Noise Insulation applying Active Elements onto Facades

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Facades of modern buildings do not only serve as the outer skin but must fulfill various tasks. Besides static strength and visual facets, attenuation of ambient noise is a significant demand. However, up to now only passive methods (e.g., double- or triple-glazed windows and sound proofing materials) are being applied. These techniques show good results in the medium and high acoustic frequency range but significantly increase weight, volume and costs in the low-frequency regime, which is often intolerable in modern lightweight architecture. Active methods can overcome this problem. In the present work a facade specimen consists of double-glazed windows and panels with a sound shielding core, implemented in a stiff frame. With Active Structural Acoustic Control (ASAC), distributed laminar piezoceramic actuators induce vibrations into the plate-like elements that decrease the structural vibrations caused by the surrounding airborne noise, thus reducing the radiated noise. Furthermore, with piezoceramic stack actuators integrated into the frame it is possible to design an adaptive interface to decouple the facade element from the building, thus preventing structure borne sound from being transmitted into the building. In order to achieve maximum results an adaptive control system is designed. Both feedforward and feedbackward control are considered. A finite element model is developed that matches the structural behavior of the facade as closely as possible. Besides the material properties the boundary conditions are of great importance. Thus, a method is presented to determine these parameters almost automatically by means of a numerical optimization procedure.

1 Introduction

In modern architecture, facades serve as the outer skin of buildings and so have to fulfill the demand of static strength. However, if facades are planned in a suitable way, they can also act as a barrier for incident sound fields. Due to the fact that nowadays noise exposure in daily life is considered as one of the main environmental pollutions, low sound immissions into buildings have gained increasing importance over the past years. For the construction of large public buildings such as hotels, hospitals, airports or conference centers, which are often situated near airports, major roads, and railway lines, the acoustic shielding of ambient noise is very important and in many cases difficult to handle with classic passive methods. Double- or triple-glazed windows and sound proofing materials are able to reduce sound immission effectively above 1000 Hz. In the low frequency domain passive sound proving methods are often inapplicable because they lead to a significant increase in weight and cost exceeding the physical and financial limitations of the construction in many cases. On the one hand, facade elements should be constructed to be as light as possible. This helps to reduce the overall weight and reduces the cost. On the other hand, heavy and stiff elements can provide better acoustic shielding. However, if active noise control techniques like Active Structural Acoustic Control (ASAC) are used, it is possible to overcome the problems mentioned. The focus of the present work lies in the development of methods to improve the acoustic insulation of facades by active means. Thus, applying the principles of ASAC, it is possible to design light facade elements providing equal or even better noise attenuation than the passive counterparts. With ASAC, actuators designed of multifunctional materials are placed at determined positions on the plate-like elements. Inducing energy into the system, they can damp the sound radiating structural modes. When an appropriate adaptive control algorithm is applied the active system is able to react on incident sound fields almost immediately. The control loop is designed in Matlab/ Simulink (M/S) environment. With M/S it is possible to simulate and test different control strategies. The success of control of the simulation strongly depends on the reproduction of the control path as it is not only used to simulate the structural response but also for filtering purposes. The control path is derived by reducing the previously developed finite element (FE) model of the facade to only a few degrees of freedom.
This procedure postulates an exact FE model that matches the structural behavior of the real facade as closely as possible. Therefore, extensive structural and acoustical measurements have to be performed. Furthermore, a numerical optimization method ensures optimal boundary conditions.

2 Acoustical Measurements

The exact knowledge of the structural properties of a construction is a basic condition for the development of an active system. For ASAC, the identification of the lower natural frequencies and corresponding structural modes is very important. This information is required at a later stage for FE modeling and for the development of an appropriate control strategy. Therefore, structural tests and acoustic measurements must be performed in order to determine this data as exactly as possible.

At the facade test facility of Schueco Int. KG (Bielefeld, Germany) a facade specimen of typical design is set up in a special testing chamber (Fig. 1 left). The dimensions are 2.8 by 3.0 m, and it is assembled out of standard elements (three double-glazed windows, six aluminum panels, stiff frame) commonly used in facade construction. This specimen is referred to as ‘reference facade’ along the ‘active facade’ project and has therefore been measured extensively. When the natural frequencies and vibration modes in the frequency range of interest were determined, it turned out that, besides the laminar elements of the facade (windows, panels), at certain frequencies the framework also began to oscillate lively. Some resonances of the frame interfered with the natural frequencies of the plate-like elements making it difficult to clearly identify all of the structural modes.

