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# A METHODOLOGY FOR PLANNING AND INSTALLING AN ACTIVE NOISE OR VIBRATION CONTROL SYSTEM

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## ABSTRACT
The application of active noise or vibration control to a real application is usually a great challenge. The step from computer simulations, or even lab studies, to the real implementation is often much larger than first expected. This is especially true for multiple-input, multiple-output control where questions like "where shall I put my control microphones?" and "where shall I put my actuators?" become relevant. In the present paper, a methodology for performing an active noise control installation is presented. This includes methods for evaluating the system performance. The importance of optimization is highly stressed as maybe the most important step in the installation process.

## 1 - INTRODUCTION
At the Department of Telecommunication and Signal Processing, the University of Karlskrona/Ronneby (UK/R), we have had the opportunity to install and test active noise and vibration control systems on a rather large number of real applications or mock-ups, see e.g. [1-3]. Some of the installations have been very successful, others have resulted in little or no attenuation. Nevertheless, all of these projects have contributed one way or another to our bag of experience and have encouraged our belief that active noise and vibration control can be a very useful noise control tool in certain applications.
According to our experience, one of the most important stages in the installation process is the planning stage. This includes determination of the number of actuators and control sensors as well as their positions. At this stage, one should also determine how the system should be evaluated, since this can affect the layout of the control system.
The work in our research group has been somewhat concentrated on the control algorithms [4], [11], which of course play a very important role in an active control system. The choice of control strategy has a major influence on the overall stability and the convergence speed. These factors are important if the control conditions vary rapidly with time, since a "slow" control system will not reach the optimum attenuation under these circumstances due to bad tracking performance. The situation is a bit different under stationary conditions. Here, the maximum achievable attenuation generally depends on the configuration of actuators and control sensors rather than on the choice of control algorithm.
The intention of this paper is to share our experiences from active noise control applications using multiple-input, multiple-output (MIMO) control. Since MIMO control is limited to the attenuation of harmonic components (there are few exceptions to this rule, one being active control of higher modes in ducts), the discussion below is based on the assumption that the sound field to be controlled is generated by some rotating mechanism, e.g. a combustion engine or a propeller.

## 2 - NOMENCLATURE
For clarity, we would like to explain the terminology used in this paper:

- Actuators - Volume velocity or force sources connected to the control system. Usually loudspeakers or force actuators.

- Control sensors - Sensors used as feedback to the control system. Usually microphones or accelerometers.
- Evaluation sensors - Sensors used to evaluate the performance of the active control system. Usually microphones. These sensors are *not* used by the control system.
- Control positions - Positions for the control sensors.
- Evaluation positions - Positions for the evaluation sensors.
- Control path - The acoustical or mechanical frequency response function between an actuator and a control sensor.
- Evaluation path - The acoustical or mechanical frequency response function between an actuator and an evaluation sensor.

## 3 - EVALUATION METHODS

One of the most important issues to clear out when discussing an active noise control installation, is how the system performance shall be evaluated. This question applies to evaluation in the space domain as well as in the frequency domain. As an example, we would like to refer to a project that was performed in the late 80s regarding active noise control in cars. The control microphones were placed at the left and right side of the headrests on the driver and the passenger seat, i.e. close to the ears of the driver and the passenger. The system performance was measured using an evaluation microphone placed at a standard measurement position, in this case at the middle of the headrest. Evaluating the control system performance at the controlled frequency showed an attenuation of about 20 dB for higher rpms. When using the dBA-measure to evaluate the performance, the maximum attenuation was about 2 dBA units. The expectations on attenuation must therefore be in relation to the selected evaluation criteria.

In this example, the control microphones were placed fairly close to the evaluation positions. In most cases, this is not possible for a number of reasons. The control sensors must be put out of sight and protected from physical damage. Microphone mounting and cabling is extremely important from a production point of view.

For this reason, one important step is to determine the positions where the control system performance shall be evaluated. This might be in just a few positions as in the example above, or by examining the sound field using a mesh of microphones that covers the area of interest. In e.g. an aircraft application, one evaluation surface would perhaps be at the passenger ear level and perhaps another at the ear level for people walking in the aisle. The results from such measurements can be presented in a 2-D surface plot that illustrates very well the distribution of potential energy over the given surface.

The mesh must be sufficiently dense to adequately represent the actual sound field. The research group at UK/R usually specifies the grid size to be less than one third of the wavelength of the highest frequency of interest. Using this criterion, an upper frequency limit of 300 Hz results in a grid size of about 0.3 m. This is a bit stricter than prescribed by the spatial sampling theorem and generally results in good data. The sound registered in all evaluation microphones is recorded for all driving conditions that are to be included in the analysis. Since the analysis is restricted to harmonic components, we are usually only interested in the relative phase and amplitude at the evaluation positions. This can be achieved by calculating the cross spectrum between a reference point and all other evaluation points resulting in a $M_e \times N$ matrix, $\mathbf{B}_e$, with the following contents:

$$\mathbf{B}_e = \begin{bmatrix} G_{01}(0)/\sqrt{G_{00}(0)} & \ldots & G_{01}(f_i)/\sqrt{G_{00}(f_i)} & \ldots & G_{01}(f_{N-1})/\sqrt{G_{00}(f_{N-1})} \\ \vdots & \vdots & \vdots & \vdots & \vdots \\ G_{0M_e}(0)/\sqrt{G_{00}(0)} & \ldots & G_{0M_e}(f_i)/\sqrt{G_{00}(f_i)} & \ldots & G_{0M_e}(f_{N-1})/\sqrt{G_{00}(f_{N-1})} \end{bmatrix} \tag{1}$$

