FIRST NUMERICAL AND EXPERIMENTAL RESULTS ON ACTIVE CONTROLLED GLAZED FACADES

B. Naticchia, A. Carbonari
Academic Centre "PCQ – Progetto Costruzione Qualità"
Polytechnic University of Marche - Engineering Faculty
Via Brecce Bianche, 60131 Ancona (Italy)
E-mail addresses: b.naticchia@univpm.it; alessandro.carbonari@univpm.it .

Abstract: Both recent legislation codes and the evidence of the effects of high noise levels on human health have directed research activities towards the problem of the improvement of Sound Transmission Loss (STL) of glazed facades in the frequency range (low-medium) where passive means were already shown to be not effective. Adopting active control systems may help to cover the lower set of frequencies, where traffic and railway noise produces high level of disturbance. In this paper it is shown how Piezoelectric (PZT) stack actuators may be installed on large curtain walls, improving their STL. They are demonstrated to be particularly useful to avoid drops of STL in correspondence of resonance frequencies of windows, hence increasing their overall STL. Some simulations on the noise reduction obtained inside a standard building enclosure were performed, comparing the noise transmitted by a standard window with the corresponding active controlled one and several technologies for the installation of PZT actuators are suggested.

Keywords: ASAC system, Sound Transmission Loss, piezoceramic stack actuators, feedback control.

1. INTRODUCTION

This work is the direct prosecution of the preliminary analyses on the integration of an automated active control system in building glazed facades that was presented in ref. [1], aimed at improving Sound Transmission Loss (STL) of glazed facades.

The 89/106/CEE European Directive, has made protection against noise a compulsory requirement for buildings. As a consequence, the importance of a good design of building envelopes, which could provide the required acoustic comfort, is increased and stressed also by national legislations on building construction. In fact, glazed facades are the preferred path followed by disturbing noise from the exterior to the interior and are often not able to respect the strict limits imposed by standards and regulations, in spite of the adoption of very expensive passive means. The two main types of passive means for improving Sound Transmission Loss (STL) presently utilized are laminated glass technology and double glazing [2]. Both of them can be useful for reducing noise transmission at high frequencies as depicted in Fig. 1. The first critical point for a single glass panel is in correspondence of the resonance effect (f_0), registered at very low acoustic frequencies: the low damping of glass let it vibrate with relatively high amplitudes. Above this value its STL increases with frequency, until the critical effect causes another drop of STL (f_c): the matching between the wavelength of flexural vibrations propagating through the glass and the projection of the disturbing noise wavelength, dramatically increases the overall radiation efficiency of the glass panel. The laminated solution, through the interposition of PVB layers between glass panels which decrease the overall flexural stiffness of laminated panels and shift up the corresponding critical frequency, is used to shift the coincidence effect at frequencies higher than the audible range (f'_c), in fact improving STL values in the range of acoustic waves higher than 1500 Hz, while determining no improvements at lower frequencies. On the other hand, in double glazing the coupling between the glass panels and the air layer adds another resonance effect at frequencies lower than 500 Hz (f'_0), while improving their STL in the range of frequencies limited between the resonance and coincidence effects.

