ENHANCEMENT OF THE TRANSMISSION LOSS OF DOUBLE PANELS BY MEANS OF ACTIVELY CONTROLLING THE CAVITY SOUND FIELD

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INTRODUCTION

Double panels are used for double-glazing windows, for aircraft fuselage shells or in car-bodies. They are distinguished by high acoustic transmission loss for the high- and mid-frequency range but are weak in the low-frequency range. The acoustic insulation is poor especially around the mass-spring-mass resonance frequency of the panel-cavity-panel system. Active methods have the potential to increase the poor transmission loss at low frequencies. In the presented work the cavity sound field between the two panels is controlled by means of secondary loudspeakers in the cavity, such that the sound field within the cavity is minimized. The influence of the minimization on the transmission loss of the double panel was investigated experimentally. Some measured results from an experimental setup with an acrylic double-glazing window and a multichannel feed-forward controller with different loudspeaker positions are presented here.

In the last five years several authors investigated the active control of sound transmission through double panels. Their main interest was to show that with active methods the reduction of the sound transmission through double panels is possible and to show which method is best suited: i) active control of one or two of the panels by vibration inputs, ii) active control of the cavity sound field between the plates by sound sources.

Sas et.al. [1], [2] showed the feasibility of actively controlling the cavity sound field with loudspeakers inside the cavity for improving the transmission loss. In their experiments they used two loudspeakers and error microphones inside and outside the cavity. The differences between the two positions of the error microphones was shown not to be significant.

Carneal and Fuller [3] showed in experimental work that the use of vibration inputs also is suited to reduce the transmission loss. They used three actuators and three microphones. So they didn't minimize the vibration of the panel but the radiated sound field. That technique used is called active structural acoustic control (ASAC). Their measurements showed that controlling the vibration of the radiating panel in the described way results in a better performance than controlling the vibration of the incident panel.

The differences between panel and cavity control were investigated experimentally by Bao et.al. [4], [5] and theoretically by Gardonio and Elliott [6]. Bao et.al. used either a single loudspeaker in a corner of the cavity or a minishaker in the center of the radiating panel, so single channel control was performed with an error microphone outside the cavity in the receiving room. Gardonio and Elliott simulated the use of loudspeakers in the cavity and the use of active mounts between the two panels. They concluded that minimization of the radiated sound power is significantly better than minimizing the cavity sound field. Both the experimental and numerical studies showed that cavity control, i.e. use of loudspeakers, results in a higher increase in transmission loss than panel control, i.e. vibration inputs.

SOUND INSULATION OF SINGLE AND DOUBLE PANEL SYSTEMS

Below the coincidence frequency (i.e. $f \ll f_c$) the transmission loss TL_{sp} of a single panel of infinite size can be calculated by

$$TL_{sp} = 10 \lg \left(1 + \frac{\omega^2 m''^2 \cos^2 \vartheta}{4 \rho_0^2 c_0^2} \right) \text{dB} \tag{1}$$

according to [7]. In eq. (1) m'' is the mass per area of the panel, ϑ is the incident wave angle, $\rho_0 = 1.2\text{kg/m}^3$ is the air density, $c_0 = 340\text{m/s}$ is the speed of sound in air and $\omega = 2\pi f$ of course is the circular frequency.

The transmission loss TL_{dp} of a double panel of infinite size and for equal panel masses m'' can be calculated by [7]

$$TL_{dp} = 10 \lg \left[1 + \frac{\omega^2 m''^2 \cos^2 \vartheta}{4 \rho_0^2 c_0^2} \left(1 - \frac{\omega^2 m''}{2 s''} \right)^2 \right] \text{dB}\,. \tag{2}$$

In eq. (2) the added variable s'' is the stiffness per area of the air volume which equals

$$s'' = \frac{\rho_0 c_0^2}{d} \tag{3}$$

where d is the distance between the panels.

Fig. 1 shows both the single panel transmission loss TL_{sp} and the double panel transmission loss TL_{dp} over a normalized frequency. The frequency is normalized to the mass-spring-mass resonance f_{dp} of the double panel. The mass-spring-mass resonance frequency which exists due to the coupling between the two panels and the air cavity equals

$$f_{dp} = \frac{1}{2\pi} \sqrt{\frac{\rho_0 c_0^2}{d} \cdot \frac{2}{m''}}\,. \tag{4}$$

As can be seen from fig. 1 or directly noticed from eq. (1) the single panel transmission loss increases with 6dB per octave (i.e. 20dB per frequency decade) in the higher frequency range. Also from eq. (1) the so called "mass-law" can easily be seen, what states that any doubling of the mass of the panel results in an increase of transmission loss of 6dB in the higher frequency range. The transmission loss for the double panel is shown in fig. 1 also. It increases with 18dB per octave (i.e. 60dB per frequency decade) in the higher frequency range so the double panel distinguishes a much higher sound insulation

