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ACTIVE CONTROL OF THE SOUND TRANSMISSION THROUGH A DOUBLE-GLAZING WINDOW

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1. INTRODUCTION

Besides their better thermal insulation characteristics, double-glazing windows generally provide also an improved sound transmission loss compared to single-glazing windows, except in the low frequency band. As active noise control techniques are very effective in the low frequency band, an experimental and theoretical analysis have been performed on the use of control loudspeakers and error microphones in the cavity of a finite double-glazing window to improve its low frequency sound transmission loss.

In most cases the effectiveness of an active noise control system strongly depends on the spatial distribution of the secondary loudspeakers and error sensors. In the present analysis the positions of both the error sensors and control loudspeakers have been optimised simultaneously using the vibration energy of the radiating glass plate as the objective function to minimise. In order to limit the computing time a data reduction is required. Therefore a numerical modal analysis has been performed using a coupled fluid-structure finite element formulation. The model has been validated by an experimental modal analysis on the test set-up. Assuming a perfectly coherent reference signal in the feedforward control algorithm, the reduction in vibration energy can be calculated for any configuration of the control actuators and error sensors. A simple genetic algorithm is used to generate a near-optimum configuration with respect to the objective function.

The resulting control configuration has been implemented on a laboratory test set-up. The obtained reductions in the transmitted sound power illustrate the potentials of active noise control for increasing the sound transmission loss of a double-glazing window at low frequencies.

2. SOUND TRANSMISSION THROUGH A DOUBLE-GLAZING WINDOW

The double-glazing window under investigation, shown in figure 1, consists of two 3 *mm* thick glass plates of 1.23 *m* x 1.48 *m*. The edges of the glass plates are attached to a concrete wall with of a layer of putty. The distance between both plates is kept constant at 0.1 *m* by means of a wooden frame. The inner dimensions of this frame (1.1 *m* x 1.35 *m*) determine the boundaries of the acoustic cavity between the glass plates.

This test set-up is incorporated in the partition of the transmission room in the Laboratory of Acoustics at the K.U.Leuven. An incident sound field was generated by a woofer in the sending room. The transmitted sound power was measured by scanning a one dimensional intensity probe along the glass plate at the reception side. The insertion loss (IL) is defined as the difference between the transmitted sound intensity of a double and a single-glazing window, obtained by removing one glass plate :

$$IL[dB]=10\log\left(\frac{I_{single}}{I_{double}}\right) \qquad (1)$$

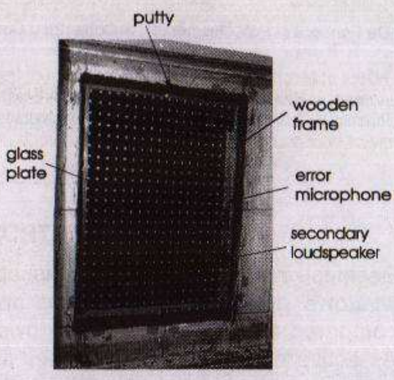

Fig. 1 : Test set-up of the double-glazing window

The sound transmission characteristics of normally and obliquely incident plane waves through double-panel partitions are well known in literature [1]. Compared to single panels, infinite double-panel partitions yield a higher sound transmission loss, except around certain resonance frequencies. On one hand, there is the low frequency mass-air-mass resonance of the double wall. On the other hand, due to the finite distance between both panels, there are cavity resonance frequencies, which appear - for common cavity thicknesses - at higher frequencies.

However, when both panels are finite in extent, the sound transmission characteristics look different, especially in the low frequency range. As the panels and the cavity are finite in all their dimensions, several coupled structural-acoustical resonances appear already at low frequencies. Together with a 'mass-air-mass'-like phenomenon these resonances result in a substantial decrease of the low frequency transmission loss [2]. For the considered double-glazing window the insertion loss at the 'mass-air-mass'-like resonance (around 70 Hz) and at the coupled resonances (e.g. around 100 and 160 Hz) decreases substantially (see figure 2). In order to improve the low frequency transmission loss, the potential of an active noise control system, implemented in the cavity, has been investigated.

