ICSV19
Vilnius, Lithuania
July 08-12, 2012

ACTIVE VIBRATION CONTROL OF A DOUBLY-CURVED PANEL UNDER PRESSURIZATION

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This study focuses on the control of doubly-curved panels which occur in aircrafts due to the deflection of fuselage panels during lateral pressure loading. This paper describes experimental work conducted toward the implementation of a feedback velocity control system on a pressurised panel of varying curvature in both the x and y directions. A thin rectangular aluminium panel was clamped to an airtight, rigid-walled enclosure and the curvature of the panel was varied through changing the interior static pressure. An electrodynamic proof-mass actuator and collocated accelerometer were implemented on the panel and in order to ensure stability and improve the performance of the feedback controller, a compensation filter was designed and added to the system. The attenuation in the structural response was investigated for increasing curvature. Current offline results show that effective control can be achieved for low levels of curvature. Furthermore, the compensated controller is robust to changes in curvature and the optimal gain of the compensated system remains more or less independent of the curvature level. However, increasing the curvature further causes modal clustering near the panel's ring frequency which reduces the performance of the controller.

1. Introduction

Shallow shells are widely used in the aircraft, naval and automotive industries. From a structural dynamics point of view, shells tend to be a more advantageous solution than plates, since in-plane strain, shear stress and bending stiffness make shells considerably stiffer than plates and more resistant to deformations, thus allowing manufacturers to produce structures that respond to both weight reduction and increased strength requirements.[1,2] However, the light weight and low damping of curved structures make them a direct transmission path to the interior of the cabin for the vibrations generated by the engine, turbulent airflow and/or road-tyre interactions. With the rising demand for quieter, lighter and more fuel efficient vehicles, active structural control may be able to achieve more effective and lighter weight solutions than passive vibration control techniques.

The coupled interaction between the transverse and in-plane motions introduces difficulties in the mathematical modelling of the structure, as the differential equations of motion include several unknown parameters. Another problem arises from the accuracy of the modelled boundary conditions and the selection of trial functions that can satisfy the equations of motion.[3] Even though a large selection of literature can be found on the topic of curved shells, most of these topics only cover specific cases such as singly-curved cylindrical shells or spherical shells.[2,4,5]

This paper describes experimental work conducted toward the implementation of a feedback velocity control system on a pressurised panel of increasing curvature. The aim of this research was to design and implement a feedback control system that remains stable and robust despite the changes in the surface shape. Such controller could be applied to flexible structures such as an aircraft fuselage, where the change in pressure caused by changes in altitude causes the fuselage to deflect. The first part of this paper will provide an overview on doubly-curved panels and the effect of curvature increase on the natural frequencies of a rectangular panel. The second section will describe the designed control unit. The last section will describe the experiments conducted on the pressurized enclosure. The measured open-loop FRFs and offline closed-loop results will be discussed for the case of a flat and a curved panel.

2. Overview of doubly-curved panels

A shell can be defined as a three-dimensional structure, confined by two parallel surfaces, where the distance between these two parallel planes is small with respect to other dimensions.[1] Based on Vlasov's definition, a shallow shell is a thin-walled structure where the rise above the based plane is relatively small, and the radii of curvature are larger than other dimensions of the shell.[5]

Analytical and numerical results have been derived for a doubly-curved panel of rectangular base and constant curvature by Nourzad et al., where the modelled panel was subjected to shear diaphragm boundary conditions.[6] The effect of curvature on the natural frequencies of the panel used for this study can be viewed in Fig. 1. The curves were plotted against zc/h, the ratio of the deflection in the centre of the panel to its thickness.

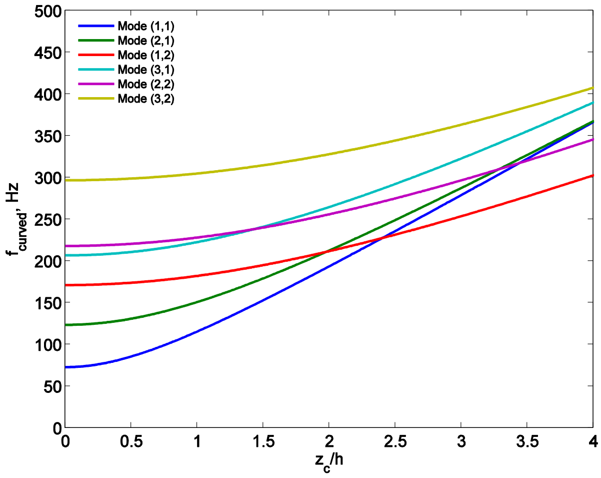

Figure 1. The 6 first modes of a simulated rectangular panel over increasing deflection-to-thickness ratio