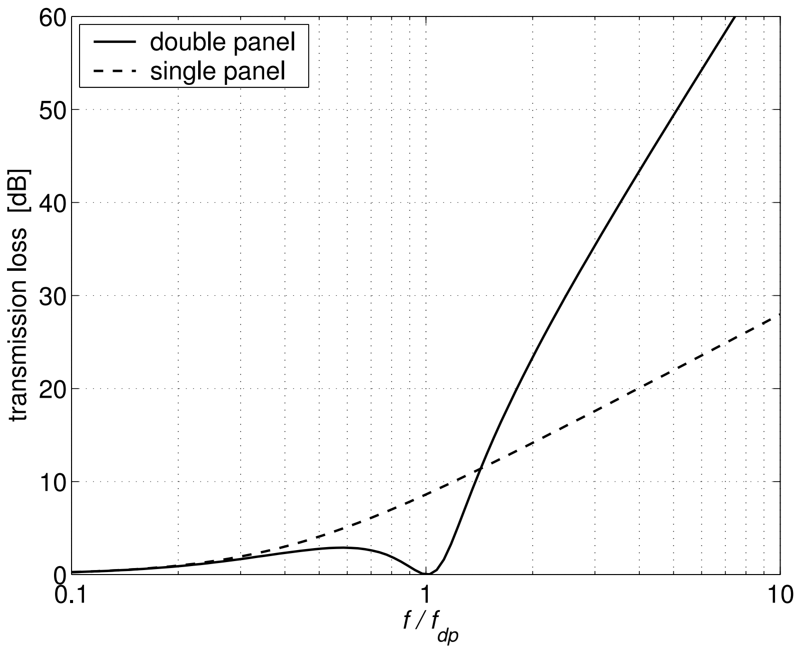

Figure 1: Comparison of the theoretical transmission losses TL_{sp} of the single panel and TL_{dp} of the double panel. The frequency f is normalized to the mass-spring-mass resonance f_{dp} of the double panel. Incident wave angle is assumed to be $\vartheta = 45°$ to the normal of the panel.

in that frequency range compared to the single panel, whereas around the mass-spring-mass resonance frequency f_{dp} the transmission loss of the double panel is lower than that of the single panel. It even decreases to 0dB in the ideal case of an infinite panel. In fig. 1 it is assumed that m'' is identical for both cases. If one would double the mass of the single panel to yield the same mass as the two masses of the double panel together, the curve for TL_{sp} would be shifted up by 6dB as said before and the difference between the two curves around the mass-spring-mass resonance frequency would be bigger.

EXPERIMENTAL SETUP

Fig. 2 shows a schematic of the experimental setup. The width of the window is $l_x = 1$m and the height is $l_y = 1.25$m. The distance between the panels is $d = 19$cm. The panels are made of acrylic glass (density $\rho_p = 1150$kg/m^3) and their thickness is $h =$ 4mm each. The panels are built into a heavy wooden frame such that the boundary conditions of simple supported plates are approximated. The whole setup is built into a rigid wall. The two rooms on each side of the wall are used for acoustic excitation and for measurements respectively. With $m'' = \rho_p h$ the mass-spring-mass resonance of the double panel according to eq. (4) is $f_{dp} = 90.7$Hz.
Table 1 lists the natural frequencies of the plates which are given by

$$\omega_m = \sqrt{\frac{B}{m''}} \left[\left(\frac{m_1 \pi}{l_x} \right)^2 + \left(\frac{m_2 \pi}{l_y} \right)^2 \right] . \tag{5}$$

The bending stiffness B is given by

$$B = \frac{Eh^3}{12(1-\gamma^2)} \tag{6}$$

where the Young's modulus is assumed to be $E = 5.6 \cdot 10^9 \mathrm{N/m^2}$ and the Poisson's ratio $\gamma = 0.3$. The coincidence frequency of the panels which is given by

$$f_c = \frac{c_0^2}{2\pi}\sqrt{\frac{m''}{B}} \tag{7}$$

equals $f_c = 7050.9$Hz.

Table 1: Natural frequencies $f_m = \omega_m/(2\pi)$ of the panels.

