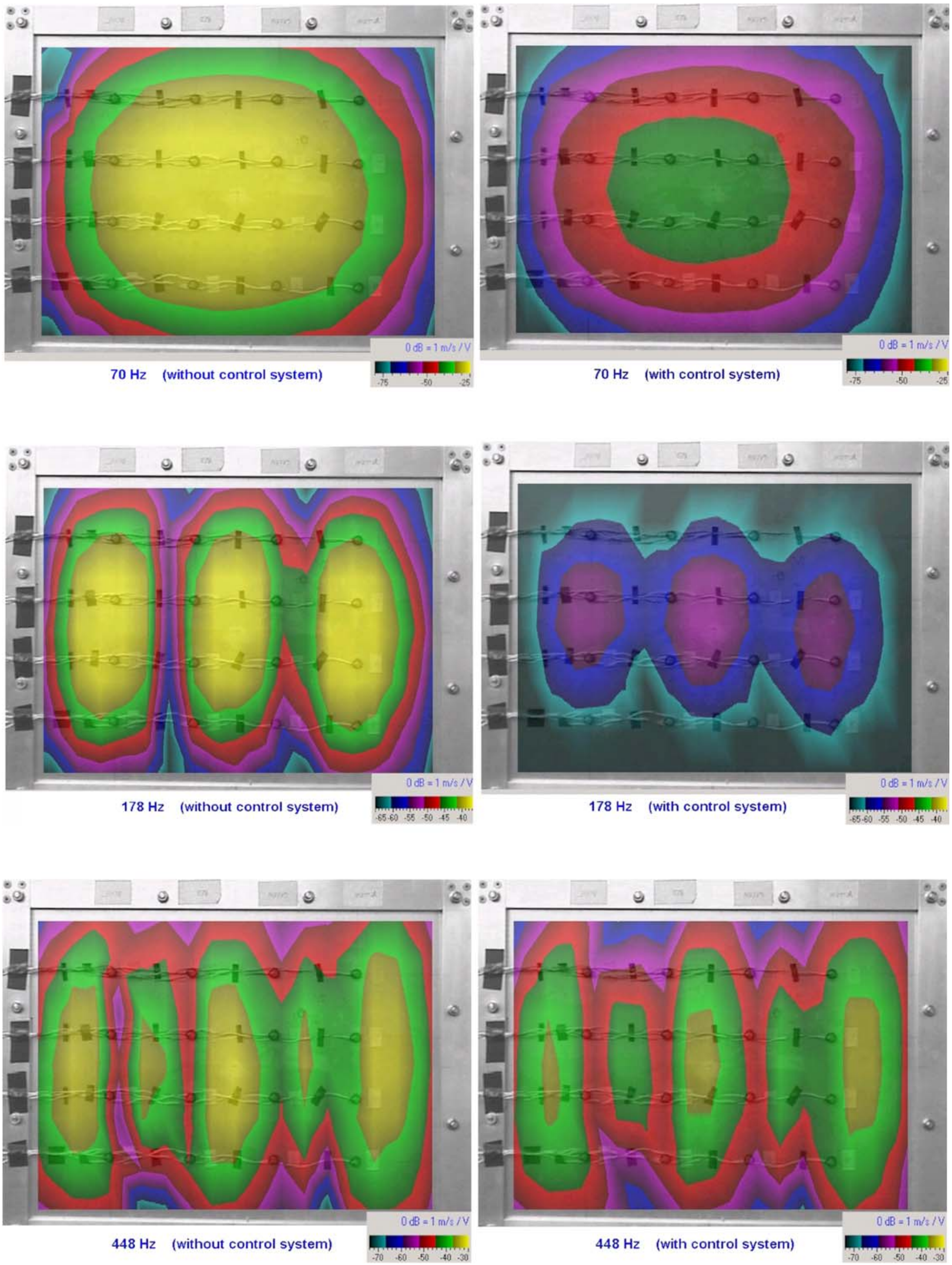

Fig. 13. Vibration amplitude of the panel per unit excitation of the loudspeaker measured with the laser vibrometer at 70 Hz (top pictures), 178 Hz (middle pictures) and 448 Hz (bottom pictures) without control (left pictures) and with 16 decentralized feedback control systems (right pictures).

