Active Control of Sound and Vibrations 12

Joachim Scheuren

12.1 Introduction

"Noise Reduction by Anti-Sound." For more than four decades, this picturesque and catchy motto claims more than the physical existence, it claims the promising and useful application of some sound field whose characteristics allow at least partially the destruction of another annoying – and thus noisy – sound field. This claim and the envisaged methodical expansion of acoustical engineering were and are highly welcome. This is because the control of the increasing noise impact on our life strongly depends on any new possibility which may contribute to reduce the noise in our environment.

Of course it was not possible to keep and realize all what had been announced and promised – sometimes in a rather careless way – by the apologists of active control. Nevertheless, previous successes as well as the justified hopes were able to uphold continuous interest in applying and further developing the method.

The above-mentioned possibility of controlling wave fields is based on the simple physical principle of interference which describes how parts of the two wave fields may cancel if they are out of phase. It was common practice to characterize such intentional, electronically controlled interference by the attribute "active."

Active control measures thus are targeted to modify given mechanical field quantities (the so-called primary field) by the superposition of additionally generated, coherent secondary fields which are generated by electrically driven actuators, the so called anti- or secondary sources.

This approach is not at all restricted to airborne sound fields. It is equally valid for arbitrary media, i.e., for fluids and gases as well as for structure-borne and vibrational fields in elastic solids. Therefore, besides direct compensation of air-borne sound, active measures may also suppress the radiation of air-borne sound by controlling structural vibrations.

The successful compensation of fields opens further, completely new possibilities. If we succeed in negatively reproducing (and thus cancelling) given sound and vibration fields, then we should equally succeed in generating and thus realizing new arbitrary field distributions. Then, instead of approximating a vanishing (zero-) field, the more general task is to approximate an arbitrary nonzero field distribution. Active sound reduction thus turns to active sound generation and active noise control to active sound design (ASD).

The high fascination of active control of sound and vibration fields is not only based on the conceptual elegance of the approach which, in suitable applications like sound fields in ducts or small volumes, was able to lead to attractive results. It equally impresses by the novel combination of classical acoustics with the innovative and rapidly developing domains of digital electronics and integrated electromechanical transducers.

However, any successful application of the technology in practice depends on its robustness and

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economic efficiency rather than on its elegance. And this, in many cases, still interferes with the required electronic and electromechanic efforts. Together with an extremely confusing patent situation which, in the past, was caused by systematic protection of all imaginable solutions this has lead to limited applications in a few special cases only. Also, only a small number of research results appeared to have a realistic chance of practical implementation. It can be assumed, however, that technological progress together with expiring patents will result in a slow but continuous growth of active methods in practice.

In the following, an overview of the present state of the art will be given for active sound and vibration control technology. Besides the most important and promising applications, the basic mechanisms as well as some advice on how to design and apply active methods shall be given. Being subject to rapid changes in development, technical details will be outlined only briefly. For more detailed treatments and discussions, the extensive literature, in particular introductory books [1–6] and surveys [7–16] as well as the proceedings of regular conferences [17–24] and the compilation of literature and patent surveys in [25, 26] should be consulted.

12.2 Some Historical Comments on Technical Development

The first written formulation of the idea of active noise abatement as "noise cancelation by (electromechanically) controlled interference" [27] can be found in the patents of P. Lueg from 1933 to 1934 [28–30]. Among others, they contain those two approaches which – some decades later – have been in the center of first practical efforts: the control of arbitrary sound fields in the vicinity of an appropriately placed secondary source and the restriction to one-dimensional wave propagation.

First results for such a control were published by Olson in 1953. He investigated arrangements which in principle can be reduced to the basic control loop given in Fig. 12.1 [31, 32].

Driving the loudspeaker with an inversely amplified microphone signal is aimed to compensate any sound pressure fluctuations at the location of the microphone. Due to differing propagation directions in the primary and secondary sound fields, the region

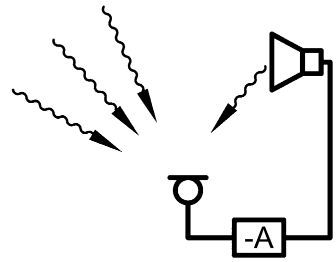


Fig. 12.1 Sound field control at the location of the microphone

of reduced sound levels is limited to some neighborhood around the microphones.

As their extension is proportional to the wavelength of the sound field, the spatial effectiveness of the anti-source is increased at low frequencies. This is an important feature of all active measures which is further emphasized by the fact that any signal processing requirements are met easier for lower sampling rates. Also, the difficulties with passive methods at low frequencies enhance the attractivity of active approaches.

As a representative result, Olson was able to reduce the sound level by some 10–20 dB within 1–3 octaves at distances up to half a meter and at mid-frequencies of some 60 Hz.

The limitation in sound level reduction is caused by instabilities occurring at high amplifications. Nevertheless, in later years this simple approach was maintained with some success for various applications. Leventhall et al. [33] have achieved sound insulations of some 20 dB by controlling the sound pressure in one-dimensional waveguides.

Although Olson was able to point out many potential applications of the method, a long way was to be gone to practical applications. This was equally caused by restrictions in fast signal processing as well as by a lack of thorough understanding of the physical mechanisms involved. The resulting underestimation of related difficulties, which can be illustrated by the confidence in simple successes for electrical transformers [34], has finally led to a temporary decline of activities.

It took another 10–15 years to stimulate new interest for the possibilities of active sound control. The temporary restriction to simple or one-dimensional fields together with the highly improved possibilities of electronic signal processing was able to promise encouraging successes.

Thus, the basic investigations of Swinbanks [35] or Jessel and his group [36] were soon followed by respectable experimental results. Typical achievements were sound level reductions of up to 50 dB for single frequency sounds [37, 38] and 10–15 dB for broadband sounds [38, 39].

A comparable but independent development could be observed in the domain of technical vibration control. Early attempts to reduce the excitation of structures by mechanically driven out-of-phase sources can be found in [7]. Since Olson had pointed to vibrational applications of his concept, Rockwell and Lawther succeeded in actively damping bending wave modes of beams by up to 30 dB [40]. Comparable, mainly theoretical investigations were carried out by Tartakowskij and his co-workers [41].

Again, it took another 10 years before the prospect of any realizability of practical measures caused growing interest. As can be seen from the literature given and the references quoted there, this interest was equally concentrated on mechanical engineering [42], space technology [43], and civil engineering [44].

Despite the close relationship between the tasks of active vibration control and active sound control, the development of both domains has hardly referred to each other. It thus took long from 1967, where the aim to reduce sounds by active control of radiating structures had been formulated [45], to concrete efforts in applying this principle [18, 46].

It then was in the eighties of the last century that the interest in active sound and vibration control and its possibilities significantly grew. Consequently, many large-scale but basic research and application-oriented development projects systematically investigated the principal realizability of the approach to nearly all, even remote fields of technical applications. However, the number of proven practical feasibilities did not correspond to the comparably low number of practical implementations.

This was mainly due to doubts in the long-term robustness of the components involved together with considerable costs to provide appropriate electronic and electro-mechanic equipment. However, any related disillusion should not obscure the fact that in fitting applications active control of mechanical and acoustical wave fields may have essential advantages and therefore can be classified as a most useful technological approach. This shall be demonstrated in the remainder of this chapter.

12.3 Structure of the General Problem

The problem of active noise and vibration control may be generally described by the structural diagram of Fig. 12.2. The source Q may represent one or even more simultaneous sound sources, thus generating the primary field quantities put together in the vector y_P. The basic mechanism of active control can be found in the fact that in some interactions points these primary field quantities y_P are superimposed by secondary field quantities y_S.

While the resulting field quantities y propagate within the mechanical system under consideration, they are subjected to multiple changes and modifications. These can be represented by the matrix of block C with output quantities z which represent the target of any active control measure. These target quantities z shall approximate and follow some given, desired quantities z_S as accurately as possible.

Depending on the application and related effort, the specification of z and z_S may realize such different design objectives as:

- Cancelation of a field or intensity component ($z = 0$) in one or several points
- Optimal approximation of one or several noise spectra at defined frequencies in given points or domains
- Minimization of the mean sound or vibration energy in a given domain
- Minimization of radiated sound power

Also, other characteristics of the sounds to be considered like:

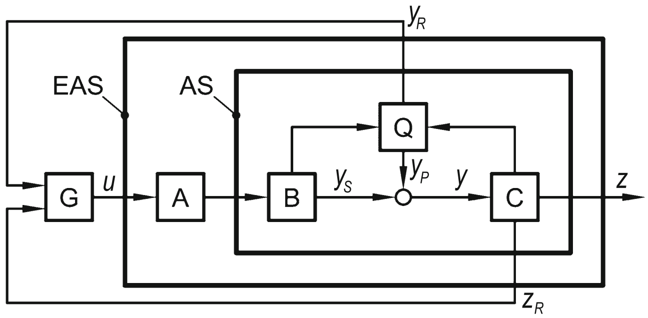
- Adjusting tonal spectral components to given amplitude values.
- Realizing given values of some psychoacoustic noise parameters (e.g., loudness or roughness).

can be obtained by appropriate selection of the target quantities z_S and the desired quantities z.

From the variety of possible definitions of target quantities it can be seen that they are not always easily accessible for direct measurements. Therefore it can be necessary to estimate the effective target quantities from measurements of other additional field quantities which are not elements of z. The complete set of measured quantities as evaluated by the signal processing unit G are put together in the feedback signal vector z_R.

The third arrow starting from C describes possible feedback effects of the generated field to the source

Fig. 12.2 Structural diagram of the general problem



mechanisms in Q. This allows the description of such sound and vibration generating mechanisms which are caused by unstable feedback effects between different mechanisms involved [11, 47].

In particular, the closed feedback loop from Q via C back to Q describes the structure and the mechanism of mechanical self-excitation which is the origin of many vibration and wave fields. By intervening in this mechanism, e.g., by adding an additional feedback loop to compensate the original one, the generation of such instabilities can be prevented (see Sect. 12.4.6).

The remaining blocks represented by A are all actuators that are needed to excite the secondary field quantities y_S and by B all mechanical transfer paths to the interaction points while the signal processing unit G provides all signals u to drive the electromechanical transducers in A.

The input signals of G which provide all information to evaluate optimal control signals u may be distinguished with respect to their content of information. In analogy to Fig. 12.2, y_R contains all signals with some (a priori) information on the sources and on the primary field quantities in y_P in the interaction points. For this reason they are often referred to as reference signals. On the other hand, z_R represents such signals which allow for estimates of the target quantities.

Apart from the paths described so far, the path from B to Q in Fig. 12.2 also takes into account the feedback effect of the secondary field to the sources. This will allow besides possible changes for any distortion of the primary field, information given in y_R caused by any secondary field action via Q. Without appropriate countermeasures, this undesired feedback may cause erroneous secondary fields and – even worse – severe instabilities as illustrated in the context of Fig. 12.1.

Typical measures against such instabilities can be selective measurements to suppress any influence of the secondary field like, e.g., directive measurements or measurements of structural quantities representing the primary field without being influenced by the secondary field. Also, instabilities may be avoided by feedback compensation as described in the next section.

The general model of Fig. 12.2 may represent and describe very different arrangements. This can be demonstrated by an example. If Q represents a combustion engine, z could equally represent the sound power radiated from some exhaust system or radiated directly from the engine. Also, the structure-borne sound power introduced into some foundation may be a useful definition of the target z. To drive the secondary transducers, y_R may contain an rpm signal as well as the sound pressure in the exhaust pipe just before (upstream of) the secondary source. Depending on the target, z_R may represent quantities like sound pressure in an exhaust pipe after (downstream of) the secondary sources, sound pressure in preselected points of the car interior or exterior and field quantities in a given base structure.

Apart from any devices to measure y_R, z and z_R, the blocks B, C and Q solely describe acoustical (or mechanical) systems which may be combined into one single acoustical system AS or into a mechanical system MS. By integrating the electromechanical transducer contained in A, systems may be extended to an electroacoustical system EAS or an electromechanical system EMS.

This formalism clearly demonstrates how the classical task of any noise and vibration reduction, or of selected control of an acoustical target quantity in AS or MS by appropriate (passive) acoustical/mechanical measures, may be transferred into the task of active control of an electro-acoustical/mechanical system EAS or EMS, just by applying electro-acoustical/mechanical transducers.

The structure of Fig. 12.2 and the functionalities and media described by the systems A, B, C and Q identify the problem at hand as a general control problem with distributed parameters [4, 48–52]. This structure includes some simple standard structures of common problems in signal processing and control theory as special cases.

In the terminology of the latter y_P represents the disturbances that must be compensated as well as possible by the anti-sources contained in A such that the target quantities contained in z are minimized. If y_R contains sufficient information on the temporal behavior of y_P, the required control signal u can be derived solely from y_R. This finally leads to a standard feed-forward control problem.

In contrast to this, the exclusive evaluation of the target quantity $z = y$ to be controlled leads to a standard feed-back problem with disturbing noise where the target signal $y = 0$ may be prescribed as desired signal.

In any application, the system design should always be aimed to compensate the measurable (by y_R) and thus predictable influence of disturbances by pure feed-forward controls. Only then does it make sense to act against the remaining components of the control signal y by a feed-back controller [53].