	$m_1 = 1$	$m_1 = 2$	$m_1 = 3$	$m_1 = 4$
$m_2 = 1$	6.9Hz	19.5Hz	40.5Hz	69.8Hz
$m_2 = 2$	14.9Hz	27.5Hz	48.5Hz	77.9Hz
$m_2 = 3$	28.4Hz	40.9Hz	61.9Hz	91.3Hz
$m_2 = 4$	47.2Hz	59.8Hz	80.7Hz	110.1Hz

The natural frequencies of a rectangular cavity with assumed rigid walls at all six sides are listed in table 2. They can be calculated by

$$\omega_n = \pi c_0 \sqrt{\left(\frac{n_1}{l_x}\right)^2 + \left(\frac{n_2}{l_y}\right)^2 + \left(\frac{n_3}{d}\right)^2}. \tag{8}$$

In table 2 only some natural frequencies for $n_3 = 0$ are listed. The lowest natural frequency for $n_3 = 1$ is $f_{001} = 905.3$Hz and is far beyond the frequency range of interest. It is obvious that only modes with constant sound pressure along the direction perpendicular to the panels will be excited.

Table 2: Natural frequencies $f_n = \omega_n/(2\pi)$ of the air cavity.

$n_3 = 0$	$n_1 = 0$	$n_1 = 1$	$n_1 = 2$	$n_1 = 3$
$n_2 = 0$	0	172.0Hz	344.0Hz	516.0Hz
$n_2 = 1$	137.6Hz	220.3Hz	370.5Hz	534.0Hz
$n_2 = 2$	275.2Hz	324.5Hz	440.5Hz	584.8Hz
$n_2 = 3$	412.8Hz	447.2Hz	537.4Hz	660.8Hz

The loudspeakers as well as the error microphones are placed near the corners of the window. The reasons being, firstly, that only a minimum of equipment should disturb the view through the window, and secondly, that in an enclosure with rigid walls the sound pressure is maximum at the corners. With the arrangement shown in fig. 2 the nodal lines of the cavity modes are avoided up to high frequencies for both the loudspeakers and the microphones.

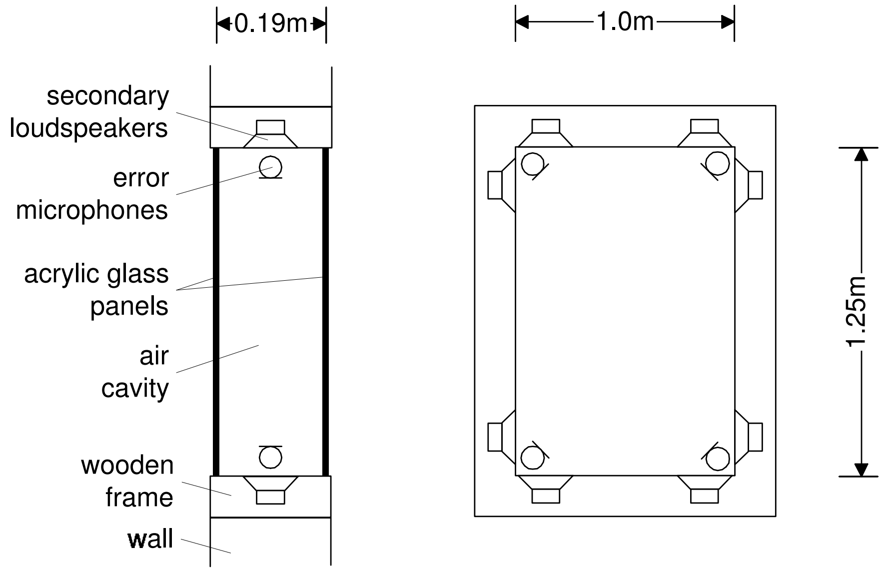

Figure 2: Schematic of the experimental setup

A four-by-four channel controller was available for the experiments, i.e. a controller which is able to minimize the mean squared error *mse* of four error sensors by means of four secondary sources. From these eight loudspeakers two loudspeakers are driven together by one channel of the controller. From the many combinations possible two arrangements were tested. In the first arrangement the loudspeakers at each corner were grouped together, in the second arrangement the loudspeakers along each side were grouped together.

For the adaptive algorithms in a feed-forward arrangement generally the secondary paths have to be identified. Each of the secondary paths include one digital-to-analog converter, one loudspeaker pair with its amplifier, the acoustic cavity, one microphone with its amplifier and one analog-to-digital converter. Fig. 3 shows the frequency response amplitudes of three secondary paths i) secondary path with a corner loudspeaker pair (solid line), ii) secondary path with a horizontal-side loudspeaker pair (dashed line) and iii) secondary path with a vertical-side loudspeaker pair (dashdot line). All three examples include the same microphone. The sampling frequency of the digital system was set to $f_s = 500$Hz. Besides the lower three resonance frequencies which are due to the experimental setup, i.e. the wooden frame and the wall, which were actually not really rigid, one can see the mass-spring-mass resonance at approximately 85Hz, i.e. the smaller one, and three higher resonance frequencies which belong to the first three cavity modes. It is noticable that all loudspeaker pairs can excite the cavity at the mass-spring-mass resonance but not all loudspeaker pairs can excite all of the cavity modes to 250Hz, which are the (0,1,0)-mode at $f_{010} \approx 160$Hz, the (1,0,0)-mode at $f_{100} \approx 194$Hz and the (1,1,0)-mode at $f_{110} \approx 237$Hz. The peaks have shifted slightly away from the frequencies calculated in table 2 due to the structural-acoustic coupling between the plates and the cavity, and due to the loudspeakers which are non-rigid irregularities in the rigid side walls. Whereas the corner pair can excite all of the cavity modes the horizontal-side loudspeaker pair can only excite modes with symmetric sound pressure field along the horizontal axis such as the (0,1,0)-mode and the vertical-side loudspeaker pair can only excite modes with

