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Broadband Active Vibration Control of a Rectangular Flexible Wall of an Empty and a Water-filled Tank

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ABSTRACT: Broadband active vibration control of a rectangular plate representing a flexible wall of a thick Plexiglas container is experimentally investigated, for both empty and water-filled tanks, using five piezoelectric patches as control actuators. Preliminary dynamical analyses are carried out on the system in order to verify the influence of different water levels and different environment conditions (in particular, temperatures) on the system dynamics. Active vibration control is then experimentally applied to the plate with a single input–single output approach in the presence of broadband primary disturbance between 0 and 400 Hz, testing both filtered-x least mean square adaptive feedforward control and direct velocity feedback control. The time delay through the disturbance and control paths is measured.

Key Words: active control, PZT, plate, vibration, fluid, feedforward, velocity feedback.

INTRODUCTION

THE use of active control techniques for the suppression of vibrations of very light structures is a very important target in many applications, since it avoids the increment of the mass of the system due to the introduction of stiffeners or dampers. Moreover, aeronautical and space structures often present thin walls containing fuel, where the fluid–structure interaction must be considered. For these reasons, the study of active vibration control of a panel in air and totally or partially wet on one side by a liquid is of interest.

In the present study, broadband active vibration control of a rectangular plate, representing a flexible wall of a thick Plexiglas container, is experimentally investigated for both empty and water-filled tanks, using five piezoelectric (PZT) patches as control actuators.

The presence of water increases the modal mass of the system and free surface waves complicate the system dynamics, as reported by Amabili (2000). Morand and Ohayon (1995) explain in depth the effects of fluid–structure interactions on the dynamics of the structure. Habault and Filippi (2004) proposed a numerical method for computing resonance frequencies

and modes of a fluid-loaded rectangular clamped plate, showing that the difference between the modal shapes of a fluid-loaded or an *in vacuo* plate increases with damping.

Recently, the internal model control (IMC) algorithm has been experimentally applied by Carra et al. (2004b) to the studied system in the presence of different kinds of perturbation, with both single input–single output (SISO) and MultiSISO approach. The filtered-x least mean square (FXLMS) algorithm has also been used by Carra et al. (2004a) for reducing vibration in the presence of sinusoidal perturbations on the first three modes of the system.

The present article refers to the new tests that have been performed on the same system in the presence of broadband disturbance, using the FXLMS adaptive feedforward technique and the direct velocity feedback (DVF) control in SISO approach and linear regime.

Some preliminary dynamical analyses have been carried out on the studied system in order to verify the influence of different water levels and different environment conditions (in particular, temperatures) on the dynamics.

Active vibration control is then experimentally applied to the plate in the presence of broadband primary disturbance between 0 and 400 Hz, testing FXLMS adaptive feedforward control and DVF control. The secondary control input is generated by

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some PZT actuators that have already been used in several applications of active control on thin-walled structures, starting from Dimitriadis et al. (1991) for applications on two-dimensional structures.

In this work the positioning of the PZT patches is based on the knowledge of the natural modes of the plate and, particularly, on their deformation energy. Some accelerometers are positioned very close to the PZT patches, in order to obtain a nearly collocated system.

The optimization of the actuators' position is, in fact, of fundamental significance for the success of the control, as already investigated in literature. The differences between collocated and non-collocated control are deeply explained by Preumont (2002). Lee et al. (2003) also investigated the problem of the optimal relative position of piezoactuators and sensors. A modified optimization criterion for the locations of actuators and sensors on flexible structures was recently proposed by Bruant and Proslie (2005), who numerically investigated the case of the active control of a simply supported plate and of a slender uniform beam. Peng et al. (2005) also proposed a performance criterion for the optimization of the PZT actuators' locations on a thin cantilever plate, showing the consequent positive effects on control effectiveness by numerical simulations. The use of self-sensing actuators (where a single piece of PZT material is a sensor and actuator concurrently) has also been extensively investigated in the last few years, as carried out by Zhang et al. (2004), who experimentally controlled multi-mode vibrations of a clamped plate through a robust μ -synthesis controller, and by Qiu et al. (2005), who performed tests for reducing noise transmission using an advanced composite board with built-in PZT elements.

In the last few years, several studies have been dedicated to the control of fluid-loaded structures. Some of them specifically concern the active control of sound radiation, while the others also investigate the application of passive or active control on structural vibrations of fluid-loaded structures or fluid-filled tanks.

In particular, Lee and Park (1996) used a near-field approach to simulate the active control of sound radiation from a fluid-loaded rectangular plate, obtaining high global reductions both in radiated pressure and plate response. Saravanan et al. (2000) also investigated fluid-filled cylindrical shells with a constrained viscoelastic layer, by identifying the regions of energy dissipation concentration and then computationally testing possible redistributions of the viscoelastic material in order to damp the vibrations. Li and Zhao (2004) numerically studied the active control of both structural vibration and acoustic radiation of a fluid-loaded laminated plate with active damping control and active constrained

layer damping (ACLD). Active control of a fluid-loaded plate has also been recently investigated by Fein et al. (2005), who proposed a modal control technique, based on a modal state-space formulation, which has been experimentally applied on a plate in order to reduce vibrations on its first natural frequencies. In such a case, the modal parameters of the modes to be controlled have to be experimentally identified and incorporated into the model, in order to take into account the fluid-structure interaction.

In this study, FXLMS adaptive feedforward technique and DVF control are tested on a fluid-loaded plate.

The least mean square (LMS) adaptive feedforward algorithm, as introduced by Widrow and Stearns (1985), is applied to the system to be controlled, as suggested by Fuller et al. (1996), in the form of FXLMS control. Following the example of Viperman et al. (1993), FXLMS control is applied in the presence of broadband excitation; the time delay through the disturbance and control paths is also measured. Control experiments have been carried out by Ma (2003) on a rectangular clamped plate in air, using both a classic adaptive feedforward control and an adaptive nonlinear method for sinusoidal and swept-frequency harmonic excitations.

Regarding feedback control, DVF has recently been applied by Gardonio et al. (2004) on a smart panel in air, by using decentralized control units, obtaining reduction of both vibrations and sound radiation/transmission. Grewal et al. (2000) also experimentally applied multi-loop feedback control on a turboprop aircraft in order to reduce cabin noise and vibrations. Huang et al. (2000) realized experiments on a feedback control system for enhancing the low-frequency transmission loss of noise into the cabin of propeller-driven aircrafts or helicopters by using ultra-light panels that were driven with PZT inertial actuators.

The aim of the present research is to experimentally test the different active control techniques on a nearly clamped plate, both in the presence and in the absence of fluid-structure interaction, in order to reduce structural vibrations due to broadband disturbance.

DYNAMICAL ANALYSIS OF THE SYSTEM

The system to be controlled consists of a rectangular aluminum plate ($400 \times 283 \times 0.8$ mm), which is connected by bolts to the thick wall of a Plexiglas container, as shown in Figure 1.

The plate has boundary conditions close to those of clamped edges and it is positioned on the wall in an opening of dimension 350×233 mm, as shown in Figure 2, at a distance of 75 mm from the bottom of the tank. Its long edges are parallel to the tank bottom.

The Plexiglas walls are quite thick (30 mm) and they are reinforced with four stainless-steel bars at the opening, so they can be considered quite rigid in the low-frequency range. The tank has internal dimensions equal to $440 \times 350 \times 500$ mm.

The control components are constituted by five PZT patches ($25 \times 25 \times 0.2$ mm, from Physik Instrumente), glued on the dry surface of the plate, and five accelerometers (model M352C66) from PCB piezotronics. Every accelerometer has been positioned close to a PZT actuator, realizing a nearly collocated control system. Every couple accelerometer actuator represents a channel of the control system and is numbered as shown in Figure 3.