With the gradual deflection of the panel, the natural frequency of each mode is significantly increased and a cluster of modes can be seen near the highest curvature. Additionally, the curves for higher order modes, such as modes (1,2) and (2,2), intersect the lower order modes (1,1) and (2,1). This can be explained by the different bow in the structure in the *x* and *y* directions, such that $R_x \neq R_y$.

3. Active control unit

Decentralised velocity feedback control can be a robust solution for the control of vibrations in flexible structures such as an aircraft fuselage. This section describes the design of the actuator and the effects of adding a simulated compensation filter.

The control unit used in this study consisted of a proof-mass electrodynamic actuator collocated with an accelerometer, signal conditioner and amplifier. The transducer unit used as a proof-mass electrodynamic actuator was a 60-mm diameter cone loudspeaker manufactured by Pro Signal, type S066M. These actuators are mass-produced and commercially available at a very low cost. They are lightweight and their high stiffness makes them fairly robust. Therefore, they are suitable for the implementation of a light weight and cost effective velocity feedback control system. The unit is composed of a Mylar diaphragm attached to a voice coil suspended in a constant magnetic field. The magnetic field is provided by a permanent magnet which also acts as the proof mass. Pictures and a diagram of the actuator can be seen in Fig. 2.

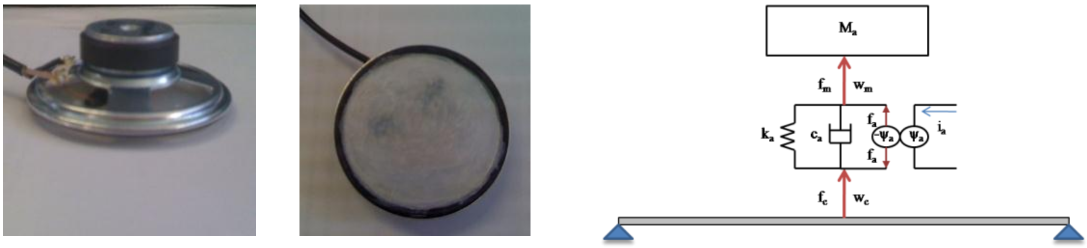

Figure 2. Actuator unit and block diagram

In order to achieve efficient acoustic coupling between the loudspeaker moving mass and the vibrating surface, the cone of the loudspeaker was filled with silicone acetate sealant. A thin plastic disk was then fixed on the filling to ensure a level and smooth contact surface. The silicone acetate was chosen after trying different possible fillings because of its light weight, ease of application and flexibility. These properties allow the loudspeaker diaphragm to move freely and react against the surface when being driven. The mechanical and electrical properties of the unit are listed in Table 1.

Table 1. Mechanical and electrical properties of the actuator

Parameter	Value	Units
Base mass, m_1	2	g
Proof mass, m_2	60	g
Suspension stiffness, k_a	6686	N m^{-1}
Suspension damping coefficient, c_a	1.92	Ns m^{-1}
Modal damping ratio, ξ_a	0.048	
Voice coil coefficient, ψ_a	0.018	NA^{-1}
Coil resistance, R_e	8	Ω
Coil inductance, L_e	88.1	μH
Actuator natural frequency, f_a	53	Hz

To characterise the designed actuator, blocked force measurements were performed up to 10 kHz when the actuator was mounted on a heavy rigid steel block acting as the blocked base. The blocked response was measured between an input voltage to the actuator and the resulting acceleration measured by an accelerometer fixed on the top of the actuator, as shown in Fig. 3.