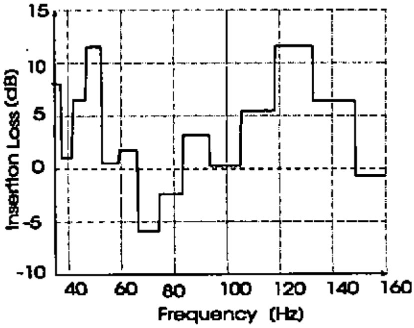

Fig. 2 : Insertion loss of the double-glazing window (in 1/6 octave bands)

3. ACTIVE CONTROL OF A DOUBLE-GLAZING WINDOW

3.1 Dynamic behaviour of a coupled vibro-acoustic system

A coupled finite element model of the dynamic behaviour of a vibro-acoustic system is (neglecting any type of damping) :

$$(K-\omega^2 M)\{X\}=\left(\begin{bmatrix} K_s & K_c \\ 0 & K_a \end{bmatrix}-\omega^2\begin{bmatrix} M_s & 0 \\ M_c & M_a \end{bmatrix}\right)\begin{Bmatrix} u \\ p \end{Bmatrix}=\begin{Bmatrix} F_s \\ F_a \end{Bmatrix} \tag{2}$$

The column vector of unknowns $\{X\}$ contains the displacements u in the structural nodes and the pressures p in the acoustical nodes. K_s, K_a and M_s, M_a are the structural and acoustical stiffness respectively mass matrices, while the fluid-structure coupling interaction is taken into account in the coupling matrices K_c and M_c. F_s and F_a are external excitation vectors in the structural respectively acoustical nodes.
By using the modal transformation

$$\{X\}=[\Phi]\{q\}, \tag{3}$$

in which the columns of matrix $[\Phi]$ are the undamped eigenvectors of the coupled vibro-acoustic system, orthonormalised with respect to the mass-matrix M, and by assuming proportional damping, the modal model becomes

$$[a]\{q\}=\left(diag(\omega_i^2)+j\omega\, diag(2\varsigma_i\omega_i)-\omega^2[I]\right)\{q\}=[\Phi]^H\begin{Bmatrix} F_s \\ F_a \end{Bmatrix}. \tag{4}$$

The ω_i are the coupled natural frequencies and the ζ_i are the modal damping factors (^{H} denotes the complex conjugate transpose).

3.2 Implementation of an active noise control system

When an active noise control system consists of c control loudspeakers, the total dynamic response of a vibro-acoustic system to the primary excitation and the secondary control excitation is :

$$\{X_{total}\}=\{X_p\}+\{X_c\}=[\Phi][a]^{-1}[\Phi]^H\left(\begin{Bmatrix}F_{sp}\\F_{ap}\end{Bmatrix}+[P]\{\mathbf{F}_c\}\right), \tag{5}$$

in which the subscript p refers to the primary disturbing excitation, while the subscript c refers to the control excitation. The vector $\{\mathbf{F}_c\}$ (cx1) contains the source strength of each control loudspeaker. The matrix $[P]$ indicates the nodal position of each control loudspeaker. The controller determines the strengths $\{\mathbf{F}_c\}$ of the control loudspeakers by minimising a cost function **E**, which is dependent on the signals of the error sensors $\{\mathbf{S}\}$. When the control configuration contains e error sensors, the vector of error signals $\{\mathbf{S}\}$ (ex1) is :

$$\{\mathbf{S}\}=[S]\{X_{total}\}, \tag{6}$$

in which the matrix $[S]$ indicates the nodal position of each error sensor. A common cost function **E** is the sum of the squared error signals :

$$\mathbf{E}=\{\mathbf{S}\}^H\{\mathbf{S}\}. \tag{7}$$

The resulting $\{F_c\}$ that minimises **E**, is ($^+$ denotes the pseudo-inverse) :

$$\{\mathbf{F}_c\}=-\left([S][\Phi][a]^{-1}[\Phi]^H[P]\right)^+[S][\Phi][a]^{-1}[\Phi]^H\begin{Bmatrix}F_{sp}\\F_{ap}\end{Bmatrix}. \tag{8}$$

When a vibro-acoustic system is subjected to a primary excitation and controlled by a certain control configuration (i.e. with a certain amount of error sensors e and of control loudspeakers c at positions $[S]$ respectively. $[P]$), the resulting response is obtained by combining (5) and (8).