Figure 1: left : Facade element at Schueco testing facility; right: Downscaled demonstrator at TUD

The results of the measurements at Schueco are used in the construction of a down-scaled model of the reference facade. This specimen is designed for testing and demonstration purposes. The physical dimensions are 1710 by 1260 mm and are chosen in such a way that it is possible to integrate the demonstrator into the semianechoic chamber at TUD (Darmstadt University of Technology) and one is able to transform the results to the dimensions of the reference facade by means of scaling laws.

This demonstrator facade has been measured with the same effort as the reference facade. Therefore, the structural responses to testing signals such as white and pink noise as well as pure sine signals have been recorded using loudspeakers (for airborne noise) and electro-dynamic shakers (for structure borne noise) as sources. Focus was given on the natural frequencies and vibration modes, the boundary conditions of the laminar elements as well as on the support between frame and wall. This information is important because the boundary conditions directly influence the structural behavior of the demonstrator along with the layout of the active system [1], as is explained later on. Besides the structural characteristics also the sound radiating behavior of the demonstrator near the resonant frequencies was examined.

Figure 2 shows the frequency response of the demonstrator to white noise in the signal room (averaged over all microphones in 1.25 m distance in the receiving room). The measurement has been done with a microphone array using 64 silicon microphones [2].

Figure 2: Frequency response in dB(A) to white noise

The areas in the receiving room showing the maximum sound pressure for each critical frequency have been located, too. This data was used for an acoustical rating of the critical modes. It turned out that only a part of the natural frequencies is of acoustical relevance.

It depends on several factors whether a resonant frequency of a structure leads to sound radiation or not. First of all, the frequency must occur in the excitation signal with sufficient strength. Furthermore, the magnitude of resonant vibrations of a structure depends on the system properties (geometry, material, etc.). The acoustic short-circuit is another factor as well as people’s hearing sensitivity. The interaction of all these factors makes it difficult to predict the acoustical relevance of the structural modes of the facade; thus, they must be confirmed by experiments. Resonant frequencies between 200 and 300 Hz have been identified as the ‘loudest’ noise emitters of the demonstrator facade.
3 Placement of Actuators

The demonstrator was reconstructed with the FE program Ansys. The results derived by the structural and acoustical measurements were thereby used as a data basis (Fig. 4). As previously mentioned, the boundary conditions of the laminar elements connected to the frame of the demonstrator along with the support of the frame to the wall exerts large influence on the structural properties of the system. The laminar elements are connected to the frame applying spring elements for all six degrees of freedom (three translatory, three rotatory) at the marginal nodes thus providing maximum variability between the two extremes “free” and “clamped”. The spring stiffnesses were derived by multiple calculus of variation. It was thus discovered that a boundary condition that is close to “clamped” meets the real boundary conditions best.

As previously explained the laminar elements of the demonstrator facade are responsible for sound radiation, implying that actuators must be attached at positions on the structure where their influence on the flexural modes is expected to be optimal. Thereby it is possible that an improved position for influencing a certain flexural mode of interest lies at a node of another one. Since the actuators are meant to have an effect on as many sound radiating modes as possible, compromises must be made regarding their positioning. Thus, the placement of the actuators is directly linked to the success of the control strategy of the active system. A commonly utilized criterion to estimate optimized actuator positions is a controllability index $\xi$ (e.g., [1]) derived by modal strains $\varepsilon$:

$$\xi_{\text{total}}(x,y) = \prod_{i=1}^{n} \frac{\varepsilon_i(x,y) + \varepsilon_{i\text{max}}(x,y)}{\varepsilon_{i\text{max}}}$$

$$\varepsilon_{i\text{max}} = \max(e_{i,x} + e_{i,y})$$

(1)

The flexural modes along the axis $x$ and $y$ are computed by the FE program, and the corresponding strains are stored for later use. The modal strains near the boundary strongly depend on the rotational degrees of freedom of the boundary conditions of the laminar elements and must thus be calculated with care.

The controllability index derived with (1) can be illustrated graphically. In Fig. 5 the results of the calculations for the window of the demonstrator facade are shown. The red peaks indicate a high controllability and thus good actuator positions whereas the dark blue areas mark nodal points. In the left image in Fig. 5, the first 12 flexural modes (between 0 and 450 Hz) are used. It is shown that only the corners remain for actuator positioning providing good controllability.