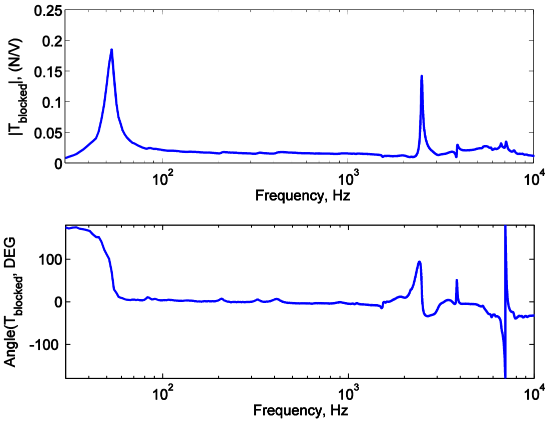

Figure 3. Blocked force transfer response when the actuator is voltage driven

The natural frequency of the actuator corresponds to the 53 Hz lightly-damped resonance peak observed in the transfer response graph and the associated 180-degree phase-lag can be seen in the lower subplot. The damping ratio of this resonance is 4.8%. For frequencies above the natural frequency of the actuator, the response drops until it becomes flat up to 2 kHz, above which the unit has internal resonances.

The fundamental resonance of the actuator influences the stability of the feedback control system. At frequencies above the natural frequency of the spring-mass system, the structure is subjected to a sky-hook damping effect, which is due to a base force being equal to the mass-spring reactive force. At frequencies lower than the natural frequency of the actuator, a negative damping effect occurs, caused by the base force being out of phase with the reactive force of the system, which may lead to instabilities in the feedback control system. Therefore, in order to achieve optimal control, it is necessary to use an actuation mechanism with a natural frequency much lower than the one of the vibrating structure, and high damping at the resonance. [7,8]

In order to achieve these requirements for the actuator used in this study, a second order compensator was added in the feedback loop. The employed compensator was based on the design proposed by Elliott et al. in [9] and is described by the following equation:

$$C(j\omega) = \frac{2j\hat{\zeta}_a\hat{\omega}_a\omega + \hat{\omega}_a^2 - \omega^2}{2j\zeta_c\omega_c\omega + \omega_c^2 - \omega^2} \qquad (1)$$

where $\hat{\zeta}_a$ and $\hat{\omega}_a$ are respectively the damping ratio and the natural frequency of the actuator, and ζ_c and ω_c respectively refer to the damping ration and natural frequency of the compensator. The natural frequency of the compensator was set to 10.6 Hz in order to reduce $\hat{\omega}_a$ by a factor of 5. The compensator was also assumed to be critically damped ($\zeta_c = 1$).

4. Experimental study of active control on a pressurized enclosure

In order to predict the performance of the active control system, a thin homogeneous aluminium panel of 1 mm thickness was mounted and clamped along its four edges on a rigid frame which was fixed on top of a rectangular enclosure. The enclosure walls were made of 3-cm thick panels of Plexiglas to provide rigid-wall acoustic boundary conditions and ensure that sound was only radiated through the panel. The dimensions of the free area of the panel are $l_x \times l_y \times h = 414 \times 314 \times 1 mm^3$ and the inner dimensions of the Perspex box are $414 \times 314 \times 385 mm^3$. The enclosure was made airtight in order to prevent any air leaking during pressurisation. The increase in the interior static pressure was achieved using a pressure compressor and a pressure regulator for controlling the gas flow into the enclosure. In order to measure the static deflection of the aluminium panel, a dial indicator was mounted on the top frame of the enclosure and positioned at its centre. A block diagram and photograph of the set-up are provided in Fig. 4.