Figure 4 shows in detail the positions of both sensors and control actuators on the plate surface; some dimensions do not appear, since they can be deduced

from the symmetry. The location of a piezoelectric force transducer (glued to the plate in the excitation point) is also shown. The position of the PZT patches has been optimized by using a numerical calculation of the deformation energy of the plate for the first natural modes in the case of the empty tank. Calculations have been performed by using a computer program based on the Rayleigh–Ritz method developed by Amabili (2004).

Every actuator receives the command signal from a high-voltage power amplifier (model E-507.00, from Physik Instrumente), and it is constituted by a single ceramic layer delimited by two metallic CuNi electrodes, respectively connected to the high-voltage command signal and to the ground signal of the amplifier. The patches are fixed on the plate surface with an electro-conductive glue (Loctite 3888); all the plate surface is therefore at the ground voltage.



Figure 1. Experimental setup (in the case of the water-filled tank).

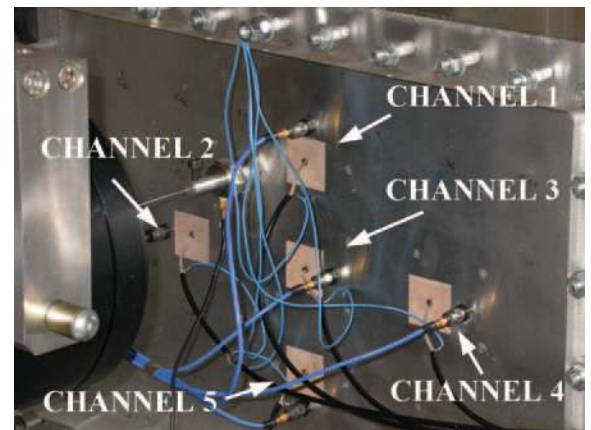


Figure 3. The control channels.

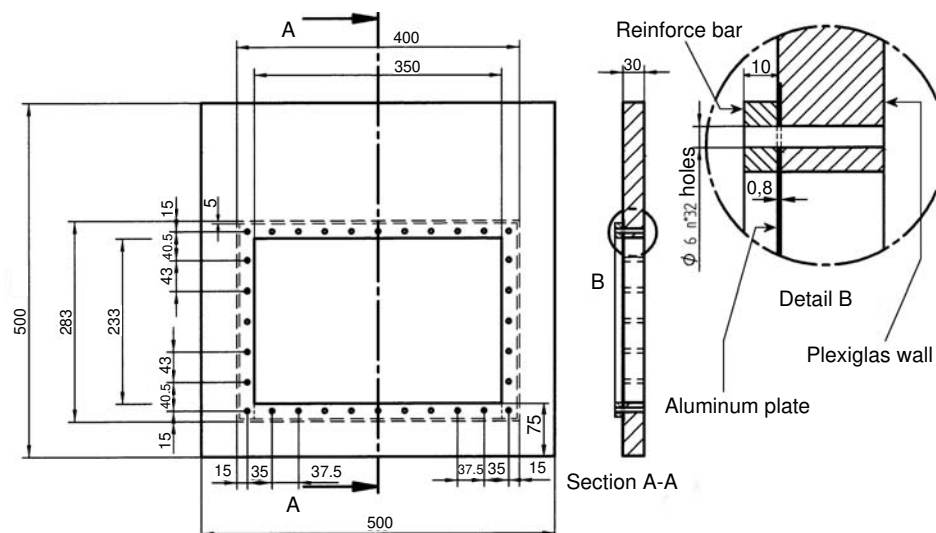


Figure 2. Geometry and dimensions of the assembly constituted by the front wall of the Plexiglas tank, the aluminum plate and the stainless-steel reinforce bars. Measures are expressed in millimeters.

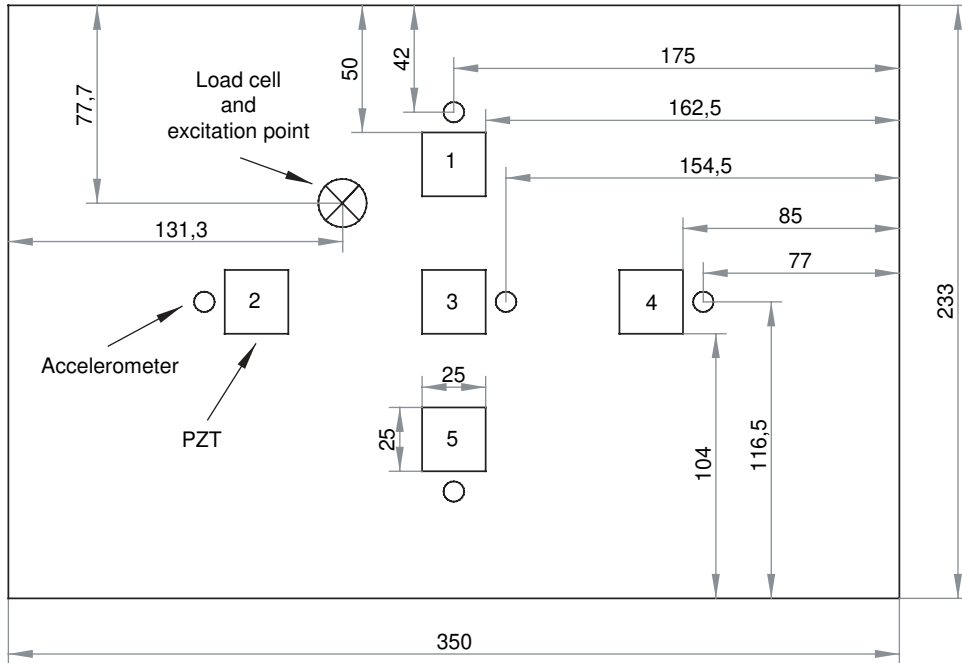


Figure 4. Location of sensors and control actuators on the plate surface. Measures are expressed in millimeters.

This kind of actuator has an in-plane deformation when subjected to a high voltage; the goal of active control is therefore to obtain a deformation of the PZT actuators that works exactly in opposition to the one produced by the vibrations of the plate surface. In particular, the used patches have a capacity of about 50 nF and are subjected to a contraction of $180 \mu\epsilon$ at -200 V, which is enough to realize control in linear field. However, these actuators were designed in order to work only between $+100$ V and -400 V; higher voltages can produce the depolarization of the patches. For this reason, a constant negative DC offset, equal to -200 V, is applied to the patches through the high-voltage amplifiers, which also multiply by a -100 gain the control signal produced by the digital controller. Keeping such a control signal between -2 and $+2$ V, it is therefore possible to obtain a global variation of $+200/-200$ V around the DC offset, without reaching the depolarization of the PZT patches.

The frequency response functions of the system have been measured in the absence and in the presence of the pre-compression of the actuators when they are subjected to the -200 V constant signal, but no important changes in the resonance peaks have emerged.

At first, a complete modal analysis, as reported in Carra et al. (2004a,b), has been carried out on the system, measuring the panel response on a grid of 9×7 points, both for an empty tank and with different levels of water, by using a sub-miniature accelerometer. The same sequence of modes has been observed before and after applying the control components on the plate

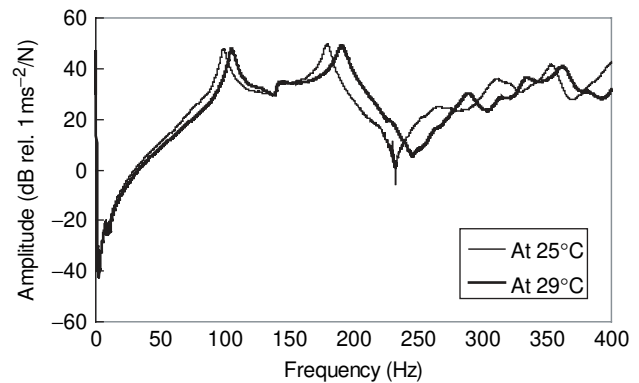


Figure 5. Measured frequency response function between error output (acceleration) and primary disturbance on Channel 3 at different temperatures (in the case of the empty tank). Measures refer to the same day.

surface. The variation of the fluid level in the tank produces a significant change of the natural frequencies of the plate: in particular, the higher is the water level, the smaller are the values of the natural frequencies.