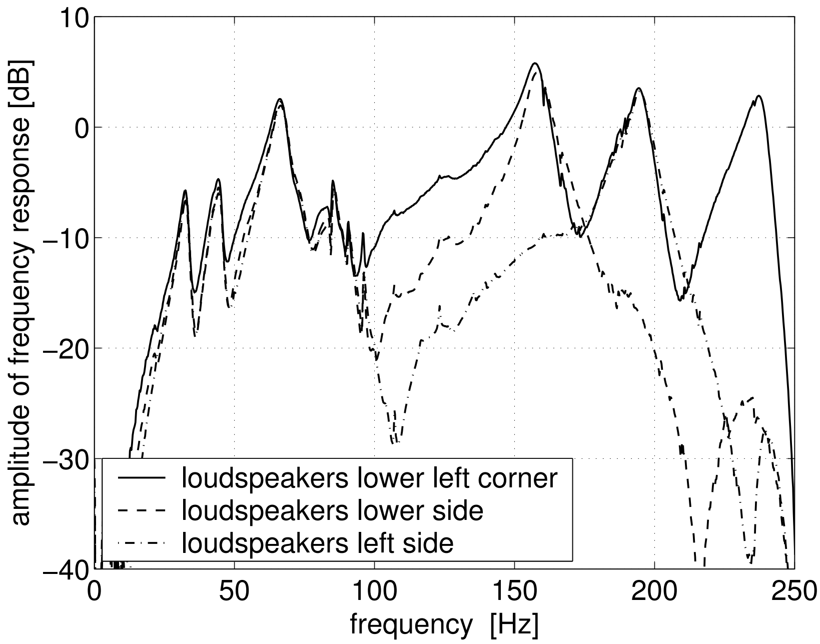

Figure 3: Typical frequency response of the secondary paths between different loudspeaker pairs and the same microphone, here: lower left microphone.

symmetric sound pressure field along the vertical axis such as the (1,0,0)-mode. None of the side loudspeaker pairs can excite the (1,1,0)-mode.

For the dimensions of the loudspeakers some general considerations were made [8]. In air the relation between changes in pressure and changes in volume is given by

$$\frac{\Delta p}{p_=} = 1.4 \frac{\Delta V}{V_0} \, . \tag{9}$$

The static pressure in air is assumed to be $P_= = 10^5\mathrm{N/m}^2$. The Volume of the cavity is $V_0 = l_x \cdot l_y \cdot d = 0.24\mathrm{m}^3$. We demanded that the loudspeakers inside a rigid wall cavity could achieve a sound pressure level of 100dB, i.e. $\Delta p = 2\mathrm{N/m}^2$. To satisfy eq. (9) a change in Volume $\Delta V = 3.4 \cdot 10^{-6}\mathrm{m}^3$ is needed. When using a loudspeaker which is able to lift its membrane about 1mm this will be achieved by a membrane area of $34\mathrm{cm}^2$ which corresponds to a circular loudspeaker with a diameter of approximately 6.6cm. So relatively small loudspeakers would be sufficient to produce high pressure in a rigid cavity. Because of the vibrating panels the walls of the cavity considered here are not rigid and so we use bigger loudspeakers with a diameter of approximately 11cm.

EXPERIMENTS

Some experiments were carried out using one, two, three and four pairs of loudspeakers each. In all tests the signals of all four microphones were used to build the mean squared error *mse*, i.e. the sum of the four squared microphone signals was minimized by the feed-forward controller. A mono-frequent excitation was used in the experiments and the frequency was varied between 60Hz and 200Hz. The results are shown in fig. 4 to 8. In

