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ACTIVE CONTROL OF THE CAVITY SOUND FIELD OF DOUBLE PANELS WITH A FEEDBACK CONTROLLER

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Abstract

Double panels 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. In the presented work the cavity sound field between the two panels of a double-glazing window is actively minimized by means of secondary loudspeakers in the cavity. The influence of the minimization on the transmission loss of the double panel was investigated experimentally. A special arrangement of loudspeakers in the window frame was used which gives a good excitation around the mass-spring-mass resonance with little control spillover into higher modes. Additionally a simple kind of a modal filter realized with symmetric arranged microphones results in a single channel controller. An adaptive feedback controller with Internal Model Control (IMC) architecture was used. Experiments with white noise and typical traffic noise samples were performed which show the feasibility of the feedback control to enhance the transmission loss of the double window.

INTRODUCTION

In the last years the focus of several authors was to investigate different active methods for the reduction of the sound transmission through double panels: 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.

In short it can be summarized that controlling the cavity sound field with loudspeakers inside the cavity is better suited to improve the transmission loss than controlling the vibration of one of the plates by vibration inputs. Sas, Bao et.al. [1, 2, 3] experimentally investigated the differences between panel and cavity control. Gardonio and Elliott [4] simulated the use of loudspeakers in the cavity and the use of active mounts between the two panels. Both the experimental and numerical studies showed that cavity control results in a higher increase in transmission loss than panel control. Panel control was investigated by Carneal and Fuller [5]. Their measurements showed that controlling the vibration of the radiating panel in a way that minimizes sound radiation (i.e. active structural acoustic control - ASAC) results in a better performance than controlling the vibration of the incident panel. Heitfeld et.al. [6] investigated cavity control of a small double-glazing window theoretically and experimentally. Jakob and Möser [7] investigated the influence of different loudspeaker positions inside a double-glazing window. Pietrzko and Kaiser [8] investigated a robust H_∞ controller which improves the transmission loss of a double window by means of cavity control too.

EXPERIMENTAL SETUP

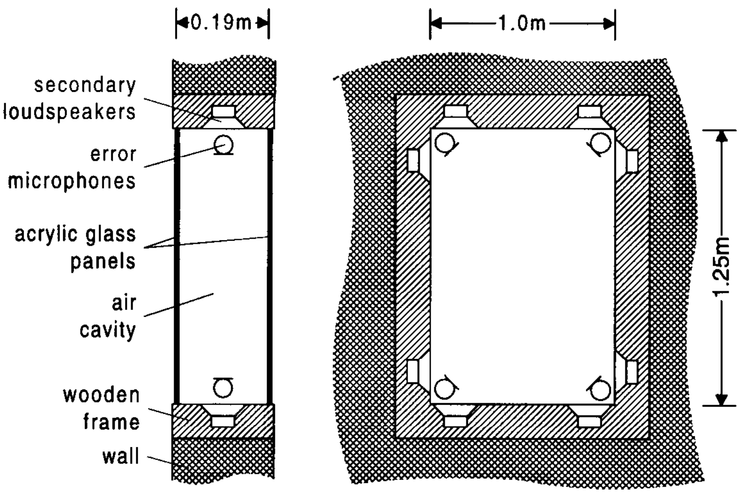


Figure 1: Schematic of the experimental setup

Fig. 1 shows a schematic of the experimental setup with dimensions given there. The panels are made of acrylic glass (density $\rho_p = 1150\text{kg/m}^3$, Young's modulus $E = 5.6 \cdot 10^9\text{N/m}^2$) and their thickness is 2mm 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 inserted into a rigid wall. The two rooms on each side of the wall are used for acoustic excitation and for measurements respectively.

For a first examination of the double panel some simple estimations of the natural frequencies will be given. The mass-spring-mass resonance frequency of infinite sized panels can be calculated according to [9] to be 126.8Hz in our case. The frequency region around this resonance defines our main region of interest.