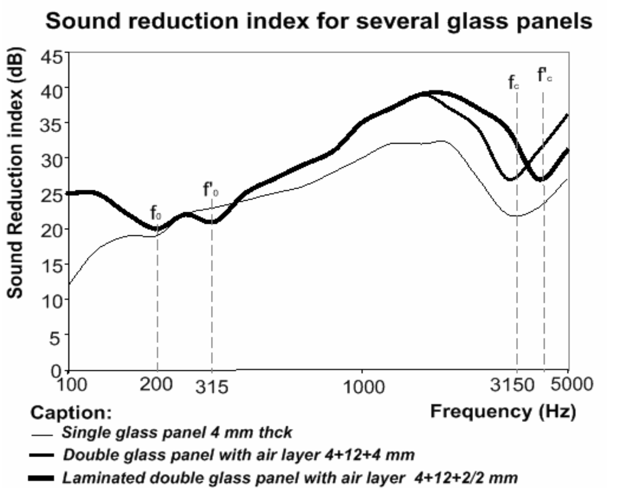


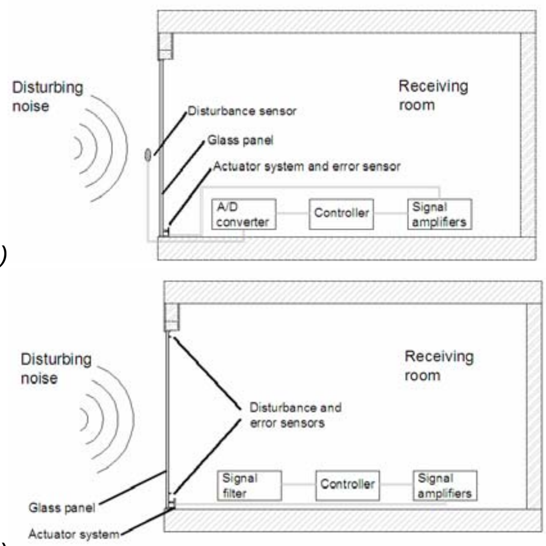
Fig. 1. Experimental STL for a particular set of single, double and laminated glass panels.

It could be shown, through the application of those calculation methods regulated by the European standard series EN 12354:2000, that when a glazed element having a sound reduction index sensibly lower than opaque walls is installed on a building façade, it acts as such a preferred path for the sound from the exterior to the interior that improving the opaque envelope's sound reduction index does not improve the overall STL of the wall, which in that case depends uniquely on the window's performances. In other words, the weakest element of a façade from the acoustic insulation point of view, determines the highest STL value that the whole facade is able to assume. In this paper, the adoption of an active control system is suggested to prevent the drop of acoustic performances of glass panels at low frequencies, showing how it is capable of controlling the vibrations of glazed panels and reducing the sound radiated as a consequence. Presently two main approaches are known for active control of sound [3]: Active Noise Control (ANC), where secondary waves interfere destructively with the disturbing noise, and Active Structural Acoustic Control (ASAC), when the source of noise is actively controlled through the reduction or modification of its vibration field, which is applied for the purpose of this paper. This paper is organized as follows: section 2 describes the automated active control system proposed for STL improvement of buildings facades; section 3 focuses on its technological feasibility; section 4 is the core of this paper and presents results obtained from experiments and numerical simulations; section 5 concludes.

2. ACTIVE CONTROL THROUGH ASAC SYSTEMS

The first type of active control system to be tested was the ANC system (Active Noise Control), that needs at least the following components: control loudspeakers to produce secondary waves, which are driven by a controller or a set of controllers, which in turn need error sensors to manage transient disturbances; one or more microphones for monitoring disturbing noise, to be located in proximity of the disturbance source (in case of feed-forward ANC) or of the disturbed receivers (in case of a feedback ANC arrangement). Therefore, when applied to buildings' glazed facades, ANC system would need an external microphone for disturbance monitoring, and internal error sensors and loudspeakers for control purposes. Therefore the ASAC (Active Structural Acoustic Control) system is preferred by the authors, because both reference sensors and actuators may be placed on the glazed panels as source of interior noise, which would interfere less dramatically with visibility. For that reason, ASAC can be easier integrated in buildings, as it does not require the use of loudspeakers or error microphones in the receiving environment. However there are not tested applications on glazed facades, and most of the experiments were carried out in the automotive and aeronautic fields of research: for example feed-forward ASAC tests have been performed in a test section comprising the aft portion of a furnished DC-9 aircraft to reduce cabin noise [4]. In addition, piezoelectric zirconate titanate (PZT) patch actuators were bonded to the cylinder surface of an aircraft fuselage excited acoustically by an exterior loudspeaker noise source to reduce interior noise [5]. Similar tests were executed for noise reduction in helicopter cabins: improvements up to 6 dB were registered in the receiving room for a range of frequencies limited between 80 and 400 Hz [6]. In the ASAC configuration there are two basic arrangements depicted in Fig. 2: feed-forward and feedback types. The feed-forward control of Fig. 2-a requires knowledge of the primary disturbance, which is derived by the use of a reference microphone: in the case of buildings this seems unpractical, because it would require the installation of a microphone on the external of the window, not feasible for functional and aesthetic issues. Therefore, the feedback type controller depicted in Fig. 2-b seems to be opportune and is detailed in the following of this paper. The whole system is made up of the following components:

- sensors to detect vibrations (e.g. strain gauges);
- electronic filters to analyze signals from sensors in order to check the vibration field induced by disturbance;
- an electronic controller to manipulate signals from the sensors and compute the most efficient control configuration at the actuator's level;
- charge amplifiers to drive the secondary actuators on glazed panels according to the outputs sent by the controller;
- actuators to control the vibration field of glazed panels.



a) b)

Fig. 2. Feed-forward (a) and feedback (b) ASAC arrangements for glazed facades.

Once the signal has come from sensors, it must be elaborated by charge amplifiers (converting voltage signals into physical variables like displacements, velocity and accelerations), and electronic filters that have the main task of separating the total vibration field into the one due to the primary disturbance from the other connected with the action of secondary sources. The controller, starting from the error signals, computes the radiated field in some positions of the receiving room, and computes the opportune voltage to be supplied to the secondary sources, whose electric power is provided by the amplifiers, in order to reduce the panel's acoustic efficiency, by opportunely changing or reducing the vibration field of the radiating glass panel. Opportune algorithms are implemented in the controller to optimize the actuator actions, like in ref. [7]. It consists of two parts, the first dedicated to the determination of actuator size and location, the second to sensors. In both parts the core algorithm exploits the quadratic linear optimum control theory to work out voltages to be supplied to actuators for every assumed location in order to reduce glass vibrations. The rest of the procedure minimizes an objective function applying the quadratic linear optimum control theory for several actuators' locations, in order to find out the best actuator configuration, upon determination of constraints relative to plate's geometry and design choices. That algorithm was applied for a bi-dimensional thin sheet metal surface, driven by PZT patch actuators. Instead this paper is concerned with the investigation of the performances of "stack" actuators, considered much more feasible for glass panels, because of their small size. Another advantage that stack actuators have over laminated is that they can be mechanically mounted to the control plant and easily dismantled. Moreover the visual impact of laminated actuators on glazed surfaces could be important, while the stack actuators and glazed stiffeners preserve the transparency. PZT stack actuators are manufactured by piling up a quantity of piezoelectric discs or plates, which are individually contacted electrically. The stack axis is the axis of the linear motion: the thickness of the layers increases by applying a voltage and thereby the total stack lengthens. The elongation of a stack is roughly proportional to stack's length, with a maximum achievable strain of 0,1-0,2 %. The absolute force/mass load of an actuator is determined by its cross section of the stack and is defined that no adverse change of actuator's properties occur. A problem are tensile forces, because even for small rates a cracking of the ceramic occurs. This is prevented by preloading/prestressing the stack by means of a case. A lot of piezoelectric stack are commercially available, starting from the minimum length of 0.01 m to some centimeters. For the purpose of glass vibration control, a stroke between 5 and 10 micron is achievable with maximum operating voltage of 150 V.

3. TECHNOLOGICAL FEASIBILITY

As far as concerns the choice of actuators, from a literature survey it was found out that two main typologies of PZT actuators are presently available on the market: piezoelectric patches, successfully experimented in [8], and piezoelectric stack actuators, already tested in [9]. As the first is a rectangular shaped patch, it may interfere with visibility, instead the second one is very small but need a stiffener to work properly. Implementing the validated analytical model to simulate the vibration field of simply supported rectangular plates proposed in ref. [3] for simply supported bi-dimensional plates in the MatLab^{TM} environment and comparing the vibration field generated by a disturbing noise of magnitude 100 dB with the maximum force provided by both kinds of actuators driven at the voltage of 100 V allowed for standard PZT actuators, it came out that single PZT thin patches were not able to contrast vibrations induced by the chosen disturbance (Fig. 3) [1]. Therefore, in the following paragraphs the effects of stack actuators on glass panels will be investigated, demonstrating that they can provide forces strong enough to contrast the action of such disturbances.

