Active reduction of sound transmission through double panel partitions - A physical analysis of the observed phenomena

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Abstract

The paper addresses the active control of the sound transmission through a double panel partition. A literature survey indicates that active cavity control is much more efficient than active structural control. The vibro-acoustic behaviour of the double wall is analysed in light of this observation. This analysis shows that the sound transmission through the double panel is larger than that through a single panel at the resonances of the coupled eigenmodes with the strongest vibro-acoustic coupling between the plate motion and the pressure in the cavity. At frequencies where the double panel radiates much sound energy, the acoustic energy in the cavity is also high. Active control simulations illustrate that the cavity control approach, reducing not only the uncoupled (0,0,0) mode, achieves a considerable reduction of the sound transmission over a large frequency band. Both symmetrical and asymmetrical double panel systems are considered.

1. Introduction

Double panel partitions are widely used in noise control applications when good sound insulation characteristics have to be achieved. Typical examples include double-glazed windows, gypsum office partitions, or aircraft fuselages. However, the acoustic performance of those double panel structures rapidly deteriorates toward lower frequencies, where it can even become worse than that of a single panel [1,2]. The ability of double panel partitions to provide significant (or any) sound transmission loss at low frequencies is nevertheless very important, because low-frequency noise is known to be more disturbing for man than indicated by the traditional dB(A)-curve. In addition, modern society produces an ever increasing low-frequency component in the environmental noise spectrum [3]. This is mainly due to the growing economic activity during the last decades, although other types of low-frequency noise sources have emerged (e.g. music).

Numerous experiments for different applications have already shown the efficiency of the active control of a sound field in the low-frequency range [4]. Therefore, a number of researchers have previously investigated the possibility of improving the low-frequency sound insulation characteristics of double panel systems by means of active noise control techniques. Possible applications for an actively controlled double-panel partition with improved low-frequency sound insulation characteristics include studios for audio recording, large conference rooms, concert halls, and discos and bars to shield the environment against the low-frequency beat which often dominates the "internal sound field".

A first class of contributions in the area of active sound transmission control through double walls employs a structural control approach, whereby structural control sources, mounted on either of the two panels, are driven so as to reduce the radiated sound field [5,6]. A second class of contributions focuses on the use of an active noise control system implemented in the cavity between the two plates [7-12]. This approach is often termed as active cavity control. In a third class of contributions, a comparison is made of the performances of the structural and the cavity control approaches. Recent papers [12-15] demonstrate that the active cavity control approach yields better results than the structural control approach on different double wall test set-ups, but they do not yet fully explain the physical reasons behind that observation. Therefore, the basic aim of the present paper is to analyse the dynamic behaviour of a double wall, and the sound transmission through it, in light of a fundamental comparison of both control approaches. The discussion concentrates on a double-glazed window because it is a relevant application, the potential of the active cavity control approach has been demonstrated experimentally on it [10], and it is easily described by an analytical model that

provides a good physical insight in its low-frequency dynamic behaviour. The glass plates in a first set-up are 1.23 m wide, 1.48 m high, and 3 mm thick, and the dimensions of the cavity are 1.1 m by 1.35 m by 0.1 m. Afterwards, the conclusions obtained from the in-depth analysis of this symmetrical double window are extended to an asymmetrical double window, consisting of a 10 mm and a 3 mm thick glass plate, separated by a 0.02 m wide air gap. The use of microphones in the radiated sound field as error sensors is deliberately discarded here, because the final actively controlled double panel partition, that is aimed at in this study, should be compact and independent of its environment in order to extend its possible application areas and reduce its installation cost.
The dynamic behaviour of the double wall is studied by means of the modal expansion method given by Fahy [16], and further developed for double panel systems by Sas et al. [8] and by Desmet and Sas [17]. This method yields a set of coupled algebraic equations in which the unknowns are the contributions of the uncoupled mode shapes (i.e. the *in vacuo* modes of the plates and the modes of the cavity *with rigid walls*) to the response of the coupled vibro-acoustical system. The sound radiation model is based on the near-field approach by Elliott and Johnson [18].
The next section of this paper analyses the sound insulation characteristic of the double window, and it is shown that the coupling of the radiating plate with the cavity sound field has a similar frequency dependence as the sound radiation of that plate into the free field. Section 3 discusses the implementation of different active control approaches on this double window, yielding as main conclusion that the active cavity control approach, reducing not only the (0,0,0) cavity mode, is, by far, superior to the other approaches. Section 4 validates the conclusions from sections 2 and 3 on the asymmetrical double window. Finally, some concluding remarks and considerations are presented in the last section.