Table 1: Calculated natural frequencies.

simple supported panel				rigid wall cavity			
	$m_1 = 1$	$m_1 = 2$	$m_1 = 3$	$n_3 = 0$	$n_1 = 0$	$n_1 = 1$	$n_1 = 2$
$m_2 = 1$	3.4Hz	9.7Hz	20.2Hz	$n_2 = 0$	0	172.0Hz	344.0Hz
$m_2 = 2$	7.5Hz	13.8Hz	24.3Hz	$n_2 = 1$	137.6Hz	220.3Hz	370.5Hz
$m_2 = 3$	14.2Hz	20.5Hz	31.0Hz	$n_2 = 2$	275.2Hz	324.5Hz	440.5Hz

For the next simple estimations of the plate and cavity resonance frequencies the double panel system is divided into its components "panel" and "cavity". No structural acoustic coupling is considered here. But some first conclusions can be drawn from the knowledge of the approximated natural frequencies. Table 1 lists the natural frequencies of a rectangular simple supported plate with the material data of the used panels together with the natural frequencies of a rectangular cavity with assumed rigid walls at all six sides. It can be seen that the first panel resonance frequencies are very low. Therefore in our main region of interest around the mass-spring-mass resonance frequency many modes contribute to the plate vibration, or in other words: the modal density is high.

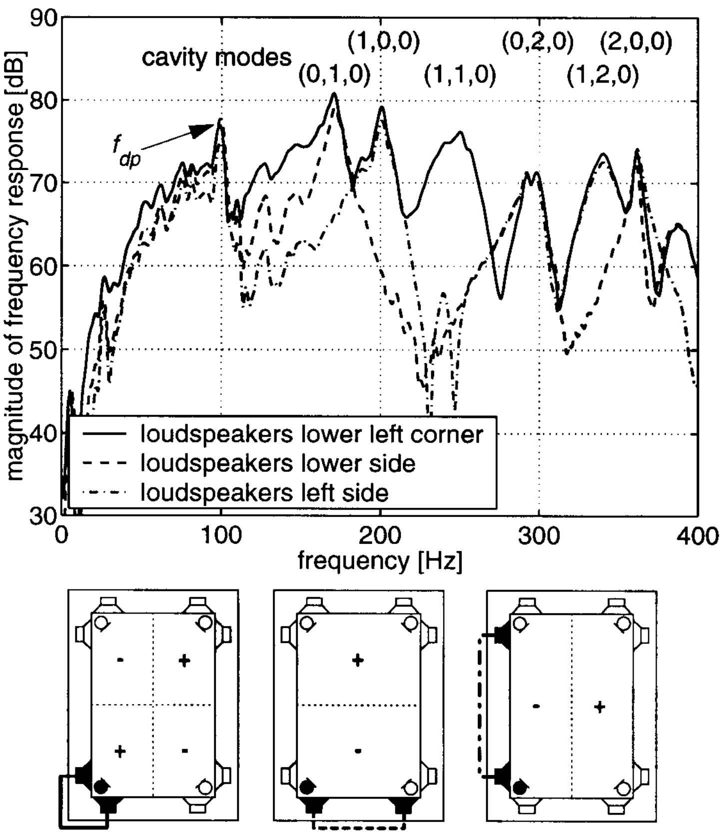


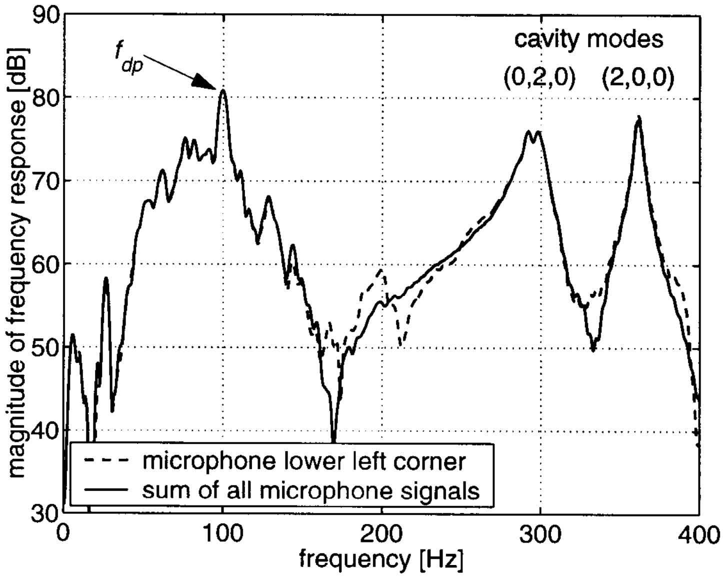
Figure 2: Typical frequency response of the secondary paths between different loudspeaker pairs and the same microphone, here: lower left microphone.