There are two basic ways to decrease the radiation efficiency of vibrating glass panels [5]:

- by decreasing the vibration amplitude of flexural waves;
- by changing the original vibration in order to obtain a vibration field where even modes dominate.

In the second case actuators should act in order to generate a vibration field with less radiation efficiency instead of reducing vibration amplitudes, like in the first case [2]. Every listed alternative determines the corresponding choice for PZT positioning: in the first case the actuators must be installed on the points where maximum vibration amplitudes are monitored; in the second case they could be moved along the border lines, where they do not interfere with the function of looking though, but should be able to modify the vibration field.

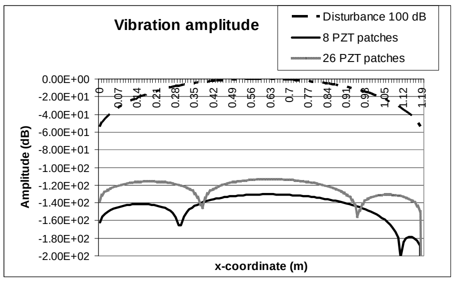


Fig. 3. Normalized amplitudes of vibration provided by disturbance and PZT patch actuators located along border lines.

Fig. 4 shows three possible technologic solutions: stack actuators are installed close to the central axis of the glass panel, where vibration amplitudes are usually very high and need to be reduced, and stiffened through a metal profile (case a); as an alternative, stack actuators may be installed along the borders and stiffened with an angular profile (case b) to modify the vibration field; finally, in case *c* stack actuators are installed along the borders and stiffened with point reaction systems. In this contribution, case *a* of Fig. 4 is analyzed, that is the basic and most straightforward type of technology. The installation of PZT actuators requires the use of stiffener profiles to contrast their strain and allow the glass panel's vibrations, as shown in Fig. 4-d. The choice of the reaction profile must assure that its stiffness is comparable with the one of PZT actuators: at this stage, the performances of a steel rectangular shaped profile were compared with the ones of an aluminum presently marketed profile for windows' frames. A proper finite element model was developed in ANSYS^{TM} environment for static analyses on stiffness computations (the magnitude of force necessary to produce a unitary displacement) for a simply supported stiffener. The extreme joints of the stiffener are supposed to be hinged to the window's frame (Fig. 4-a).

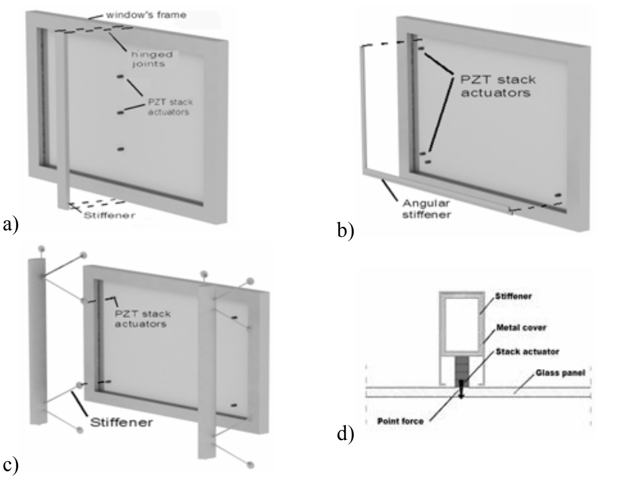


Fig. 4. Technologic solutions suggested for the installation of actuators and example of reaction profile (d) for case a.

