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Smart panel with decentralised inertial actuator active damping units

Cristóbal G. Díaz^a, Christoph Paulitsch^b and Paolo Gardonio^c

ISVR, University of Southampton

Highfield

Southampton, SO17 1BJ

United Kingdom

ABSTRACT

This paper presents a study of decentralised velocity feedback control on a rectangular panel using five electrodynamic inertial actuators. The aim is to generate active damping in such a way as to control the low-frequency lightly damped resonances of the panel and thus reduce the sound radiation. It has been shown that collocated direct velocity feedback (DVFB) with an ideal point force actuators mounted on a structure generates active damping and is unconditionally stable. However, when an inertial electrodynamic actuator is used to implement velocity feedback control, the fundamental resonance frequencies of the actuator determine stability and control bandwidth limits.

In this paper the design of a small scale inertial actuator with a low mounting resonance frequency is first presented. The main design issues related to the stability of a velocity feedback control loop are discussed. The simultaneous implementation of five control units is then studied. The stability of the five control systems is assessed with reference to the locus of the eigenvalues of the 5x5 fully populated matrix with the open loop frequency response functions between the sensors and actuators. Finally the control performance both in terms of vibration reduction and radiated sound is presented.

1 INTRODUCTION

Sound radiation and transmission by thin, lightly damped, partitions is a relevant problem both in land and air transportation vehicles. For instance, in order to improve the fuel consumption efficiency, the new designs of aircraft fuselage and automobile bodywork involve stiff and light panels which therefore efficiently radiate noise generated by external acoustic sources (i.e. jet noise or reciprocating engine noise), by aerodynamic sources (i.e. turbulent boundary layer pressure fields on aircraft skins or on cars bodyworks) and by structure borne paths (i.e. engine induced vibrations or road induced vibrations) [1-Ch 6 7,2]. Passive treatments, such as stiffening, mass or damping treatments, can be used to reduce this problem [3] although, in order to be effective at low frequencies, they tend to be bulky and to introduce extra weight that contrast with the fuel consumption requirement. Recent research work has shown that decentralized active vibration control with point force actuators provides an efficient and comparatively convenient solution to the low frequency sound transmission problem [4].

At low frequency, the sound radiation by a lightly damped panel is characterised by well separated resonances of the low order modes of the panel [3, 5 Part-I]. Thus, steady state broad band sound radiation can be effectively reduced by tackling the low frequency resonances of the panel itself without the need of involving control arrangements with complicated sensor and actuator pairs that operate on the sound radiation modes of the

^a Email address: cgd@isvr.soton.ac.uk

^b Email address: cpaulits@gmx.de

^c Email address: pg@isvr.soton.ac.uk

structure [1,6,7]. In general, the resonant response of structures is controlled by damping. Thus in order to control resonances it is necessary to implement active systems that produce damping actions [4]. This can be achieved with simple direct velocity feedback (DVFB) control systems [8,9]. If ideal point force actuators are used, then collocated velocity feedback is bound to be unconditionally stable [10,11]. In practice, force actuators are constructed with electrodynamic actuators that react off a proof mass. These types of actuators produce a constant force above the fundamental resonance frequency of the resiliently mounted mass. Thus they must be designed with relatively low fundamental resonance frequency which must be well below the first resonance of the structure to be controlled. Also, Elliott et al. [12] have shown that in order to guarantee large control gains the amplitude of this fundamental resonance must be as small as possible. This can be achieved by enhancing the damping of the mounted inertial actuator [13] and by lowering as much as possible the fundamental natural frequency of the actuator [14].

Theoretical work has shown that in order to efficiently produce damping on panels arrays of point actuators must be used [4]. The control bandwidth tends to rise with the number of control units [7]. Thus it is necessary to design small scale control systems which withstand the necessary stability requirements for the implementation of direct velocity feedback loops and produce the largest possible damping effect. In this paper the design of a small scale inertial actuator for the implementation of direct velocity feedback loop on a thin panel is introduced. The stability of such single control system is first considered using the Nyquist criterion. Next the stability is assessed when five of these independent control units operate simultaneously. Finally the control performance is investigated both in terms of vibration reduction and radiated sound pressure reduction when the panel with the five control systems is mounted on a box and it is excited by a shaker placed within the box.

2 SMALL SCALE ACTUATOR

2.1 Actuation mechanism

In order to build a smart panel whose sound insulation effect competes with passive sound insulation treatments, the inertial actuators used to generate active damping must be light, provide large control forces and guarantee large control gains when velocity feedback is implemented. As schematically shown in Figure 1a, the control system considered in this study uses a coil-magnet electrodynamic linear motor. The coil is fixed to the base of the actuator and the magnet is suspended on springs so that it provides the inertial reaction necessary to generate a point force f_a on the structure where the actuator is fixed [8].

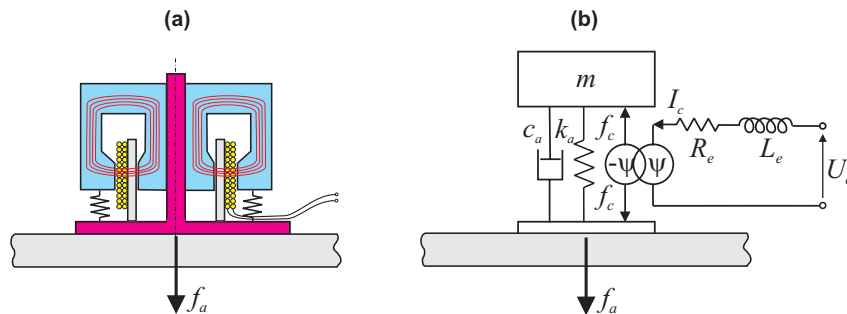


Figure 1: Proof mass electrodynamic force actuator: a) sketch b) electro-mechanical schematic.