The definition of the controller's structure and its appropriate parameters should be based on the many methods of analog and digital control theory with one or more control signals as well as on design rules for signal processing systems as described in [4] or in the relevant system and control theory and signal processing literature.

12.4 Principal Considerations on Working Mechanisms of Active Systems

Any arbitrary change of given wave fields requires their previous compensation. This is because any change or replacement of field distributions cannot be effective unless the original field can be compensated or reduced at least. For this reason, apart from direct control of the sound-generating mechanism, all active control approaches are based on the negative (anti-phase) reproduction of mechanical sound and vibration fields. Here, field reproduction generally means the approximation of a given primary field quantity $f_p(x,t)$ by a secondary field quantity $f_s(x,t)$ for an infinite number of points t in time and $x = [x_1,x_2,x_3]^T$ in space.

However, as the generation of $f_s(x,t)$ is limited by a finite number of control elements and controllable degrees of freedom, the demand for equality of f_p and f_s cannot be fulfilled for all values of x and t but only for a finite subset.

Any field distribution may be described by rather different parameters. Therefore, the definition of this subset is not necessarily restricted to the space/time domain, but can be based on any parametric description like modal decomposition (modal transformation) or superposition of plane waves (Fourier transformation). The reproduction of waves allows the control of their propagation while the reproduction of modal amplitudes directly aims at the active reduction of the related vibration amplitudes.

To get the relevant field information from as few parameters as possible directly corresponds to the important requirement to enable the related field control by as few actuators as possible. Thus, the decision whether to use a modal or a wave-based approach may be directly deduced from the concrete type of problem. As long as only a few modes of a structure contribute to the disturbing noise component, the modal approach limiting all active measures to the relevant modes may be recommended. By contrast, for one dimensional wave fields e.g., it is advisable to characterize the field by forward and backward propagating waves.

As both approaches describe or – for a finite number of parameters – approximate the same field, they mutually imply each other. Therefore, any immediate control of propagating waves may suppress the modes and thus the resonances of the structure, if propagation loops closed by mechanical feedback are cut off by absorption of incoming waves [54]. For this reason, the concept of propagating waves has been adopted by vibration theory which traditionally is dominated by modal concepts. Here, first applications were focused around large space structures [43, 55].

Apart from concentrating on modes and plane waves, other concepts have been used to theoretically describe the possibilities of active field control. Among these, the direct control of power flow [56] and the control of actively realized impedances [57] shall be mentioned here. Impedance control takes advantage from the fact that its effect only depends on the local impedance of the medium under consideration.

Therefore, any modeling of wave propagation becomes unnecessary. But this advantage may be compensated by the difficulty of finding local criteria which sufficiently describe the global field behavior. An example of how high-damping values may be based on the minimization of wall impedances by using a skillfully chosen arrangement can be found in [58].

A more direct link to the global behavior is found for the targeted control of power flows. Because all related power flow strongly depends on any secondary measures, respectively, it seems to be advantageous to control power flows indirectly only by controlling other target quantities like the amplitudes of propagating waves [59]. In the following, with respect to the most important concepts of field modeling and superposition, a systematic guidance to active system design will be given.

12.4.1 Concepts for Active System Design

As active sound and vibration control is based on the superposition of appropriate secondary fields, the essential task in designing active systems is to define favorable, i.e., most effective secondary sources together with their interaction points. This is because the kind, the number, and the location of secondary sources crucially determine the controllability of the system under consideration (EAS or EMS), i.e., the quality of approximation of the target quantities z to their given desired values.

It is good practice therefore to base the controllability of target quantities on physical considerations, e.g., by systematical analysis of the primary field on its propagation path from the source to the target points, and by investigating the effects of additional secondary sources.

This approach fully corresponds to the proven concept of passive noise control which is given by investigating the propagation path of the sound to be controlled from the source to the receiving point in order to find the optimal location for effective noise control measures. Thus, the following sections of this chapter will exemplarily explore, i.e., by how many sources at which locations and to what effect, sound and vibrations may be controlled

- at source, where the field is introduced
- on the propagation path of the field or, finally
- at a given target volume.

Also, defining the measurement chain that acquires the target quantities to be controlled as well as the available input signals z_R and y_R may crucially influence the success of an active measure. This is because this success is directly related to the quality of estimating the desired behavior or, in the terminology of control theory, to the observability of the system (EAS or EMS) under consideration. With respect to the delay time required by the signal processing unit G, it is important to register appropriate and, above all, early signals y_R and z_R with sufficient lead time.

From these principles, a design strategy for active systems may be derived:

1. Target formulation for an active measure in terms of a finite number of parameters. As an example, these parameters may be given by the complex amplitudes of certain waves or modes or just by the complex amplitudes of the sound pressure at given points.
2. Specification of such a source arrangement (A) that is best capable of achieving the target defined under 1 with acceptable effort.
3. Specification of a measurement arrangement (within C) that again is most suited for achieving the target defined under 1 with acceptable effort.
4. Specification of the structure and the parameters for the signal-processing unit G to optimally implement the control possibilities given in 1.3.

The proven principle to try to fight any sound and vibration as close as possible by the source stays most valuable for active systems too. Besides preventing the introduction of energy by reflecting or absorbing it at the excitation point, in some cases the generation of the field itself can be prevented efficiently.

In particular, if the acoustical or vibrational field is generated by self-excited limit cycles caused by unstable interaction, simple measures may force stability and thus suppress the generating mechanism.

In the following, the most important physical concepts of active sound control are shortly illustrated. Because of some limitations resulting from idealized assumptions, these concepts do not always refer to arrangements realizable in practice. However, they do offer some physical insight which helps to estimate necessary source efforts or limitations in frequency. Also, the most important question as to the occurring flows of energies and/or power can be answered in this context.

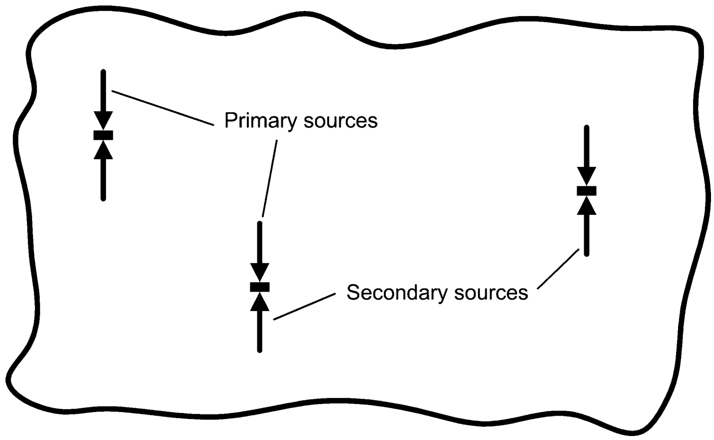


Fig. 12.3 Principle sketch of source reproduction

As said before, active approximation of arbitrary fields requires first, by principle, the compensation of the existing primary field. Therefore, the following considerations focus on such field compensation. This can further be justified by stating that any approximation of arbitrary fields formally can be treated like the approximation of a vanishing zero field.

12.4.2 Source Reproduction

The control of multidimensional wave propagation as well as the compensation of vibrational fields with high modal density often require a high number of sensors and actuators hardly to be realized in practice (see Sects. 12.4.3 and 12.4.4). Therefore, it often is preferrable to try to counteract the field by a copy of the original sources.

The sketch given in Fig. 12.3 illustrates this symbolically by using primary (acting downwards, ↓) and secondary (out of phase by 180° and therefore acting upwards, ↑) source arrows which e.g., may represent loudspeakers in a room or forces acting on a plate.

The smaller the distance between two identical sources in antiphase is, the more the source-generated fields are cancelled; the sources mutually hinder themselves in radiating sound. Besides simple, easily reproducible sources with low directivity, effective field reductions require therefore distances between primary and secondary source of less than half a wavelength [1, 60]. And this applies not only to free fields but also equally to closed spaces with high modal densities [1].

To the extent of successful source reproduction (e.g., in a certain frequency interval), the related field reproduction can be global, i.e., successful within the

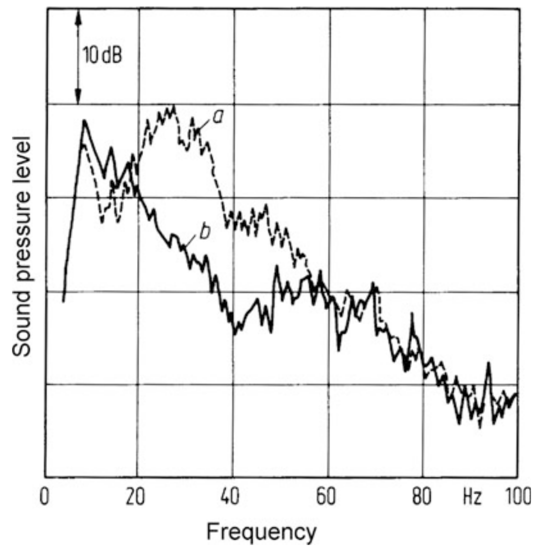


Fig. 12.4 Sound pressure level in the far field of a gas turbine exhaust (**a**) without and (**b**) with active compensation sources at the outlet [7]

complete volume under consideration. As can be seen from Fig. 12.3, such global compensation can be achieved with locally arranged sensors because the reproduction of sources in amplitude and phase can be controlled by local measurements.

As in the first reliable out of lab realization at all, such suppression of sound radiation has been successfully applied in practice to the exhaust outlet of a gas turbine [61, 62]. By applying 72 bass loudspeakers on a circle around the outlet it was possible to reduce in the far field the sound pressure level between 20 and 50 Hz by more than 10 dB (see Fig. 12.4).

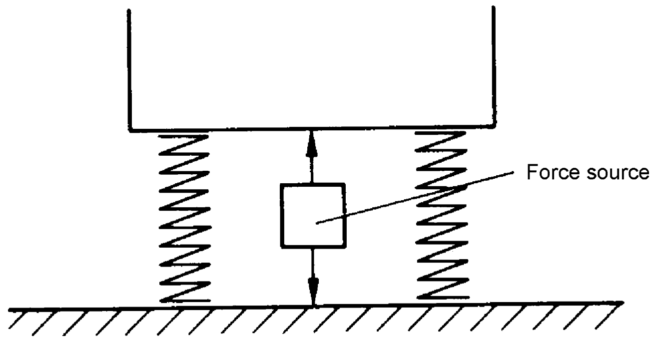


Fig. 12.5 Compensation of the force applied by a spring by an additional force source applied in parallel

Equally this approach can be applied to structure-borne sound problems if the original primary force is compensated by another secondary force acting close by. This is especially effective if the primary force can be considered as a point force.

For low frequencies, the latter requirement normally is fulfilled at the coupling points of resilient mounts. By appropriate control of a second force which is applied in parallel to the springs (see Fig. 12.5) the force transmitted through the springs can be compensated. The resulting force introduced into the foundation thus vanishes, the aggregate is effectively decoupled and vibrates freely.

If compared to alternative arrangements [148], the advantage of the arrangement given in Fig. 12.5 is that only the dynamical part of the forces has to be generated and applied because the static part is taken over by the passive spring. Also, the good vibration attenuation of the spring at high frequencies is preserved and therefore the remaining task of the secondary force is to improve the vibration attenuation at low frequencies including the domain of mass/spring resonance. By appropriate control of the force actuator, this can be achieved by shifting the resulting resonance frequency to lower frequencies and thus reducing the resonance peak of the transmission curve [63].

For harmonic excitation, the predictability of the signal wave form allows attenuation values of vibration which in many cases of passive mounts cannot be achieved. An impressive example will be given in Sect. 12.6.

Thus, active vibration isolation systems, often called active mounts, offer the well-founded perspective of becoming an indispensable engineering tool for vibration control in special cases of high demand. Possible applications go from mounts for complete vibration suppression of highly sensitive processes [63, 64], where active mounts are frequently used today already, to improved mounts for engines and other vibration sources. In addition the operating staff of vibration intensive machinery thus can be protected. Further hints to two active engine mount systems realized in standard cars are found in [65].

General information on the possibilities and on the design of active mounts are given in [2, 3, 6] and in a recent VDI guideline [66].

As concrete example, we refer to the application of secondary forces to the car body of an ICE passenger train car in close proximity to the bogie's secondary spring. By this active measure, the low frequency noise components excited by wheel harmonics under special track conditions could be essentially reduced in the passenger compartment around 90 Hz. Comparing the spatial sound pressure distributions of Fig. 12.6 (measured at the rolling test site of Deutsche Bahn AG at a speed of 200 km/h) shows that the related reductions were up to 20 dB for some seats. The average reduction for a group of six seats was up to 12 dB [67, 68].

The use of sliced piezo elements placed in the interior volume of the secondary springs resulted from a systematic feasibility study investigating various active concepts for the bogie and the passenger compartment with respect to their applicability and effectiveness. It turned out that immediate control of the sound field in the passenger compartment by loudspeakers would also be possible. However, in summary, the global compensation by compensating forces applied to the excitation points was shown to be advantageous.