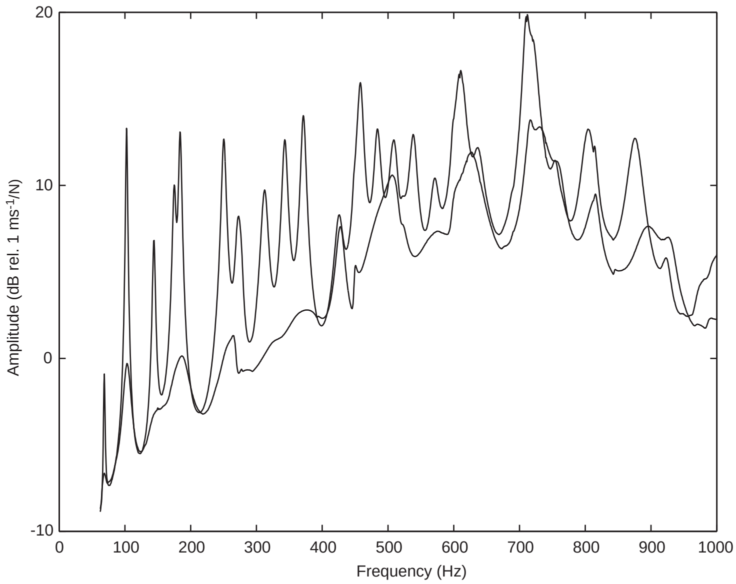

Fig. 14. Average frequency response function of the panel velocity (0–1 kHz) per unit excitation of the shaker calculated as the average of all the 133 frequency response functions measured at the grid points: no control system (solid line) and with decentralized feedback control (faint line).

Fig. 15 shows the vibratory field of the panel without control (left-hand side pictures) and with control (right-hand side pictures) when it is excited directly by the shaker at the frequencies of 70 Hz (top pictures), 250 Hz (centre pictures) and 343 Hz (bottom pictures).

As for the loudspeaker primary excitation, the two top pictures show the vibration of the panel in correspondence to the resonance frequency associated to the (1,1) mode of the smart panel. In this case, the shaker produces a point force at the top right-hand corner of the panel and therefore the vibratory field is characterized by the superposition of the (1,1) mode and the local response to the point excitation. Considering Fig. 13, it can be noticed that the acoustic excitation by the cavity underneath the panel does not produce any localized effect and therefore the response of the panel at 70 Hz is exactly characterized by the vibration field of the (1,1) mode of the panel. The left-hand side top picture in Fig. 15 shows that as the 16 control gains are turned up the amplitude of the vibratory field is reduced over most of the smart panel surface except in correspondence to the excitation point. Probably a much larger control gain should be implemented on the top left control unit in order to damp the local response of the panel to the concentrated force excitation exerted by the shaker. Similar behaviour is noticed for two centre pictures in Fig. 15 which shows the vibration of the panel in correspondence to the resonance frequency associated to the (2,3) mode of the smart panel. This mode is not seen when the panel is excited by the acoustic field in the cavity underneath it. Indeed, at such low frequency the cavity produces a volumetric excitation on the panel that cannot excite the modes of the panel without a net volumetric displacement as for example is the (2,3). The point force can instead excite this type of mode and therefore the resonance frequency at 250 Hz is measured as shown in Fig. 14. Also in this case the 16 control units produce good reductions of the vibration over the panel surface except in the vicinity of the point excitation exerted by the shaker. The two bottom pictures in Fig. 15 show the