First, the parameters of the finite element model were set as in the case of Fig. 5-a, comparing its solution with the corresponding analytic well-known solution [10], and no differences were noticed in the two cases (in the middle point of the beam the same displacement of $2.68 \cdot 10^{-5}$ m due to a 1 N force was recorded for section of Fig. 5-c) showing that the adopted finite element model has proper parameters. Then the case object of the study in paragraph 4, having three actuators applied along the main axis (Fig. 5-b), was analyzed, computing the stiffness provided by a reaction profile with the two cross sections showed in Fig. 5-c and 5-d in correspondence of the points were actuators are applied.

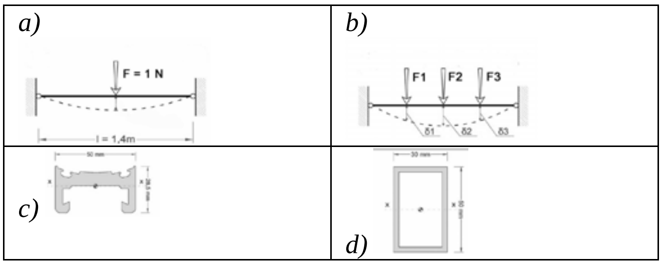
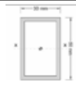


Fig. 5. Finite Element models and cross sectional areas for the cases of interest.

Assuming for PZT stack actuators the characteristics provided by commercial suppliers, Tab.1 lists all the stiffness values obtained for the case of Fig. 5-b, when equipped with the *c* type (Fig. 5-c) or *d* type section (Fig. 5-d). For safety reasons, actuators should be driven by low voltages and it can be noticed by Tab.1 that the cross section of Fig. 5-d is the most suitable, which can be easily produced with already available manufacturing processes. From a parametric study it came out that forces of about 0.5 N are necessary to contrast disturbing waves higher than 80 dB: Tab.1 shows that low voltages are required to drive that plate thanks to the use of the stiffener in Fig. 5-d, while those actuators would not be large enough for the other aluminum profile. In general, we can state that every time the reaction profile stiffness is comparable with the one of the actuators (the actuators of Tab.1 have a stiffness of $1.63 \cdot 10^8$ N/m) and much higher than the controlled system's one (the plate of glass used for experiments in the following has a stiffness in the centre of $8.80 \cdot 10^4$ N/m), it gives back opportune reaction forces.

Tab.1. Stiffeners' computation results.

Profile	Force positioning	Stiffness (N/μm)	Voltage to be supplied (V) for 0.5 N
	F1 and F3	$0.30 \cdot 10^8$	40
	F2	$0.22 \cdot 10^8$	52
	F1 and F3	$2.20 \cdot 10^4$	> 150
	F2	$1.56 \cdot 10^4$	> 150