Here, $G_{00}(f)$ is the power auto spectrum at the reference point and $G_{0m}(f)$ is the cross spectrum between the reference point and evaluation point $m$. Each row in $\mathbf{B}_e$ contains the cross spectrum for the $N$ frequencies calculated by the FFT and there are as many rows as evaluation sensors. The cross-spectral values are normalized with the *rms*-value at the reference point, in order to obtain the correct amplitude at the evaluation point. Letting the $M_e \times 1$ vector $\mathbf{b}_e(f_i)$ represent one column of $\mathbf{B}_e$, this vector will contain the complex amplitude values for the corresponding frequency, $f_i$, at all evaluation points and thus represents the *primary* or original sound field at the frequency $f_i$ as measured by the evaluation sensors.

## 4 - OPTIMIZING THE SYSTEM

Unless the application has extremely simple geometry and total flexibility in terms of actuator positions, it always pays to do some kind of optimization of actuator and control sensor placement, as well as their numbers. The purpose of the optimization is generally to find the "best" configuration of actuators and control sensors from a given set of positions that are possible to use from a construction/design standpoint.

### 4.1 - Calculation of attenuation

Assume that the number of *possible* (or practically useful) actuator positions is $L_p$ and the number of *possible* control sensor positions is $M_p$. Then, the goal is to find the configuration of $L$ actuators ($L \leq L_p$) and $M_c$ control sensors ($M_c \leq M_p$) that will give the maximum attenuation according to the given evaluation criterion. If the criterion is to minimize the energy as measured by the evaluation sensors, we obtain the cost function $J$ as

$$J = \mathbf{e}_e^H \mathbf{e}_e \tag{2}$$

where $\mathbf{e}_e$ is the $M_e \times 1$ vector describing the controlled sound field as measured by the $M_e$ evaluation sensors at one particular frequency, $f_i$ and $H$ denotes the conjugate transpose. The vector $\mathbf{e}_e$ is calculated from

$$\mathbf{e}_e = x\mathbf{F}_e\mathbf{c}^\infty + \mathbf{b}_e \tag{3a}$$

where $\mathbf{b}_e$ is the $M_e \times 1$ vector describing the uncontrolled sound field as explained above, $\mathbf{c}^\infty$ is the $L \times 1$ vector of optimal (complex) controller weights (the LMS solution) and $\mathbf{F}_e$ is the $M_e \times L$ matrix of frequency response values from each actuator to each evaluation sensor (the evaluation path). Since $\mathbf{F}_e$ describes the frequency response at one single frequency, each element of $\mathbf{F}_e$ is just a complex number. The $x$ in equation (3a) is a complex reference signal generated internally within the controller. Since this variable has no effect on the LMS solution other than a complex scaling, we can set this to an arbitrary value, e.g. $x$=1, which leads to

$$\mathbf{e}_e = \mathbf{F}_e\mathbf{c}^\infty + \mathbf{b}_e \tag{3b}$$

Given $L$ and $M_c$, a *selection* must somehow be made from the total set of $M_p \times L_p$ possible positions. The matrix $\mathbf{F}_c$ is obtained as a selection from the $M_p \times L_p$ matrix $\mathbf{F}_p$ containing the frequency response functions (the control paths) between all possible actuator positions and all possible control sensor positions. For a given frequency, $f_i$, the vector $\mathbf{b}_c$ is obtained by selecting $M_c$ rows from one column of the $M_p \times N$ matrix $\mathbf{B}_p$ (similar to $\mathbf{B}_e$), that contains the uncontrolled sound field measured in all possible control sensor positions. Once this selection is made, the vector of optimal controller weights for this particular configuration can be calculated using the cost function for the controller, $J_c$, obtained from

$$J_c = \mathbf{e}_c^H \mathbf{e}_c \tag{4}$$

where $\mathbf{e}_c$ is the $M_c \times 1$ vector of controller errors, as measured by the control sensors. The controller error is in turn calculated with an expression similar to equation (3), as

$$\mathbf{e}_c = \mathbf{F}_c\mathbf{c} + \mathbf{b}_c \tag{5}$$

where $\mathbf{b}_c$ is a vector similar to $\mathbf{b}_e$ that describes the uncontrolled sound field as measured by the control sensors. The $x$ has been omitted in equation (5) for the same reason as above in (3b). The LMS solution for $\mathbf{c}$ is found by setting the derivative of $J_c$ with respect to $\mathbf{c}^*$ (* denotes conjugate) to zero and noting that the derivative of $\mathbf{c}$ is zero [5], i.e.