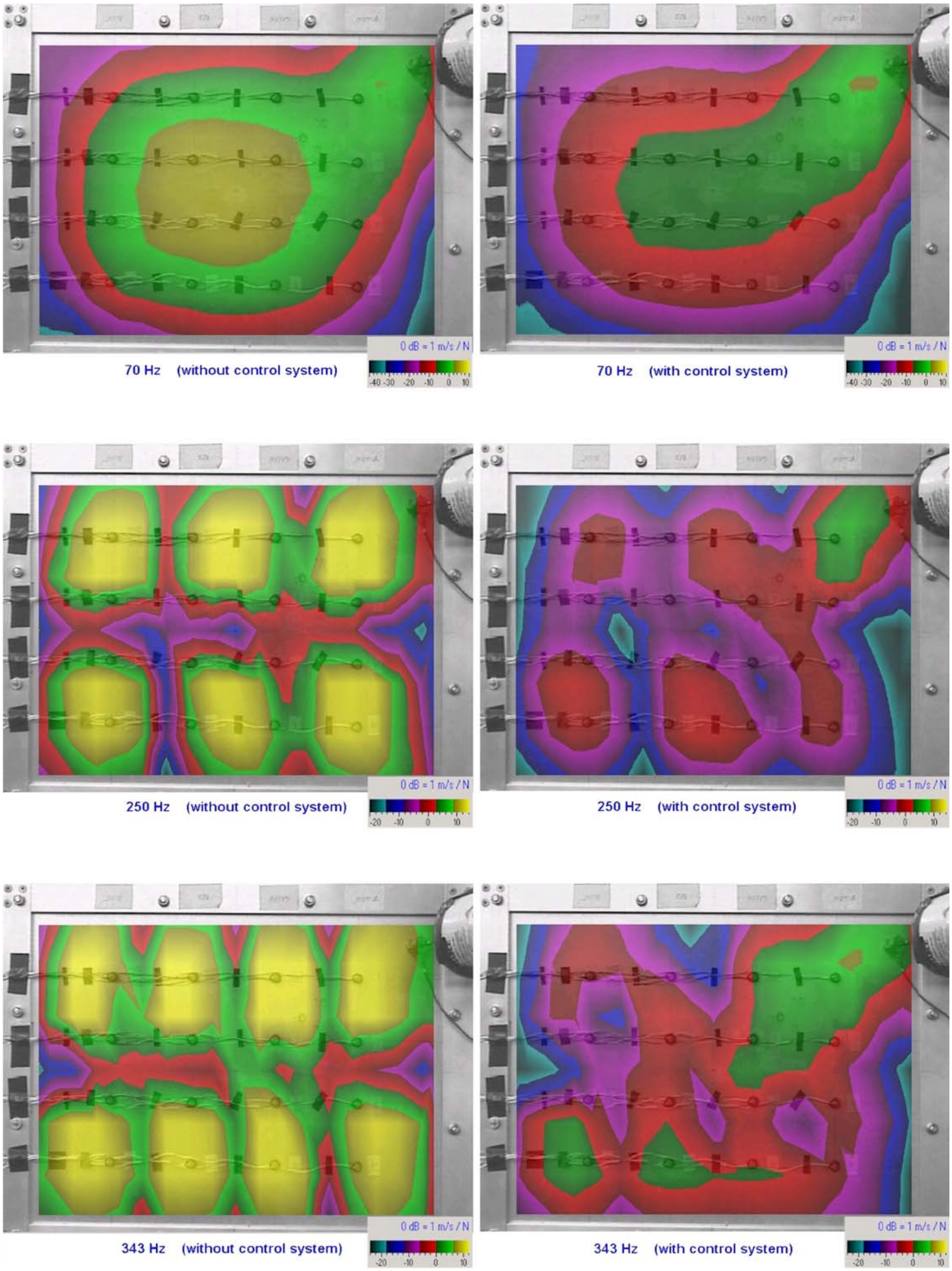

Fig. 15. Vibration level of the panel per unit excitation of the shaker measured with the laser vibrometer at 70 Hz (top pictures), 250 Hz (middle pictures) and 343 Hz (bottom pictures) without control (left pictures) and with 16 decentralized feedback control systems (right pictures).

vibration of the panel in correspondence to the off-resonance frequency of 343 Hz where the response is characterized by the (2,4) mode. Also in this case, when the 16 control gains are turned up, good reductions of the overall vibration are achieved. Moreover, after control, the vibratory field is more irregular since the response of the panel is characterized by a set of modes rather than the (2,4) mode only.