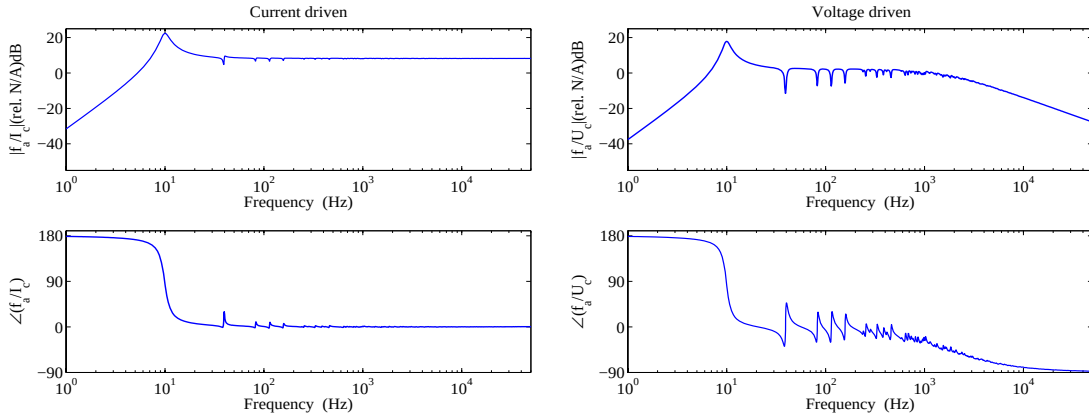


Figure 2: Force transmitted to the base structure per unit driving current (left) and voltage (right).

Figure 1b shows the equivalent electro-mechanical schematic that has been used to model the response of this actuator when it is fixed on a simply supported plate. The details of the fully coupled mathematical model can be found in reference [15]. As shown by the two plots in Figure 2, the spectrum of the transmitted force f_a per unit driving current I_c or voltage U_c increases monotonically with frequency up to a maximum value at the fundamental resonance of the actuator system at about 10 Hz. At higher frequencies, the transmitted force f_a levels down to a value that is approximately constant and equal to the reactive force f_c generated by the coil–magnet system [8]. A number of troughs are shown in correspondence to the low frequency resonances of the plate, which are due to the fact that at these resonance frequencies the plate produces little reaction to the actuator excitation. Above about 1 kHz, the transmitted force by the voltage driven actuator is characterised by an amplitude roll off and a constant phase lag which are due to the inductance of the driving coil. This tends to decrease the current in the driving coil, consequently lowering the actuation force. At frequencies below the fundamental resonance frequency of the actuator, the transmitted force is out of phase with the control signal while, at frequencies above the actuator resonance, it is in phase with the control signal. Thus, if negative velocity feedback is implemented, the desired damping action is produced only above the fundamental resonance of the actuator. In contrast, below this frequency negative damping is produced which tends to destabilise the control loop. In conclusion in order to guarantee a stable control loop with large bandwidth it is necessary to design an actuator with low fundamental resonance.

2.2 Stability of a direct velocity feedback loop

The plots in Figure 3 show the Bode and Nyquist plots of the open loop frequency response functions (FRFs) between the velocity sensor and current or voltage driven inertial actuators mounted on a simply supported rectangular aluminium plate of thickness 1 mm and dimensions $414 \times 314 \text{ mm}^2$. The Bode plots indicate that, for both current and voltage driven actuators, the phase of the open loop sensor–actuator FRF starts from $+270^\circ$ at low frequency, drops to $+90^\circ$ beyond the resonance of the actuator and then alternate between $+90^\circ$ and -90° for the resonances of the plate. The first 180° phase drop, from $+270^\circ$ to $+90^\circ$, is due to the dynamics of the actuator which, as discussed above, moves the actuation force from being out of phase to in phase with the driving current or voltage signals. In the voltage control case, at higher frequencies above 1 KHz, the phase plot goes to values beyond -90° because of the coil inductance. This inductive effect also decreases the amplitude of the open loop response function. The Nyquist plots

show that the loci of the open loop sensor–actuator FRFs are characterised by one circle in the left hand side quadrants, which is due to the resonance of the actuator, and many circles in the right hand side quadrants which are due to the resonances of the plate. Thus, according to the Nyquist stability criterion [16], the control system is bound to be only conditionally stable since, for relatively high control gains, the circle on the left hand side due to the fundamental resonance of the actuator can enclose the Nyquist instability point.

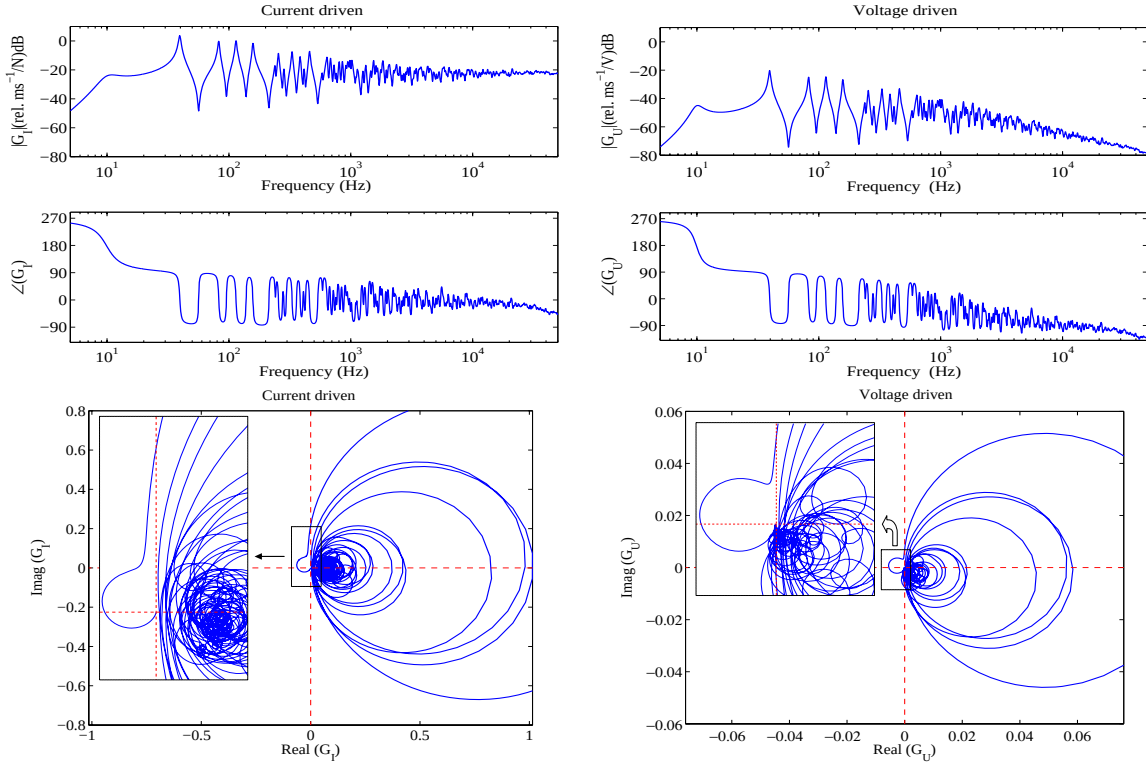


Figure 3: Bode (top) and Nyquist plots (bottom) of the open loop sensor–actuator FRF when a proportional feedback loop is used for current (left plots) and voltage (right plots) control.