Only some natural frequencies corresponding to cavity modes with a constant sound pressure field along the direction perpendicular to the panels are given in table 1, because the lowest natural frequency with non-constant sound pressure along that direction is above 900Hz and therefore is far beyond the frequency range of interest. The first cavity mode with non-constant sound pressure along all directions occurs at 137.6Hz. Therefore around the mass-spring-mass resonance frequency only a few modes contribute to the cavity sound field, or in other words: the modal density in the cavity sound field is low in our main region of interest.

Because the modal density of the cavity sound field is lower than that of the plate vibration, the overall cavity sound field is easier to control than the overall plate vibration. So it can be assumed that controlling the cavity sound field yields in higher improvements of the transmission loss than controlling the plate vibrations. That means that using loudspeakers and microphones inside the cavity promises to be better than using vibration inputs and accelerometers on the plates. This of course is in good agreement to the results of the authors mentioned in the introduction.

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. 1 the nodal lines of the cavity modes are avoided up to high frequencies for both the loudspeakers and the microphones.

From these eight loudspeakers some loudspeakers are driven together by one channel of the controller. Maybe the most basic combinations of loudspeakers are shown in the



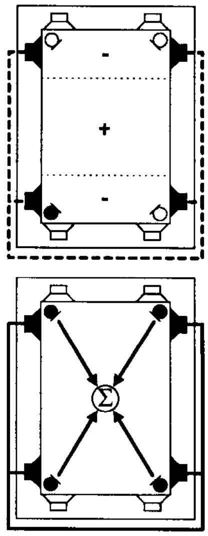


Figure 3: Typical frequency response of the secondary paths of other loudspeaker and microphone combinations (compare text).

lower part of fig. 2. In the first arrangement the loudspeakers at one corner were grouped together, in the second and third arrangement the loudspeakers along one horizontal side and one vertical side respectively were grouped together.

Usually with adaptive algorithms the transfer functions of the secondary paths have to be identified. The upper part of fig. 2 shows the magnitude of the frequency response of three secondary paths with the loudspeaker arrangements shown below. One can see the mass-spring-mass resonance at approximately 100Hz and some higher resonance frequencies which belong to the cavity modes. The peaks have shifted away from the frequencies calculated in table 1 to higher frequencies due to the structural-acoustic coupling between the plates and the cavity. 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. Whereas the corner pair can excite all of the cavity modes the side loudspeaker pairs can only excite modes with some symmetric sound pressure field along one of the axes as indicated for some examples in the lower part of fig. 2.

With these arrangements some tests where done which are described in [7]. The maximum achievable improvement of the transmission loss in the mass-spring-mass resonance is mainly limited by the control spillover into higher modes. As can be seen from the frequency responses in fig. 2 the control spillover of the side arrangements is lower than that of the corner arrangements because the side loudspeaker pairs cannot excite all modes. This behavior even can be made better when two opposite side loudspeaker pairs are used in parallel as indicated with "a)" in fig. 3. Now the symmetry of the excitation is more perfect and even fewer modes can be excited, which are shown in the frequency responses in fig. 3. When one is only interested in the frequency region around the mass-spring-mass resonance such a loudspeaker arrangement can be a suitable choice. The controller used would be one with one output channel and four input channels for the microphones.