Additional tests, realized in the laboratory of the University of Parma in the presence of different environment conditions, have shown that the studied structure is very sensitive to temperature and atmospheric pressure. In particular, the higher the temperature of the plate, the higher the resonance frequencies are, as shown in Figure 5 for Channel 3.

In the presence of water, the trend is the same, but frequency variations are smaller and slower because of the thermal inertia of the liquid.

The effect of temperature on natural frequencies is related to the boundary condition, which creates internal stress in the plate with the change of temperature.

Thus, the control techniques should take into account the extreme variability of the dynamics of the system with the environment conditions.

THE CONTROL SYSTEM

For every SISO channel of the complete control system, as shown in Figure 6, the acceleration signal is sent to a dSPACE real-time digital controller (one DS1104 processor Board at 250 MHz) passing through a 12-bit A/D converter. The controller generates the command input u for the correspondent nearly collocated PZT actuator: such a signal is reconstructed by a 16-bit D/A converter. Every actuator receives the command signal from a high-voltage power amplifier. Smoothing and antialiasing filters (eighth-order filters, with cut-off frequency at 400 Hz) have been used. The control algorithm has been implemented on the processor board working in a Matlab–Simulink environment.

Some amplification gains have been opportunely introduced into the control-loop in order to increase the ratio signal/noise.

The primary disturbance is given by an electro-dynamical shaker LDS V406, with an LDS PA100E power amplifier, that excites the plate in an eccentric position through a thin stinger with random excitation.

A PZT force transducer B&K 8200 is glued to the plate in the excitation point in order to measure the actual force given to the plate by the shaker.

The Control Algorithms

Active control is experimentally applied to the studied system in the presence of broadband disturbance using the FXLMS adaptive feedforward technique and the DVF control, in a SISO approach, for both an empty and a completely water-filled tank.

In SISO control, the acceleration signal measured by each sensor is influenced only by the control action of the corresponding collocated PZT actuator. The other patches are connected to the power amplifiers, in order to keep into account the effect of their constant pre-compression, but they are not transmitting any additional secondary control input to the plate.

In the case of FXLMS control, a reference signal is derived from the primary excitation and it is used directly to drive the secondary control input (Fuller et al., 1996). Theoretically, in the absence of noise, this technique could bring a perfect cancellation of the primary disturbance. In real cases, because of the continuous changes in the excitation signal and in the primary path, it is necessary that the frequency response of the controller is adjusted with time, using an adaptive technique. The time delay on the control path should also be small enough to guarantee the causality of the control system (Kuo and Morgan, 1996).

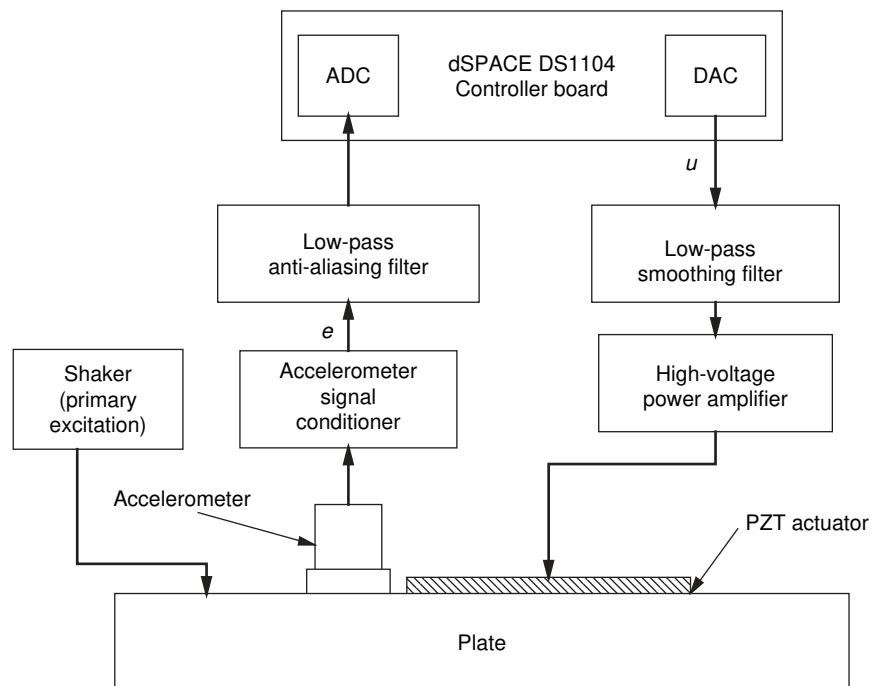


Figure 6. Schematization of the control system hardware for a single SISO channel.

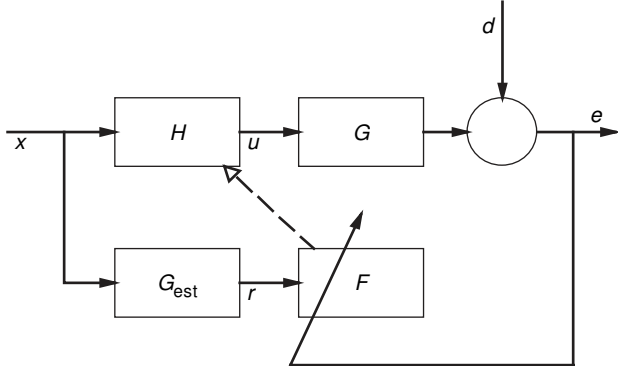


Figure 7. Block diagram of the FXLMS algorithm.

The model of a practical implementation of a single-channel feedforward control with FXLMS adaptive algorithm is shown in Figure 7, where x is the primary excitation, d is the response of the mechanical system in the absence of control, u is the secondary control input and e is the error signal (that is, the measured acceleration signal).

The controller is represented by H , that is, a finite impulse response (FIR) filter that acts on the primary disturbance x in order to generate the control secondary input u . The coefficients of such a filter must be continuously changed with time in order to have optimal control performances. For this reason, the cost function J must be minimized at every sample time n :

$$J = e^2(n). \quad (1)$$

Such minimization is realized by the LMS adaptive filter F that calculates the coefficients w_i of the H filter

$$w_i(n+1) = w_i(n) - \mu \frac{\partial J}{\partial w_i(n)} \quad (2)$$

where $w_i(n)$ represents the i th controller coefficient at the n th sample time and μ is a convergence parameter.

Following Fuller et al. (1996), Equation (2) can be rewritten as

$$w_i(n+1) = w_i(n) - 2\mu e(n)r(n-i). \quad (3)$$

In order to obtain the so-called *filtered reference signal* r , the primary disturbance x must be filtered with an appropriate filter G_{est} , which represents an estimated model of the secondary control path G (transfer function between the sensors and the PZT patches in the absence of any other external perturbation). In the present study, a preliminary off-line identification of G_{est} as a FIR filter is realized.

Other tests are performed on the studied system by applying the DVF technique, where the velocity signal v from the sensors is multiplied by an opportune feedback gain k and is sent back to drive the secondary PZT actuators with changed sign (Figure 8). For each

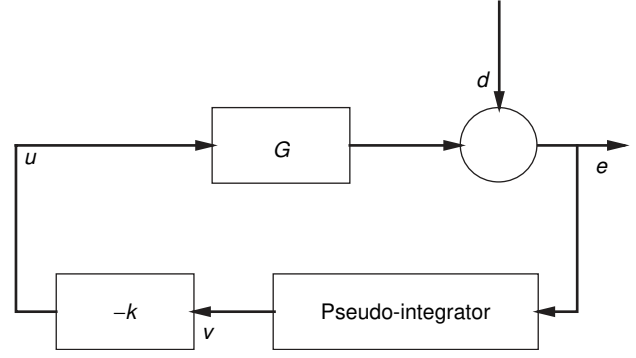


Figure 8. Block diagram of the DVF control algorithm.