In conclusion in order to obtain a stable proportional velocity control loop with large control gains it is necessary to have a low-amplitude actuator resonance frequency. This condition is also important to minimise the control spillover effects at low frequencies around the resonance of the actuator. This low-amplitude resonance of the actuator is normally obtained by designing an actuator with natural frequency well below the first resonance of the structure. There is however an intrinsic limit to achieve this result since the stiffness of the actuator must remain high enough to hold the inertial mass without a big static deflection; therefore a compromise must be found between a stiff enough spring to support the static weight of the suspended mass and a soft enough spring to guarantee a low fundamental resonance frequency of the actuator [12]. The amplitude of a given fundamental resonance frequency can be lowered by enhancing the internal damping of the mounting system. This can be effectively done by adding a relative velocity feedback loop to the actuator which generates internal active damping [13]. An alternative approach to this problem is to design an actuator with very low fundamental resonance frequency in combination with an integral displacement feedback control loop, which acts as a self leveling system that limits the extent of the static displacement [14].

2.3 Actuator Design

This section presents a brief summary of the design of the small scale inertial actuator shown in Figure 4 that has been built for this study. The details of the design are presented in reference [17]. As schematically shown in Figure 1a, this actuator is composed of a base disc with a cylindrical former on which the coil is wound up. A cylindrical permanent magnet, of mass m , is mounted on three springs and a vertical bush, which forces the magnet to oscillate in axial direction. As shown in Figure 4, the three springs are made of small circular rings which guarantee a relatively larger stiffness in transverse direction than in axial direction (k_r and k_a respectively). In this way the mount axial natural frequency, $\omega_a = \sqrt{k_a/m}$, can be kept rather low with a good transverse guiding effect which prevent non linear effects due to stick slip friction on the axial bushing. As shown in the Figure 1a, the cross section of the magnet is shaped in such a way as to have a magnetic circuit that generates a field oriented in direction

orthogonal to the coil winding. In this way a current flow through the coil produces a reactive axial force between the coil and the magnet which is given by $f_c = -\psi I_c$, where ψ is the transduction coefficient and I_c is the current in the coil circuit. The design of a small scale inertial actuator is relatively complex since for each part of the actuator there are contrasting requirements. First the number of coil windings should be large for a high transducer coefficient

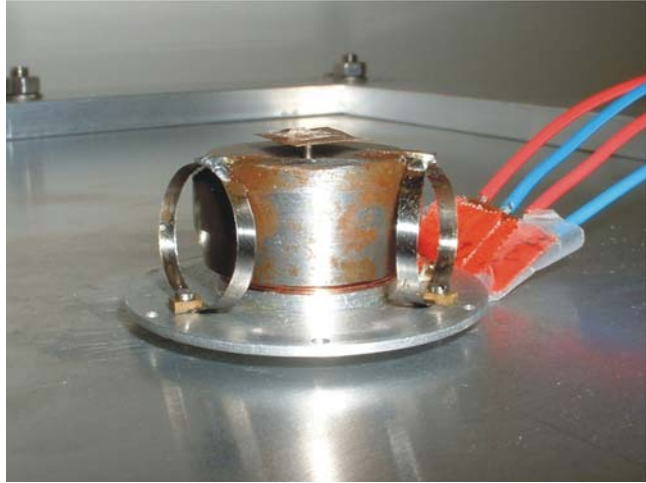


Figure 4: Photo of the small scale inertial actuator.

ψ , but the coil has to fit in the magnetic circuit and its resistance should not be high in order to minimize electrical energy dissipation. This can be achieved by appropriately designing the number of coil windings, the coil diameter and the diameter of the wire. Second, despite the small dimension requirements, a relatively large actuation force should be generated by the coil–magnet actuation system. This can be achieved by maximizing the magnetic field perpendicular to the coil windings by appropriately choosing the size of the permanent magnet, the air gap between the coil and the magnet, the shape and material of the magnetic circuit. Third, the mounting springs must have very low axial stiffness in order to keep as low as possible the axial fundamental natural frequency although they must also bear the weight of the magnet without breaking and without too large static displacements. The design of the actuator shown in Figure 4 has been carried out by setting first the dimensions and shape of the magnetic circuit, second the coil dimensions and third the shape and dimensions of the circular springs. For a moving mass of 20 grams an actuation force up to 4 N at an electrical power input of 5 W was aimed. Different shapes of the magnetic circuit were considered and, because of the possibility to use a strong Nd-Fe-B permanent magnet, the one shown in Figure 5 with the given geometric dimensions was further investigated. For this geometry the actuation force is found to be [17]

$$f_c = \frac{B_r H_l p h_g (A + 2t) b \mu_o (A^2 - A_i^2)}{4 H_l p h_g (A + 2t) b \mu_o + B_r q (A^2 - A_i^2) s} \sqrt{\frac{P}{\rho_{wi}}} \sqrt{\frac{1}{2d + h_g}} \sqrt{\frac{\pi s}{(A + 2t + s)}} \quad (1)$$

where P is the electrical power, d is the actuator stroke, ρ_w is the resistivity of the coil wire, B_r , H_l , μ_0 are the electromagnetic properties of the permanent magnet and other mediums of the magnetic circuit and p , q are non-dimensional leakage factors depending on the actuator shape. The force can be plotted against the moving mass for a large number of geometric values and as a result a linear relation between the maximum force and moving mass has been determined in the considered size range. This general trend has been verified by finite element simulations [17] where the leakage factors p and q are more accurately estimated. The actuation force scales like $f_c = 125 \rho_{mag} l^3$ with ρ_{mag} the mass density and l a characteristic dimension of the moving mass. The parameter values compiled in Table 1 in mm have been determined for a moving mass of about 20 grams.