$$\frac{\partial}{\partial \mathbf{c}^*}\mathbf{c} = \mathbf{0} \tag{6}$$

The derivative of equation (4) is set to zero to find the optimum solution, which results in

$$\mathbf{F}^H\mathbf{F}\mathbf{c}^\infty + \mathbf{F}^H\mathbf{b}_c = \mathbf{0} \tag{7}$$

The LMS solution is now obtained as

$$\mathbf{c}^\infty = -\left(\mathbf{F}^H\mathbf{F}\right)^{-1}\mathbf{F}^H\mathbf{b}_c = -\mathbf{F}^+\mathbf{b}_c \tag{8}$$

where $\mathbf{F}^+$ is the pseudo inverse of $\mathbf{F}$. This means that the LMS solution for $\mathbf{c}$ *only* depends on the frequency response functions from the actuators to the control sensors and on the primary sound field as measured by the control sensors. The overall attenuation for this particular configuration is finally obtained as

$$D_{LMS} = 10\log\frac{\mathbf{b}_e^H \mathbf{b}_e}{\mathbf{e}_e^H \mathbf{e}_e} \tag{9}$$

Another selection of $L$ actuators and $M_c$ control sensors is now made and the calculations are performed over again to obtain the overall attenuation for the new configuration. If the set of possible positions for the actuators and control sensors is small, all possible combinations of $L$ actuators and $M_c$ control sensors can be tested and the true optimum is found. As the set of possible positions is increased, the number of possible combinations of actuators and sensors increases rapidly and it becomes practically impossible to calculate the attenuation for each selection.

### 4.2 - Search algorithm

A number of optimization algorithms have been presented over the last years, see e.g. [6-8] and the purpose of such an algorithm is to find a configuration of actuators and control sensors that is close to the true optimal configuration. Since all possible combinations of actuators and sensors cannot be tested, some form of randomness has to be built into the algorithm. At UK/R, such an algorithm was developed based on simulated annealing [9]. This algorithm has proven to be very fast and arrives generally at a solution that is as good or better than other algorithms.

### 4.3 - The number of actuators and sensors

By varying $L$ and $M_c$ and redoing the calculation scheme described above for each $L$ and $M_c$, the optimal attenuation can be plotted as a function of $L$ and $M_c$. It is generally found that, starting with small numbers for $L$ and $M_c$, as more actuators and sensors are used, the optimal attenuation is increased up to a certain level where the curve flattens out and eventually decreases. Thus the cost per dB attenuation increase is low at the beginning up to a level where each extra dB of attenuation becomes very expensive. With this data as a background, a decision can be made on what is the most suitable and cost effective size for the control system.

### 4.4 - Varying the rpm

If the primary noise source has a rpm that varies within a given range (such as in a car), the procedures describe under $4.1 - 4.3$ will have to be run through for a number of frequencies within the rpm range. The necessary frequency resolution depends on the properties of the acoustic field. Lightly damped modes might require higher frequency resolution while highly damped sound fields require less resolution.

### 4.5 - Harmonics

Usually, it is desired to attenuate not only the fundamental frequency, but also a number of harmonics. Preferably, all of the procedures described above should be performed for each harmonic to be attenuated. It is not unusual that the optimal configuration for attenuation of the harmonics is quite different from the best configuration for the fundamental [10].

### 4.6 - Multiple references

If more than one reference source is to be controlled, all of the above mentioned procedures should be executed for each reference source [11,12].

### 4.7 - Actuator constraints

All actuators have an output limit that must be considered and this limit may well be frequency dependent... For loudspeakers, there is a limitation in volume velocity and force actuators have limited force output and stroke. These limitations can be built into the optimization algorithm, so those configurations that require more output than can be delivered are automatically discarded.

### 4.8 - Other requirements

There are other requirements that might have to be considered in a production design. Typical issues are the overall size and weight of the active control system and the size and weight of actuators. Heat dissipation might be a problem in some applications.

## 5 - PERFORMANCE PREDICTION CHECKLIST

Finally, we would like to end with a small checklist that has proven to be very useful. The following is to be considered (according to our experience) when planning to predict the performance of an active noise- or vibration control installation:

- Determine the positions for the control system evaluation;
- Determine practically useful positions for the actuators;
- Determine practically useful positions for the control sensors;
- Measure the frequency response functions with the proper frequency resolution between all actuators and all evaluation sensors;
- Measure the frequency response functions with the proper frequency resolution between all actuators and all control sensors;
- Record the sound field in all practically useful evaluation positions and all practically useful control positions. This has to be done for all working conditions (speed, load, etc.);
- Calculate the matrices $\mathbf{B}_e$ and $\mathbf{B}_c$ (similar to $\mathbf{B}_e$ but for the control sensors) from the recorded data;
- Determine ranges for the values of $L$ and $M_c$. Determine the number of harmonics to be included;
- Find the optimal attenuation for the different values of $L$ and $M_c$ according to the given evaluation criterion;
- Recalculate the attenuation for different working conditions.

## 6 - CONCLUSION

In MIMO active noise- and vibration control applications, finding the optimum configuration of actuators and controls sensors can turn out to be the most important step in the process. In this paper we have discussed some procedures that might be of interest for anyone planning to experiment with or install such a system.

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