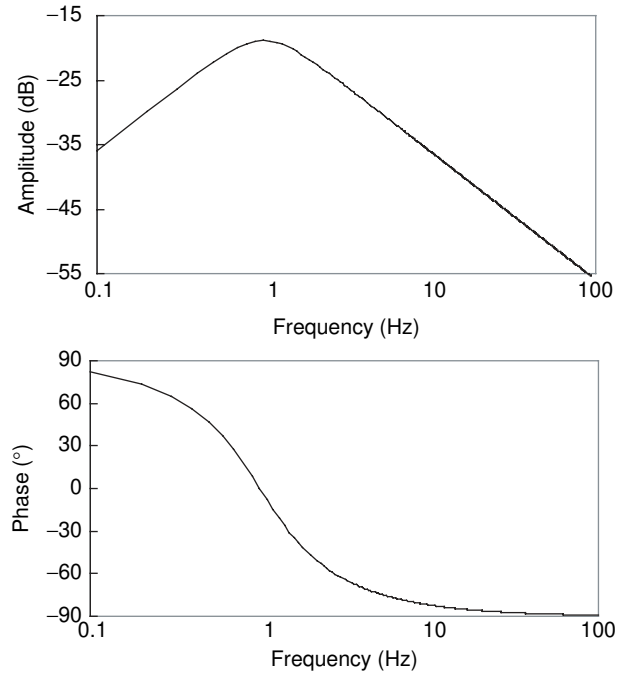


Figure 9. Amplitude and phase of the implemented pseudo-integrator for DVF active control.

channel, the maximum value of k , which allows the best control performances without reaching instability, is found out by comparing the results of some preliminary experimental tests and is then adopted for the final control test.

In this study, accelerations are measured by sensors as error signals, therefore a digital integration is necessary in order to obtain velocities. For this reason, a pseudo-integrator with a cut-off frequency at 1 Hz has been implemented on the digital controller, using a sampling frequency of 30 kHz. Such a solution allows the simulation of the behavior of a perfect integrator starting from about 10 Hz (Figure 9). The practical implementation of an ideal integrator is in fact impossible, since the integration of any

DC value (that is, the signal offset) would provide infinite values.

Preliminary Control Tests

Some preliminary experimental tests have been performed in the presence of broadband primary disturbance, in order to find out the best control parameters and the optimal measurement setup for the used control algorithms.

The level of the primary disturbance force has been chosen high enough to guarantee a good signal/noise ratio, and therefore a good coherence of the measurements, but also low enough to avoid reaching large amplitude vibrations (nonlinear behavior).

The propagation times between the primary disturbance (as measured by the load cell) and the error output (primary path), and between the secondary control input and the error output (secondary control path) have been measured by computing the cross-correlation functions across each path in the presence of white-noise excitation. In particular, on every control channel, white-noise propagates along the primary path passing through the load cell, the plate, and the accelerometer. The secondary path is instead constituted by an antialiasing filter, A/D converter, digital controller, D/A converter, smoothing filter, high-voltage power amplifier, PZT patch, plate, and accelerometer.

Table 1 shows the measured values of the propagation times in the presence and in the absence of water in the tank. The propagation times on the primary path are different on each control channel (especially in the presence of water), probably because of the different relative positions between the accelerometers and the excitation point. The propagation times on the secondary path are instead quite constant, since they are measured between nearly collocated components.

In order to realize a preliminary off-line identification of control path G , three different measurement conditions have been compared: (i) shaker mechanically

unconnected from the plate (with load cell still glued on the plate), (ii) shaker mechanically connected to the plate but not active, and (iii) shaker mechanically connected to the plate and active.

In case (iii), under the hypothesis of the linear system, the measured total acceleration can be considered as the sum of the acceleration given by one PZT patch and the one given by the shaker. Modelling the primary path with an opportune FIR filter, the acceleration due to the control path can therefore be obtained as the difference between the total error signal and the primary path acceleration component. G is then simply obtained as frequency response function (FRF) between such an acceleration component and the disturbance given by the PZT patch.

Results show that in case (ii), where the measurement is influenced by all the additional mass and damping of the shaker, the structure does not show exactly the same resonance poles that it has when it is excited by the shaker. Consequently, control results are affected by the possible effects of the shaker on the system dynamics, especially in the case of the empty tank.

Function G as calculated in case (iii) seems to be very similar to case (i) at low frequencies and it starts to follow case (ii) from about 220 Hz (Figure 10). Experimental comparisons between shaker excitation and hammer excitation of the plate have confirmed that after about 220 Hz the measurement is quite influenced by the presence of the shaker.

For these reasons, method (iii) has been adopted to realize the off-line identification of G .

The calculated impulse responses of G show that the water-coupled system needs a G_{est} filter with much more coefficients (450 in the case of an empty tank, 1500 coefficients for the water-filled tank) in order to avoid excessive phase differences between the identified and the real control system and thus the instabilities of the controlled system (Kuo and Morgan, 1996). This means that the digital realization of the feedforward controller is much heavier, from a computational point of view, when the tank is water-filled.

Table 1. Measured delays on primary and secondary paths (sampling frequency 2000 Hz).

	Empty tank				Water-filled tank			
	Primary path delay		Secondary path delay		Primary path delay		Secondary path delay	
	(10^{-3} s)	(Samples)	(10^{-3} s)	(Samples)	(10^{-3} s)	(Samples)	(10^{-3} s)	(Samples)
Channel 1	9	18	5.5	11	2.5	5	5.5	11
Channel 2	9	18	5.5	11	3	6	5.5	11
Channel 3	8.5	17	5.5	11	11	22	5.5	11
Channel 4	9	18	5.5	11	6	12	6	12
Channel 5	8	16	5.5	11	18.5	37	5.5	11

EXPERIMENTAL ACTIVE CONTROL RESULTS

In the past, as reported by Carra et al. (2004a), good reductions (up to about 45 dB for the second mode in case of an empty tank) have been obtained by

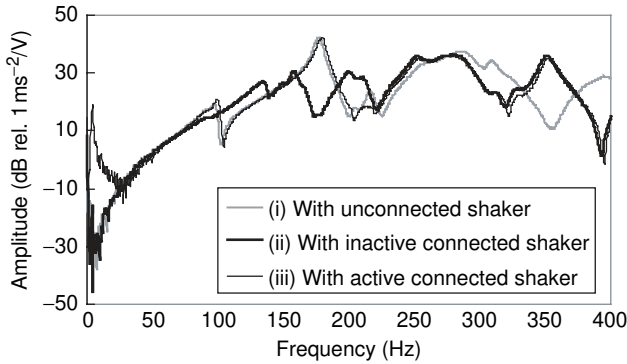


Figure 10. Measured frequency response function between error output and control input on Channel 4 for different setup configurations (in the case of the empty tank).

the authors on the studied system in the presence of sinusoidal excitation using FXLMS control. Upon the introduction of the control action gives lower effects on the first three vibration modes. Similar results were obtained by using IMC control (Carra et al., 2004b), where tests were extended to the case of a MultiSISO approach and broadband control.

In the present study, several experimental control tests have been performed in the presence of a primary broadband excitation between 0 and 400 Hz, for both an empty and a water-filled tank. The FXLMS technique and the DVF control have been applied in the SISO approach.