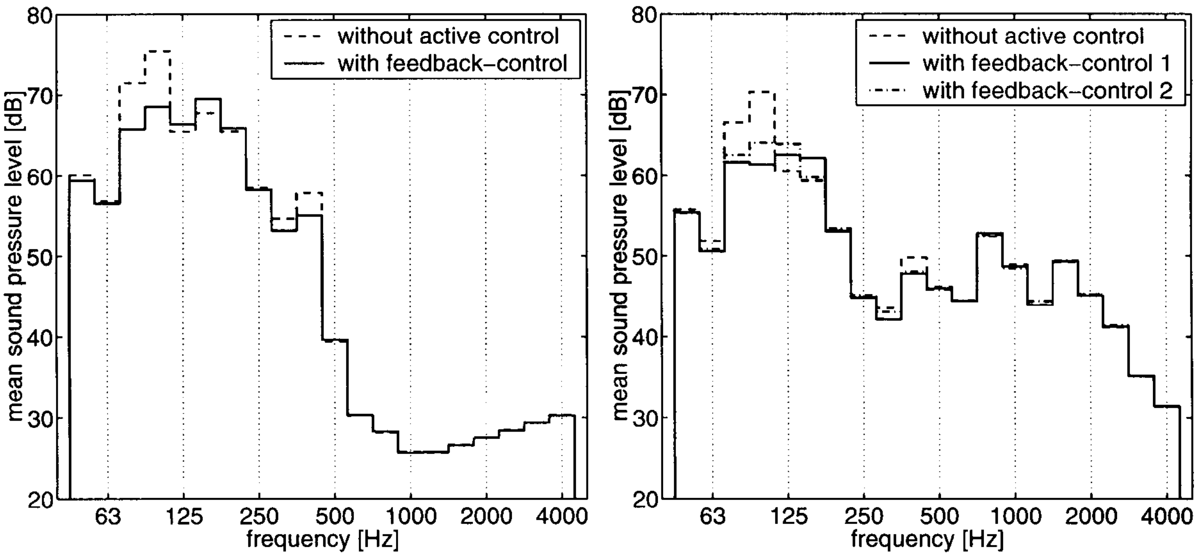


Figure 4: Measured third-octave level of the mean sound pressure in the receiving room with and without active control. Excitation with band limited white noise (left) and with typical freight train noise (right). Controllers 1 and 2 are explained in the text.

The number of input channels can be reduced when summing up the signals of the four microphones. This is motivated by the fact that only these modes needs to be observed that can be excited by the controller. This arrangement is indicated with "b)" in fig. 3 and the corresponding frequency response of the secondary path is shown too which is nearly identical to arrangement "a)". This sum of signals of symmetric arranged microphones is a simple kind of a modal filter. In [10] the results with the 1-by-4 and the last mentioned 1-by-1 controller are shown for monofrequent excitation. Both controllers give nearly identical results as can be expected.

With this single-channel-control-arrangement "b)" of fig. 3 all the following experiments with a feedback controller were performed.

ADAPTIVE FEEDBACK CONTROL

In the experiments presented here an adaptive feedback controller was used. The controller architecture used is well known as Internal Model Control (IMC) [11]. If the model of the secondary path is known with high accuracy the feedback control system can be rearranged to result in a feedforward control system. With this feedforward control system the well known filtered-x Least Mean Squares (LMS) algorithm can be used to optimize the filter coefficients of the controller. The algorithm is given in e.g. [12, 13] with a modification given in [13] which makes the controller robustly stable.

The most advantage of the adaptive IMC controller is that it is a ready to use technique, i.e. no calculations have to be made off-line like solving Riccati equations, no model has to be build like a state-space model. The main disadvantage is the high effort of online calculations. It is even higher than that of the feedforward filtered-x LMS algorithm due to the needed reference input signal estimator [12] which is an additional filtering with the secondary path models.