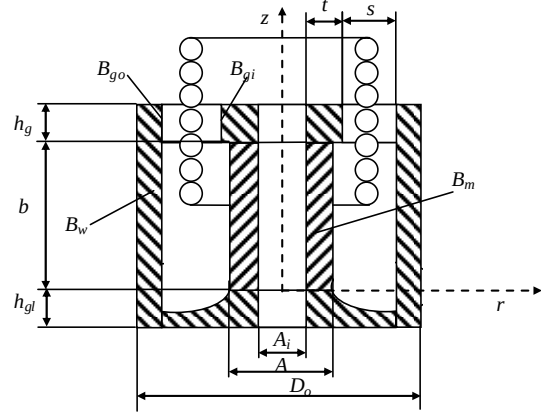


Fig 5: Characteristic dimensions of the coil-magnet system.

Table 1: actuator design parameters

Design	A	A _i	h _g	s	b	D _o	B _g (T)	m (g)	F _a (N)	h _{gl}
final	12.5	2	4	3	4.8	23.5	0.2972	20.3	3.66	2.3

The number of coil windings and the coil diameter are chosen based on the geometry given by the magnetic circuit design. At a given magnetic field in the air gap B_g the actuation force

$$f_c = \psi I_c = B_g I_c l_{cg} = B_g \sqrt{\frac{P \pi}{4 \rho_{wi}}} \left[d_w \sqrt{\pi (A + 2t + s) N n + l_{con}} - \frac{d_w l_{con}}{\sqrt{\pi (A + 2t + s) N n + l_{con}}} \right] \quad (2)$$

depends on the coil diameter d_w and the number of turns in vertical direction $N = (h_g + 2d)/d_w$ and in horizontal direction $n = ((s - t_{ol})/d_w - 1)2/\sqrt{3} + 1$ where t_{ol} is a constant for the thickness of the coil former and gap clearance, e.g. $t_{ol} = 0.0005$, and l_{con} is the length of the connecting wire. In order to reduce gaps between individual wires, the wire diameter should be chosen as small as possible. Thus it is a manufacturing constraint that determines the smallest thickness of the wire. Also, the number of turns should be as big as possible according to the manufacturing and construction constraints. Only a limited amount of current of 2 A is normally generated by off the shelf electric amplifiers. It is therefore possible to determine an upper limit of the wire diameter for a given power input of 5 W and a maximum heating limit of about 150°C. As a result of all these constraints, a wire diameter of 3.15 mm with 18 vertical and 8 horizontal windings was chosen [17]. The supporting springs have to be designed in such a way that the inertial actuator resonance ω_a is about 3 times lower than the lowest flexible resonance frequency of the plate $\omega_1 \approx 60$ Hz. As seen above, in this way, the

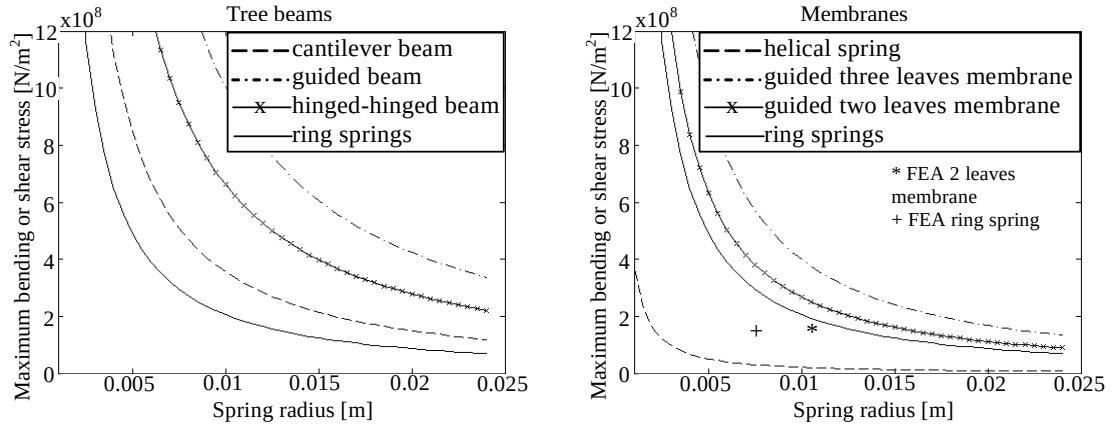


Figure 6: Internal stress for a 316N/m suspension stiffness as function of spring radius.

actuator transmits a constant force, which is proportional to the reactive force f_c generated by the coil–magnet system, over the range of frequencies where the resonant response of the panel has to be controlled. The resonance frequency of the inertial actuator $\omega_a = \sqrt{E_s I_d / l^3 \rho_{mag} l^3}$ scales like l^{-3} provided that the magnitude of the moment of inertia of the suspension I_d can always be made small enough [17]. The fulfilment of the last condition is not obvious since a small moment of inertia I_d and a large stroke result in a large mechanical stress that potentially cannot be withstood by the spring material. Figure 6 indicates that in comparison to other spring types, ring shaped springs are predicted to have the lowest internal mechanical stress for the size determined by the moving mass.