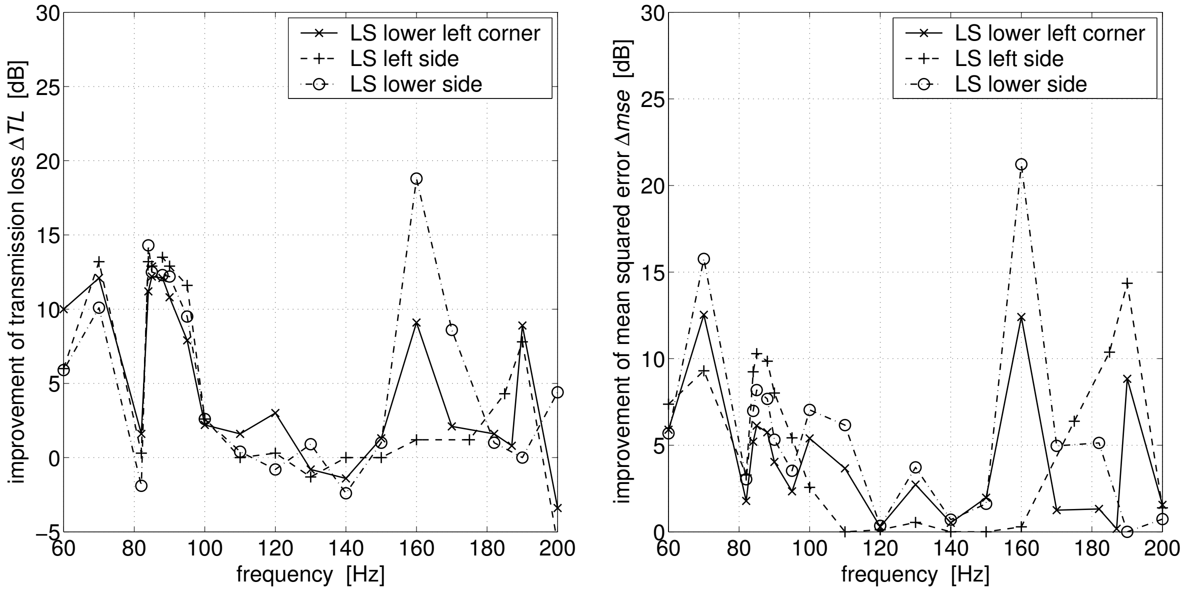

Figure 4: Improvement of the transmission loss ΔTL (left) and improvement of the mean squared error Δmse measured by the four microphones (right) after actively minimizing the cavity sound field with one pair of loudspeakers (LS) respectively.

fig. 4 to 7 the improvement of the transmission loss ΔTL is shown on the left half, i.e. the difference (in dB) between the transmission loss with control and the transmission loss without control. Clearly if ΔTL is positive it means that the transmission loss becomes higher with control and vice versa if ΔTL is negative. On the right half of fig. 4 to 7 the improvement of the mean squared error Δmse is shown. Clearly Δmse always must be positive because the minimization should result in a decreasing of the mse. ΔTL is a measure for the improvement of the sound insulation of the double panel whereas Δmse is a measure for the change of the sound field inside the cavity.

Fig. 4 shows the results of the minimization with one loudspeaker pair each. Three cases are investigated: i) one loudspeaker pair at a corner, ii) one loudspeaker pair at one horizontal side and iii) one loudspeaker pair at one vertical side. Around the mass-spring-mass resonance frequency in all three cases an approximately equal improvement of transmission loss of nearly 15dB was measured. In the frequency range of 100Hz to $130Hz$, i.e. the range between the mass-spring-mass resonance and the first cavity resonance only poor improvement of the transmission loss could be achieved and even a slight worsening was measured. The best improvement of the transmission loss was measured in the first cavity resonance (160Hz) after minimizing with the loudspeakers at the lower side. At this frequency the loudspeakers at the corner gave a medium improvement and the loudspeakers at the left side gave only poor improvement. This is easy to explain when looking at the diagram for the improvement of the mean squared error on the right half of fig. 4[1]. The (0,1,0)-mode, which is dominant at this frequency, cannot be excited by the

[1]For a similar example compare e.g. the theoretical results of the work of Heitfeld et.al. [8], [9].

left-side loudspeakers, so Δmse is nearly zero and therefore ΔTL also. The loudspeakers at the corner do excite the (0,1,0)-mode but also excite the (1,0,0)-mode. So some control spillover (compare e.g. [10], [11]) into the (1,0,0)-mode and of course in all higher modes limits the control effect, which results in a medium Δmse and therefore in a medium ΔTL. The lower-side loudspeakers mainly excite the dominating (0,1,0)-mode, allowing only little control spillover into higher modes. This occurs because these modes are all in a higher frequency range, e.g. the next mode which can be excited is the (0,2,0)-mode and the corresponding natural frequency is at $f_{020} = 275.2$Hz according to table 2.