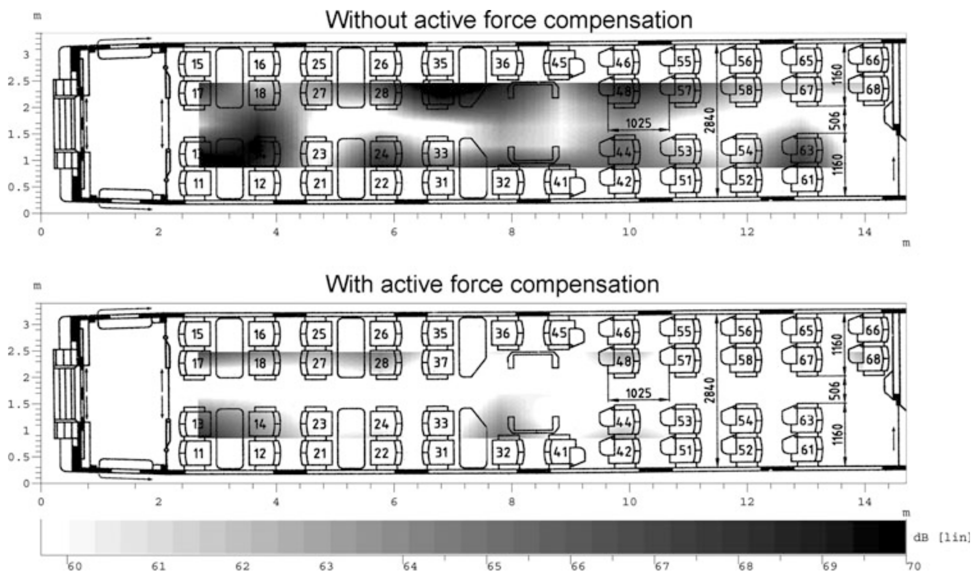


Fig. 12.6 Spatial sound pressure distribution in the passenger compartment of an ICE passenger train car without (*upper*) and with (*lower*) active force compensation at the bogie's secondary springs

12.4.3 Active Control of Wave Propagation

Apart from stability problems, the propagation delay times within the secondary sound field and the electronic controller may limit the performance of the simple standard feedback controller mentioned in Sect. 12.2. It therefore appears useful to take advantage of the finite velocity of wave propagation or the related delay time of the primary wave field to be controlled, respectively. This allows a temporal prediction if the field quantities can be measured before arrival at the point of superposition.

It is possible then to use the delay time of the propagating wave to determine the optimal waveform for an appropriately placed interfering field source. As can be seen from the principal sketch of Fig. 12.7, the propagation of the wave can be hindered essentially or even suppressed completely.

The possibility of determining field quantities from their values on a preceding incoming wavefront follows from Huygens' principle. This principle in the formulation of Kirchhoff has been the starting point for basic considerations first published by Malyuzhinets [69] and Jessel [36].

According to this principle, any field within a volume *V* generated by exterior sources outside *V* can be equally generated by substitute sources *Q** distributed on the closed surface *S* which limits the volume *V*. Figures. 12.8 and 12.9 show this for the case of *V* being free of any interior sources.

Apart from the change in sign, this is exactly what active control aims to do: to define source distributions -*Q** which completely counteracts and thus eliminates a given field within a volume *V* according to Fig. 12.8.

However, this requirement is not unique, i.e., it may be fulfilled by various field distributions. This allows additional requirements like defining that, according to Fig. 12.9, the sources *Q** should provide a field that reproduces the given field within *V* but vanishes outside *V*.

For airborne sound fields, such sources are given by monopoles and dipoles continuously distributed over *S*; they fully compensate the original sound field inside *V* without any feedback effect to the exterior volume, i.e., without changing the field outside *V* (see Fig. 12.10).

Any arrangement driven this way describes an active absorber with antisources -*Q** just receiving but leaving unchanged the complete incoming wave field. Also, the power transmitted from the primary field to the secondary sources is extracted by these secondary sources and, if necessary, reintroduced at some other location. Any original power absorption within *V* will now be taken over by the substitute sources -*Q**.

Other than in the arrangement of Fig. 12.10 which not only absorbs incoming power but also emits power

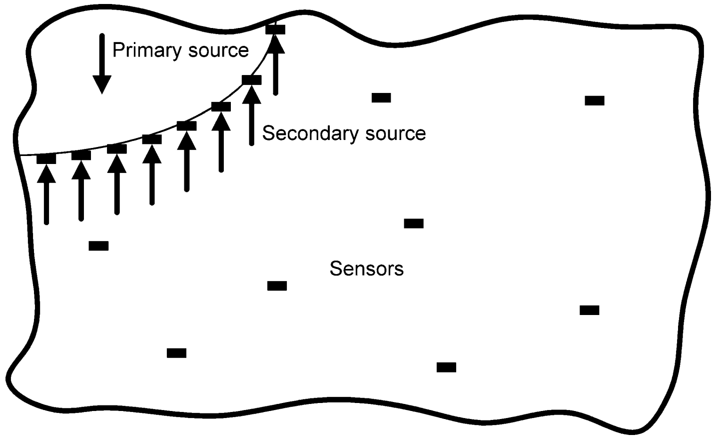


Fig. 12.7 Principal illustration of active control of wave propagation

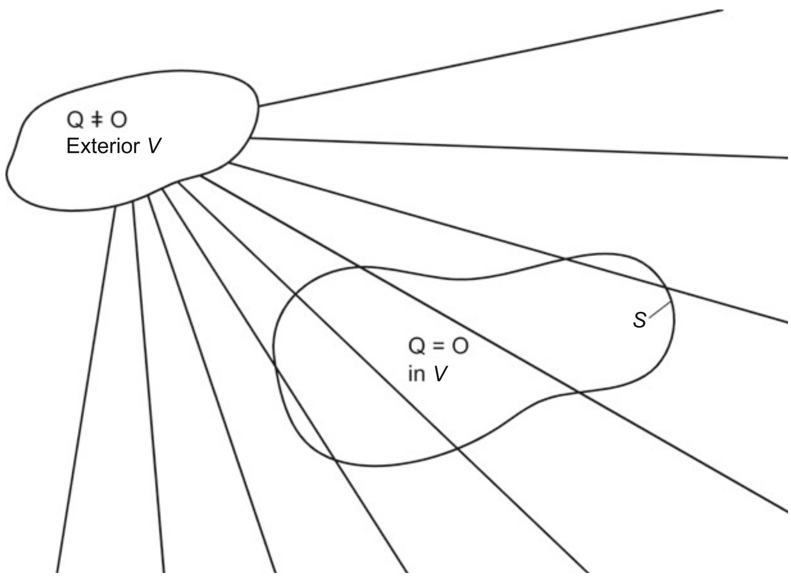


Fig. 12.8 Original field distribution with sources Q

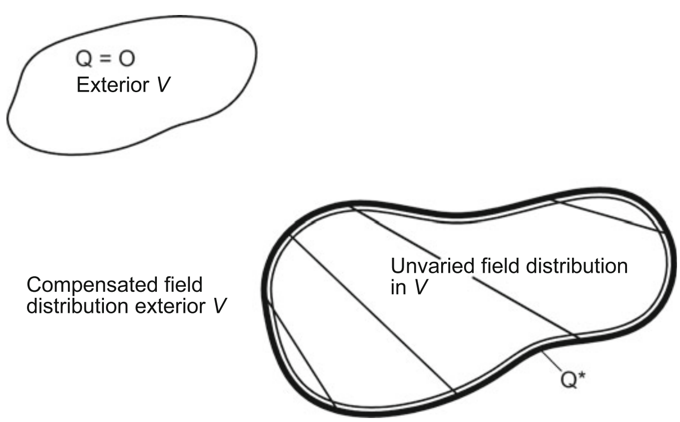


Fig. 12.9 Reproduction of field distribution by substitute sources Q^*

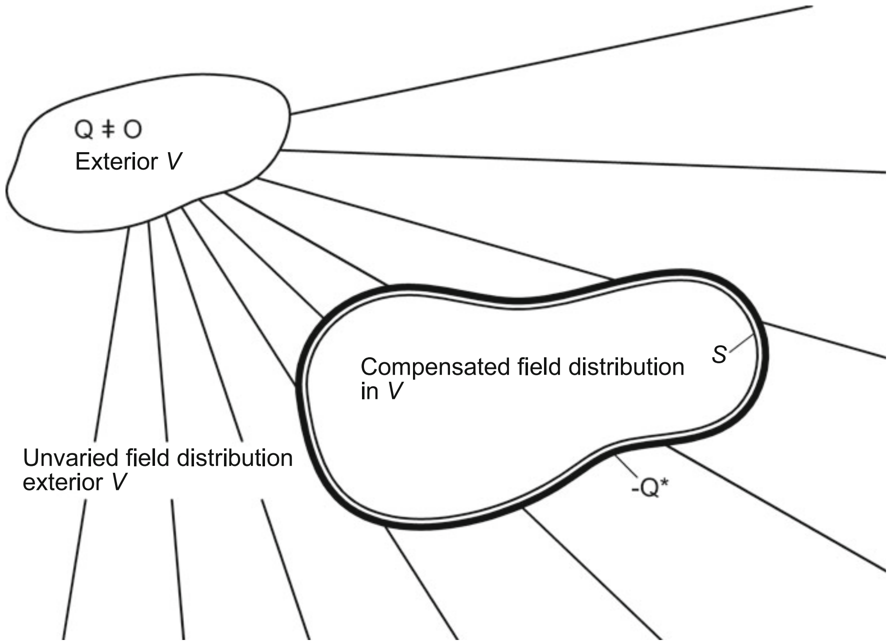


Fig. 12.10 Compensation of the field in *V* without feedback to the outer volume outside *V*

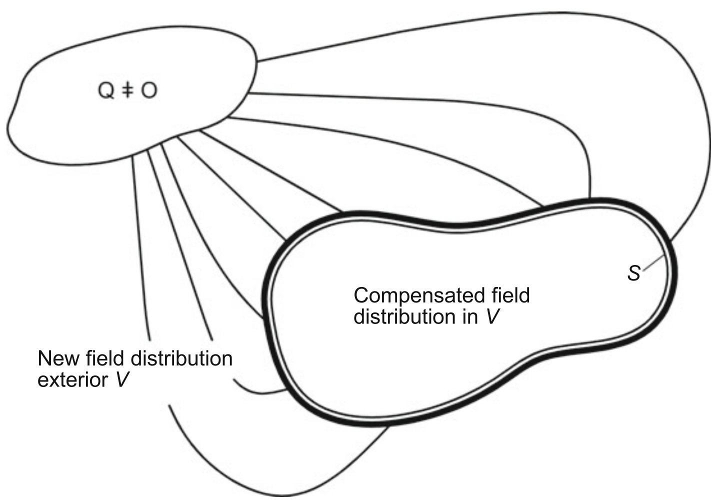


Fig. 12.11 Purely absorbing field compensation in *V*

in parts of the surface *S* in order to leave the outer field unchanged, another source distribution could be characterized by sources -*Q** on *S* which only absorb power. An exemplary field caused by such an active absorber is given in Fig. 12.11.

Here, the energy relations are a bit more complicated and strongly determined by the concrete case. The field generated by the antisources acts back on the primary sources and thus changes their radiative environment. Some simple and illustrative examples of such energetic interaction can be found in [8, 54, 70, 71]

The reproduction of the field within *V* can also be achieved with simpler source arrangements but then without additional restrictions on the field outside *V*. This compensation no longer is free of feedback, the field outside *V* changes and the arrangement thus incorporates an active reflector scattering back the incoming wave field (see Fig. 12.12).

These considerations show that secondary source distributions on the boundary surface *S* may be configured such that they can absorb (by extracting incoming power) or reflect (by redistributing incoming power) primary wave fields. Even if this reflection is

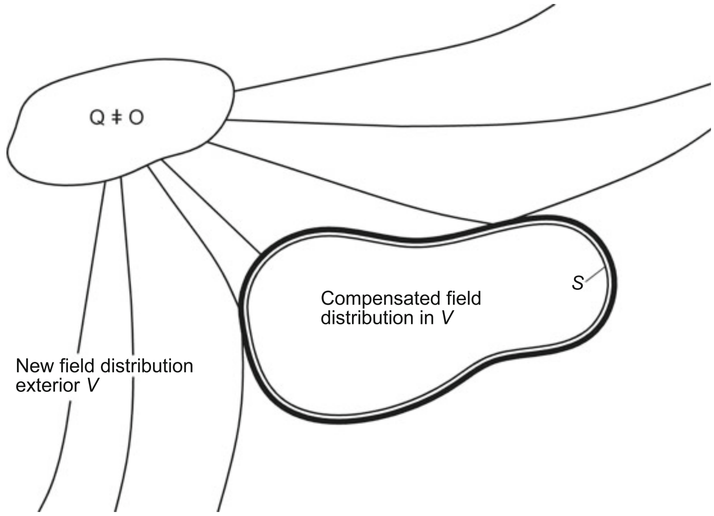


Fig. 12.12 Reflecting field compensation within *V*

complete in the sense of neither extracting nor introducing any mechanical power at the boundary *S*, the overall arrangement may change the energetic relations.

By acting back into the volume outside *V* where the primary sources are located, the radiation impedance at these sources is changed. As a consequence, this in general will change the power emission of these sources. For the analysis of power and energy relations in practical situations this effect tends to be more important than direct power extraction by secondary sources.

The formalism based upon Huygens's principle may be transferred to other media. Theoretically such analogies start from integral relations equivalent to the Kirchhoff formula which can be found for any boundary value problem defined by self-adjunct differential operators. For the general case of elastic solids with shear deformations such relations are found in [72]. For the case of flexural waves in thin plates which are important with reference to acoustic radiation, an analogous equation has been derived in [73].