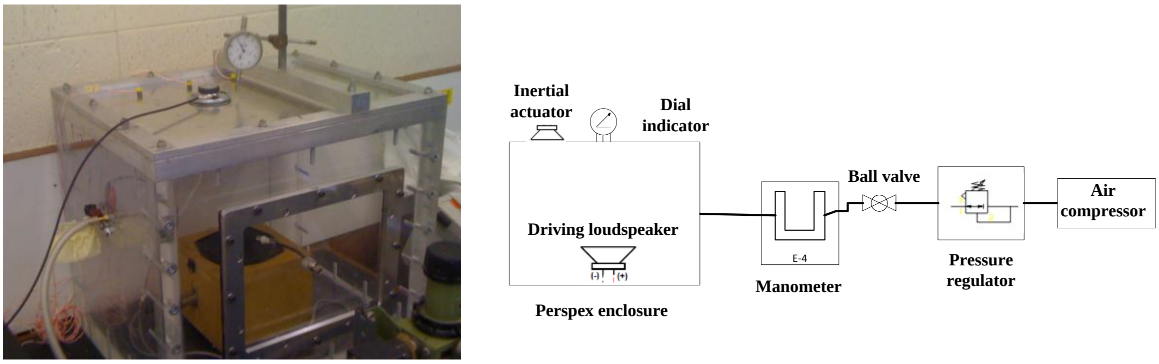

Figure 4. Experimental set up and diagram of the pressurised enclosure

A loudspeaker was placed inside the enclosure to generate the primary disturbance. The actuator was mounted on the surface of the panel at a position ($l_x - 0.62l_x, 0.53l_y$) and the corresponding accelerometer sensor was positioned directly under. This slightly off-centre position is optimal for controlling more modes when there is only one controller. Measurements were performed over 11 different curvature levels corresponding to $z_c/h = 0, 0.2, 0.4, ..., 2$. In order to obtain a static deflection of 2mm in the centre of the panel, the interior pressure was increased to 715 Pa. The experimental results for a flat and curved panel with $z_c/h = 2$ will be shown in the following subsections. The frequency range has been measured up to 2 kHz but plotted up to 500 Hz as the response rolls off above this frequency.

4.1 Open-loop response

The open-loop response between the voltage input to the actuator and the acceleration measured by the error sensor is presented in Fig. 5, for flat and curved panels. The effect of adding the compensator on the natural frequency of the actuator and the stability of the system can be seen in the Nyquist and Bode plots. In both cases, the results have been plotted for a gain level corresponding to the 6-dB gain margin of the uncompensated control system.