In the case of feedforward control, the force measured by the load cell, which is glued on the plate surface in the excitation point is adopted as the reference signal (Fuller et al., 1996). Its spectrum and amplitude (about ± 1.5 N) are shown in Figure 11 for the case of an empty tank. Similar graphs can be obtained in the case of the water-filled tank, where the spectrum moves towards lower frequencies and the amplitude

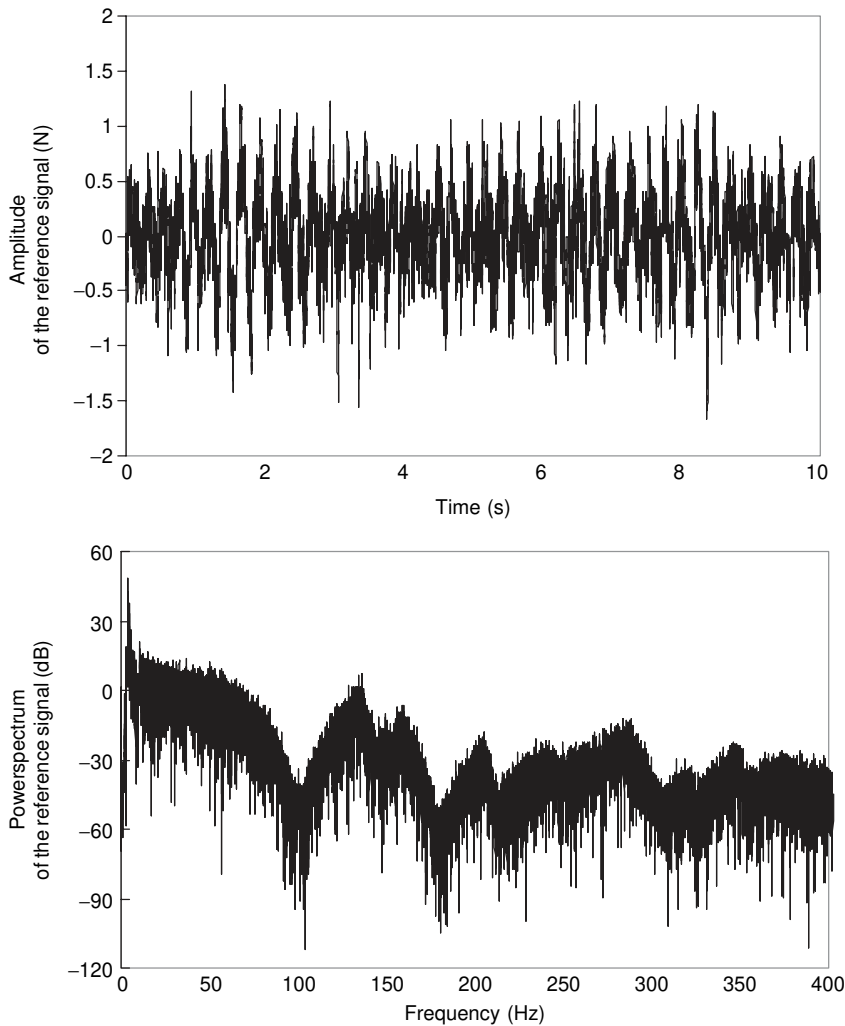


Figure 11. Amplitude and powerspectrum of the reference signal for FXLMS control, measured in the case of the empty tank.

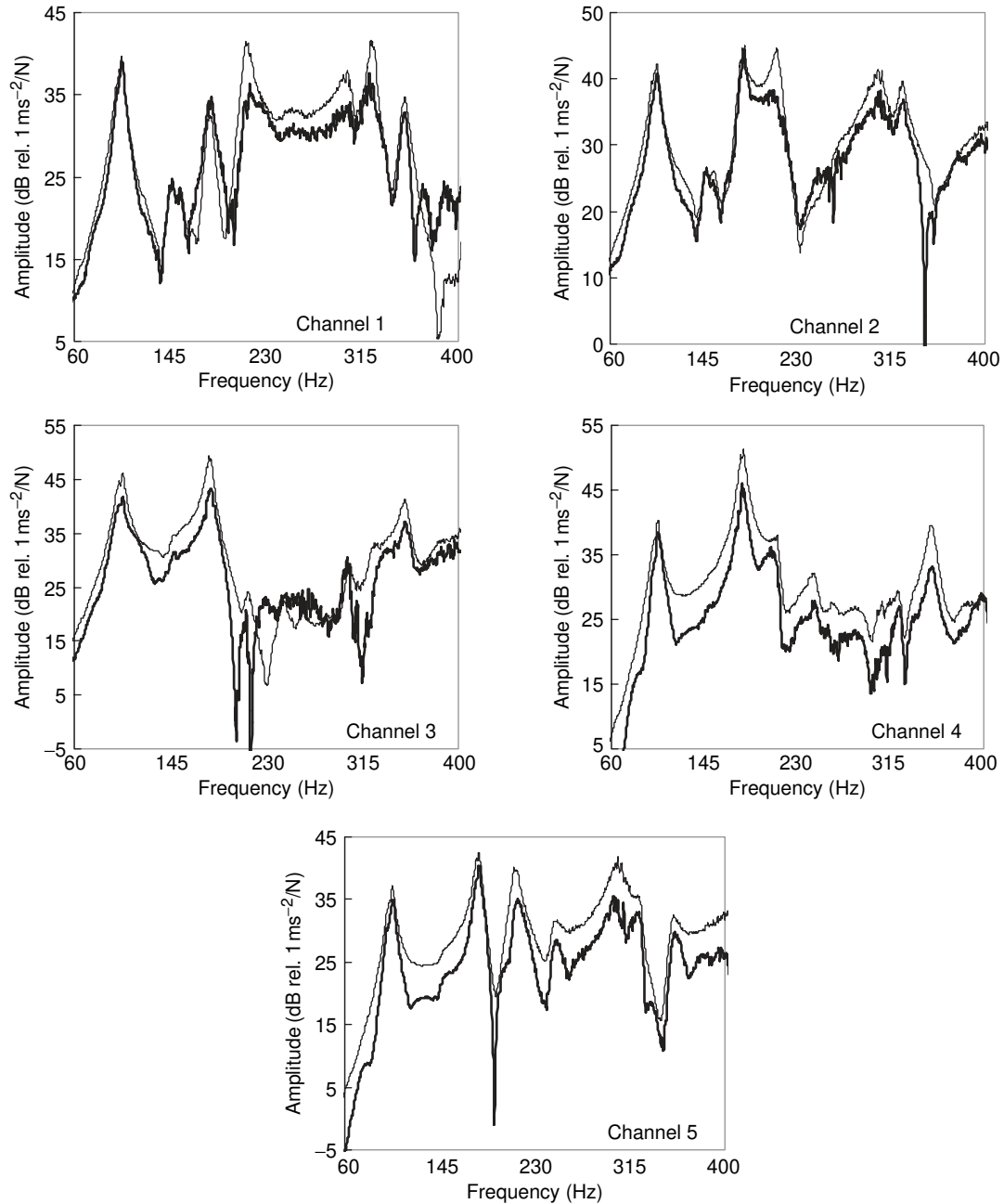


Figure 12. Measured frequency response function between error output and primary disturbance on all the control channels in the case of the empty tank: without control (thin line), in the presence of FXLMS control (thick line).

is about ± 3 N. It can be observed that the flat random signal between 0 and 400 Hz that has been sent to the shaker is influenced, when it reaches the plate surface, by the dynamics of the structure, as one would expect in real cases.

An LMS filter with 80 and 200 coefficients and a G_{est} model with 450 and 1500 coefficients have been used in the case of the empty and the water-filled tank, respectively, using in both cases a sampling frequency of 2000 Hz. The lengths of such FIR filters have been chosen by experimentally comparing the control performances obtained by using a different

number of coefficients. The convergence parameter μ , as defined by Widrow and Stearns (1985), has been adopted as a fifth of its theoretical optimal value, in order to guarantee both control effectiveness and stability.