4. PRELIMINARY ESTIMATION OF ACOUSTIC IMPROVEMENTS

4.1 Setting of the Finite Element Model

The development of an appropriate finite element model will be described in this sub-paragraph, preliminary to its application to a test case for estimating the acoustic improvements that could be determined by the application of such a system, which is the object of study of the next sub-paragraph. By means of that approach it will be shown that the magnitude of acoustic relief determined inside building rooms is appreciable for pursuing the acoustic comfort required by European standards and regulations. Previous to the numerical analyses that will be carried out in the following sub-paragraph, it is necessary to set a proper finite element model, relative to its subdivision into well refined elements, to the inputted material and geometric parameters and to its boundary conditions. As test case a rectangular (1.4x1.0) m glass plate, 0.006 m thick, simply supported along its edges was chosen for numerical and experimental analyses. A numerical modal analysis was carried out, and used to set the parameters of the finite element model through a comparison with the corresponding experimental results. The model was implemented in ANSYS 8.0^{TM} environment, subdivided in squared shaped finite elements of 0.02 m of side. The nomenclature of the glass natural modes is chosen according to the number of troughs along respectively the major and the secondary axes of the plate. The material-geometric parameters inputted for this analysis are: elasticity modulus $E = 6.9 \cdot 10^{10}$ Pa; Poisson coefficient $\upsilon = 0.23$; density $\rho = 2457$ kg/m^3. The choice of those values derived from an iterative refinement of that model, through comparison with experimental data: a prototype having the same geometric dimensions of the one numerically simulated was built, as shown in Fig. 6, arranging a uniform tightening of the glass panel all along the border lines to simulate a simply supporting constraint, whose manufacturing was pursued through the interposition of two cylindrical Teflon bars between the glass panel and the two window frame profiles (Fig. 6-c and 6-d). Every screw fixing the glass panel in the window's frame was subject to a torque equals to 0.1 N·m, in order to guarantee a good and uniform contact between glass and Teflon. Before starting measurements, all the system was positioned over dumping supports to avoid the influence of external actions on the glass's vibrations, as required by the numeric model (Fig. 6-b). Fig. 7 is devoted to show how experiments were performed. The window was located horizontally at such a distance form the ceiling that it could be zoomed by the terminal of a Laser-Doppler vibrometer [11], placed 3.30 m high from the ground by means of a steel bar-joint structure, allowing in this way a 3 m distance between that terminal and the tested window. Through the experimental setup shown in Fig. 7-a it was possible to perform the modal analysis for the prototype of window object of this study: first the panel was excited by an impulsive disturbance (at the points marked in Fig. 7-c) and then the Laser-Doppler terminal monitored vibrations at the vertices of a squared grid of 0.1 m of side and spread over the window. Then, those measures were elaborated through a dedicated software to infer mode shapes and frequencies, whose values were compared in Tab.2: absolute shifts are generally very low, and when they seem to be a bit high it can be noticed that the relative error is in every case lower than 6% (in the last two cases). Thanks to the overall agreement between numeric and experimental results, this model was considered suitable for further numerical simulations, detailed in the following sub-paragraph.

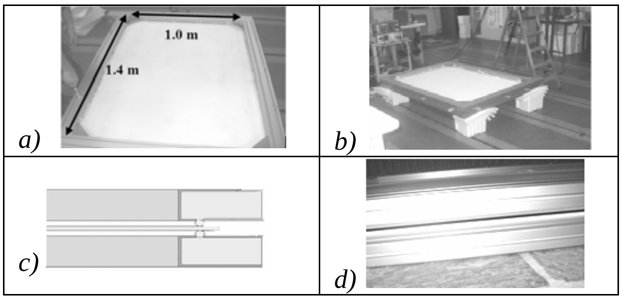


Fig. 6. Prototype used for experiments (a), dumping supports under the window frame (b) and simply supporting constraint (c, d).

4.2 Improvements of STL due to active control

It is assumed that the aforementioned rectangular simply supported window is stroked by a harmonic wave at 140 Hz, whose noise level is reasonably the one that could be generated by a lorry traveling at low distance and at a speed of 70 km/h equals to 85 dB, and the mode (1,3) is the most excited. Fig. 8-a and b sum up the harmonic motion generated in the uncontrolled and in the controlled case. In the uncontrolled one the plate's maximum displacement reach the value of $1.73\cdot10^{-6}$ m in the central troughs, whose value is very close to the other two ones. Actuators were supposed to be located at the maximum displacements of the three toughs located along the minor axis, according to the shape of the mode (1,3) mainly excited by the disturbance. Fig. 8-c depicts the locations for the controlling stack actuators, that are supposed to act along the minor axis, to generate a harmonic force amplitude of 0.383 N at a frequency of 140 Hz. The voltage to be supplied for the production of such forces can be computed through the model in ref. [3]. It avails itself of the data relative to the lass panel, stack actuators and stiffening structures (Fig. 5-d). Then applying the equation:

$$F_a = \frac{d_z \cdot V \cdot K}{1 + \dfrac{K}{K_a}} \qquad (1)$$

it came out that voltage V lower than 40 V is adequate for each actuator. In eq. (1) d_z is the z-axis elongation, K and K_a are the stiffness of the reaction system and actuator respectively.