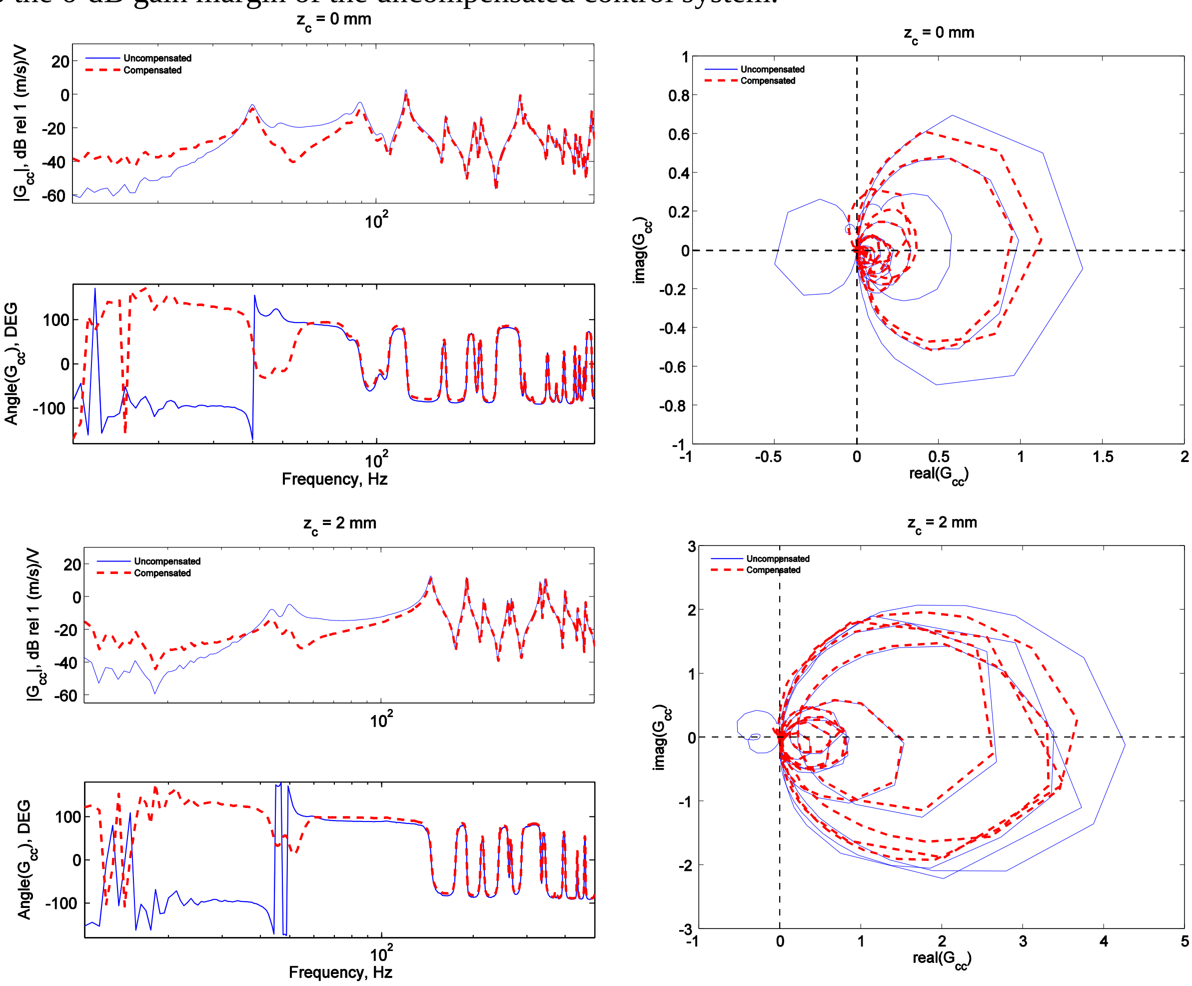

Figure 5. Bode plots (left) and Nyquist plots (right) for the flat and doubly-curved panels, before and after compensation

The uncompensated bode plots show the coupled interactions between the actuator and the panel. In both flat and curved panel figures, two closely spaced peaks of opposite phase can be seen in the low frequency domain: one located below the actuator natural frequency and the other one above the panel's first resonance. The first peak, which occurs near 40 Hz in both flat and curved panels, corresponds to the coupled response where the actuator dominates, while the second peak is the coupled response where the panel dominates. The actuator natural frequency is lowered due to the decrease in stiffness when being coupled to the plate. In the case of the curved panel, the second peak is shifted towards higher frequencies, such that it is near 88 Hz for the flat panel and around 140 Hz for the curved panel. This is caused by the clustering of modes due to increased curvature, as explained in Section 2. A low-amplitude peak can be seen around 50 Hz in the uncompensated open-loop FRFs, which corresponds to the blocked natural frequency of the mass-spring system in the actuator and its passive effect on the panel.

For the uncompensated system, the natural frequency of the actuator appears on the left-hand side of the nyquist plots, while the lobes on the right-hand side are due to the resonances of the panel. The stability and performance of the control system can be assessed by the ratio of the size of the right-hand side lobe to the left-hand side lobe.[10] The larger this ratio, the greater the distance between the natural frequency of the actuator and the 1st mode of the panel. It can be seen from the uncompensated Nyquist plots, that this ratio is much larger in the case of the curved panel than the flat panel. It can be deducted that the introduction of curvature in the structure may improve the stability of the system.

When the compensator is added in the feedback loop, a dip occurs at the location of the actuator's resonance and a phase lag is introduced to the system at this frequency. Consequently, the lobe corresponding to the actuator natural frequency is shifted to the right-hand side in the Nyquist plot, which corresponds to a significant increase in the stability of the system and allows the implementation of feedback control at higher gain levels. The Nyquist plot of the curved panel still shows a larger ratio after compensation, showing that the actuator-compensator system is robust to changes in curvature.

4.2 Control performance

The control performance and robustness of the system were assessed through offline simulations based on the measured open-loop responses, the measured transfer mobility between the actuator and a grid of 16 points on the panel and the measured transfer mobility between the primary disturbance and the grid of 16 points.

The uncontrolled velocity per unit voltage at the position of the actuator for the uncompensated and compensated closed-loop systems on the flat and curved panels is shown in Fig. 6. The uncompensated results are plotted with a 6-dB gain margin and the compensated results are plotted for an optimal gain level corresponding to the maximum attenuation in the overall structural response as discussed below.