As mentioned before, not all of the flexural modes radiate sound equally. Regarding the results of the acoustical relevance evaluation some of the natural frequencies can be excluded from the examination. With the window of the demonstrator facade, modes between 200 and 300 Hz have been identified as ‘loud’ sound emitters. The corresponding index is shown in the right part of Fig. 5.

A restriction related to the project is that the elements of the active system (actuators, sensors) attached to the laminar elements must be placed in direct proximity of the boundary. Figure 6 shows the controllability at a distance of 50 mm from the boundaries. As is shown in Fig. 5 the index near the boundary shows quite good results. This assumption has been proven by experiments where actuators were applied at such positions.
4 Adaptive Control

The controller design is realized in Matlab/ Simulink (M/S) environment. M/S is an ideal tool for simulating various control strategies because, besides the digital controller itself, it is also possible to reproduce all the hardware that is needed in the experiment (like piezo amplifiers, analog filters, etc.). By utilizing adaptive filters, the complete signal path can be identified including control path and connected hardware. With M/S it is possible to verify the stability of the active control response in the virtual computer environment. Both complex load collectives and coincidental unique load cases can be considered in that context. Extensive time-consuming experiments during controller design are thus avoided. Being able to simulate an experiment completely in M/S, several conditions must be satisfied whereby the exact reproduction of the mechanical structure is crucial. Regarding the size of the simulated model and the efficiency of the active control mechanism, importing the complete structure into the simulation environment is impracticable. Therefore, the system has to be reduced to only a few degrees of freedom. This is realized by modal reduction of the equation of motion (2). The remaining degrees of freedom (master degrees of freedom – mDOF) must be chosen that way in order to be able to characterize the system behavior in the frequency band of interest, containing a maximum of vibrational information.

\[ M \ddot{\mathbf{w}} + D \dot{\mathbf{w}} + K \mathbf{w} = F \]  

(2)

In lightly damped systems, damping has only limited influence on the location of the natural frequencies. In (2) the damping matrix D along with the vector of the external forces F can be neglected within the reduction process [3]. Later on damping can be reimplemented into the reduced M/S-model by means of Rayleigh-damping or a damping ratio. The reduced form can be expressed as [4]:

\[ \mathbf{K} \tilde{\mathbf{\Phi}}_i = \lambda_i \mathbf{M} \tilde{\mathbf{\Phi}}_i \]  

(3)

In (3) \[ \mathbf{K} \] and \[ \mathbf{M} \] are the reduced stiffness- and mass-matrices, \( \tilde{\mathbf{\Phi}}_i \) are the shortened eigenvectors and \( \lambda_i \) the eigenvalues of the undamped system.

The reduced matrices are calculated within the Ansys FE program and transferred into the M/S environment. The correlation between the resonance frequencies of the FE model and the reduced system depends on the position and count of the selected mDOFs. By rule of thumb their number should be at least twice the count of the flexural modes in the frequency band of interest. A digital adaptive control system is characterized by the application of timevariant control algorithms permitting the controller to react on changes inside the control path immediately. Control is realized via an adaptive FIR (finite impulse response) filter. The coefficients of the adaptive FIR are modified by means of an LMS (least mean squares) algorithm [5] (see principle in Fig.7).

![Figure 7: Principle of LMS adaptive FIR filtering](image)

Actually a lightly altered derivative of the adaptive LMS is utilized. The filtered-X-LMS [6] applies the transfer function from actuator to sensor in the secondary path. The same transfer function is used for filtering the input signal of the LMS-block, thus reducing the effect of the error path on the controller signal. The performance of the controller is increased by applying the filtered-X strategy and the resulting in a higher quality of the output signal. In the following the adaptive controller concept is presented utilizing the double-glazed window of the demonstrator facade as control path. After the reduction the window maintains a set of eight mDOFs. A perturbation signal is applied to one of the nodes.

![Figure 8: Block diagram of adaptive control system](image)

The objective of the control is to minimize the motion of an observed second node, applying an actuator signal at the same point. This is the case if sensor and actuator are located at the same position (e.g., collocated). In Fig. 8 a Simulink block diagram illustrates this case.
In Fig. 9 the frequency response of the window for the controlled and the uncontrolled case is shown. At frequencies below 250 Hz magnitude reductions of about 20 dB and more are achieved. The position of the actuator was chosen in such a way that it was possible to excite many flexural modes. Nevertheless, some of the modes (3rd, 5th) can neither be observed nor controlled from this position. Consequently control shows no effect at these frequencies. Results show that adaptive control applying the filtered-X-LMS algorithm is a good alternative. But other derivates of the LMS algorithm are imaginable and will also be considered.