Results show that, in the case of the empty tank, the FXLMS control can be effective in both resonance and non-resonance zones, as shown in Figure 12. A satisfactory overall reduction of vibrations on almost all of the studied frequency range is also obtained on Channels 4 and 5. This kind of result is different from the one that can be typically reached by using

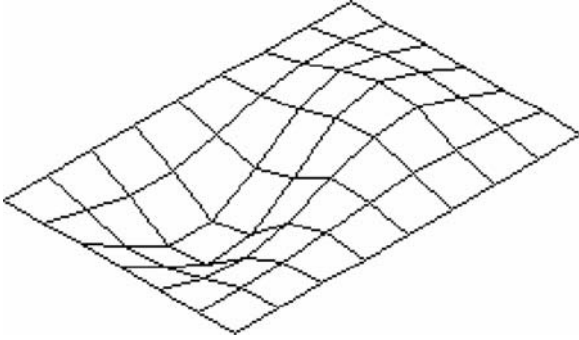


Figure 13. Experimentally identified modal shape ($m=2$, $n=1$) for the second mode in order of frequency, at about 180 Hz (in the case of the empty tank). Integers (m , n) identify the mode shape: m = number of half-waves along the long edge; n = number of half-waves along the short edge.

Table 2. Experimental mean square error (MSE) reductions (measured on the acceleration signal) in the presence of FXLMS control.

	Empty tank MSE reduction (dB)	Water-filled tank MSE reduction (dB)
Channel 1	-4.8	-8.2
Channel 2	-6.0	-8.0
Channel 3	-7.7	-6.1
Channel 4	-7.4	-10.1
Channel 5	-9.6	-11.0

feedback approaches as DVF control, which produces, in fact, active damping (and therefore a reduction of the structural response) only on the resonance peaks.

In particular, this work investigates the effectiveness of active control in reducing the amplitude of the low frequency resonance peaks of the plate. Each PZT patch is, in fact, located on an anti-node (position of maximum oscillation) of one of the first three modal shapes of the plate, so it is expected to optimally control it. A particular effectiveness of the optimally located actuators for each corresponding modal shape can therefore be observed on most control channels. For example, as expected, even if the reductions are quite well-distributed on all of the studied frequency range, the control effectiveness of Channel 4 on the second vibration mode at about 180 Hz is higher than the one obtained on the first and the third peak (Figure 12). This mode is in fact characterized by a modal shape with two half-waves along the long edge, i.e. mode (2,1) shown in Figure 13, and Channel 4 is located on one of its anti-nodes.

Significant reductions of the mean square error (that is the mean square value of the acceleration signal) are obtained on all the control channels in the case of an empty tank, as reported in Table 2, with a maximum attenuation of 9.6 dB of the acceleration

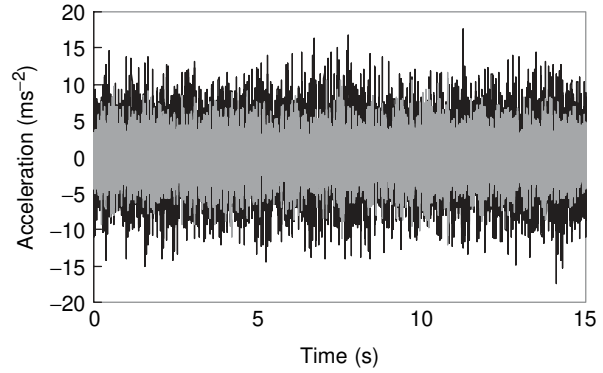


Figure 14. Measured acceleration signal on Channel 4 in the case of the empty tank: without control (—), in the presence of FXLMS control (grey 40%).

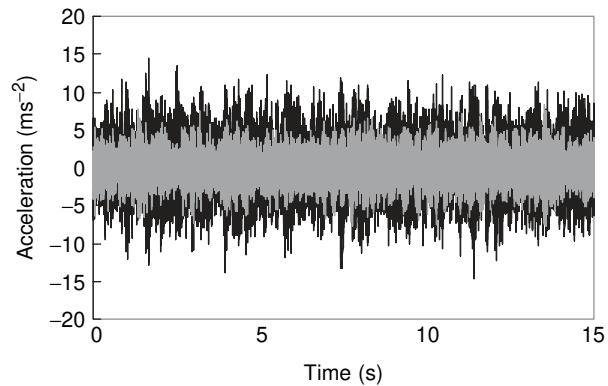


Figure 15. Measured acceleration signal on Channel 4 in the case of the water-filled tank: without control (—), in the presence of FXLMS control (grey 40%).

signal on Channel 5. Figure 14 shows the reduction on Channel 4.

With the introduction of water in the tank, significant reductions of the mean square error (up to 11 dB on Channel 5, as reported in Table 2) are still obtained. Figure 15 shows the reduction on Channel 4. Active control is anyway less effective than in the case of the absence of water at frequencies below about 35 Hz, as shown in Figure 16 for all control channels. The graphs do not report all the studied frequency range, in order to give more legibility to the first resonance peaks. The reduced effectiveness could be due in general to the increased modal mass of the system. Moreover, the location of the PZT patches has been optimized only considering the case of the empty tank. Figure 17 shows, in the case of Channel 4, that better results can be obtained at higher frequencies, where reductions are more uniformly distributed on both resonance and non-resonance zones.

In all the tests, the control signal sent to the PZT actuators stays inside the maximum acceptable operative range (about ± 200 V) to prevent depolarization of the PZT material. As an example, Figure 18 shows the transient control signal u during the first 10 seconds