3 SMART PANEL WITH FIVE INERTIAL ACTUATORS

The inertial actuator described above has been designed in order to implement decentralised velocity feedback control loops that produce active damping on a thin panel. As shown in Figure 7, five prototype actuators have been built and mounted on an aluminum panel of dimensions $414 \times 314 \text{ mm}^2$ and thickness 1 mm. For each actuator

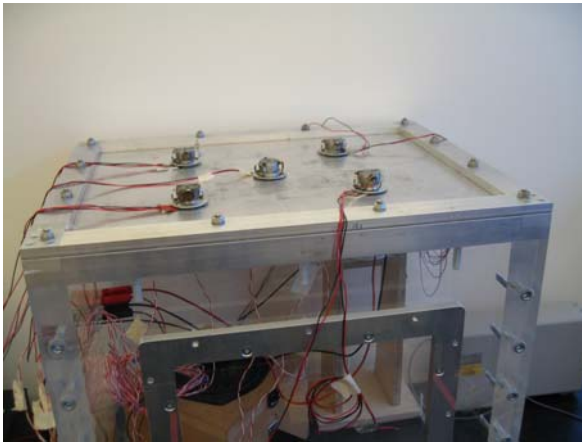


Figure 7: Photo of the box test rig and smart panel with five inertial actuators for the implementation of five decentralised velocity feedback loops.

an accelerometer sensor has been fixed on the other side of the panel in correspondence to the axis of the actuator itself. In order to get the desired negative velocity feedback control loops, for each control unit, the output signal from the accelerometer has been inverted, integrated and amplified with an analogue control system and then fed back to the actuator. Additionally in order to reduce the influence of the fundamental actuator resonance a high-pass filter with a cut-off frequency of about 50Hz is used in the feedback loop. The panel is mounted on a rigid frame

which is fixed to the top open side of a rectangular box with thick Perspex walls and internal dimension $414 \times 314 \times 360 \text{ mm}^3$. Four actuators have been arranged along the diagonals of the panels at distances $d_x = 0.35l_x$ and $d_y = 0.35l_y$ from the four corners of the panel and the fifth actuator in the center of the panel. The panel is excited by a shaker, mounted on a wooden stand within the rectangular cavity, which generates a point force at position $x_p = 0.66l_x$ and $y_p = 0.85l_y$. The box has been built with thick walls in such a way as to enclose the sound radiated by the bottom side of the panel. In this way it has been possible to measure the response of the panel and its sound radiation in presence of the primary excitation only and when the five active damping control units are also set in to work.

4 STABILITY OF SISO AND MIMO-DECENTRALISED VELOCITY FEEDBACK

The stability of the five decentralised feedback control loops has been assessed in two stages: first by analyzing the open loop sensor–actuator FRF of each control unit and then by analyzing the locus of the eigenvalues of the 4×4 fully populated matrix with the open loop, FRFs between the sensors and actuators.

4.1 Stability of a single control unit

Although the stability analysis of each control unit does not provide a clear indication on the stability of the whole set of five control systems, it is instructive to start with this type of analysis, which gives a physical understanding of the causes of instability in each control unit. The stability of a single control unit has been assessed by considering the Bode and Nyquist plots of the open loop FRF between the error sensor signal and the input signal to the analogue control system that drives the control actuator; the two plots are shown in Figure 8 for the control unit number 2, which is located close to the primary shaker.

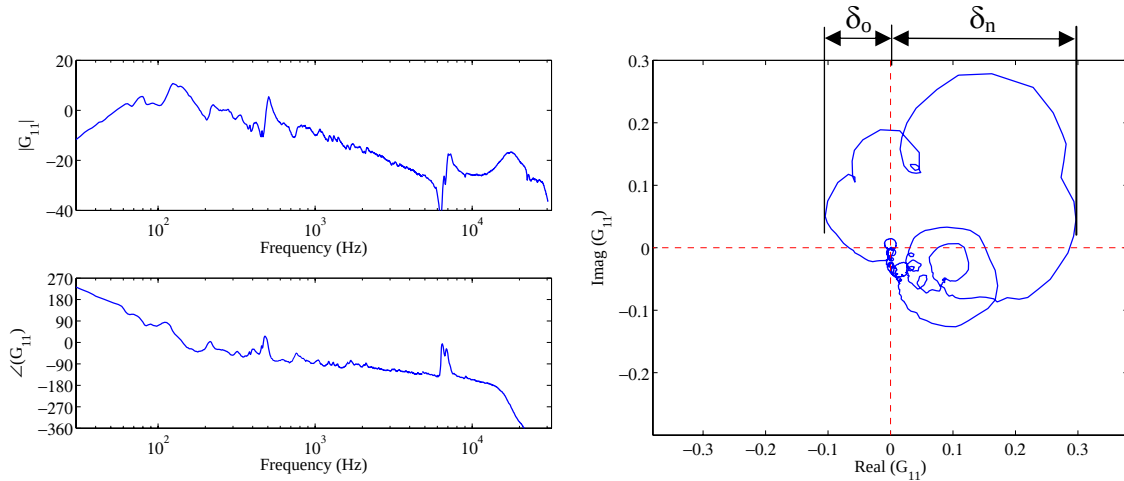


Figure 8: Bode (left hand side) and Nyquist (right hand side) plots of the open loop frequency response function between the error sensor signal and the input signal to the analogue control system.

The Bode plot in Figure 8 shows that there is no peak of the fundamental resonance of the actuator at about 20 Hz. This is due to the fact that it is well below the first resonance of the panel but also to the rather high internal damping effect in the actuator and to the high pass filter used to implement DC decoupling. As anticipated in section 2.2, the response is characterised by a sequence of plate resonances between 60 Hz and 2 kHz. At higher frequency there is a constant amplitude roll off due to the inductive effect of the coil. At about 6 kHz there is a deep trough followed by a sharp resonance peak. A finite

element analysis of the base disc and cylindrical former has shown a resonance frequency in a similar range. At higher frequencies two other resonances are visible which are due to the flexible modes of the circular springs and the roll-off in the amplifier.