2. Sound transmission through double walls

The sound transmission characteristics of normally and obliquely incident plane waves through infinite double-panel partitions are well known in literature [1,17]. Compared to single panels, infinite double-panel partitions generally yield a higher sound transmission loss, except around the low-frequency mass-air-mass resonance of the double wall, where the glass plates oscillate in anti-phase against the stiffness of the compressible acoustic medium in the cavity:

$$\omega_0 = \sqrt{\frac{\rho_0 c_0^2}{L_c^z}\left(\frac{m_{p_1} + m_{p_2}}{m_{p_1} m_{p_2}}\right)}, \qquad (1)$$

where L_c^z is the distance between both plates (0.1 m), ρ_0 the density of the acoustic medium in the cavity (1.225 kg/m^3 for air), c_0 the sound speed in that medium (340 m/s), and m_{p_1} and m_{p_2} represent the mass per unit area of both plates (7.5 kg/m^2 for the 3 mm thick glass plates). For the double window being considered in this study, the mass-air-mass resonance occurs at 97 Hz. This frequency coincides with the eigenfrequency of the first cavity controlled coupled mode of this symmetrical double panel partition (see [19]).
When both panels have finite dimensions, the sound transmission characteristics look different, especially in the low-frequency range. Several coupled structural-acoustical resonances appear at low frequencies, which, together with a 'mass-air-mass'-like phenomenon, result in a substantial decrease in the low-frequency transmission loss [17]. This is illustrated in figure 1, which shows the calculated insertion loss of the double-glazed window being considered here. The insertion loss (IL) is defined as the difference between the average radiated sound power of a double- and a single-glazed window, obtained by removing one glass plate:

$$IL[dB] = 10\log\left(\frac{W_{single}}{W_{double}}\right) \qquad (2)$$

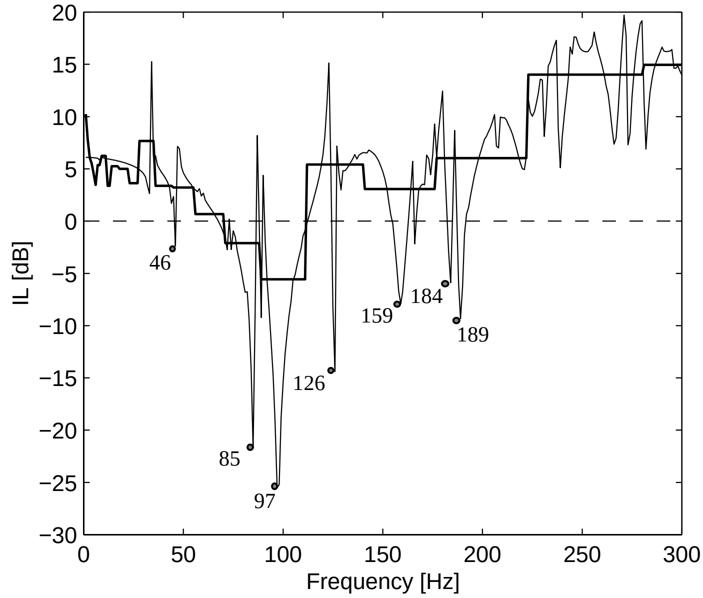

Figure 1: Calculated insertion loss of the second glass plate (thick line: in 1/3 octave bands, and thin line: continuous spectrum).

A diffuse field excitation was modelled by a superposition of 20 plane waves with random amplitudes and random angles of incidence.

At very low frequencies, both panels move in phase with an identical vibration pattern, and, as the sound transmission is concerned, they behave in the same way as a single panel of twice the thickness of the glass plates considered here.