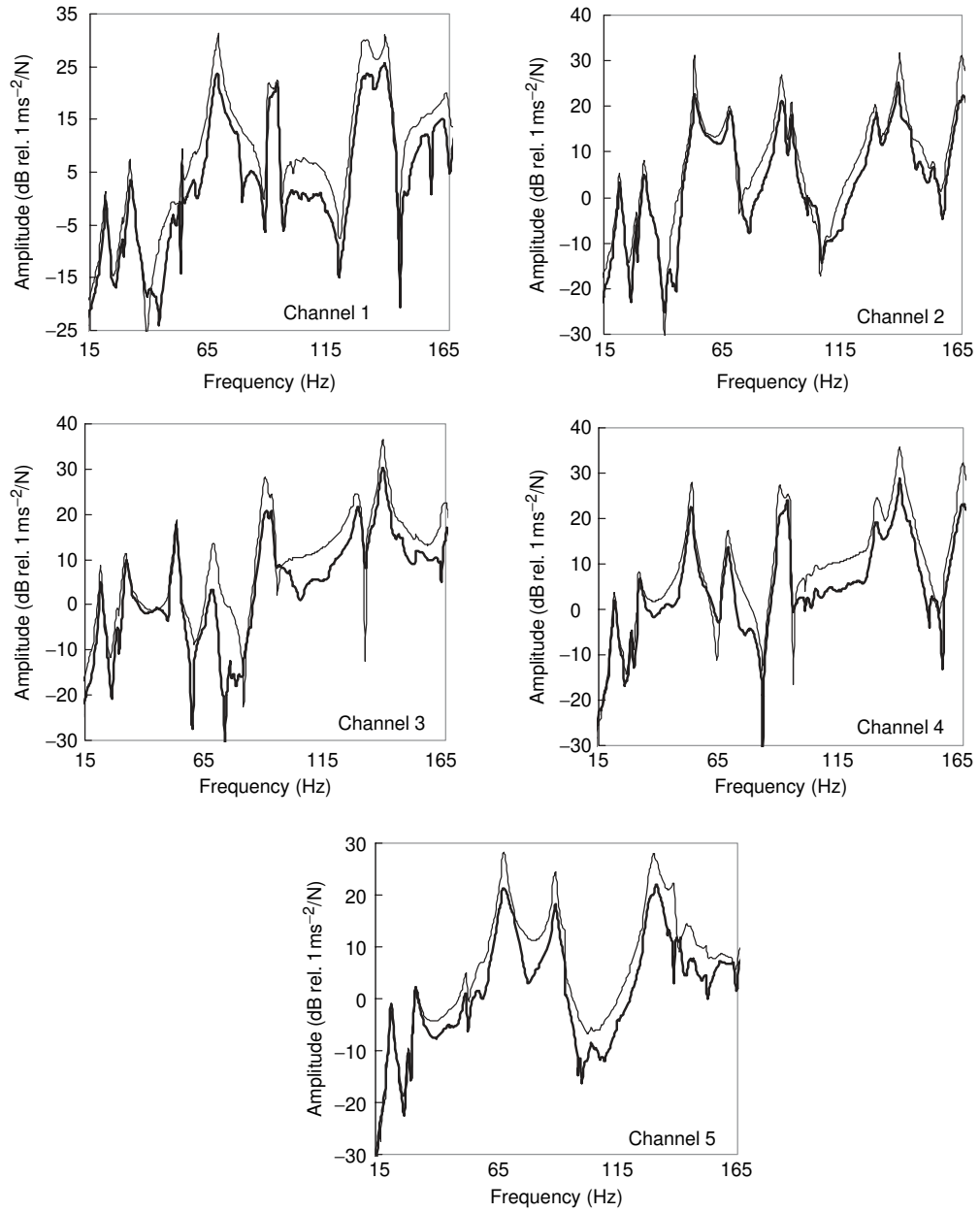


Figure 16. Measured frequency response function between error output and primary disturbance on all the control channels in case of water-filled tank: without control (thin line), in the presence of FXLMS control (thick line).

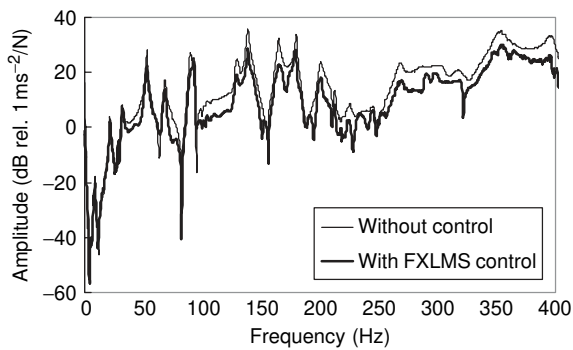


Figure 17. Control effectiveness (FXLMS algorithm) on the measured frequency response function between error output and primary disturbance on Channel 4 (in the case of the water-filled tank).

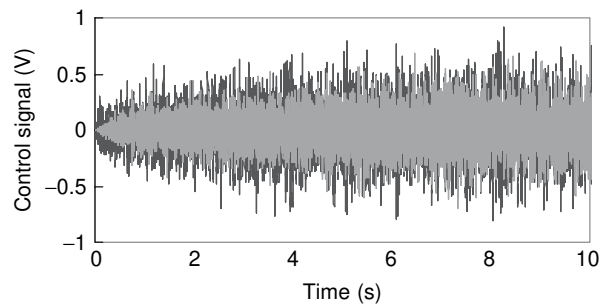


Figure 18. Measured transient control signal u on Channel 4 during the first 10s of FXLMS control: in the case of the empty tank (\rightarrow), in the case of the water-filled tank (grey 40%).

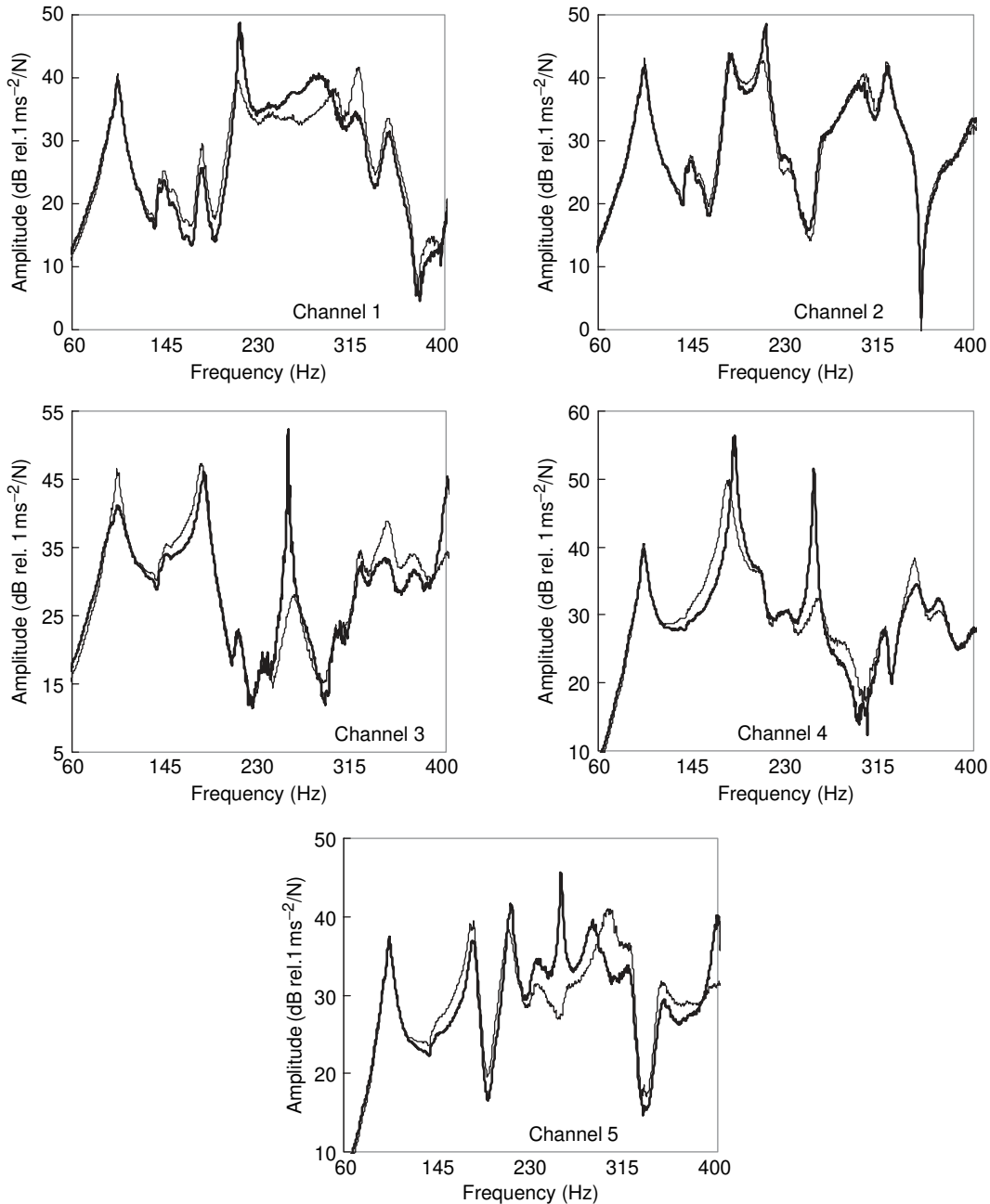


Figure 19. Measured frequency response function between error output and primary disturbance on all the control channels in the case of the empty tank: without control (thin line), in the presence of DVF control (thick line).

of FXLMS control on Channel 4; the graph refers to the signal as generated by the controller, before the amplification (gain -100) given by the high-voltage amplifier. The progressive increment of the control signal during the adaptation of the algorithm can be observed; in the case of a water-filled tank, such adaptation seems to be a little bit slower.

Applying DVF control, a small active damping effect can be observed on some resonance peaks in the case of the empty tank, as shown in Figure 19. A particular

effectiveness of the optimally located actuators for each corresponding modal shape can be observed on some control channels. For example, in the case of Channel 3, reductions are concentrated on the first peak, at about 105 Hz, corresponding to mode (1,1) (one half-wave along both long and short edge). Patch 3 is in fact the optimally located actuator for controlling this modal shape, since it is located in the center of the plate, on the anti-node of mode (1,1). The control effectiveness on the mean square error is anyway not satisfactory, since some amplified peaks also appear.