Although such considerations essentially contributed to the theoretical analysis of active systems, they rarely served as an immediate starting point for practical realizations. This may mainly be due to the difficulty to predict the effects of discretization and of the spatial extension of the secondary sources.

Nevertheless, basic experiments in shielding the multidimensional propagation of sound fields with loudspeakers have been based on this idealized theoretical approach so far. As an example, laboratory arrangements using 16 loudspeakers and 24 measurement microphones were able to obtain in parts of the shielded domain level reductions of up to 30 dB between 200 and 500 Hz [74].

However, like with other experiments, e.g., acoustic shielding of transformers, it became clear that effective level reductions in considerable areas require high efforts. Therefore, although proven to be functional, any practical applications of active barriers are hardly to be expected for three-dimensional wave propagation.

There is, however, one important exception given by the approach described in [75] to decrease the diffraction angle and thus to increase the shielding effect of sound barriers by actively realizing an optimal impedance at the edge of the barrier. As the local criterion "impedance" sufficiently describes the global behavior of the barrier with respect to diffraction it is not necessary to model the spatial propagation of waves in three dimensional space.

Figure 12.13 shows a representative measurement point beyond the barrier where the sound pressure level may be reduced at frequencies between 200 and 600 Hz by some 4–8 dB if an acoustical soft edge of the barrier is realized by active means. This gives an additional improvement by some 2–5 dB in comparison to a barrier with a sound absorbing cylindrical edge.

The restrictions put forth against controlling multidimensional wave propagation do not apply to the one dimensional special case. In particular, if, below some

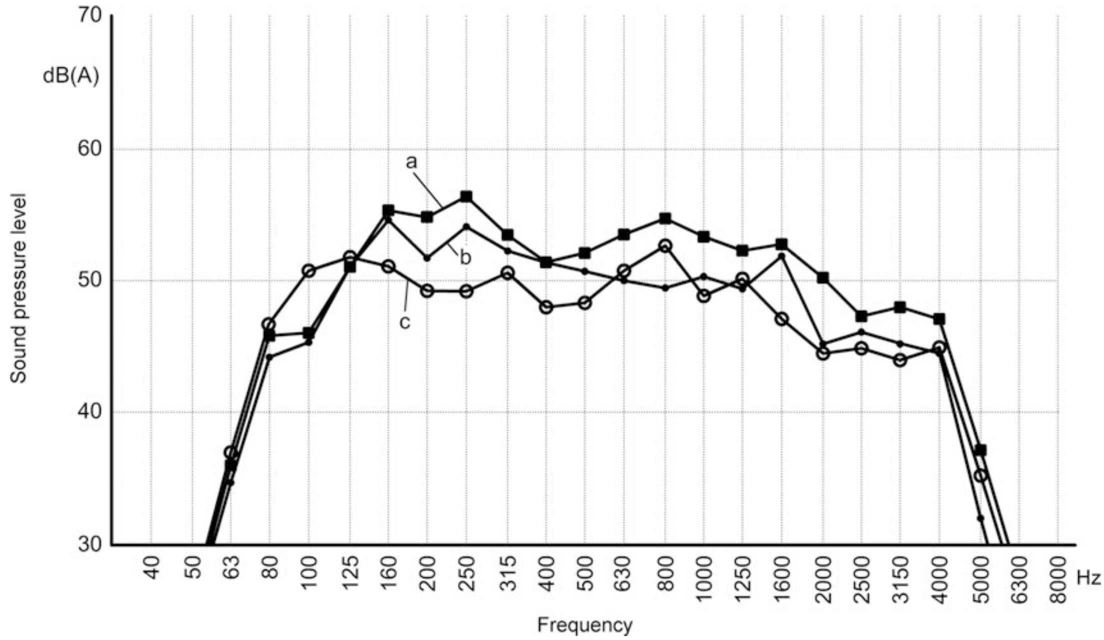


Fig. 12.13 Comparison of sound pressure levels behind a sound barrier (**a**) without and (**b**) with passive absorbing edge and (**c**) with active realisation of a soft edge [75]

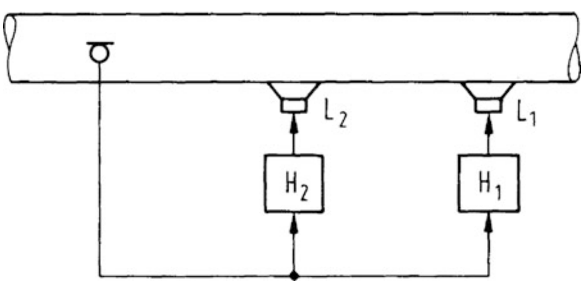


Fig. 12.14 Active sound field control in channels

cut off frequencies, only a limited number of wave types or modes is propagative, this propagation may be influenced effectively with lesser efforts.

The conditions in channels are most favorable if the related frequencies do not exceed the lowest cut off frequency. Then, only plane waves propagate and it doesn't matter where the generating volume flow is introduced to the channel. Instead of monopoles and dipoles distributed over the cross-section of the channel it is sufficient to apply independently controlled loudspeakers to the channel wall as shown in Fig. 12.14.

Approximating a dipole by two closely placed monopoles, results in a limited bandwidth. Swinbanks has shown how this bandwidth may be increased by additional loudspeakers.

Figure 12.14 includes the special case of one loudspeaker only ($H_2 = 0$) which initially was preferred for experimental implementations. The loudspeaker L_1 is driven such that the incoming wave field is reflected and the channel is kept free behind L_1. In analogy to the considerations at the beginning of this section, this arrangement implementing an active reflector may be extended to an active absorber by using and driving L_2 such that the wave reflected by L_1 will be absorbed by L_2 [35, 60].

After the realizability of these approaches had been proven experimentally [38] many attempts have been made to improve the results with respect to level reduction and to bandwidth and to consecutive transfers in practical applications. This may be demonstrated here by the work of La Fontaine/ Shepherd [76], who came up with broadband (30–650 Hz) level reductions of 20 dB. Roure [77] used an adaptive filter to obtain the results of Fig. 12.15 for simultaneous flow.

At low frequencies the limitation of active sound attenuation is caused by turbulent pressure fluctuations at the input microphone, while at high frequencies the effectiveness is reduced by the higher modes.

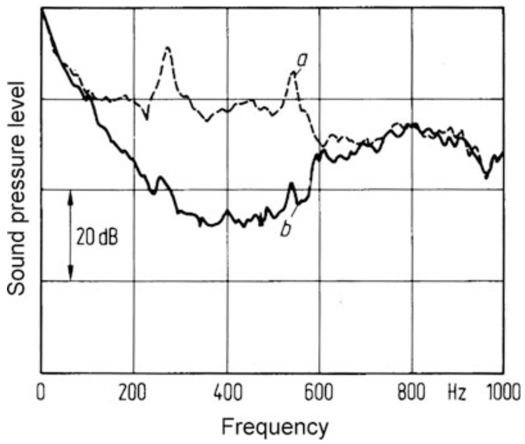


Fig. 12.15 Sound pressure level in a duct with flow (**a**) without and (**b**) with active sound attenuation [77]

Besides increasing the bandwidth, an increased number of loudspeakers may be used for compensating additional modes. In the past, this has not only been investigated theoretically but has also been practically realized in industrial plants[11].

Typical applications for active measures with one-dimensional wave propagation are air conditioning and exhaust systems. As an example, it is referred again to the gas turbine of Sect. 12.4.2 (Fig. 12.4). In this application, the suppression of external radiation means, with reference to the duct channel, a total reflection at the exhaust outlet. Further examples for active systems in ventilation systems are found in [78–80]. Also, direct suppression of ventilation noise by loudspeakers close to the source has been proven first in [81].

The control of wave propagation is not restricted to airborne sound waves. Figure 12.16 illustrates this for bending waves in beams by giving an example of where incoming waves may even be absorbed by appropriately driven secondary sources [60, 70]. The reduction of the reflection coefficient by 32 dB (in the mean) could be obtained by an electrodynamical shaker at the free end of a beam and by suitable reproduction of the passively reflected primary field. Thus, 99.95% of the incoming power could be absorbed.

This also illustrates the possibility (mentioned at the beginning of Sect. 12.4) of active structural damping resulting in multimodal resonance suppression. This approach has some advantages if compared to direct control of modal amplitudes, and may thus be helpful for some structural dynamic applications.

A practical application showing how suppression of mechanical vibration transmission through beamlike elements may be used to reduce the gearbox noise in helicopter cabins is described in [82].

Besides industrial exhaust systems the smaller exhaust and air-intake systems of motor vehicles may also be subject to active measures [83–88]. Like with all channels with flow, the additional pressure losses of passive measures thus might be reduced, the related power being saved or used otherwise.

As powerful engines generate high sound pressures in the exhaust system, any compensating actuators have to meet high requirements with respect to power and temperature. Therefore, besides loudspeakers, alternative actuators like oscillating flaps [87] directly modulating the exhaust flow have been taken into account. Because of their higher flexibility there is a tendency, however, to prefer loudspeakers skillfully coupled via airstream-cooled pipes.

Figure 12.17 shows that a series-proven active silencer (curve 2) has a better noise-reducing performance than a passive silencer (curve 1). Compared with the system without silencer (curve 5) level reductions of some 6–12 dB below 4,000 rpm and some 2–6 dB above 4,000 rpm are obtained [85].

The remaining curves 3 and 4 describe two actively realized sound variants clearly differing in character from each other as well as from the other sounds. Besides a slight reduction of the third engine order and some of its multiples, additional orders have been added and thus raised. While the amplification of the 2.5th and 3.5th order generated a low frequency oscillation resulting in an eight-cylinder-like sound (curve 3), the sportive character of the six-cylinder engine could be further emphasized by raising the 4.5th and 7.5th order (curve 4, [85]).

Besides – in a wide sense – free realizability of sounds the active system was able to save nearly 50% of volume and weight if compared to a standard muffler. However, although the system showed promising mechanical and acoustical robustness, the related cost prevented it from being introduced on a large scale until now.

Another application of wave propagation control could be the suppression of sound propagation in fluid-filled pipes [89], especially because the higher impedance of fluids enables a more effective coupling of actuators.

Finally, for completeness sake, a totally different application area should be mentioned. The acoustic

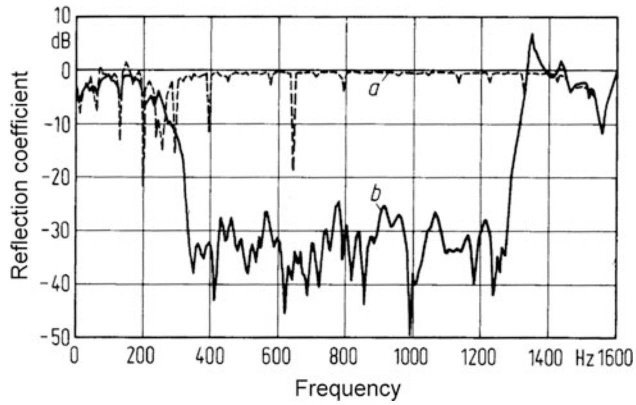


Fig. 12.16 Reflection coefficient (**a**) without and (**b**) with active absorber at the free end of a beam [60, 70]

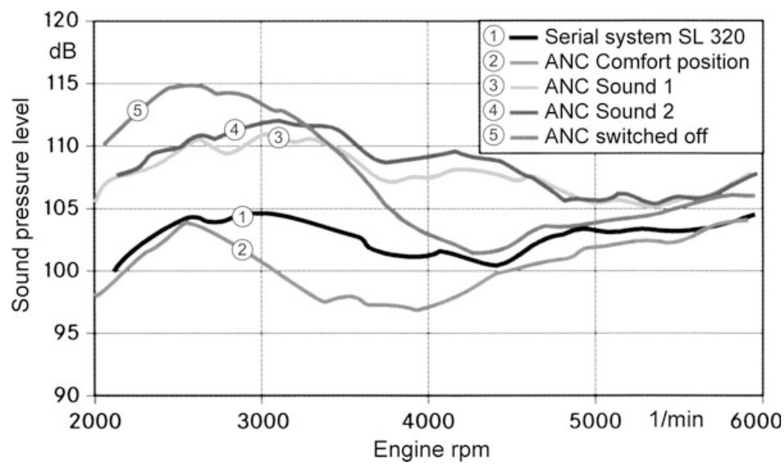


Fig. 12.17 Total exhaust noise level of a passenger car: Comparison of various sound variants of an active muffler with a passive standard muffler [85]

properties of rooms strongly depend on the reflective properties of the surrounding walls; considerations of influencing these by active elements are found in [90].

12.4.4 Active Control of Enclosed Domains

The concepts described so far dealt with directly controlling the excitation and the propagation of waves independent of the space defining the targeted area or volume. These approaches fail where the sources cannot be accessed, or where the sound field is characterized by many multidirectional waves. Such fields typically can be found in bounded areas with reflecting boundaries, thus being capable of resonances. In most such cases, it is easier to restrict active-control measures to a subspace or to base it on a modal decomposition.

Figure 12.18 illustrates this for the case that the effect of several sources distributed over a volume can neither be controlled at the excitation points nor be shielded actively. The only possibility remaining then is to control by applying secondary sources appropriately distributed over the accessible volume.