5 Numerical Optimization of the Boundary Conditions

As mentioned in Sec. 3 the correct modeling of the boundary conditions is crucial to the quality of the FE model. To adapt the behavior of the FE model to that of the real structure a numerical optimization technique [7] is developed, which optimizes the stiffness of spring elements along the edge of the structure in such a way that the natural frequencies of the FE model \( f_{i,FE} \) and of the real structure \( f_{i,real} \) match as closely as possible.

A rectangular plate (635 mm by 355 mm by 2 mm) made of aluminum is used as an example structure. The plate is supported by torsional spring elements alongside its edges (see Fig. 10). These springs have 26 different stiffnesses (nine at each edge along the long side of the plate, four at each edge along the short side). Thus, there are 26 design variables to be controlled and optimized by the optimization algorithm.

The optimization algorithm tries to minimize the objective function \( F \), which is the difference between the first five natural frequencies of the FE model \( f_{i,FE} \) and the first five natural frequencies of the real structure \( f_{i,real} \)

\[
\min F = \sum_{i=1}^{5} (f_{i,FE} - f_{i,real})^2. \tag{4}
\]

For this purpose Matlab’s optimization algorithm “fminsearch” (a variant of the simplex algorithm) is linked with the FE software Ansys. The user must provide an initial guess for the 26 design variables (spring stiffnesses), then Matlab initiates an FE calculation of the natural frequencies of the plate using Ansys. Ansys returns the objective function value to Matlab, then Matlab adapts the design variables appropriately and starts a new FE calculation. The iterations stop if the maximum number of iterations is reached or if there is no significant progress in the objective function anymore.

Fig. 11 shows the iteration history of one of the optimization runs. As can be seen the objective function value is reduced from about 5020 to about 390. The measured first five natural frequencies of a real plate supported by some frame were 87.5 Hz, 126 Hz, 177 Hz, 190 Hz, and 262 Hz. At the end of the optimization run, the first five natural frequencies of the corresponding FE model were 75 Hz (−12.5 Hz, −14%), 119 Hz (−7 Hz, −6%), 183 Hz (+6 Hz, +3%), 201 Hz (+11 Hz, +6%), and 258 Hz (−4 Hz, −2%).

These first preliminary results are not very satisfying yet but they are just a starting point. The next steps for improvement will include the testing of other Matlab optimization routines, a variation of the number and type of design variables, and the choice of another initial set of design variables. Finally, this method will be applied not only to simple example structures but to the real facade element described in Sec. 2 above.
6 Conclusions

A simplified facade element for detailed studies and demonstration purposes at TUD was constructed using the results of the measurements of the reference facade at the Schueco testing facility. During the measurements of the demonstrator facade, the fundamental data for the FE modeling work was collected. The data provides the basic information on which the design and number of the actuators is based. It has been shown that adaptive control strategies are suitable for active control and are capable of providing an efficient system for active noise insulation of facades. The effectiveness has been proven on the basis of simulations. The control concept can now be transferred into a real-time DSP environment. With rapid-prototyping systems like dSPACE it is possible to directly import the controller previously designed in M/S into the DSP-system. The dSPACE hardware transforms the digital signals into the analog world. Providing appropriate hardware (piezo amplifiers, analog filters for reconstruction, anti-aliasing, etc.) it is thus possible to directly drive actuators and connect sensors to confirm control strategies at the real world test stand (demonstrator facade).

In this paper only the noise emission (airborne noise) of the laminar elements of the facade has been considered. A second problem actually worked on is the transmission of structure borne noise and its active damping. Besides the flexural vibrations of the plate-like elements of the facade, exterior noise sources also excite longitudinal vibrations that are transmitted into neighboring facade elements and have undesired effects on the applied control system. Furthermore, structure borne noise can travel along the mounting of the facade deep into the structure of the building causing vibrations and sound emission at another place. The assumptions of active systems are equally suited for attenuation of structure borne sound as for flexural transversal modes. The adaptive control can be designed according to the same principles [9]. The constructional design and realization of such an adaptive interface for decoupling the facade from the building is a task actually worked on. In addition to the controller design there are continuing studies on structural intensity. Software development in the area of acoustic nearfield holography may also become helpful in the project.

References