Following the same scheme the results at 190Hz can be explained. Despite the fact that no measurement was made exactly in the resonance frequency $f_{100} = 194$Hz it can be seen quite well in the right half of fig. 4 that the lower-side loudspeakers give $\Delta mse \approx 0$ and therefore $\Delta TL \approx 0$ too. Again the loudspeakers at the corner give a medium Δmse and here the left-side loudspeakers give the highest Δmse. Certainly the same improvement of transmission loss was measured here with the corner and the left-side loudspeakers, which was as high as in the case of the corner loudspeakers at the first cavity resonance frequency.

In the next series of measurements two pairs of loudspeakers both on the sides were used to minimize the four microphone signals. Fig. 5 shows the measured results. The highest improvement of transmission loss was achieved at the mass-spring-mass resonance frequency at 85Hz by the loudspeakers at the left and right side. More than 30dB were measured. Besides that peak the high values between 80Hz and 100Hz are in approximately the same range and are $10 - 15$dB higher than in the case of only one loudspeaker pair. As can be seen from Δmse in the right half of fig. 5 the symmetric arrangements with the loudspeakers either on the left and right side or on the lower and upper side yields in a much better decrease of the mse than the non-symmetric arrangement with the loudspeakers at the left and lower side. Nevertheless, in this case only little influence on the improvement of the transmission loss was observed. In the higher frequencies a greater worsening was observed with the symmetric arrangements than in the case with only one loudspeaker pair. In that frequency range the use of the non-symmetric arrangement results in the highest ΔTL especially in the resonance frequency at 160Hz and only positive ΔTL was measured over nearly the whole frequency range of interest.

Fig. 6 also shows measured results when using two pairs of loudspeakers each, but here loudspeaker pairs at the corners where used. In the range of the mass-spring-mass resonance frequency the use of the lower left and upper right corners as well as the use of the lower left and upper left corners results in a slightly lower improvement of transmission loss than with the loudspeakers at the sides. The use of the lower left and lower right corners don't give better results than in the case with only one loudspeaker pair, but in the range at 110Hz it shows better performance than the other corner pairs. Compared to the loudspeakers at the sides the worsening at the higher frequencies is lower with the loudspeakers at the corners and all three corner arrangements give medium improvements of the mean squared error at both cavity resonancies.

Fig. 7 shows the measured results obtained by minimizing the four microphone signals by i) three loudspeaker pairs at the corners, ii) four loudspeaker pairs at the corners and iii) four loudspeaker pairs at the sides respectively. Here only the frequency range around the mass-spring-mass resonance is shown. It can be seen that in the mass-spring-mass

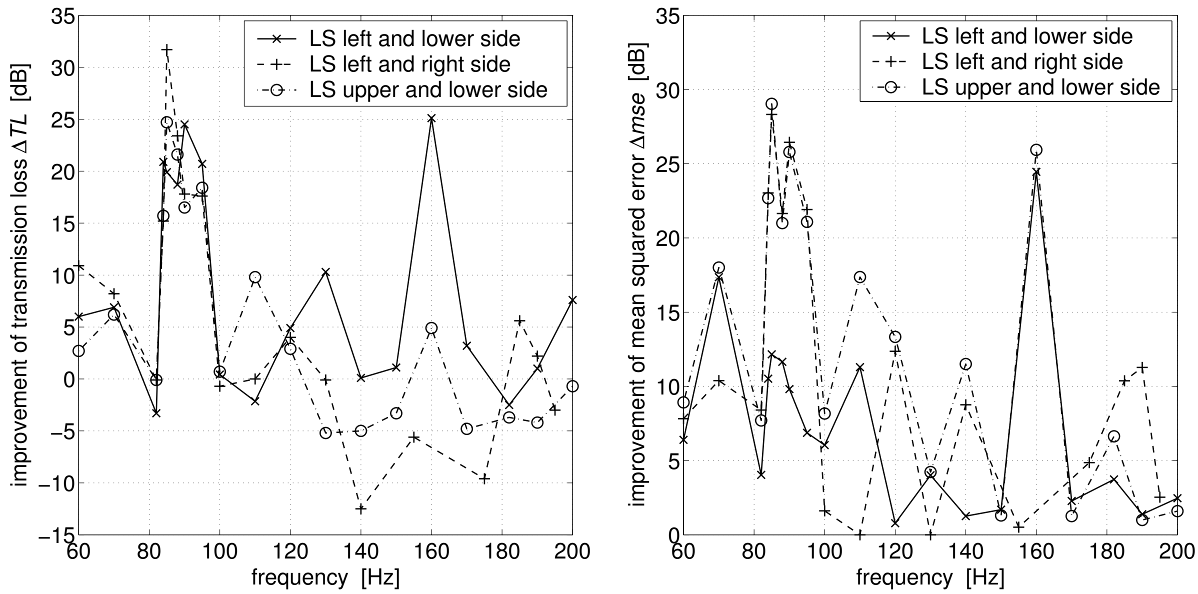

Figure 5: Improvement of the transmission loss ΔTL (left) and improvement of the mean squared error Δmse measured by the four microphones (right) after actively minimizing the cavity sound field with two pairs of loudspeakers (LS) at the sides respectively.