7. Concluding remarks

This paper summarizes the experimental tests carried out to assess the control effectiveness of a prototype smart panel with 16 decentralized control units, which implement velocity feedback control, for the reduction of sound radiation/transmission. Two types of measurements have been taken: first, the total sound power radiation has been measured in an anechoic chamber with and without the 16 control units turned on and second, the vibratory field over the panel surface has been measured with a vibrometer so that the variation of the vibratory field averaged over the panel surface and the vibration of the panel at some specific frequencies have been considered. Both types of measurements have been taken with reference to the acoustic excitation produced by the loudspeaker placed in the cavity underneath the panel and the point force generated by a shaker mounted on the panel. The first types of tests have highlighted the following conclusions.

1. When the smart panel is excited by the acoustic field generated by the loudspeaker placed in the cavity underneath it, the control system is able to damp down the sound radiation in correspondence to the first three resonance frequencies by about 8–13 dB. Between 350 and 1200 Hz there is little control effect; in particular, the resonance frequency controlled by the first few cavity modes can not be controlled. However, between 1200 and 1400 Hz reductions of about 10 dB of the measured sound level are registered. The measurements in third octave bands between 0 and 5 kHz have shown frequency averaged reductions of about 6 dB in correspondence to the third octave bands at 63, 80 and 1250 Hz. Smaller reductions are measured for the other third octave bands.
2. When the panel is excited by the point force generated by a shaker, the control system is able to damp down the sound radiation in correspondence to the first five resonance frequencies by about 12–18 dB. Between 350 and 1300 Hz and between 1600 and 1800 Hz there are significant control effects that go from a minimum of 3 to a maximum of 8 dB. Reductions of the sound radiation are measured in correspondence of all third octave bands between 0 and 5 kHz and there are at least seven bands with reductions of about 5–8 dB.
3. Very little control spillover effects have been found in these tests and in most cases they correspond to antiresonance frequencies as one would expect with active damping.
4. When the panel is excited by the point force generated by a shaker the low-frequency sound power radiation is characterized by a larger number of resonance frequencies while at higher frequencies there are less resonance frequencies than in the case of the acoustic primary excitation. Considering the loudspeaker primary excitation, then the smart panel is excited by the volumetric acoustic field in the cavity which is well coupled only with the panel modes with a net volumetric displacement, i.e., with either one or both odd modal order. However, at

higher frequencies the sound radiation is characterized not only by the resonance frequencies associated to the natural modes of the panel but also those associated to the natural modes of the acoustic cavity.

The measurements of the vibratory field over the panel surface with the laser vibrometer have highlighted the following features of the control system with 16 decentralized feedback units.

1. When the smart panel is excited by the acoustic field generated by the loudspeaker placed in the cavity underneath it, it has been found that the 16 decentralized control units uniformly damp down the overall vibration associated to the low frequency (1,1) and (1,3) modes of the panel. In contrast the 16 control units cannot produce any control effect on the (1,4) mode of the panel because they are exactly arranged along the nodal lines of this mode.
2. Also, it has been highlighted that the point force excitation produced by the shaker can excite even mode number as well as odd mode number. Also, this excitation is characterized by a local vibratory field that could not be contrasted by the decentralized control unit close to it.
3. Finally, the spectrum of the variation of the vibratory field averaged over the panel surface has highlighted that relatively high reductions up to 18 dB could be achieved for all resonance frequencies except those controlled by the response of the cavity. This problem is particularly important when the test rig is excited by the loudspeaker while it is negligible for the shaker excitation. Therefore, when the system is excited by the shaker all resonance frequencies are damped down by 5–13 dB.

It is important to emphasize that the testing configuration with the panel mounted on the top of the Perspex box has not given a complete picture of the effective control performance of the smart panel since the 16 control units are not able to reduce the vibration of the panel corresponding to well coupled cavity and panel resonating modes. Also, at some frequencies, the sound radiated by the panel is equivalent to the flanking sound radiation through the side walls of the Perspex box and therefore it was not possible to assess the true control effects. The control effectiveness of the smart panel is therefore expected to be relatively higher when tested on a proper sound transmission suite where the panel can be excited by a diffuse acoustic field without any low-frequency volumetric effect or higher frequency modal pattern of the incident acoustic disturbance.

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