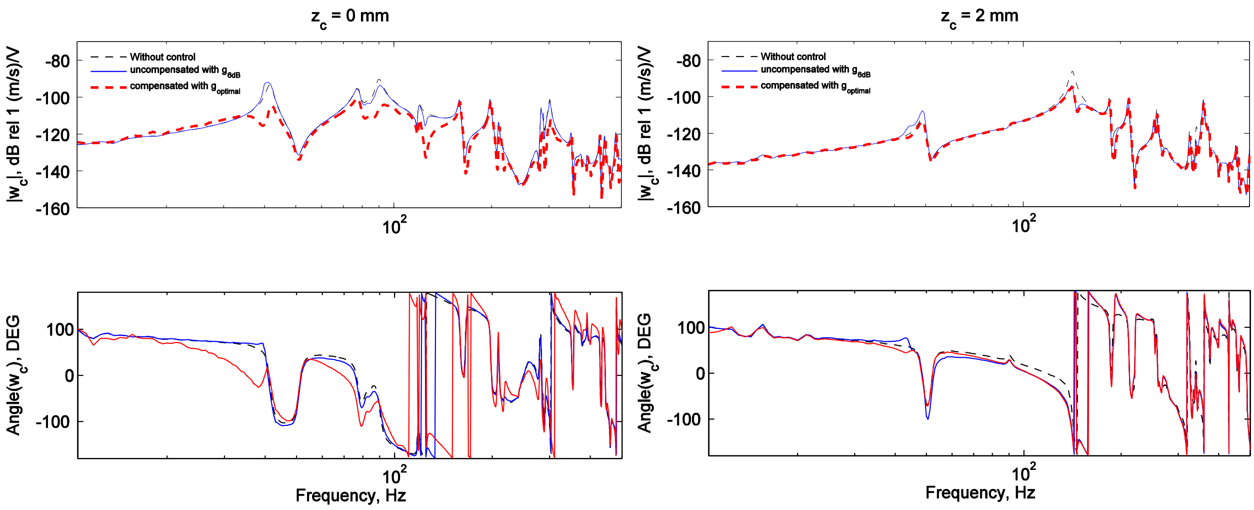

Figure 6. Control velocity per unit voltage for flat and curved panels, before and after compensation

In the absence of compensator, when the feedback loop is closed, the level of control is very limited. The uncompensated system has a low gain that cannot be increased without pushing the system towards instability. However, when the compensator is introduced in the feedback loop, the control system can more successfully attenuate the response. Even though the increase in curvature may reduce the attenuation levels in comparison with the flat panel, the panel is still controlled and no spill-over or noticeable enhancement occurs.

In order to further assess the control performance, the overall kinetic energy was calculated up to 500 Hz both with and without compensator. Figure 7 shows the kinetic energy and optimal gain graphs for both flat and curved panel. The increase in the natural frequency of the first panel mode with pressurization, from 88 Hz to 150 Hz, is clear from this figure. The optimal gain is the point at which maximum attenuation in the overall kinetic energy level is achieved and can be seen on the right-hand plots in Fig. 7.

The changes in kinetic energy level with feedback gain for the uncompensated system give an indication of its low gain margin. Because of the low gain margin of the controller and the lack of sufficient distance between the optimal gain and the maximum stable gain, most of the attenuation in the structural response is due to the passive effect of the actuator rather than the feedback controller. However, when the compensator is added to the feedback loop, the maximum stable gain and the gain margin are both significantly increased. The optimal feedback gain is also at least a factor of 10 away from the limit of instability.