Another choice to be made then is whether field control is intended for the entire space (global field control) or for a selected subspace only (local field control). It is immaterial then whether this space is given as a closed volume (e.g., an aircraft cabin) or as structural element (e.g., beam or plate).

If the whole space is considered, modal description of the field has the advantage of providing independent parameters for field description and – control: the complex modal amplitudes.

Any targeted suppression and modification of single modal amplitudes assumes that they can be registered as well as excited. The simple example that a mode cannot be registered nor modified in its nodal points or lines illustrates that this requirement must be fulfilled by appropriate selection of all sensing and actuating points first.

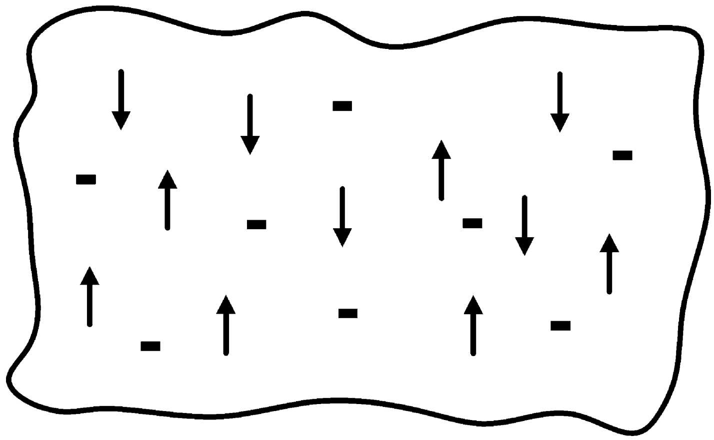


Fig. 12.18 Principal sketch of global active control of bounded areas

When fixing the number of measurement and actuator points, besides the number of modes to be controlled it must be taken into account that a single source even may excite all modes. Any desired suppression of modes therefore often brings about an undesired excitation of other modes.

This fact, often called "spillover" [6, 91], may require that the number of sources exceeds the number of controlled modes to be able to cope with the constraint not to excite some undesired modes. A comparable mix-up of modes can also occur while registering and analyzing modes. Therefore, similar precautions must be taken there.

If undesired modes are to be included into the considerations, the total number of modes in general exceeds the number of degrees of freedom given by the transducers. The thus over determined system of equations as well as the corresponding control law have a unique solution provided the mean square error of the modal amplitudes is minimized.

As a special case, this includes the possibility of controlling uncoupled modes vibrating with clearly separate frequencies by a single force only if this force is driven by the required line spectrum [60].

The modal approach fails as a conceptional basis for active measures if the number of relevant modes or the modal density do not allow a separate treatment. Then, like in the open space global field, compensation only can be obtained by reproducing or shielding the sources. Otherwise, the effect is limited to small zones around one or several measurement points.

This immediately leads to the pragmatic approach of directly minimizing the mean energy in all measurement points without any modal separation. For sound fields this means a reduction of the registrable mean square sound pressure while for structures a reduction of the radiated power can be obtained by minimizing the mean square velocity.

Potential applications of these approaches are widely spread concentrating, within the domain of vibration engineering, on large structures to ensure their dynamic properties in spite of extreme weight reductions. Besides basic investigations [6, 44, 52] how to stiffen selected structures like aircraft wings [92] by active means, much work was motivated by the perspective of being used in large space structures [6, 43, 55, 93, 94].

At least as long as all structural elements may be modeled as homogeneous continua the concept of wave propagation has some advantages over the modal approach like local limitation of registration and excitation or a smaller number of degrees of freedom [54, 95, 96].

For airborne sound fields in closed volumes it can be stated from the above considerations that their control is so much the easier the smaller the dimensions of considered volumes in comparison to the wavelength are. As higher modes for small dimensions only build up at higher frequencies, active measures can be limited to some fewer modes which depend on the highest frequency to be considered. If harmonic waveforms of the field quantities then allow a good prediction of their future behavior, all conditions for effective active field control are met [97].

An early, most useful application for active measures was given by active earphones because they enclose an extremely small volume which, below some kHz, is characterized by a basic mode with spatially constant sound pressure only. Within the last 30 years this has lead to various standard products being offered today by various established manufacturers of headsets. These solutions differ between pure ear protectors and headsets with improved low-frequency broadband sound insulation (typically below 1 kHz) against undesired exterior noises. By such active measures, the reception of any desired signal (music or speech) can be essentially improved.

As arbitrarily incoming broadband sound signals are generally nonpredictable, these ear-related systems require feedback control where the signal delay time determines the upper frequency limit. By integrating a loudspeaker, all active ear protectors are converted into ear phones. Also, some error microphone must be provided.

Figure 12.19 shows that the sound insulation of a headset for pilots can be improved by adding an active noise suppression system by up to 35 dB below some 600 Hz.

For larger volumes, things are more complicated because the larger the volume the lower the frequency where spatial dependence of the sound field has to be taken into account. Then, any active field compensation necessarily requires the reproduction of this spatial dependence in magnitude and phase. This generally requires several loudspeakers and their number must be the larger the higher the upper frequency limit has been defined.

Attempts to control the interior noise of cars by active means have been made since 1980 and led 10 years later to first successful demonstrations (see e.g., [1, 99]). It subsequently turned out that for passenger cars an equipment of 4–6 loudspeakers is appropriate and manageable and that this enables global reductions and/or changes of the sound field below some 300 Hz which can be registered at all seats and which are widely independent of head positions and head movements.

For engine-related sounds having a harmonic spectrum of multiples of half engine orders, the predictable sinusoidal waveform of the noise components allows good compensation and, in consequence, good sound modification. Figure 12.20 illustrates typical input signals and components.

The first and most important step in designing such a system is to find out whether the loudspeakers and power amplifiers of the standard audio system are capable of providing the required sound field (which is the case for most of today's systems) or if additional secondary sound sources are needed. These loudspeakers are driven by signals which an adaptive signal processing unit determines from a given rpm-signal and error signals (difference between actual and desired signal). The respective algorithm realized in the signal processing unit aims to minimize the error signal or to approximate the target signal, respectively. In most cases, the present sound pressure value (actual

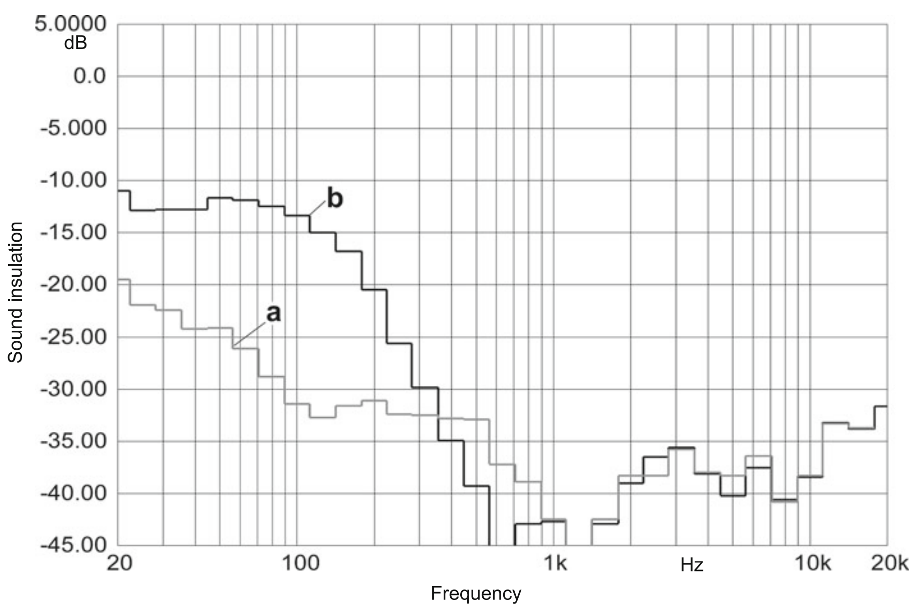


Fig. 12.19 Sound insulation of a pilot's headset (**a**) with and (**b**) without active noise compensation [98]

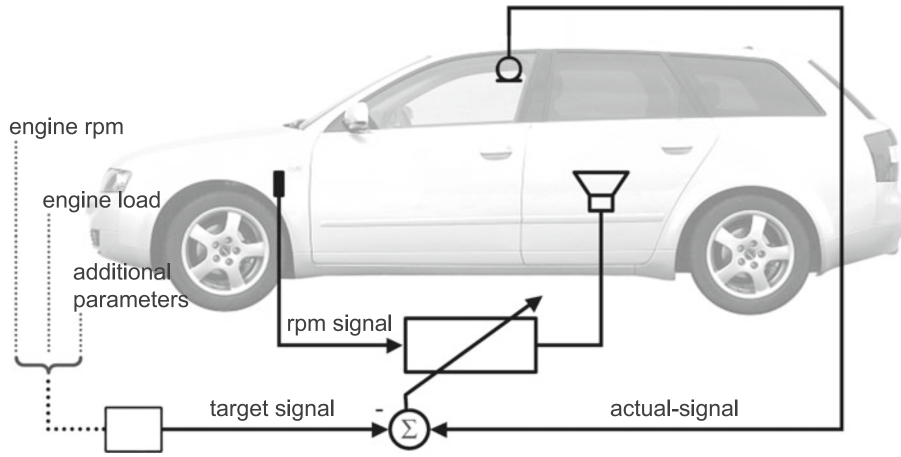


Fig. 12.20 Principal arrangement of a system for active control of car interior sound

signal) is registered via six microphones appropriately distributed over the car's roof area.

In case of a pure noise suppression system, the target signal is zero. By minimizing the mean square error, the signal processing algorithm then equally minimizes the signal itself. By defining differing target signals, it is also possible to realize differing interior sounds in the car [16, 100]. However, to be realistic, such target signals have to be evaluated from important operational parameters like engine rpm or engine load.

An example for the noise reduction which can be achieved by such an arrangement is given in Fig. 12.21. It can be seen that the dominant second engine order of a four cylinder passenger car could be reduced essentially to a comparable level at the two front seats. The reductions in level were up to 20 dB.

By increasing the number of loudspeakers and microphones involved, this approach can be transferred to larger volumes, especially if sinusoidal noise components are to be controlled. This is particularly true for the noise components caused by turbo-propeller engines in aircraft cabins. Starting from proven evidence of the realizability of active noise cancelation in aircrafts [101–103], such systems have been developed for mass production and can be purchased on the world market [104].

Other than for tonal engine sounds stochastic driving sounds are related to the difficulty of finding coherent input signals which provide enough delay time to evaluate suitable secondary signals. Therefore, any broadband level reductions obtained in practical experiments were limited so far to some 5 dB.

Again, higher level reductions may be obtained by limitation of the frequency range to be considered. For low frequency rolling noise with a dominance around some 40 Hz, level reductions of some 10 dB could be obtained with a ready for production system using skillfully a combination of feed-forward and feedback control [65, 105].

As stated above, for global field control the number of secondary sources increases with the size of the volume to be controlled and the highest frequency to be considered. For many problems it is sufficient, however, to control the respective sound and vibration field locally only, in a smaller part of the total volume. As such a limitation in space – for a constant upper frequency limit – needs less secondary sources or – for a constant number of sources – allows higher upper frequencies, it is worthwhile to limit the considered volume according to the sketch of Fig. 12.22.

This can be concretized by an example of how the frequency range for active car interior sound control may be extended. Of course, this can be achieved by increasing the number of loudspeakers and sensors at positions which have to be checked carefully with respect to the specific situation. However, restricting larger head movements of the passengers and allowing

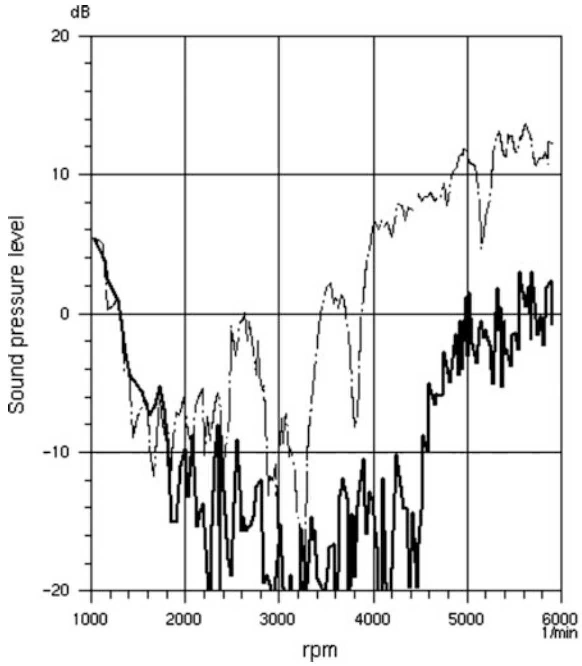


Fig. 12.21 rpm dependence of the sound pressure caused by the second engine order at the driver's (*left*) and co-driver's (*right*) seat without (*upper curve*) and with (*lower curve*) active noise cancelation (ANC)

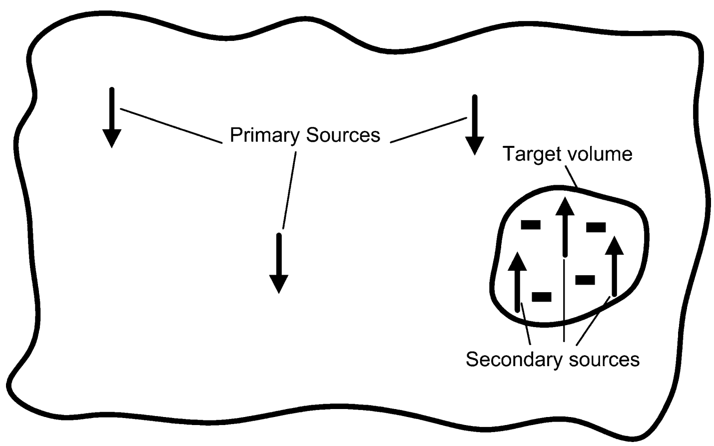


Fig. 12.22 Principal illustration of local control for partial volumes

small movements only, it is sufficient to keep the number of sources constant and to control small volumes only around the heads.