resonance frequency there is no difference in the improvement of sound insulation between three or four loudspeaker pairs at the corner, despite the fact that the mean squared error was reduced more substantially by the four loudspeaker pairs as shown in the right half of fig. 7. The improvement of transmission loss even decreases a little bit faster to zero with increasing frequency when four loudspeaker pairs are used than with three pairs. This can be explained in terms of control spillover. In general with four independent sound sources it is always possible to drive the signals of four error sensors to zero. The theoretical Δmse would tend to infinity for that case. This total silencing of single points in the sound field often results in an high increase in the sound field between the controlled points because of strong excitation of higher modes, that is called "spillover". If the number of error sensor used is higher than the number of sources used it is more likely to control a mean value of the whole field. That is the case with three loudspeaker pairs and four microphones. Only a slightly higher improvement with the four side pairs could be achieved. Not shown on the plot is that with the three corner arrangement good results were achieved between 160Hz and 190Hz with a maximum peak of $\Delta TL = 23$dB at 170Hz.

The amplitudes of the complex gains of the digital filters, which can be used as a measure of the control effort, are in the same range as with corner arrangements and side arrangements when using one or two loudspeaker pairs. When using three or four corner pairs the amplitudes of the complex gains are in the same range also, whereas the amplitudes of the gains with the four sides arrangement results in the highest values observed here. This is illustrated in fig. 8 where the mean values of the amplitudes of the three or four digital filters calculated from the filter coefficients at the corresponding frequencies

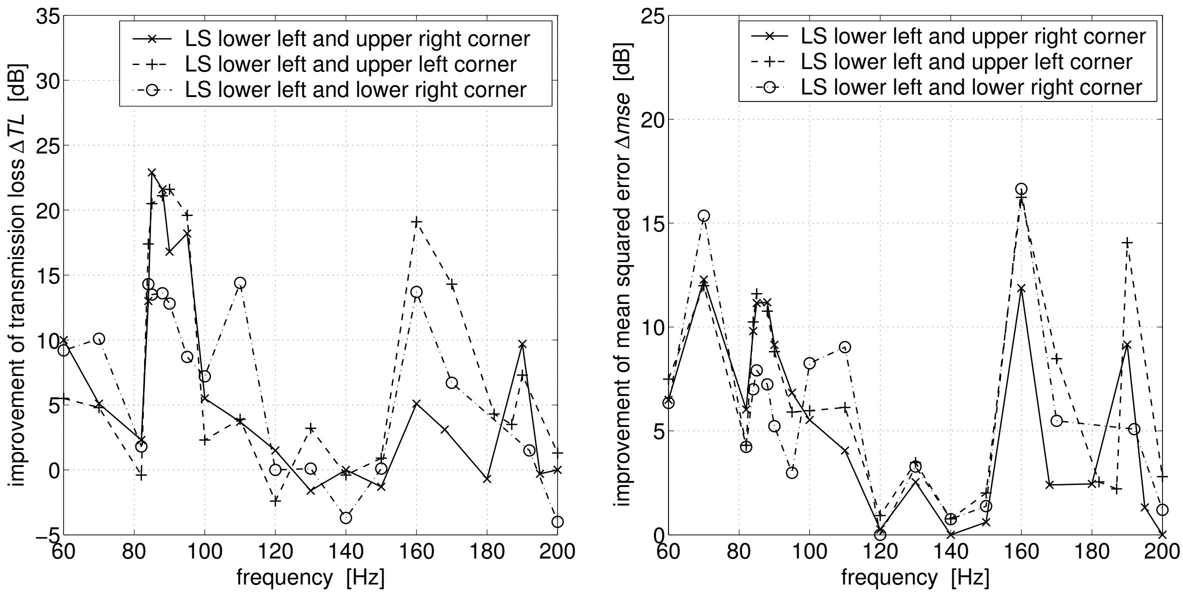

Figure 6: Improvement of the transmission loss ΔTL (left) and improvement of the mean squared error Δmse measured by the four microphones (right) after actively minimizing the cavity sound field with two pairs of loudspeakers (LS) at the corners respectively.

are displayed. Especially at the mass-spring-mass resonance the loudspeakers were driven relatively hard with the four side arrangement.