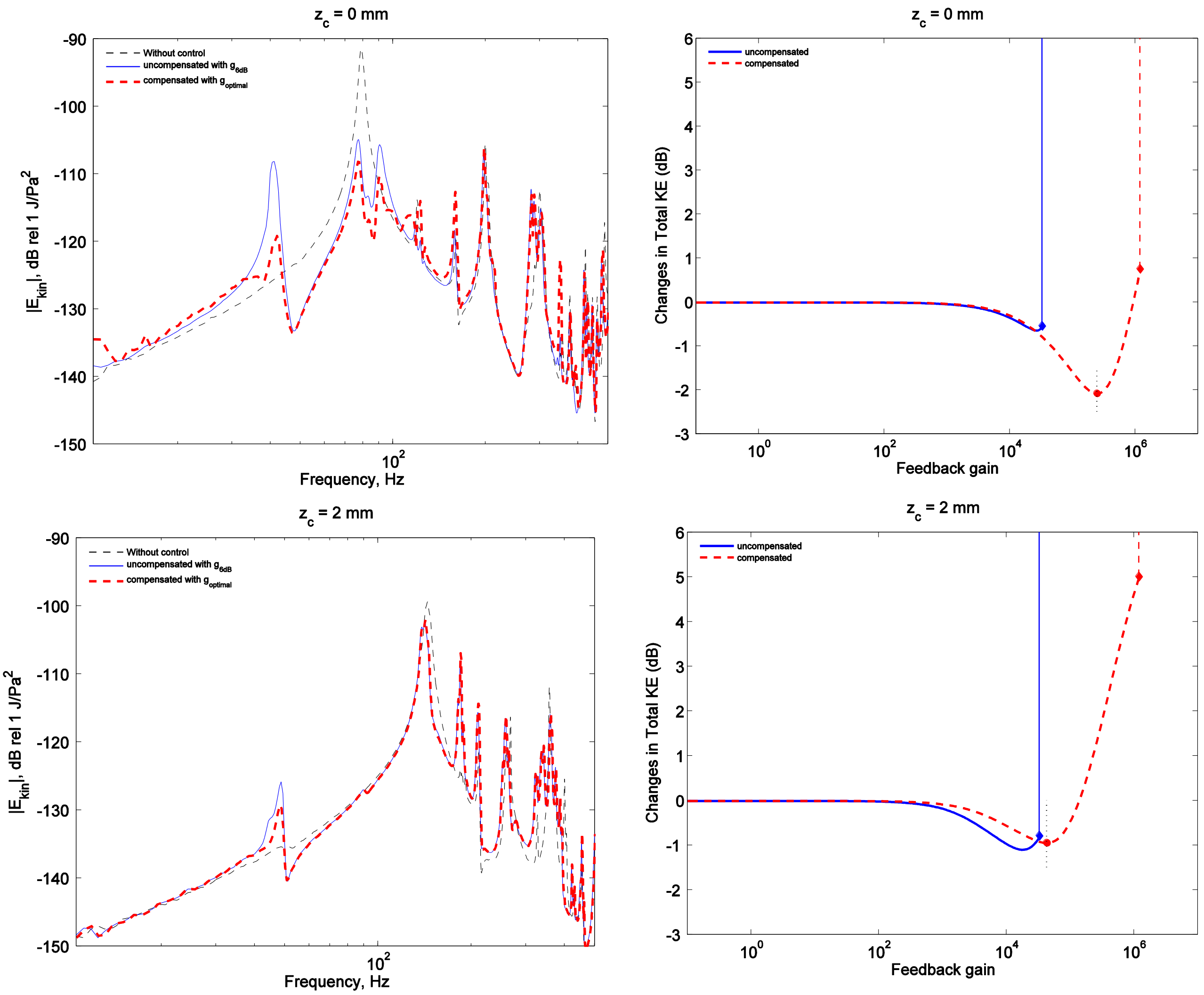

Figure 7. Kinetic energy (left) and optimal gain plots (right) for flat and curved panels: maximum stable gain (diamond) and optimal gain (circle)

For the flat panel, attenuation in the kinetic energy level of the first few modes is significant and the first peak shows an attenuation of 20 dB. In the case of the curved panel, even though the overall attenuation is not as high as with the flat panel, and the first peaks only show a 2 dB reduction, the feedback gain curve shows an optimal feedback gain well below the maximum stable gain.

5. Conclusions

Active vibration control of a doubly-curved structure has been investigated using a pressurised panel. As expected, the curvature significantly changes the natural frequency of the first few modes. A velocity feedback controller was developed using a proof-mass electrodynamic actuator and electronic compensator. The compensator allowed significant attenuation to be achieved on the flat panel, with a 15 dB gain margin, even though the natural frequency of the actuator was close to that of the first mode of the panel. When the panel was pressurised, although the natural frequency of the first panel mode was then well above that of the actuator, much less reduction was possible in the panel's global response. This is due to the fact that with the increase in curvature, several of the modes cluster together, making control difficult for a single channel system.

Despite the change in curvature, the stability of the system was not compromised and the compensator-actuator pair remained robust to the changes in the surface shape due to pressurisation. The optimal gain of the compensated system did not seem to be very sensitive to the increase in curvature and it remained more or less the same, in both flat and curved panel cases. This is because the compensator has been designed to only modify the physical properties of the actuator without needing to account for the dynamics of the panel.

Future work will involve online measurements of feedback velocity control using the designed actuator and compensator. The robustness of the controller will be further assessed using aluminium panels of increasing thickness, for both cases where the first structural mode is situated before and after the actuator's natural frequency. Finally, the possibility exists of self-tuning a decentralised multiple-channel feedback control, using the method cited in Zilletti et al.[10] This will allow more efficient control of the cluster of modes on the curved panel.

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