An analysis of the operational deflection shapes near the dips in the insertion loss curve shows that essentially two types of coupled modes are responsible for the bad sound insulation characteristics of the double panel system in the mass-air-mass frequency region (see [19] for a more complete discussion). The first type are the cavity controlled modes (e.g. at 97, 159 and 189 Hz). The pressure field at those resonances is dominated by the uncoupled cavity mode shapes, from which the corresponding cavity controlled coupled modes originate (the (0,0,0)-mode at 97 Hz, the (0,1,0)-mode at 159 Hz and the (1,0,0)-mode at 189 Hz). The second type comprises a few out-of-phase plate controlled modes, often dominated by a volumetric uncoupled plate mode (i.e. with an odd number of nodal lines in both directions, like the (3,1)-mode at 46 Hz, the (1,5)-mode at 85 Hz, the (5,1)-mode at 126 Hz, ...). The pressure field at those resonances mainly consists of the uncoupled (0,0,0)-mode, with some higher order modes superimposed on it.

This analysis indicates that, on the one hand, the pressure field in the cavity, at frequencies where the insertion loss is negative, is not only dominated by the (0,0,0)-mode as previously assumed by many authors (for example [7,9,20-22]). Therefore, for optimal efficiency, an active cavity control system should not only focus on reducing that particular mode. On the other hand, the sound transmission reduction of the double panel falls short of that of a single panel at the resonances of those coupled eigenmodes with the strongest vibro-acoustic coupling ([19]) between the plate motion and the pressure in the cavity. This means that at frequencies where the double panel radiates much sound energy, the acoustic energy in the cavity will also be high.

This is further demonstrated by figure 2 which shows, in the upper part, the ratio between the radiated sound energy and the kinetic energy of the radiating panel, and, in the lower part, the ratio between the acoustic energy in the cavity and the kinetic energy of the radiating panel (corrected for the quadratic frequency dependence of the radiated sound power). The behaviour of both curves in figure 2 is very similar from 50 Hz on, indicating that the acoustic energy in the cavity is a good measure for the radiated sound energy. Consequently, an active control system reducing the potential energy in the cavity, not only caused by a (0,0,0)-type of pressure field, can also be expected to effectively reduce the sound transmission through the double panel from that frequency on.

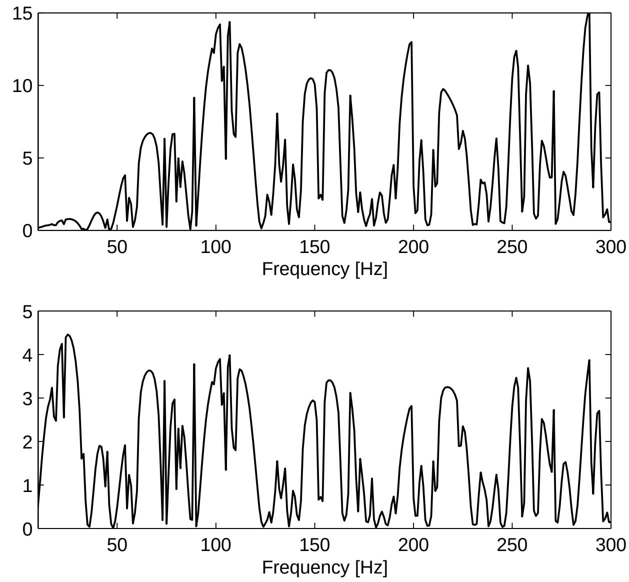

Figure 2: Upper part, ratio between the radiated sound and the kinetic energy of the radiating panel; Lower part, between the acoustic energy in the cavity and the kinetic energy of the radiating panel.