For experimental sound tests in testing vehicles, this is absolutely sufficient and it thus became customary to expand the frequency range for car interior sound control in such special (test) situations by close-to-the-ear positioning of error microphones from some 300 Hz to some 600 Hz.

12.4.5 Active Compensation of Sound Radiation

Many technical noise control problems deal with sound transition through and sound radiation from vibrating structures. In such cases, it is obvious to try to control the radiation itself instead of controlling the radiated sound field. The principal sketch of Fig. 12.23 illustrates this [2, 3].

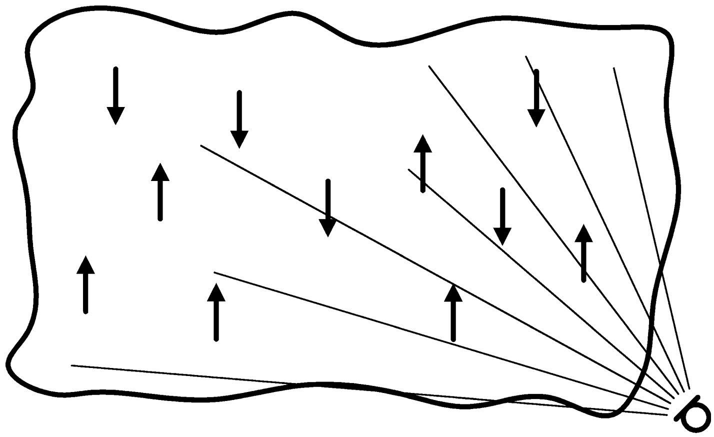


Fig. 12.23 Principal sketch of active radiation compensation

Basically there are two approaches to solve this problem; controlling the vibrating structure which excites the surrounding medium, or controlling the surrounding medium to reduce the introduction of power from the vibrating structure. While the first approach acts on the vibration pattern of the structure, the second approach aims at changing the radiation impedance seen by the structure in a way that the total radiated power will be reduced [106].

This can only be achieved by arranging secondary sources directly at the radiating structure and then reproducing that part of the volume flow that contributes to radiation which then results in a hydrodynamic short circuit. Apart from basically considering the realizability of the concept by discrete [107] or distributed (active skin, [108]) sources, integrating piezoelectric actuator elements into passive absorbing materials (acoustic foams, [106]) might open interesting perspectives. However, their effective practical applicability still has to be proven finally.

The other possibility is to influence the sound radiation via the radiating structure. The easiest way to do this is to reduce the vibrations of the radiating structure in total, e.g., by minimizing the mean square velocity which is proportional to the kinetic energy of the structure. With this definition of solely structure-related target quantities the complete task has been restricted to the structure and therefore must be analyzed systematically in structural terms according to Sects. 12.4.2–12.4.4.

This certainly makes sense if the structural vibrations can be controlled globally by a few sources only, e.g., by reproducing all effective structure-borne sound sources. As an example for this approach we refer again to the compensation of low frequency interior sound components in an ICE train car as mentioned in Sect. 12.4.2. There, compensating the exciting forces at the secondary spring connecting points resulted in significant airborne sound reductions.

In particular, if such simplifying limitations of the secondary source effort are not possible it is worthwhile to differentiate the structural vibrations with respect to their radiation efficiencies. Especially for low frequencies, below the critical frequency defining coincidence with the wavelength of the surrounding medium, the radiation efficiency of various vibration patterns may be very different. Any definition of target quantities directly related to radiation inevitably leads to a weighting of the vibration patterns according to their radiative coupling to the surrounding medium.

As a result, only the modes which contribute to radiation are controlled. It is therefore evident that this approach needs fewer secondary sources than the previous one in order to minimize the mean square velocity of the radiating structure. This applies the more, the less modes contribute to radiation.

Because this active measure fully concentrates on the structural acoustic coupling with the surrounding medium, it usually is called “Active Structural Acoustic Control” or, in short, ASAC.

The difficulty with this approach can be seen in the necessity to register all radiation relevant parameters by appropriate sensors. In some cases, this can be achieved by microphones placed appropriately in the radiated sound field, but often this is not feasible. It is therefore desirable to derive the radiation-relevant parameters, e.g., the complex amplitude of an efficiently radiating mode, from structural measurements only.

As each vibration mode corresponds to a characteristic wave number, this derivation can be reduced to evaluating the wave number spectrum of the radiating vibrational distribution by measurement where the mean value ($k = 0$) of this distribution specifies the total volume flow.

A detailed explanation of these relations may be found in [2] and [3], a recent overview in [106] as well as in the literature given there.

As a representative example for this section we again refer to active control of aircraft cabin interior noise caused by propeller-driven pressure waves acting on the fuselage. Instead of using loudspeakers distributed all over the aircraft cabin, this sound field also can be reduced by force actuators acting on the fuselage.

Figure 12.24 gives the result of such a measure fitted by standard into an aircraft. Here, 42 electromagnetic force actuators (active tuned vibration absorbers, ATVA) tuned to multiples of the blade passage frequency (bpf) act such on the structure that the sound pressure, which is registered by 80 microphones, is minimized at the bpf and its first harmonics [109].

As can be seen from the comparison of Table 12.1 which was taken from some other aircraft, the radiation suppressing system acting on the structure (ASAC) comes up with better results at all four frequencies than a comparable system based on loudspeakers (ANC, [104, 106]).

Table 12.1 Comparison of level reductions (in dB) in a propeller driven aircraft [104]

	1 bpf	2 bpf	3 bpf	4 bpf
ASAC	10.5	7.6	4.4	3.0
ANC	8.0	6.6	3.6	0.4

12.4.6 Stabilization of Self-Excited Systems

Many sound and vibration fields emerge from an unstable interaction of different physical processes mutually affecting themselves. Increase of one field quantity results in the increase of another second field quantity which then causes further increase of the first again until nonlinear restrictions enable a stationary limit cycle in the end.

Typical examples are flutter vibrations caused by flow [110], compressor surge [111] or vibrations resulting from mutual interaction between combustion and air/fuel supply in combustion chambers [112–114].

In the third section it was pointed out that this mutual excitation can be described by two coupled systems (Q and B in Fig. 12.2) in a feedback loop. This arrangement immediately suggests a concept for stabilization: the compensation of unstable feedback by an additional system G (Fig. 12.25).

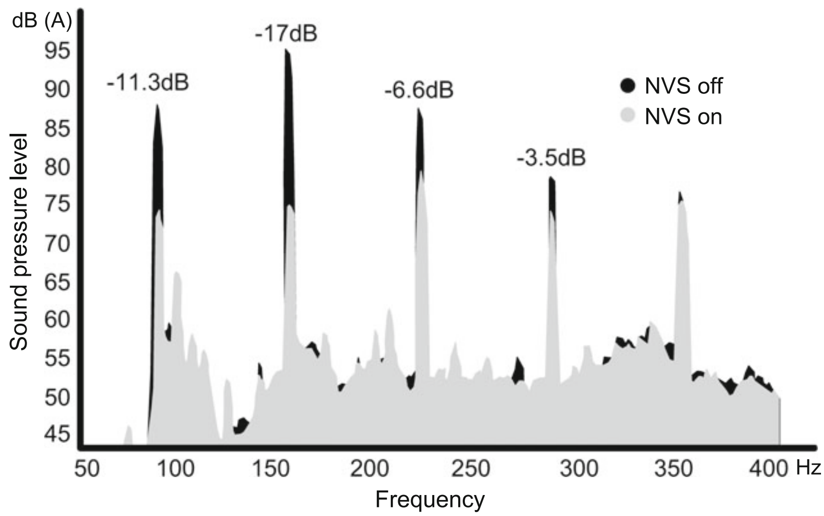


Fig. 12.24 Sound spectra in a turbo-prop driven aircraft without (NVS out) and with (NVS in) active suppression of sound radiation [109]

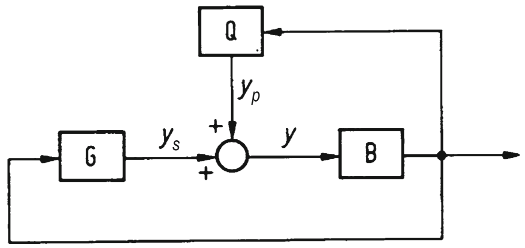


Fig. 12.25 Structural diagram illustrating the compensation of unstable feedback

Unlike the field reproductions considered so far, G does not have to provide a negative copy of the primary field quantity y_P. Because y_P itself depends on y and thus also on y_S, it is sufficient to ensure stability of the total resulting system. Therefore, the transfer characteristics of G can be followed from requirements to the position of zeroes of the characteristic equation

$$B \cdot (Q + G) = 1. \qquad (12.1)$$

In practice this means that amplitude and phase of the secondary field quantity just have to stay within wide stability intervals instead of fulfilling specific values [11, 115]. Also, early interaction with the sound-generating mechanism will avoid large amplitudes. Therefore, like powerful signal processing units, powerful actuators may turn out to be unnecessary. The secondary field quantities remain small but may have huge effects [116].

This simple concept of stabilization may directly be transferred to physical reality. At first this can be followed from the good agreement of computation and experiment for simple systems like the blown Helmholtz resonator [115] or the Rijke-tube [116].

Besides these rather simple systems, the approach described also seems to be promising for other technical problems, especially in flow acoustics [11, 117]. However, to cope with the needs of industrial practice, the requirements to sensors, controllers and actuators are considerably increased. Nevertheless, it was possible to successfully apply the method to practically important technical systems and processes.

As an example, self-excited combustion vibrations of a 170 MW gas turbine at 433 Hz could be successfully reduced by an active control system. These vibrations are caused by feedback between the sound field in the combustion chamber, the supply of fuel and air to the burner and the burning reaction itself. Sound pressure fluctuations cause pulsations in fuel supply which in turn force the flame to fluctuations in the heat release rate. Fluctuating heat release leads to additional pressure fluctuations which then further stimulate the sound field.

Applying the stabilizing method means to monitor the pressure fluctuations in the combustion chamber and to apply them to a controller which then drives a special valve in the fuel supply. This valve modulates the mass flow of fuel such that the pulsations resulting from the cycle of self-excitation are compensated [118].

Figure 12.26 shows the pressure amplitudes measured in the combustion chamber of a gas turbine with and without such active control. It can be seen that the vibrations at 433 Hz in the combustion chamber could be reduced by more than 80%, from some 210 mbar (177 dB) to some 30 mbar (160 dB).

Another example is active control of low-frequency pressure and velocity fluctuations which often are a real problem in open jet wind tunnels. This so-called wind tunnel buffeting is excited by vortex sheds from the nozzle convecting downstream. Reaching the collector, they generate a pressure disturbance which then may excite an acoustic mode of the duct. At resonance, a standing wave occurs inside the duct which then in turn triggers further vortex generation.

As wind tunnel buffeting may seriously distort acoustic and aerodynamic measurements, different measures to diminish this problem have been considered in the past. Unfortunately, many of these passive measures generate disturbing noises and therefore are useless for applications in aeroacoustic wind tunnels.

On the other hand, by applying an active control system it was possible to significantly reduce wind tunnel buffeting without generating additional noise perturbations. The system uses appropriately driven loudspeakers within the tunnel causing a change in impedance which then influences the feedback process at the resonance frequencies of the channel.

Figure 12.27 gives the spectrum of a sound pressure level measured in the test section of a tunnel under normal conditions and with the control system switched on. At the resonance frequency, the sound pressure level interfering with the operational measurement is seen to be reduced by more than 20 dB

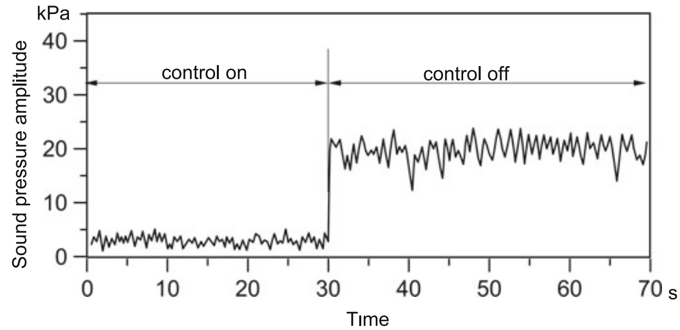


Fig. 12.26 Sound pressure amplitudes in the combustion chamber of a gas turbine (170 MW, some 17 bar statistical combustion chamber pressure) with (0–30 s) und without (30–70 s) stabilizing control [118]

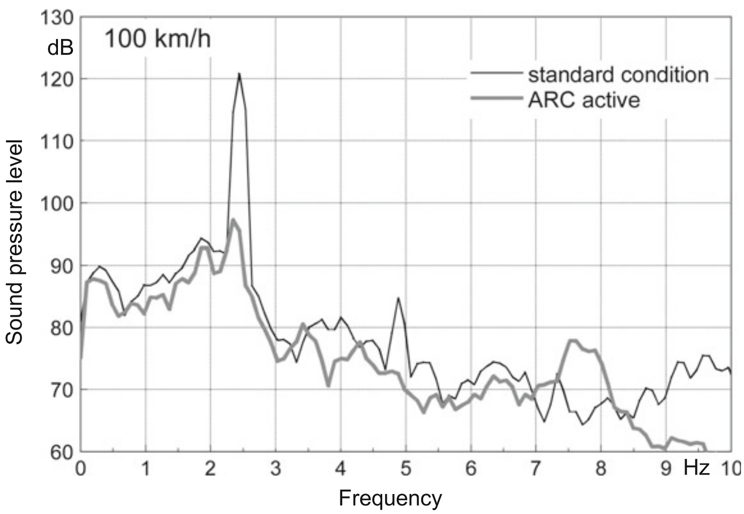


Fig. 12.27 Sound pressure level spectrum in the test section of a wind tunnel at a flow velocity of 100 km/h under normal standard conditions and with active control ARC [119]

when the active system is applied (Active Resonance Control, ARC, [119].