The Nyquist plot shows a small circle on the left hand side which is due to the heavily damped fundamental resonance of the actuator and the high-pass filter for the DC decoupling. There are then a number of circles on the right hand side which are due to the plate resonances. The phase lag effect generated by the inductive effect in the coil gradually drifts the locus towards the imaginary axis in the bottom right quadrant so that the large circle for the base disc and cylindrical former resonance is oriented along the imaginary negative axis. At higher frequencies the locus enters the left hand side quadrants but, because of the inductive roll off effect, the amplitude is relatively small compared to the low frequency part located on the positive real quadrants. In conclusion the control system is bound to be conditionally stable with a gain margin $g_m = 1/\delta_o \cong 3$. The Nyquist plot in Figure 8 can also be used to estimate the maximum reduction of vibration that can be achieved at the control position. In fact, as shown in reference [18], for the n -th resonance frequency, the reduction of vibration in dB is given by $R_n = 20\log_{10}|1 + \delta_n/\delta_o|$, so that for the largest resonance at about 480 Hz the reduction is about 11 dB. In order to further enhance the controllability of the system several actions could be taken. First, the high-pass filter could be tuned such that an anti-resonance frequency lies at the intersection of the loop with the real axis. Second, the cut-off frequency of the high-pass filter could be lowered so that the low frequency bandwidth is dominated by the inertial actuator resonance at about 20Hz. Then the amplitude of the fundamental actuator resonance, at about 20 Hz, should be further lowered by either pulling down the resonance frequency or by enhancing the damping of this vibration mode. Third, the amplitudes of the two flexible resonances of the circular springs should also be lowered by enhancing the structural damping effect in the ring springs. It probably would also be helpful to increase the amplifier bandwidth or increase damping at the cut-off frequency. In this way, the part of the locus on the left hand side quadrants of the Nyquist plot is further squeezed towards the imaginary axis and larger control gains can be implemented in the feedback control loop.

4.2 Stability of the five decentralised control units

In order to assess the stability when the five decentralised control units operate simultaneously, the 5x5 fully populated matrix with the open loop frequency response functions between the five sensors and five actuators has been measured [8,19]. Figure 9 shows the modulus of these twenty five response functions. At first sight, two important features are observable. First, the amplitudes of the diagonal FRFs, i.e. the point FRFs, are much larger than the off-diagonal FRFs, i.e. the transfer FRFs. Second, above about 1 kHz, the off-diagonal frequency response functions are characterised by a much larger amplitude roll off than the diagonal frequency response functions. These two aspects are of great importance since the control stability of the five decentralised control units is determined both by the local dynamics of each control unit and by the cross dynamic effects between different control units. In fact as highlighted in Section 2, the stability of feedback control is determined by the phase lag between the error signal and the input signal to the actuator. Thus collocation between the sensing and actuating transducers is a key issue in order to guarantee stability. When multiple control systems operate, then the collocation property is affected by the neighbor control systems. Although the off-diagonal plots in Figure 9 indicate that the cross

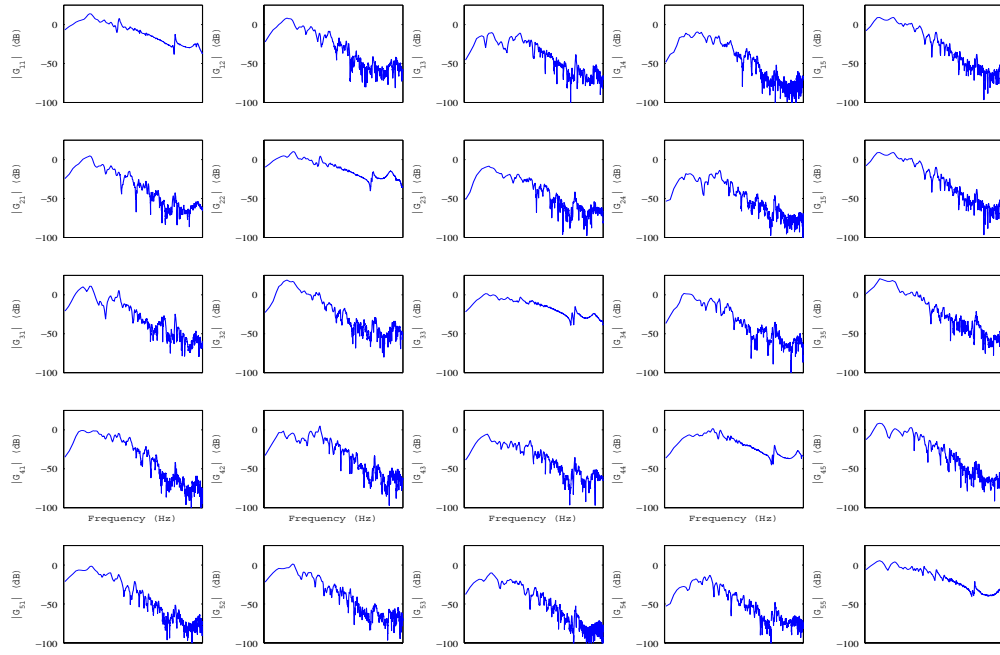


Figure 9: Modulus of the twenty five open loop frequency response functions between the five sensors and five actuators of the decentralised control systems.