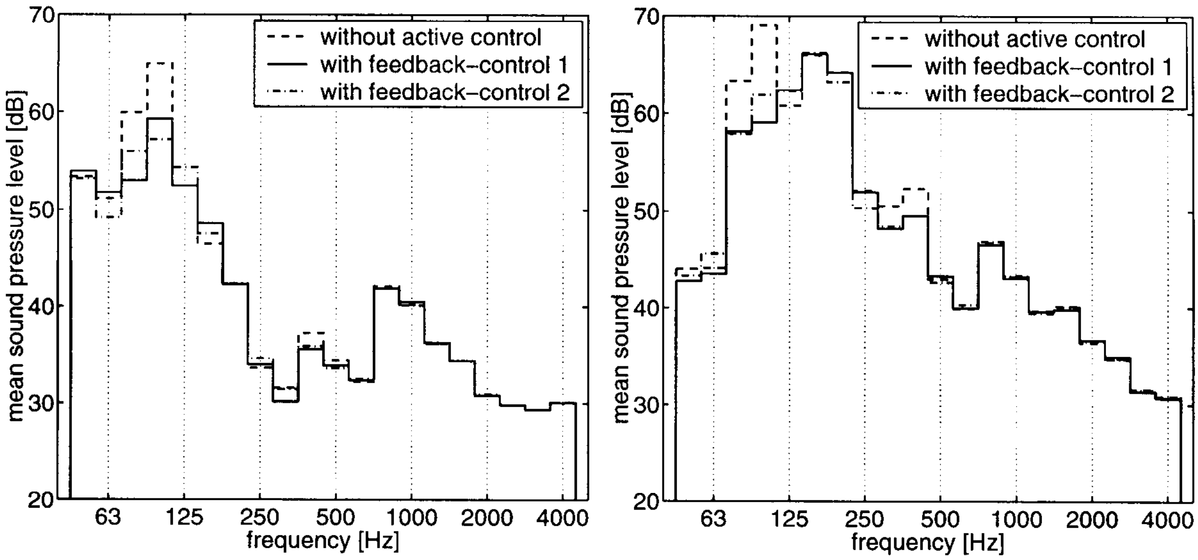


Figure 5: Measured third-octave level of the mean sound pressure in the receiving room with and without active control. Excitation with highway noise (left) and with jet aircraft noise (right). Controllers 1 and 2 are explained in the text.

EXPERIMENTS

Some experiments were carried out using the single channel adaptive IMC feedback controller. The results are shown in fig. 4 to 6. Fig. 4 and 5 each show the mean sound pressure level in the receiving room with and without active control measured in third-octave bands. The first test was done using band-limited white noise. The frequency range of the controller was limited to approximately 400Hz due to the anti-alias filters of the digital system. As can be seen from the left half of fig. 4 the highest mean sound pressure level without active control was measured in the third-octave band with center frequency 100Hz because there the mass-spring-mass resonance frequency is included (dashed line). After adaptation of the feedback controller the sound level was measured again and the solid line resulted. As could be expected the main improvements in transmission loss were achieved in the third-octave of the mass-spring-mass resonance, i.e. nearly 7dB and 5.7dB of improvement were achieved in the 80Hz third-octave band. Some worsening in the 160Hz third-octave band could be observed due to the fact, that the controller cannot work well because no modes in that frequency region can be excited (compare frequency response of fig. 3). The overall improvement of the transmission loss with the active controller, i.e. the difference between overall sound levels with and without control, was calculated to be 3.3dB.

Because the double panel was designed to be a double-glazing window and intended to protect against traffic noise some tests were performed with typical noise samples of a freight train, of a highway and of a jet aircraft. The right half of fig. 4 shows the measured sound levels for the freight train test with and without control for two "different" controllers. Controller 1 names the controller which was adapted with the excitation with white noise while the test before. Controller 2 names the controller which adapted to the sound of the freight train. In both cases again the most improvements were found in the

100Hz third-octave band and in the 80Hz third-octave band. In the case of controller 1 the improvement in the 100Hz third-octave band was nearly 9dB and in the 80Hz third-octave band it was nearly 5dB. The overall reduction was 3.9dB. When the controller was adapted to the train noise the improvements were found to be slightly lower. This was due to the fact, that the train signal is much more instationary than the white noise. It is more difficult to adapt the controller even in the non-realistic case of the repeated pass-by of the freight train. In practical applications it could be helpful to "pre-adapt" the controller with some suitable test signal like white noise. Maybe other algorithms (here the LMS was used) like Fast Transversal Filters could be another solution.