3.3 Application to the double-glazing window

The response vector (5) for the considered double-glazing window was determined by calculating the matrix $[\Phi]$ in a coupled finite element modal analysis, while the natural frequencies ω_i and the modal damping factors ζ_i of [a] resulted from an experimental modal analysis. As the translation of the incident sound field in the sending room into nodal structural forces on the glass plate is not straightforward, the uncontrolled modal response vector $\{q_p\}$ was identified in a least squares form from a vector $\{X\}_{meas}$ of measured glass plate displacements and cavity pressures :

$$\min\left\|\{X\}_{meas}-[\Phi]\{q_p\}\right\|_2. \tag{9}$$

4. OPTIMISATION OF THE CONTROL CONFIGURATION

The effectiveness of an active control system is strongly dependent on the number and positions of both the error sensors and control loudspeakers (i.e. [*S*] and [*P*]). Instead of using a very time consuming trial-and-error procedure, the optimum control configuration can be determined from the vibro-acoustic model by optimising a certain objective function. Although the sound energy, radiated into the receiving room would be the most appropriate objective function, the reduction of the vibrational energy was used to limit the computational effort.

This objective function has been optimised by means of a genetic algorithm [3,4]. Genetic algorithms belong to the so-called directed random search techniques. The form of direction is based on Darwin's "survival of the fittest"-theories.

The feasible positions for the control loudspeakers and the error sensors (microphones) were restricted to 72 discrete positions in the wooden frame between the glass plates (see figure 1). Each position was coded in a 7-bit binary representation. One gene consisted of the binary representations for the respective control loudspeakers and error sensors one after each other. The initial mating pool contained 26 randomly generated genes. They are all attributed a weight based on the value of the corresponding objective function (their *fitness*). Depending on its weight each gene has a chance to be involved in a cross-over process, generating two new genes whose properties are determined by their parents properties. Besides the cross-over operator also genetic translation and mutation operators were included in the genetic algorithm.

The positions have been optimised for different types of control configurations with up to four error microphones and four control loudspeakers.

5. EXPERIMENTAL RESULTS

Some of the theoretically optimised control configurations have been implemented in the test set-up. For the experiments an adaptive multi-channel feedforward controller, developed at K.U.Leuven [5], has been used. Figures 3 and 4 show the measured sound intensity spectra in the receiving room for two different control configurations, both with two error microphones and two secondary loudspeakers.

Clearly, the performance of the active noise control system is much better for the optimised control configuration than for the arbitrarily chosen one, especially in the frequency band from 90 to 160 Hz. The obtained reduction in the mass-air-mass region (around 70 Hz) is almost insensitive to the control configuration (± 9 dB).

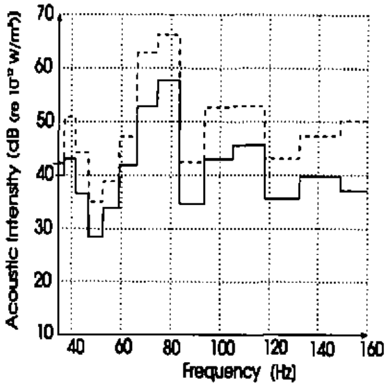

Fig. 3 : Acoustic intensity without (dashed line) and with active noise control (full line) for an optimised control configuration.

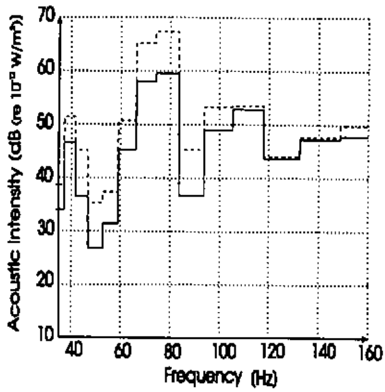

Fig. 4 : Acoustic intensity without (dashed line) and with active noise control (full line) for an arbitrarily chosen control configuration.

6. CONCLUSIONS

Compared to the single-glazing windows, finite double-glazing windows have a high sound transmission loss, except at low frequencies. This paper illustrates that an active noise control system, implemented in the cavity between both glass plates, yields an improved low frequency sound transmission loss. The effectiveness of the control system is strongly dependent on its configuration, i.e. on the number and the positions of the error sensors and control loudspeakers. A theoretical procedure to determine the optimal control configuration has been described and experimentally validated.

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