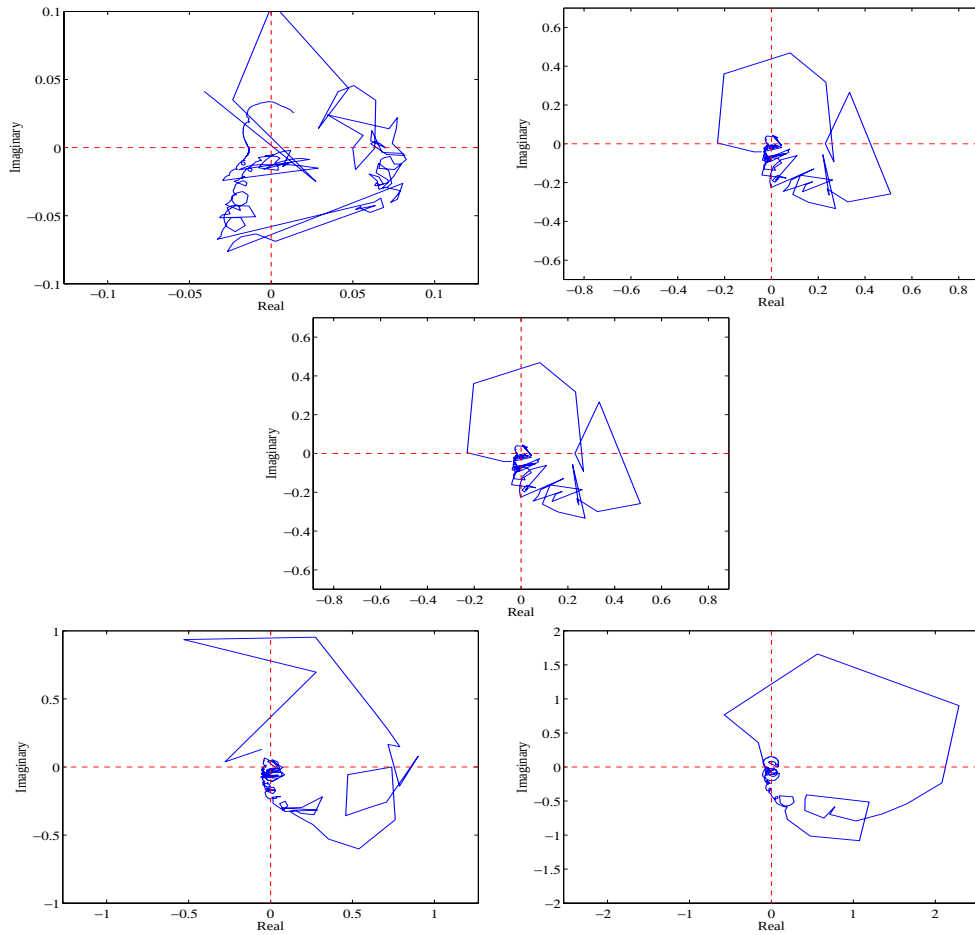


Figure 10: Loci of the Eigenvalues of the measured sensors-actuators transfer matrix between 0–32 kHz.

effects between the five control units are relatively small, particularly above 1 kHz, they may still introduce small phase shift effects that could lead to instability when the control gains that guarantee stability of the individual control units are implemented. This is an important issue that has been assessed by plotting the Loci of the Eigenvalues of the measured sensors-actuators transfer matrix between 0 and 32 kHz. The five plots in Figure 10 indicate that indeed the simultaneous implementation of the five feedback loops can be carried out but with relatively smaller control gains than those possible when the control units operate independently.

5 CONTROL PERFORMANCE

The control performance of the system with five decentralised active dampers shown in Figure 7 has been assessed in two steps. At first stage, the control effectiveness has been assessed when the control units operate independently by measuring the local reduction of vibration at the control points. The control effect produced by the five control units has then been evaluated by measuring the local reductions of vibration at the five control points and the reduction of the sound pressure measured at about 50 cm from the panel in correspondence to the centre of the panel.

5.1 Effectiveness of a single control unit

The plot in Figure 11 shows the spectrum of velocity measured by the error sensor of the control unit number 1 per unit primary force generated by the primary shaker. The low order modes of the plate give rise to well separated resonances although they are not characterised by sharp peaks since the five control units generate quite large passive damping effects. The amplitude of the response tends to decrease with frequency. This is due to the combined mass effect of the force gauge used to measure the point force of the primary excitation and the actuators. When the feedback loop is closed with the maximum control gain that guarantees stability, then, as shown by the dashed line, between 60 and 250 Hz the vibration at the error sensor goes down by about 8 dB. However at very low frequency in correspondence to the resonances of the low order modes of the panel, there is a small control spillover effect which is due to the smooth 180° phase transition of the high-pass filter at about 50 Hz.

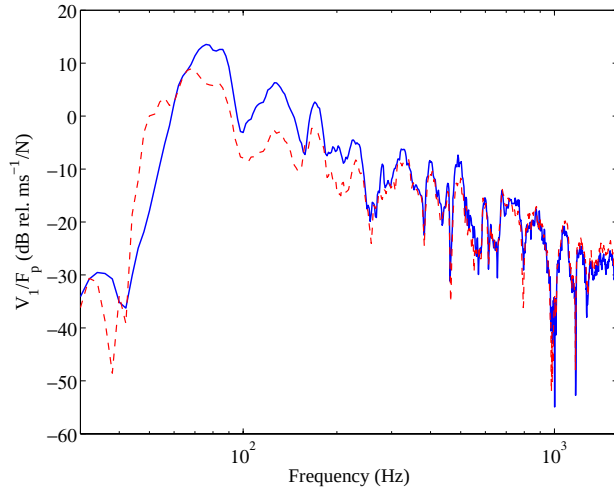


Fig 11: Velocity at the error sensor Nr. 2 per unit primary force without (solid line) and with (dashed line) control.

5.2 Effectiveness of the five decentralised control units

The control effectiveness when the five control units operate simultaneously is also assessed by considering the velocity spectra measured by the five error sensors per unit primary force generated by the primary shaker. The dotted lines in the five plots of Figure 12 show the measured error signals when the five decentralised control units operates simultaneously. Alternatively the dotted lines show the error signals produced by each control unit when the others are not running. Comparing these two curves, it is clear

that, when the five decentralised control units are implemented simultaneously, higher control effects are measured at the error sensors. In particular, at low frequencies, below about 60 Hz, the control spillover effects noticed above are much more marked. This is due to the interaction effects between the five control units which alter the stability and also controllability effects of the five systems. In particular the phases of the open loop response functions of the five sensor–actuator pairs are modified in such a way as the feedback loop produces a positive, rather than negative, velocity feedback control loop.