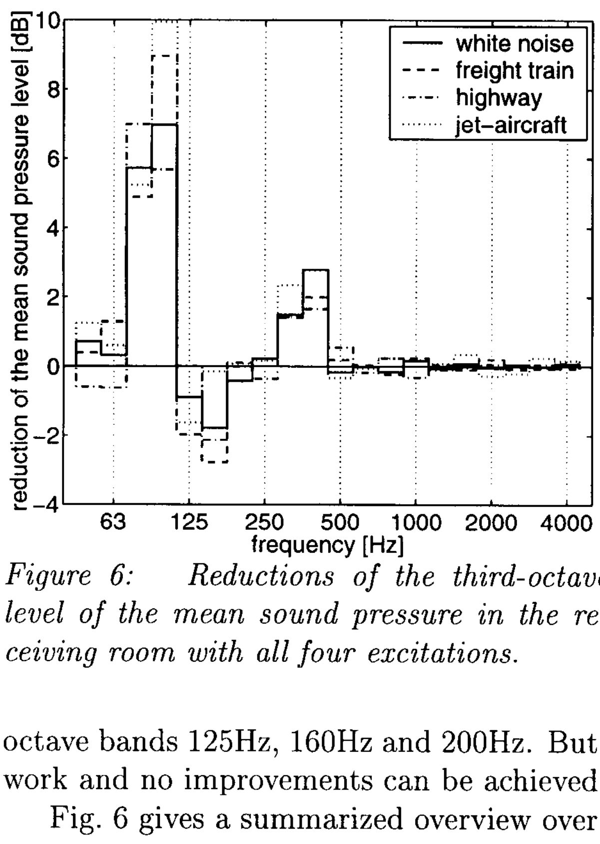


Figure 6: Reductions of the third-octave level of the mean sound pressure in the receiving room with all four excitations.

In the case of the highway noise (left in fig. 5) the adaptation to highway noise performed better because 1) of the more stationary signal of the noise from the busy highway compared to the train and 2) of the longer time the controller was able to get an input signal from the highway. The overall performance was 4.3dB with the pre-adapted controller and 4.8dB with the highway noise adapted controller.

The last case tested was noise from a jet aircraft (right in fig. 5). Again the most improvements were achieved in the 100Hz third-octave band which was nearly 10dB in the pre-adapted case. In spite of it the overall improvement was only approximately 2.5dB with both controller 1 and 2. Besides the high sound levels in the 100Hz third-octave band also high sound pressure levels were measured in the third-octave bands 125Hz, 160Hz and 200Hz. But in this frequency region the controller cannot work and no improvements can be achieved.

Fig. 6 gives a summarized overview over the frequency regions in which the transmission loss was to be improved or worsened. The differences between the measured sound level without and with control shown there clearly reflect the frequency response of the secondary path shown in fig. 3, i.e. high improvements in the region of the mass-spring-mass resonance frequency, worsening in the region where no cavity mode can be excited (mainly 125Hz and 160Hz third-octave band) and some medium improvements in the region of the cavity modes which can be excited.

CONCLUSIONS

The active feedback 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 four loudspeakers on the vertical opposite sides were used in parallel as the secondary source. The sum of the signals of the four microphones was used as the error signal. The controller used was a single channel adaptive feedback controller with LMS algorithm in the Internal Model Control (IMC) architecture. The experiments where performed with white noise and different traffic noise samples. The highest improvements of the transmission loss can be achieved around the mass-spring-mass resonance frequency. Reductions of 7-10dB can be achieved in the "best" third-octave band. The overall sound level reductions were measured to be 2.5-4.8dB. Due to the instationarities of real traffic noise sources it can be useful to pre-adapt the controller with white noise excitation. Further investigations will be done to improve the transmission loss in a sense that the improved frequency region will be wider.

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