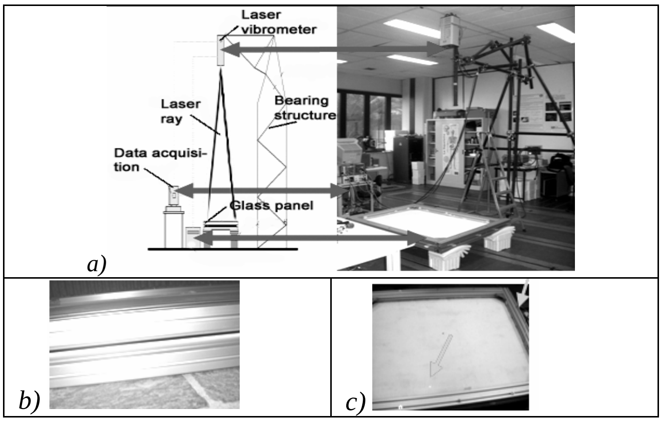


Fig. 7. Experimental apparatus for measurements (a), Laser-Doppler terminal (b), measure and excitation points on the glass (c).

Tab.2. Comparison of numeric and experimental results.

Mode	Freq. exp. (Hz)	Freq. num. (Hz)	Shift (Hz)
1,1	25.0	22.4	2.65
3,1	87.5	82.8	4.61
2,2	90.0	89.4	0.64
3,2	127.5	127.0	0.48
3,3	190.0	200.9	10.88
5,3	320.0	330.9	10.92

It can be noticed that there is a strong reduction of vibration amplitudes from the uncontrolled maximum value of $1.73\cdot10^{-6}$ m (Fig. 8-a) to $4.13\cdot10^{-7}$ m, that are the residual vibrations, due to the difference of shape between the vibration field induced by a disturbing wave and the other field induced by control point forces induced by PZT stack actuators, as depicted in Fig. 8-b.

However the release produced from an acoustic point of view may be estimated only analyzing the effects inside a test room. For that reason the previously tested window was supposed to be installed on one wall of a test room having the following characteristics: plastered walls and ceilings (whose absorbance coefficient may be estimated equals to 0.04) and back stalls on moquette floor, having an estimated overall absorbance coefficient equals to 0.7.

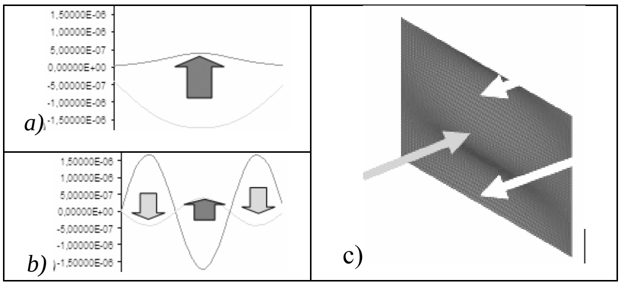


Fig. 8. Active control of a window subject to harmonic disturbance.

Assuming the same harmonic disturbing wave of 85 dB level and propagating at 140 Hz, like in the previous study, the resulting acoustic harmonic disturbance inside the room will be equal to the one depicted in Fig. 9-a, whose average level is 66 dB, the minimum is 34 dB and the maximum is 78 dB. In case the window is controlled by the three stack actuators along the minor axis, with a consequent decrement of vibrations, a strong decrement of acoustic pressure level is obtained, as shown in Fig. 9-b: the maximum peaks drop from 78 to 63 dB, thanks to the reduction of vibration amplitudes generated by the actuators, with a final drop of 15 dB. The average and minimum values drop respectively to 53 and 32 dB.