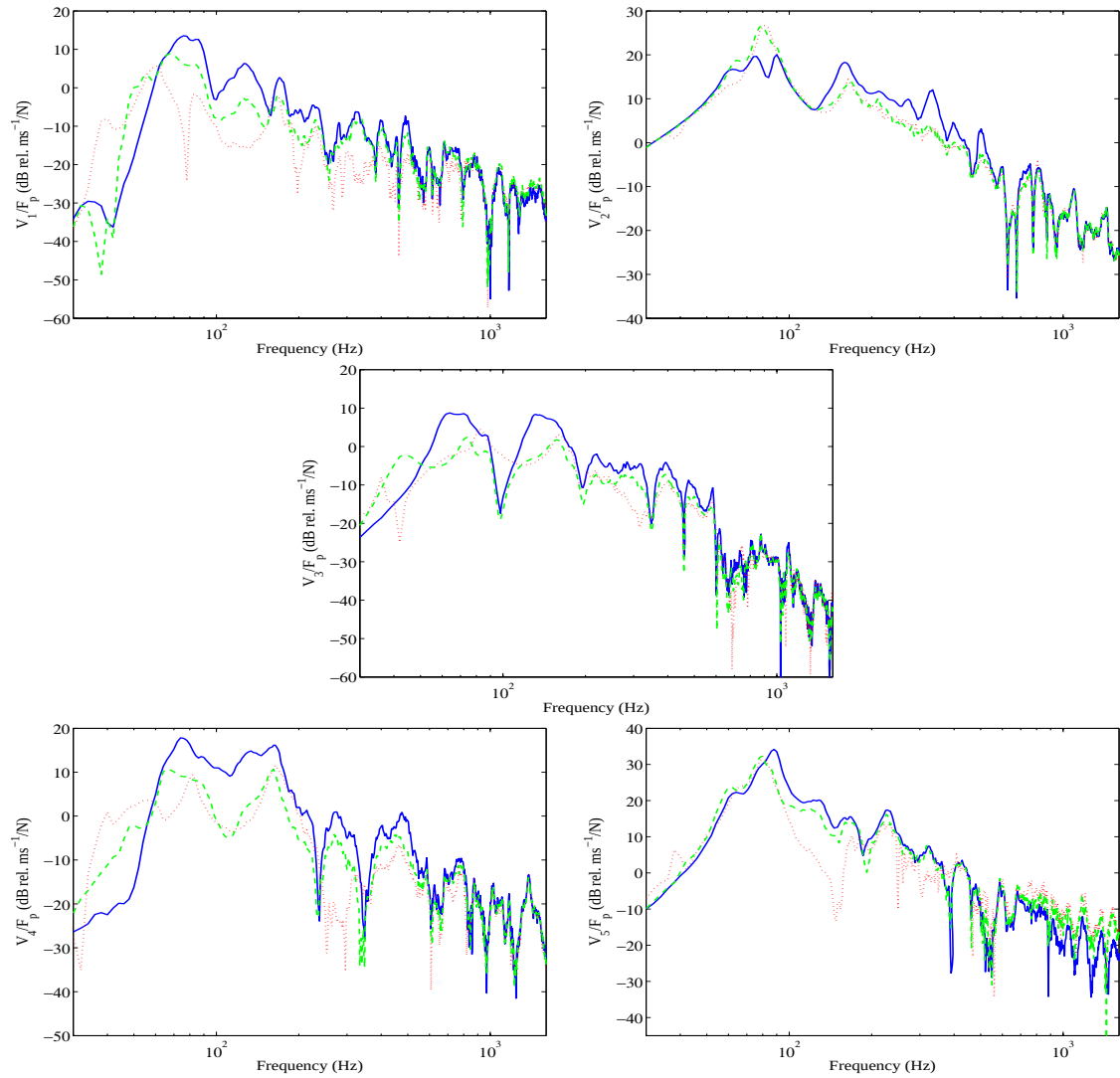


Figure 12: Velocities at the five error sensors per unit primary control force when there is no control (solid lines) and when the feedback control loops operates individually (dash lines) or simultaneously (dotted lines).

The overall control effect of the five control units has been assessed by measuring the radiated sound pressure about 50 cm above the centre of the panel. It must be noted that, although the smart panel is a symmetric structure, the primary excitation is positioned close to a corner. At the low frequencies considered in the plot, the bending near field generated by the point excitation is rather large and thus the near field sound radiation is not symmetric. Thus it is likely that the measurement is strongly linked to the acoustic

fields radiated when the panel is excited by the primary excitation only or both primary and five control systems. Nevertheless, this type of measurement is presented as a first simple assessment of the overall reduction of sound radiation. Further measurements of sound power radiation in anechoic room will be taken to provide a clear picture of the control performance. The plot in Figure 13 shows that the five control units generate good sound reductions of about 10 dB in the frequency range between 50 and 400 Hz.

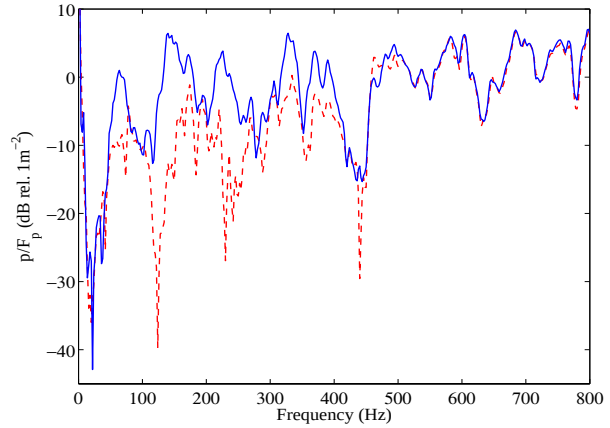


Figure 13: Sound pressure per unit primary force measured by a microphone placed at 50 cm from the centre of the panel when there is no control (solid line) and the five control units operate (dashed line).

6 CONCLUSIONS

In this paper the design and operation of a smart panel with five decentralised velocity feedback control systems using electrodynamic inertial actuators has been presented. These control systems are meant to generate active damping on the panel in such a way as to control the low-frequency lightly damped resonances of the panel and thus reduce the sound radiation.

At first the principal design issues for the actuator have been discussed. In particular the stability of velocity feedback using an inertial force actuator has been examined in view of collocation and duality properties between the actuator and a point sensor. The system is only conditionally stable because of the low frequency fundamental resonance of the actuator. The stability of a single control unit has then been assessed using a prototype actuator specifically built for this study. It has been shown experimentally that the system is only conditionally stable because of three principal effects: 1) the low frequency fundamental resonance of the actuator, 2) the high pass filter used for DC decoupling and 3) the flexible resonances of the mounting springs. Nevertheless it has been possible to implement a relatively high control gain so that large vibration reductions have been measured at the control point. Stability and control performance experiments have also been carried out with five decentralised control units using the prototype inertial actuator. The stability of the five control loops has been assessed with reference to the locus of the eigenvalues of the 5x5 fully populated matrix with the open loop frequency response functions between the sensors and actuators. It has been found that when the five control units operate simultaneously smaller control gains can be implemented. This is due to cross effects between the five control units which perturb the collocation property which is necessary to guarantee stability. Also in this case it has been possible to implement closed loop experiments which have given interesting preliminary results both in terms of the vibration reductions at the control points and the measured sound pressure at about 50 cm from the centre of the panel.

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