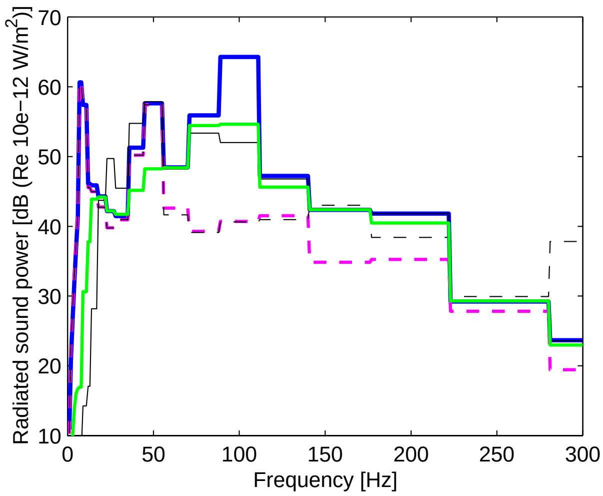

Figure 3: Radiated sound power (in 1/3 octave bands) without control (thick solid line), and with an active control system with one loudspeaker, reducing either the total acoustic potential energy (thick dashed line) or only the contribution of the (0,0,0)-mode (thin dashed line), or with one shaker, reducing either the structural kinetic energy of the radiating plate (grey solid line) or only the contribution of its uncoupled (1,1) mode (thin solid line).

3. Active sound transmission reduction

The simulation results of section 2 indicate **why** the acoustic potential energy in the cavity is a good control objective to minimise for reducing the sound transmission through a double wall. This thesis is further verified by the simulations presented in this section, and, of course, by many experimental results. However, many authors [7,9,20-22] focus only on the (0,0,0) uncoupled cavity mode when implementing an active cavity control system. The discussion in section 2 already indicated that also other uncoupled cavity modes are responsible for the deficiencies in the sound insulation properties of a double panel partition, and, consequently, that a cavity control system should reduce the total acoustic energy in the cavity, rather than only this (0,0,0) mode. This is illustrated in figure 3, which shows the radiated sound power by the double panel system with an active control system with one loudspeaker, reducing either the total acoustic potential energy or only the contribution of the

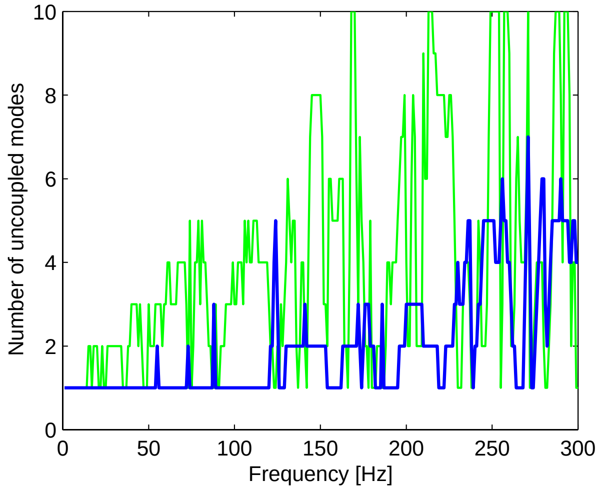

Figure 4: Number of uncoupled cavity modes (thick black line) and number of uncoupled plate modes (thin grey line) required to account for 99 % of the acoustic energy in the cavity, respectively the structural kinetic energy of the radiating panel.

(0,0,0)-mode, or with one shaker, reducing either the total structural kinetic energy of the radiating plate or only the contribution of its uncoupled (1,1) mode. In order to allow a fair comparison, all sensor and actuator locations are optimised for maximal sound transmission reduction in the frequency band of interest.

This figure clearly demonstrates that a control system with a loudspeaker and an acoustical control objective successfully reduces the radiated sound power from about 50 Hz on. This observation is in good agreement with figure 2, from which it was concluded that the acoustic energy is only a good measure for the radiated sound power from that frequency on. However, the acoustical control system that only reduces the (0,0,0) cavity mode (in thin dashed line), yields only a significant improvement of the transmission loss in a limited range around the mass-air-mass frequency. The shaker-based control system achieves only considerable reductions in the radiated sound power in the very low frequencies (in the sub-audio range) and near the mass-air-mass frequency. The first reason therefor is that a high reduction of the kinetic energy of the radiating panel does not necessarily correspond to a comparable reduction of the radiated sound power. The second reason is that the panel is much more difficult to control with one single actuator than the cavity, because the (uncoupled) modal density of the radiating panel is much higher than that of the cavity, as illustrated in figure 4. This figure shows the number of uncoupled cavity modes and the number of uncoupled plate modes required to account for 99 % of the acoustic energy in the cavity, respectively the structural kinetic energy of the radiating panel.