12.4.7 Energy and Power Considerations

In general only little can be said about energy and power relations for active methods of field control. Unlike successful control of sound-generating mechanisms, where the control of field quantities correlates with control of power radiation, the energy requirements related to interference of reproduced fields are not evident. This is because linear superposition of coherent field quantities does not cause additive superposition of the related quadratic power quantities.

This may be shortly illustrated in the following for the example sketched in Fig. 12.28, where N point forces F_i, $1 \leq i \leq N$, act on an arbitrary structure.

The total power introduced to the structure by the forces is given by

$$P = \sum_{i=1}^{N} \frac{|F_i|^2}{2} \left[\mathrm{Re}\{\Upsilon_{ii}\} + 2 \sum_{k=1}^{i-1} \mathrm{Re}\left\{\frac{F_k}{F_i}\right\} \mathrm{Re}\{\Upsilon_{ik}\} \right], \tag{12.2}$$

whereby v_i is the velocity at the point of action of the i-th force F_i and

$$\Upsilon_{ik} = \frac{v_i}{F_k}, \tag{12.3}$$

generally describes the transfer admittance from the excitation point k to the receiving point i. The point admittances are included in this definition as special cases $i = k$.

It is obvious from the second term of Eq. (12.2) that the power introduced does not only depend on the amplitudes $|F_i|$ of the forces and the admittances Υ_{ik} of the structure but also on the differences in amplitude and phase between the force amplitudes. This

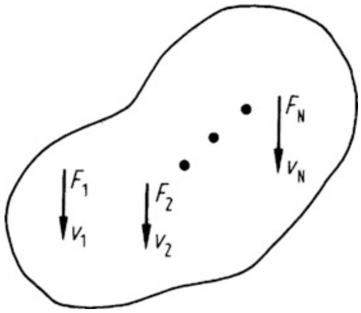


Fig. 12.28 Application of several point forces to a structure

interaction becomes particularly clear if the effective admittance at the point of the ith force, given by

$$\Upsilon_{i,\mathrm{eff}} = \frac{v_i}{F_i} = \sum_{k=1}^{N} \Upsilon_{ik} \frac{F_k}{F_i}, \tag{12.4}$$

is considered. This quantity, which determines the introduction of power depends from the relation of the complex force amplitudes.

Following this equation, the point admittances may have negative imaginary parts. Then, the force F_i absorbs power from the structure. However, it cannot be concluded from this that the total energy introduced or stored will be reduced because absorption at point i may be coupled with increased power injection at other points.

These considerations illustrate that the effects of secondary sources on energy relations can be explained by two mechanisms: by control of the radiation impedance seen by the primary source and by injection or absorption of power.

For this reason, any assessment of an active measure based on local power balances must consider the totality of all sources involved or – similar to the active absorber of Sect. 12.4.3 – be sure that the given arrangement of secondary sources is free of any feedback. Any exclusive consideration of the power absorbed by anti-sources may generally not be sufficient. Illustrative examples confirming this generally may be found in [8, 54, 60, 71, 120].

Active absorption, i.e., power extraction by electroacoustic and electromechanic actuators, is not only a theoretical possibility but has also been proven by practical measurements. For an appropriately driven loudspeaker, such proving evidence was described in [121].

12.5 Active Sound Design

As stated in the introduction, the compensation of fields also gives the possibility to replace the compensated sounds by other sound impressions and thus to change freely – within wide limits – existing sounds. Even if – apart from the related cost – doubting the acceptance of “artificial” electroacoustic sounds may argue against bringing it on a product level: as a tool, useful to preliminary realize and test attributive sounds, active sound control offers enormous possibilities.

This is because any reliable assessment of such attributive sounds requires their reception within their total multimodal context, where other perceptive sensations add to the related sound. As this cannot be realized in a studio, the importance of electroacoustic sound manipulations which change given sounds at the product continuously grows. And this is exactly what active sound design (ASD) can do. In the following this shall shortly be illustrated for car interior sounds.

Historically, the sound of a car is one of the most important attributes for the user. Consequently, such sounds were designed more and more consciously. But such design needs a clear specification of the noises and sounds to be targeted. For success, known or somehow given sounds may be manipulated, new sounds may be defined, both then may be analyzed and assessed psychoacoustically in hearing experiments.

However, because hearing sensation and sound impressions are strongly influenced by the driving experience, such new car sounds can finally not be assessed in the lab. Therefore any final assessment must be based on comparable in-situ tests in driving cars. The availability of related prototypes with various, thoroughly tuned sound variations generally demands very high efforts.

Here, active design and realization of rpm-related car interior sounds provides a development tool which varies the sounds of engine-related components without modifying the components themselves. Thus enabling subjectively justified tests and specifications, differing sounds can be assessed and further developed in the context of a full driving experience without building prototypes [122].

As an example, Fig. 12.29 shows for the car, previously mentioned in Fig. 12.21, how the sound

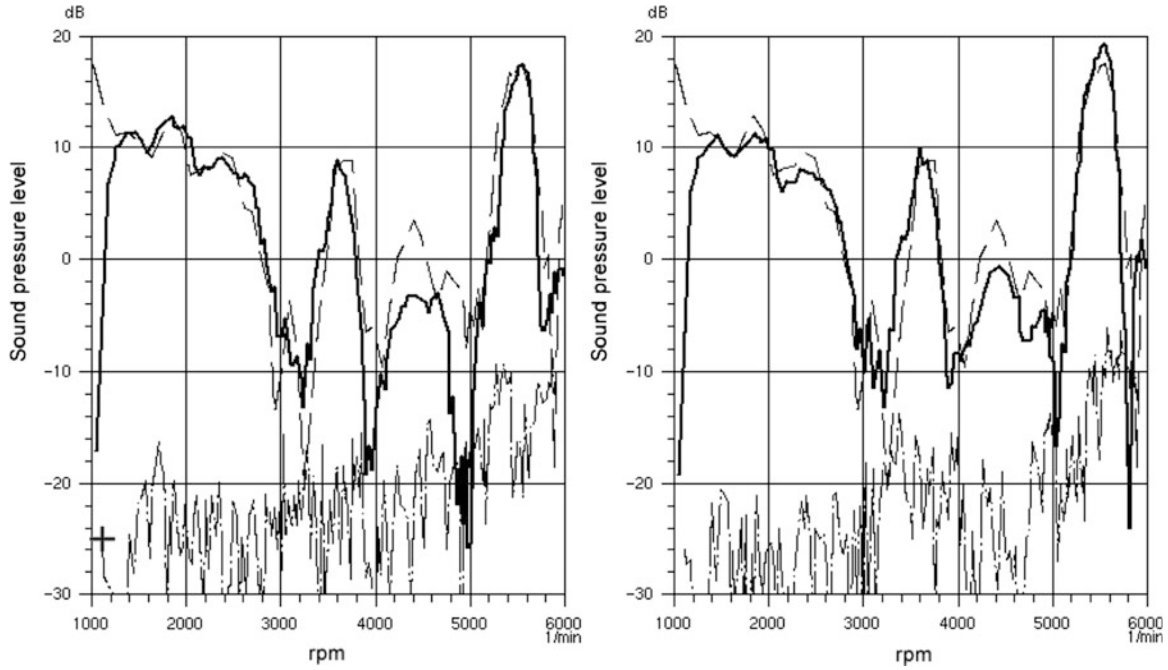


Fig. 12.29 rpm dependence of the sound pressure related to the third engine order at the driver's (left) and co-driver's (right) seat without (*lower dash-dotted line*) and with (*upper continuous line*) active sound design (ASD). The dashed upper line describes the target to be met by ASD

character of this four cylinder engine can be modified [100]. It can be seen that the third engine order which typically can be neglected for four cylinder engines (dash-dotted line) well approximates a given rpm curve (dashed line) by using ASD. Spectra dominated by the third engine order are typical for six cylinder engines. Thus, by comparing Fig. 12.29 with Fig. 12.21 where the suppression of the second order is demonstrated, one can conclude that the sound impression of a six cylinder engine can be generated by a four cylinder engine properly controlled.

This can also be demonstrated for the order analysis of another car equipped with ASD, Fig. 12.30. Here it can be seen again how the reduction of the second engine order in the middle figure enhances the third, sixth, and ninth engine order which are typical for six cylinder engines (see the lower figure).

Exterior noise of cars also can be modified by active mans. For exhaust mufflers this had been shown already in Fig. 12.17. Further realizability of comparable results for engine air intake systems is demonstrated in [86].

Besides evidence by measurements the authenticity of actively realized engine sounds in cars is proven by many subjective assessments. Even if sound variations in practice work with very fine gradations only; together with the authenticity, the examples given clearly demonstrate the possibilities of active sound design when specifying and developing product sounds.

12.6 Aspects of Signal Processing

Realizing effective control of mechanical vibration and wave fields requires not only appropriate selection and arrangement of the actuators but also their proper driving.

Unlike directly influencing the sound-generating mechanism as described in Sect. 12.4.6, all signals provided by the signal processing unit G must exactly meet their desired values in amplitude and phase to obtain considerable level reductions by interference. This can be illustrated easily by considering the superposition of two harmonic vibrations with small deviations α in amplitude and δ in phase. The level reduction with respect to one of these vibrations alone is given by

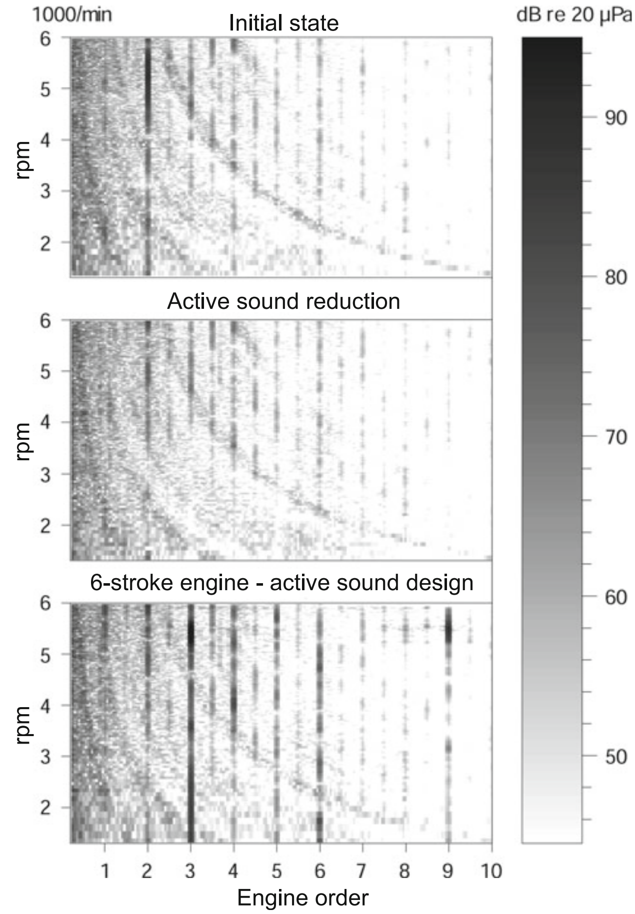


Fig. 12.30 rpm dependence of engine order related sound pressures (order analysis) at the driver's seat of a car at the initial state (*above*), with active sound reduction (ANC, *middle*) and with active sound design (ASD, *below*)

$$\Delta L = -10 \lg\left[\alpha^2 + 4(1+\alpha)\sin^2\frac{\delta}{2}\right]\text{dB}. \qquad (12.5)$$

From this equation, it can be seen that an error in amplitude of 10% ($\alpha = 0.1$) and in phase of $\delta = 10°$ reduces the level reduction ΔL to 13.6 dB only. For $\delta = 20°$ and constant $\alpha = 0.1$ only 8.5 dB remain.

Together with the fact that large errors in phase even may increase the resulting level this shows that realizing good approximations of the primary field quantities is an important issue. The precision of this approximation is limited by what information on the temporal behavior of the primary fields can be taken from the input signals y_R and z_R.