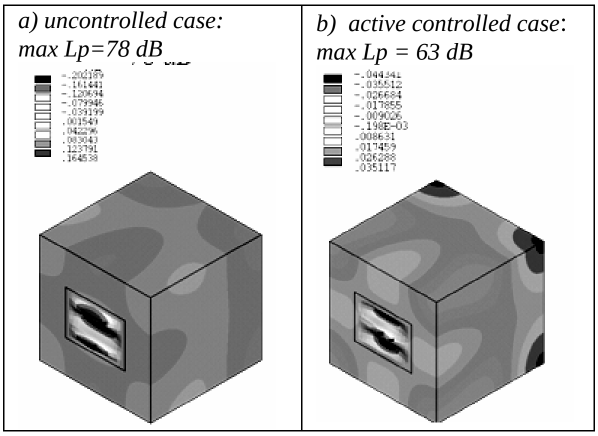


Fig. 9. Harmonic acoustic field generated by a lorry (a)inside the test room and reduction due to the active structural acoustic control system (b).

5. CONCLUSION

Thanks to the application of an active structural acoustic control system, it is possible to strongly improve the STL of window panels in the low frequency range: the presence of actuators drops dramatically the noise transmitted from the exterior to the interior, even if the disturbing wave is near its resonance effect. Coupling this system with laminated technology, that are effective at high frequencies, would allow to obtain good insulation properties all over the range of audible acoustic frequencies, determining an increment of transmission loss performances for all the wall were such active controlled windows are inserted. In the particular case considered in this paper, where a relatively complex mode was excited, it was possible to obtain a sound reduction of the highest value up to 15 dB inside the chosen test room. Moreover it was shown that several technologic solutions are available for its installation on windows and that its functioning requires the use of very low voltages, that cannot be considered dangerous for users. Further research will be devoted to check other positioning of actuators and the possibility to control vibration fields with high modal density .

REFERENCES

[1] Naticchia, B., Carbonari, A. (2005) Integration of an automated active control system in building glazed facades for improving sound transmission loss, *Proceedings of the 22nd ISARC 2005, September 11-14th, Ferrara, Italy.*

[2] Fahy, F. (1985) Sound and structural vibration: radiation, transmission and response. *Academic Press, London - San Diego – New York – Berkeley – Boston – Sydney – Tokyo.*

[3] Fuller, C. R., Elliott, S. J., Nelson, P. A. (1997) Active Control of Vibration. *Academic Press, 2nd Edition, San Diego-London- Boston-New York.*

[4] Simpson, M. A., Luong, T. M., Fuller, C. R., Jones, J. D. (1991) Full-scale demonstration of cabin noise reduction using active vibration control. *Journal of Aircraft, 28(3), 208-215.*

[5] Fuller, C. R., Snyder, S. D., Hansen, C. H., Silcox, R. J. (1992) Active control of interior noise in model aircraft fuselages using piezoceramic actuators. *AIAA Journal, 30(11), 2613-2617.*

[6] Petitjean, B., Legrain, I., Simon, F., Pauzin, S. (2002) Active control experiments for acoustic radiation reduction of a sandwich panel: feedback and feedforward investigations. *Journal of Sound and Vibration, 252(1), 19-36.*

[7] Clark, R. L., Fuller, C. R. (1992) Optimal placement of piezoelectric actuators and polyvynidilene fluoride error sensors in active structural acoustic control approaches. *Journal of the Acoustic Society of America, 92(3).*

[8] Wernli, L. (2001) Regelung der Schall¨ubertragung einer Doppelverglasung. *Term project, Automatic Control Laboratory, Swiss Federal Institute of Technolgy Zurich.*

[9] Simpson, M. A., Luong, T. M., Fuller, C. R., Jones, J. D. (1991) Full-scale demonstration of cabin noise reduction using active vibration control. *Journal of Aircraft, 28(3), 208-215.*

[10] Merritt, F. S. (1993) Standard Handbook for Civil Engineers. *McGraw-Hill, 3rd edition, New York - St. Louis - San Francisco – London – Madrid – Milan – Paris – Sidney – Tokio.*

[11] Castellini, P., Revel, G. M., Tomasini E.P. (1998) Laser Doppler Vibrometry: a Review of Advances and Applications. *The Shock and Vibration Digest, 30(6), 443-456, November, Sage Science Press, Thousand-Oaks, CA.*