The same conclusions can be drawn from the application of the radiation mode concept, as introduced by Elliott and Johnson [18], to double panel partitions [19]. The projection of the radiation modes of the two plates on the coupled modes of the double panel partition shows that the piston type of radiation mode, which is dominant at low frequencies, couples with the (0,0,0) cavity mode near the mass-air-mass frequency. Essentially, controlling the volume velocity of the radiating plate comes down to controlling only the (0,0,0) cavity mode. At higher frequencies, many radiation modes contribute equally to the total radiated sound power, and consequently, reducing only the piston type of radiation mode (the volume velocity cancellation approach, see [12,23]) is not an effective strategy for reducing the total sound transmission. Thus, at higher frequencies also other cavity modes should be controlled for optimum performance.

4. Asymmetrical double panel partitions

Up to this point, the discussion focused on a symmetrical double window set-up with a relatively large cavity thickness (0,1 m). In this section, the application of the cavity control approach is investigated for a slightly modified set-up. In this set-up, the thickness of the glass plate at the incident side is 10 mm, instead of 3 mm, and the thickness of the cavity is reduced to 0,02 m. The

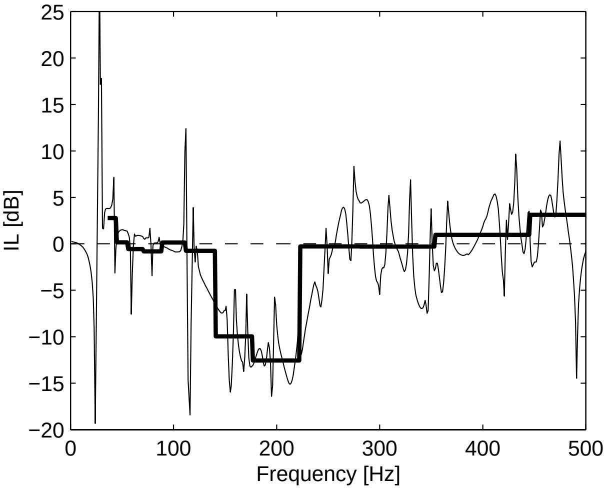

Figure 5: Insertion loss of the 10/20/3 double wall partition.

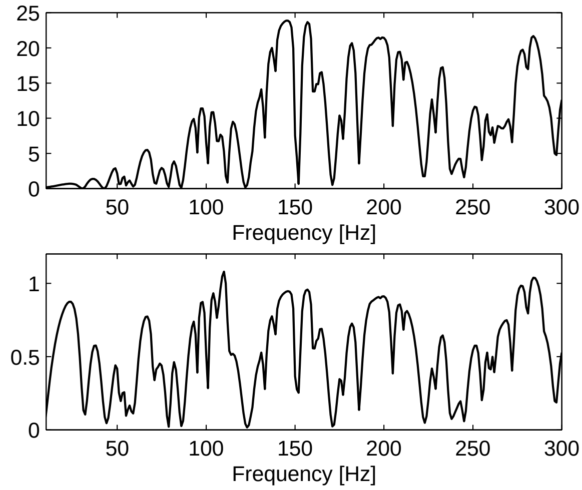

Figure 6: Upper part, ratio between the radiated sound and the kinetic energy of the radiating panel; Lower part, between the acoustic energy in the cavity and the kinetic energy of the radiating panel.

mass-air-mass resonance frequency, as predicted by equation (1), lies at 176 Hz for this double window.

Figure 5 shows the insertion loss, as defined by equation (2), for the 10/20/3 double window, where the reference single panel is the 10 mm thick single glass plate. The frequency range around the mass-air-mass resonance, where the transmission loss of the double wall drops below that of the single panel, is much larger for this set-up than for the original 3/100/3 double window.