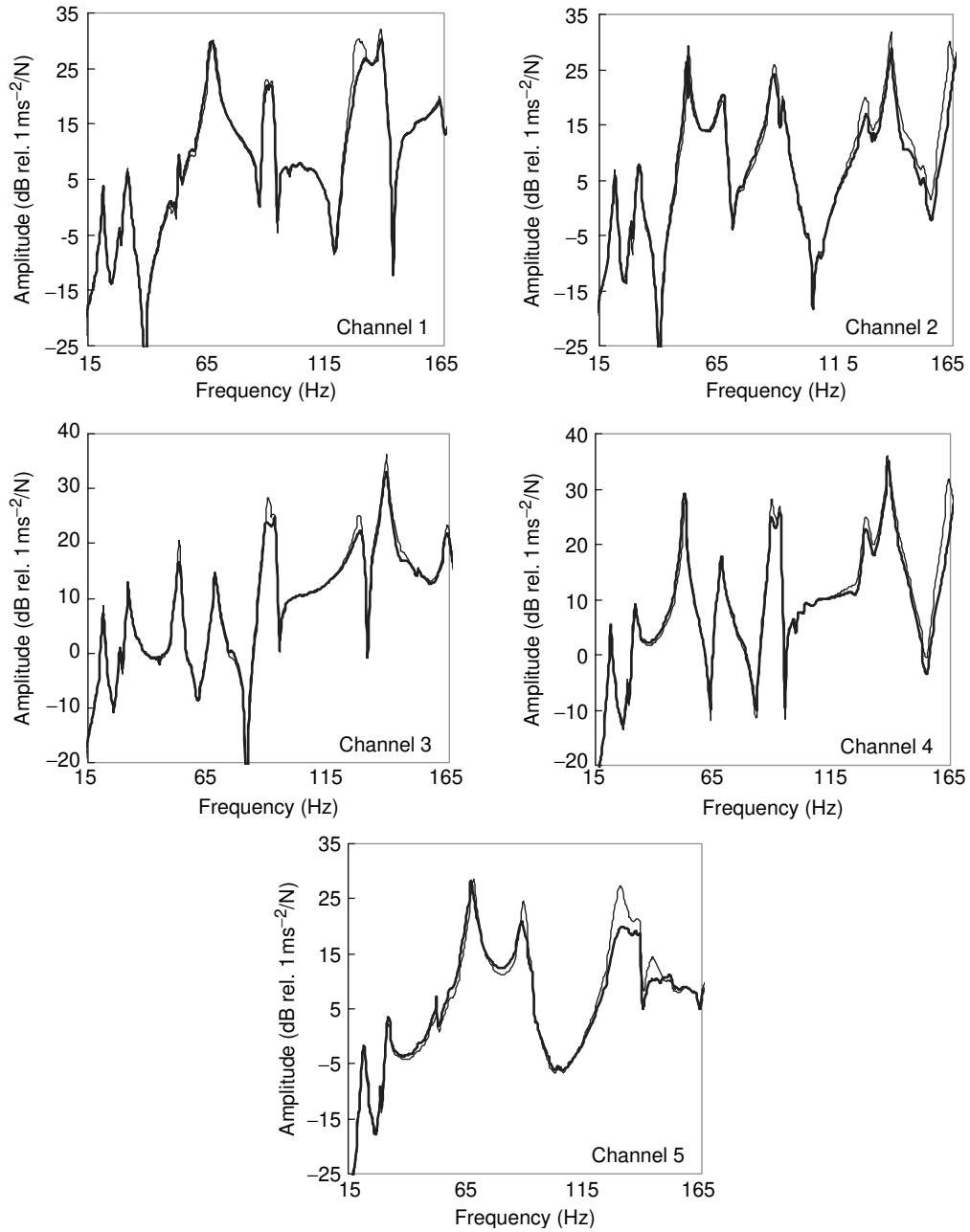


Figure 20. Measured frequency response function between error output and primary disturbance on all the control channels in the case of the water-filled tank: without control (thin line), in the presence of the DVF control (thick line).

The acceleration signal is therefore globally amplified by the control action.

Moreover, the experimental tests have shown that no very high values of control gain k can be adopted without reaching instability, therefore damping effects cannot be very significant. This problem can also be due to the non-perfect collocation of the sensor/actuator pairs. Different locations of accelerometers and PZT patches could be tested in future in order to improve the results. The introduction of classical compensator circuits (Preumont, 2002) could also be advisable.

Similarly, in the case of a water-filled tank (Figure 20), only some small reductions have been observed, and no significant DVF control effectiveness appears under about 35 Hz, as in the case of FXLMS control. The graphs in Figure 20 do not report all the studied frequency range in order to give more legibility to the first resonance peaks; anyway it can be observed that no amplification effect appears up to 165 Hz. Figure 21 shows instead, in the case of Channel 3, the control effectiveness at higher frequencies, where some peaks are amplified by the control action.

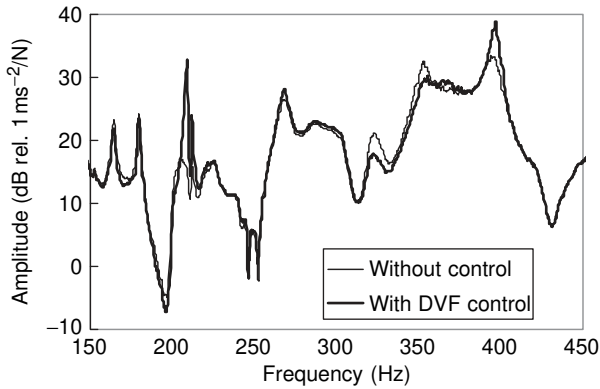


Figure 21. Control effectiveness (DVF algorithm) on the measured frequency response function between error output and primary disturbance on Channel 3 (in the case of the water-filled tank).

CONCLUSIONS

Broadband active vibration control of a rectangular plate representing a flexible wall of a thick Plexiglas container is experimentally investigated, for both the empty and the water-filled tank, using five PZT patches as control actuators.

Modal analyses have shown that the system dynamics is influenced by the environment conditions (particularly temperatures) and that the natural frequencies are significantly reduced by the introduction of water in the tank.

Active control has been experimentally applied on the plate in linear field with a SISO approach in the presence of broadband primary disturbance between 0 and 400 Hz, testing FXLMS adaptive feedforward control and DVF control. In particular, active control of the first three vibration modes of the plate, which can be theoretically optimally controlled with the adopted actuators' location, has been investigated.

In the case of feedforward control, satisfactory results have been obtained (with an overall control effectiveness on both resonance and non-resonance zones) for both a plate in air or in contact with water, even if, in the presence of fluid–structure interaction, no significant vibration reduction has been observed under about 35 Hz. A particular effectiveness of the optimally located actuators for each corresponding modal shape can be observed on most control channels. The delays on primary and secondary paths have been evaluated.

The DVF control has generally produced a small vibration damping on the first resonance peaks but some amplifications also appear. In particular, no control effectiveness has been observed at very low frequencies in the presence of fluid–structure interaction, as in the case of FXLMS control. Results could be improved

in the future by introducing classical compensators and by changing the relative position between actuators and sensors in order to improve the collocation of the system. A further optimization of the values of the control gains would also be advisable in order to reduce the amplification effects.

The absence of control effectiveness of both algorithms on the first resonance peaks in the case of the water-filled tank can be due to the limited power of the actuators. Results could be improved in future by optimizing the control effort required for the PZT patches.

Different excitation methods, based on non-contact external forces (such as acoustic loads) could also be tested, in order to avoid the observed influence of the actual exciter (shaker) on the plate dynamics.

The realization of finite-elements models of the system is already in progress in order to evaluate the effects of the tank and of the hydrostatic pressure on the dynamics of the plate. Calculations could also be useful in order to realize further optimizations of number, locations and dimensions of the PZT patches, especially in the case of the water-filled tank.

Finally, the FXLMS and DVF algorithms will be experimentally applied to the plate also in a MultiSISO approach in the near future, as often required in practical and industrial applications of the active control of plates.

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