CONCLUSIONS

The active control of the transmission of sound through a double panel system was investigated experimentally. The system under investigation was a window made of two acrylic glass panels. To enhance the transmission loss the air cavity between the panels was actively controlled by means of loudspeakers mounted in the window frame. The influence of the number of active loudspeakers as well as the influence of two different types of loudspeaker positions, i.e. loudspeaker pairs at the corners and loudspeaker pairs at the sides, where considered. High improvements of the transmission loss can be achieved around the mass-spring-mass resonance frequency. With one loudspeaker pair $10 - 15$dB improvement of the transmission loss where observed. When using more than one secondary sound source improvements of more than 25dB are possible. With the corner arrangements all cavity modes can be excited, whereas with different side arrangements different modes can be excited. If one needs high improvements at one special cavity mode the choice of a suitable side arrangement could be preferable, because the control spillover into higher modes could be lower than with the corner arrangement. The latter can be used when medium performance in all cavity modes should be achieved.

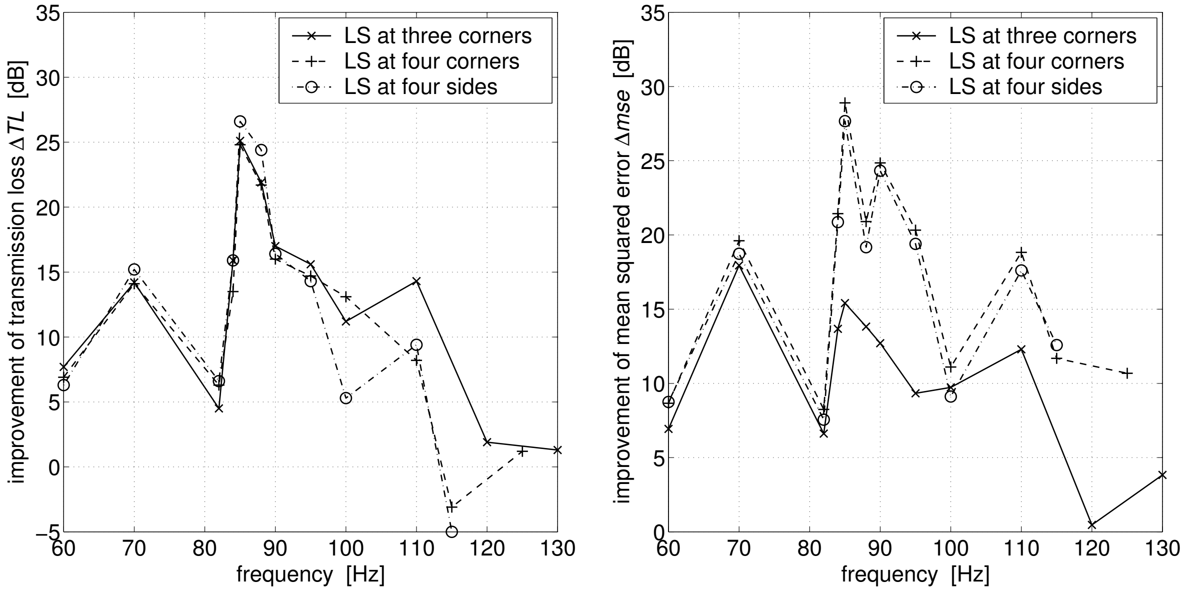

Figure 7: Improvement of the transmission loss ΔTL (left) and improvement of the mean squared error Δmse measured by the four microphones (right) after actively minimizing the cavity sound field with three or four pairs of loudspeakers (LS) respectively.

ACKNOWLEDGEMENTS

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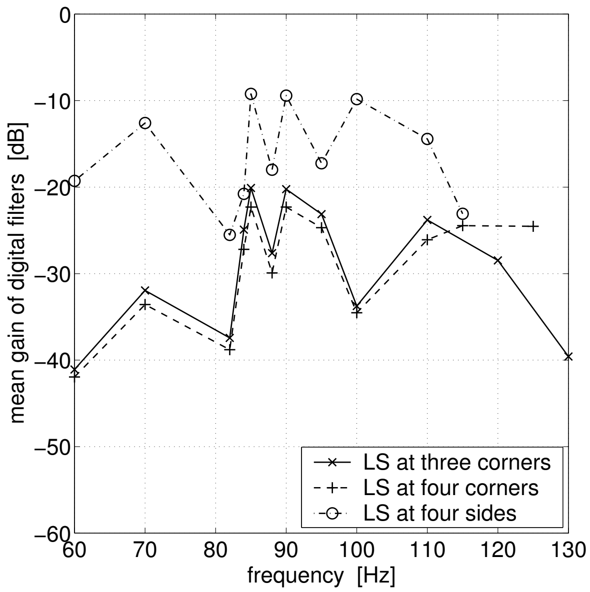

Figure 8: The mean amplitude of the complex gains of the digital filters calculated at the corresponding frequencies after actively minimizing the cavity sound field with three or four pairs of loudspeakers (LS) respectively.

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