Figure 6 shows the ratios between the radiated sound and the kinetic energy of the radiating plate (upper part) and between the acoustic potential energy in the cavity and the kinetic energy of the radiating plate (lower part). These ratios show a similar behaviour from about 120 Hz on, indicating that also here the acoustic potential energy in the cavity is a much better measure for the radiated sound power than the kinetic energy of the radiating panel from that frequency on. Hence, the active cavity control approach can be expected, also for these modified set-ups, to yield better performances than a structural control approach, from 120 Hz on.

Figure 7 shows the performances of an active control system with two shakers and four accelerometers, with two loudspeakers and four error microphones, and of a control system with the same two loudspeakers minimising the (0,0,0) cavity mode only. Again the pure active cavity control approach with control loudspeakers in the cavity appears to be the best solution for reducing the sound transmission through this asymmetrical double panel set-up. The purely structural control system (thick grey line) only achieves a significant reduction of the radiated sound power below 30 Hz, which is negligible in noise control applications. Its

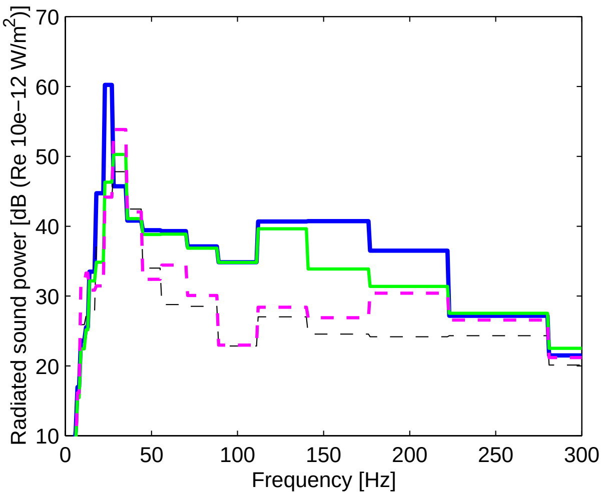

Figure 7: Radiated sound power (in 1/3 octave bands) without control (thick solid line), and with an active control system with two loudspeakers, reducing the (0,0,0) cavity mode (thick dashed line), or with four microphones (thin dashed line), or with two shakers and four accelerometers (grey solid line).

performance in the mass-air-mass frequency region is less significant than that of the cavity control systems. The cavity control system with four error microphones yields higher reductions over a wider frequency range than the control system which only reduces the (0,0,0) cavity mode. This latter system only achieves significant reductions in the mass-air-mass frequency region, which is now larger than in the previous set-up (compare figures 5 and 1). Thus, for optimal performance the cavity control system should not only reduce that particular (0,0,0) cavity mode.

5. Conclusions

This study focuses on the possibilities of applying active noise control techniques to improve the inherently bad sound insulation characteristics of double panel systems in the low-frequency range.

A literature survey shows that an active cavity control approach yields better results than a structural control approach on different double wall test set-ups, but it does not fully explain the physical reasons behind that observation. Therefore, the basic aim of the present study is to analyse the dynamic behaviour of a double wall, and the sound transmission through it, in light of a fundamental comparison of both control approaches.

A detailed response analysis of the double panel indicates that the sound transmission reduction of the double panel falls short of that of a single panel at the resonances of the coupled eigenmodes with the strongest vibro-acoustic coupling between the plate motion and the pressure in the cavity. This means that, at frequencies where the double panel radiates much sound energy, the acoustic energy in the cavity is also high. The strongest vibro-acoustic coupling occurs for the cavity controlled coupled modes of the symmetrical double wall. This demonstrates that the pressure field in the cavity, at frequencies where the insertion loss is negative, is not only dominated by the (0,0,0)-mode as commonly assumed. Therefore, an active cavity control system should not only focus on reducing that particular mode.

These observations are further confirmed by the active control simulations. The two main conclusions from these simulations can be summarised as follows:

- An acoustical active control system is superior to a structural control system, except in the very low frequency range where the double panel behaves like a single panel of double thickness.
- An active cavity control system can produce reductions in the sound transmission over a wide frequency range extending well beyond the mass-air-mass resonance, provided that it aims not only at reducing the (0,0,0) uncoupled cavity mode.

These conclusions remain valid in the case of an asymmetrical double panel partition with a substantial difference in stiffness between the panels on the incident and on the radiating side.

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