To quantitatively describe this information the stochastic frequency domain relation between these input signals and the primary field quantities y_P can be taken into account. This relation is given by the coherence function γ^2 and allows to specify an upper limit for the level reduction which, independent of the signal treatment, cannot be exceeded.

If S_g describes the power spectral density of the field resulting from the superposition of primary and secondary field and S_p the power spectral density of the primary field alone, the minimally obtainable lower bound for the ratio of S_g and S_p is given by [4, 123]

$$\frac{S_g}{S_p} = \left(1 - \gamma^2\right). \qquad (12.6)$$

For a coherence of 90% ($\gamma^2 = 0.9$), the maximal level reduction amounts to 10 dB, for 99% coherence to 20 dB therefore, considerable reductions of primary fields require a strong correlation between the primary and the compensating fields. Hence, after a careful specification of the source arrangements, is the selection of such signals a second very important issue in the designing of active measures or in assessing their performance.

Together with the transducers, the original mechanical part of the system and the signal processing components (like low pass filters, measurement, and power amplifiers), the source and measurement arrangements define a global system M whose input and output signals are connected to the signal processing unit G according to Fig. 12.31.

The remaining task in specifying such G is to assure that all input information available is exploited to drive the selected actuators optimally with respect to the target quantities.

This problem has to be solved in two steps. First, the structure of G must be specified in terms of a sufficient number of suitably linked parameters. This structure describes the relations between input and output quantities in general terms, e.g., by a specific form of equations or a structural diagram. Only then any concrete realization of the optimal transfer behavior between input and output quantities can get started by specifying all related parameter values.

Any definition of the structure of G should take advantage from given knowledge about the real structure of M. For example, this may essentially facilitate the approximation of any relation between measured quantities and the primary field quantities or the consideration of internal feedback. Bad structural adjustment between M and G reduces the quality of signal adjustment ultimately obtainable by optimal parameter selection. An illustrative example for this difficulty is found in the difficulty to approximate recursive (IIR) structures by non-recursive (FIR) filters and vice versa [124–126].

An important application of structural adjustment of G to M is given by the compensation of closed (via M) feedback loops which cause actions of the outputs u on the inputs x of G. Such compensation can be achieved by additional feedback loops realized within G [13].

With this measure, the stability problems caused by feedback and mentioned in Sect. 12.2 can be reduced. Therefore, the compensation of feedback (feedback cancelation) is widely used for tests and for applications of active methods [4, 9, 60, 78]. Also, it is shown in [60] how feedforward and feedback loops can be identified in the frequency domain from different input/output measurements while, in [78] a possibility of adaptive identification of both loops is presented.

There are many ways of defining the structure and the parameters when designing the signal processing unit. Unfortunately, it is not possible to deal thoroughly with them here, especially because there is a general lack of systematic procedures. Instead, only some important terms and principles shall be mentioned. Further considerations of system design [4, 48, 53, 124–126] and of concrete applications [1–5, 25] are found in the literature.

The global properties of M, like the transfer behavior, observability and/or controllability [48, 50, 53], not only depend on the total mechanical and acoustical arrangement but also on the number and position of the actuators. If, in addition, the coherence requirements Eq. (12.6) together with the fact, that only causal relations between x and u can be realized, are taken into account, the importance of selecting appropriate locations for measurements and actuations is obvious.

Optimal transfer properties of G may be obtained by minimizing certain field quantities put together in the target quantities z. Here, quadratic criteria are to be preferred because they combine good theoretical insight with easy implementation. Above all, optimal linear filter theory [127] can be applied to support and assess the design of G.

Then, continuous monitoring of the remaining error signal allows an equally continuous adaptation of the transfer behavior aiming at further error reduction.

This adaptive approach, schematically given in Fig. 12.32, ends up with optimal transfer characteristics and the related transfer elements therefore are called adaptive filters. Their ability to automatically

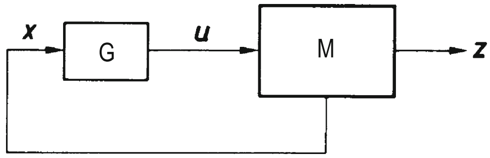


Fig. 12.31 Principal sketch of the total system and its respective separation into an electromechanical system M and a signal processing unit G

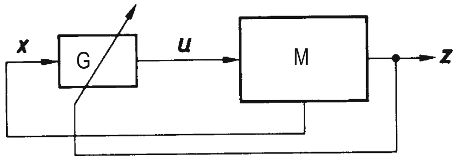


Fig. 12.32 Principal arrangement with an adaptive signal processing unit

compensate any fluctuations of important system parameters like temperature, flow velocity, or rpm has given them a dominant role in applying active methods.

The theory of adaptive signal processing is highly advanced and may be used for designing active systems. Comprehensive treatments are found in [128–131], typical examples for concrete realizations of adaptive algorithms in [4, 5, 78]. Early difficulties related to adaptive implementations of IIR-filters could be overcome by well-converging algorithms with real time capabilities [78, 132].

Of course, the particular choice and efficiency of such computational rules strongly depend on the statistical signal properties. Many sound and vibration sources, e.g., all rotational machinery, generate periodic time histories whose future behavior can be well predicted from their past history. Based on given rpm information this allows the development of efficient, well-converging algorithms which, following a computational proposal first formulated in [128], can be implemented in compact form.

Many of the intuitively derived algorithms for a single input and one output quantity as well as their multidimensional extensions can be considered as generalizations of the LMS-algorithm with filtered input signal (filtered-x-LMS-algorithm). This enables global convergence properties with even many input and output quantities [4, 5, 129].

The predictability of revolutionary signals enables adaptive algorithms to come up with high-level reductions. This can be demonstrated by a laboratory experiment of active vibration isolation. As shown in Fig. 12.5, four electrodynamic vibration sources (shakers) had been arranged in parallel to four passive springs at the four coupling points between an electric motor and its foundation. These shakers were driven such that the forces introduced into the foundation at the coupling points and measured as error signals were minimized.

Figure 12.33 shows the result without and with application of secondary forces [133]. As can be seen, adaptive control is able to fully compensate the force caused by some unbalance to the level of broadband noise. The related force level reduction at the basic frequency is 81.6 dB. Also, all further harmonics of the line spectrum generated by the electric motor are reduced accordingly while the amplitude at some 100 Hz, uncorrelated to the line spectrum, remains unchanged.

Despite all advantages offered by adaptive control schemes, the benefits of using or combining simple basic signal processing elements to more complex control structures have retained continuous interest in feedback control loops. Examples how this approach may be used to suppress structural vibration and radiation can be found in [134–136].

12.7 Electromechanical Transducers as Actuators

Before the 1980s, any limitation in realizing active measures was mainly caused by limitations of electronic signal processing. After the immense development of digital signal processors in the last 20–30 years this does hardly apply today. Instead, present limitations are rather caused by restrictions of electro mechanic transducers.

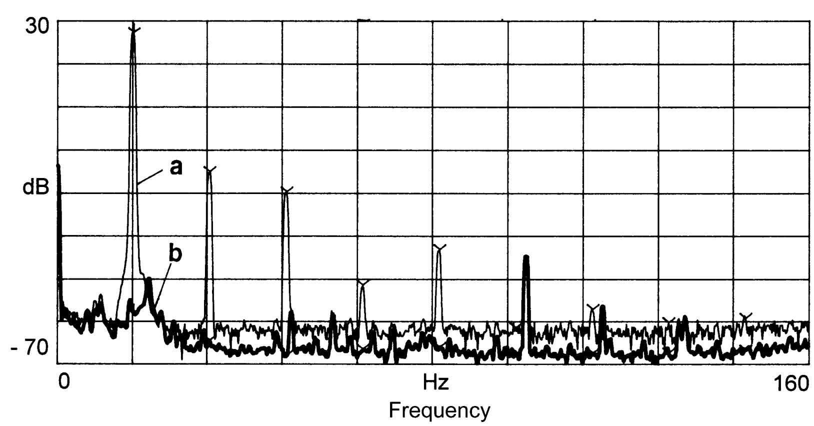


Fig. 12.33 Frequency spectrum of the force introduced into the foundation (**a**) without and (**b**) with active isolation

Such transducers typically are characterized by low efficiencies. This is particularly true for low frequencies as the frequency transfer characteristics of closed loudspeaker boxes fall off with 6 dB/octave below the mechanical resonance frequency. The resulting intention to tune the system to lower frequencies directly leads to larger volumes because otherwise the resonance frequency will be determined by the stiffness of the air enclosed.

If the application provides limited space for transducers only, this tends to be the decisive restriction. For this reason, providing powerful transducers is an essential condition for increasing applications of active measures.

Also, the actuating devices used to directly control structure-borne sound may refer to proven principles. Thus, any application of forces in practice often is realized by electrodynamic or electromagnetic vibration exciters.

Besides that, other transducer principles, not being considered for airborne sound excitation because of insufficient deflections, may be very efficient for structural applications. Both the piezoelectric as well as the magnetostrictive effect allow for applying very strong forces [137–139]. The small deflections related to these effects, however, limit their application to cases where the force application points are subject only to small travel distances.

In special cases this limitation may be overcome, Fig. 12.6 illustrates this for the case of active compensation of force introduction at an ICE bogie. This force being supplied by sliced piezo elements and being arranged together with their seismic masses within the secondary springs has been freely applied to the car body. Despite this combination of high force potential and compact arrangement, problems with linearity finally led to the insight that preference would have been given to electrodynamic actuators in case of practical applications.

For surface structures, active control is not tied to the application of point forces. In particular, with rods and plates it is possible to integrate suitable actuators into or onto these. This application of distributed actuators, e.g., by coating with piezoelectric foils, is not limited to flexural and longitudinal excitation [2, 3, 6, 106]. By using corresponding shaping, it can further be achieved that registration and excitation are confined to specific modes.

The integration of electrically shapable elements into structures gives rise to multiple perspectives and hopes as put into words like "intelligent" or "adaptive" structures. Despite some evidence of realizability mainly based on controlling single modes and waves so far, the practical potential of surface distributed or structure integrated actuators cannot be predicted reliably yet. If such approaches would mature to an applicable technology in the future, they could serve as a promising basis for active control of sound radiation and transmission from or through structures.

12.8 Further Applications

Without claiming completeness, this overview shall give some completing examples for the applicability of active methods being disregarded before.

It was in the early days of active sound control that people tried to take advantage of the predictability of sound fields with single discrete frequencies to effectively cancel them. An ideal technical realization to test the new method was seen in those days in electric transformers with their discrete harmonic frequency spectrum [34].

However, the dimensions of transformers cause rather complicated, moreover load-dependent spatial radiation patterns which therefore require spatially variable secondary fields. Experimental successes therefore were confined to distinct volumes [140, 141], particularly spatial angles [142] or simplified reference radiators [143].

For this reason, any global compensation by secondary sources of the sound field radiated from transformers has been judged unsatisfactory on account of complexity and the obtainable level reductions. Nevertheless, [106] reports on a system which successfully compensates the basic frequency by a loudspeaker and higher harmonics by piezoceramic actuators fixed to the casing. Also, active measures at transformers may complement passive measures, e.g., by supplying indispensable openings of shielding elements with active sound attenuators.

For open windows of buildings, comparable solutions could not be found. Although quite a few corresponding investigations had been made in the first years of active sound control research, they finally failed because of the difficulty of reproducing three dimensional incoming wave fields with acceptable efforts.

Things are more handsome if the sound insulation between two rooms shall be improved by acting on the separating structure. Corresponding investigations for (closed) windows are found in [144, 145].

In vibration control, all approaches requiring electronic control are classified as active methods anyway: magnetic bearings and lightweight robot devices. Active magnetic bearings use appropriately controlled magnetic forces for position control and, because of reduced friction losses, are mainly applied in rotating systems. Their advantages as well as further applications and problems are described in [146] and in the references given there.

Progress with robot devices had to cope with increased moving speed of more and more lightweight arms. Therefore, their control could not only be restricted to their positioning. Instead, the dynamics of the structure to be moved had to be taken into account or even to be compensated by considering them in the signals controlling the robot movements.

In the end of this section reference shall be made to applications where, instead of the sound and vibration field quantities themselves, the dynamic properties of the elements involved (e.g., the stiffness of a spring) are controlled. Typical examples of this approach, which often is referred to as semi active, are shock absorbers and actively supported dynamic vibration absorbers [6].

12.9 Summary and Perspectives

This chapter attempts to present the state of the art of active noise and vibration control methods. Special emphasis has been given to essential physical mechanisms because any use of highest development standards in algorithms and hardware even is limited by the fundamental limits of physics. Furthermore, thorough knowledge of any physical possibilities is the best platform for systematic exploration of the most promising concepts towards satisfactory solutions.

Besides emphasizing these basics, the many successful application exemplify the surpassing of provisional laboratory experiments and tests.

Like any other technical method, active control of sound and vibrations offers a general approach capable of solving many, yet not all, problems of noise and vibration. This may have been overseen by some early, mostly enthusiastic promises stirring unrealizable hopes.

However, the results as well as partial results so far justify further confidence. If further progress is made with materials and concepts to generate sound and vibrations, with electronic units to process signals and with clear and robust control concepts for multiple inputs and outputs, it should be possible to improve development and handling and to reduce effort and cost of active systems to an acceptable level.

Against this background, some confidence and optimism may be justified that the fascinating approach of active sound and vibration control may further flourish in the future by a growing number of fitting applications.

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