ACTIVE CONTROL OF SOUND AND VIBRATION – A SYSTEMATIC APPROACH WITH TECHNICAL EXAMPLES

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ABSTRACT

After some general comments and assessments with respect to the current status of active noise and vibration control measures in practice, the basic structure of any active approach is used to illustrate the various steps of any system design process. Within this process, a logical differentiation of underlying physical principles may help in finding optimal locations and mechanisms of interference. This is presented in a systematic way and demonstrated with some basic but illustrative examples which include the fascinating possibility of implementing and experiencing arbitrarily designed sounds in situ, i.e. outside the laboratory under real life and real time conditions.

INTRODUCTION

As long as technical historians don't agree on a final classification, the definition of technological periods is always characterised by personal impressions and assessments. This definitely holds for active sound and vibration control, where the period of serious technology-oriented and technology-promising efforts was started almost 30 years ago with the pioneering work of M. Swinbanks, who – in a unique way – was able to combine a thorough understanding of the physics involved with the necessary need for technical pragmatism ([1]). Together with the upcoming prospect that almost all signal processing needs might be met sooner or later by the new and fast progressing digital signal processors, this was the starting point of an era which unfortunately cannot be concluded yet by a final breakthrough: the era of developing, introducing and using active sound and vibration control as a broadly accepted acoustic technique.

This technique was first formulated in writing by P. Lueg in the thirties of the last century([2]) and had an early exploration phase in the fifties ([3],[4]), where systematic laboratory experiments were used to investigate the feasibility of the novel approach. In the later sixties first attempts were made towards an improved understanding by theoretically modelling physical interaction on the basis of various field descriptions ([5]).

But then, since about 1970, serious work was started at different places and soon was accompanied by an optimism which sometimes could be observed and even was urged to degenerate into loud and unrealisable promises. This was – in the long term – paid by a loss of confidence and reputation and can be held jointly responsible for some frustration coming up in the last years. Other reasons for disappointment can be seen in a largely spread patent policy which, by covering over the territory of potential implementations without accompanying solutions - hinders progress rather then supporting it.

But it definitely would be too easy to put the blame on these secondary effects only. Of course some well justified disappointment originates from the fact that the many and enormous efforts to establish the technique were hardly repaid by economical solutions meeting the needs of superior performance and robustness. Thus, besides multiple laboratory successes, many expectations could not be met and many investments could not be returned. But it would be completely unjustified to base an overall assessment of active sound and vibration control mainly on these shortcomings. There are successes and there is (and will be) continuous further progress to add the technique to the toolbox of acoustic engineers.

This text will try to substantiate this and thus help in making it comprehensible. It is therefore intended to be introductory as well as describing some state of the art. It is obvious that this only can be done in an exemplary way here – without claiming any completeness. However, taking into account this incompleteness, emphasis is given to the typical, representing basic insight and thus allowing to be applied to other problems in the broad area of active sound and vibration control. Those however who want more completeness are referred to the introductory textbooks ([6]-[9]) as well as to the extensive literature.

GENERAL PROBLEM STRUCTURE

When analysing an active system as well as when designing one it can be extremely helpful to have a clear understanding of the unique structure and the elements of active sound and vibration control systems. It may be illustrative to start with a very general look to any passive sound or vibration control problem, where the common goal is to control acoustic or vibrational field quantities or mechanisms of a mechanical system MS such that some target quantities **z** are influenced in a desired way. This is illustrated in Fig.1. Many tools and techniques are available to fulfil this task, e.g. damping or absorbing materials, changing or adding masses and stiffnesses, disturbing propagation paths or protecting specific areas.

Fig.1 Scheme to illustrate the control of target quantities z of a mechanical system MS

Whenever difficulties in achieving the targets lead to considerations whether to apply an active approach, the first step is to examine thoroughly the controllability of the target quantities by appropriately applied electromechanical transducers. These generate a so-called secondary field which is aimed to interfere with the existing primary field in a prescribed way. This can be a most ambitious task because several issues have to be taken into account. To illustrate this, some typical questions to be answered at this stage shall be mentioned here:

- How many actuators should be placed where, in order to achieve maximum effect with minimal excitation effort?
- How many sensors should be placed where, in order to supply sufficient information early enough for the actuators to be processed with the desired effect?
- Is the resulting system – with respect to the target quantities – observable?
- Is the system – with respect to the target and excitation quantities – controllable?

- Is the necessary controlling unit causal and realisable?

If these considerations can be concluded successfully, the mechanical system MS has been extended to an electromechanical system EMS with new inputs **y** and additional outputs **z'** which may be combined with the former outputs to a new ensemble **z** of output signals. The remaining problem is to evaluate the necessary input signals **y** from the available output signals **z** by properly designing a signal processing unit SPU, as shown in Fig. 2.

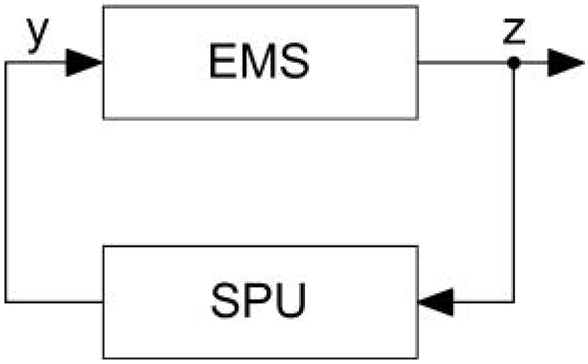

Fig. 2 Scheme to illustrate the control of target quantities **z** of an electromechanical system EMS by suitably driving the control inputs of EMS.

Although the variety and complexity of the questions involved prevent this paragraph from giving any details, it should be emphasised here that the sequence of the two design steps was chosen intentionally. Any signal processing is confined to what the physical relation between the available observations **z** and the necessary control signals **y** permit. Appropriate design of the electromechanical system EMS, and i.e. appropriate selection and arrangement of suitable actuators, is crucial to the successful design of the signal processing unit SPU and thus of the overall approach. For this reason, the first task normally is to do the described

- electromechanical design of the system EMS including sensors for the signals to be supplied and actuators for the signals to be applied.

The second task then will be to do the necessary

- design of the controller units within the signal processing unit SPU by

defining and realising an algorithm for which the unit SPU - together with the hard- and software around - will be able to cope with the requirements of the system EMS. Having given a rough context of the steps involved, this text will concentrate on pointing out various physical concepts of mechanical interference which may be very helpful in designing and analysing active control measures.

ACTIVE NOISE CONTROL AND ACTIVE SOUND DESIGN

Up to now, no assumptions have been made with respect to the target signals and the values or signal characteristics they should be targeted at. From the very beginning of active sound and vibration control, one obvious and highly motivating aim was to use the system for noise and vibration reduction. However, active systems are inherently conditioned to do the much more general task of generating and relating arbitrary sounds to technical processes. This is because sound design differs from sound reduction by adding sound instead of just reducing it. Active techniques, which - by its very definition - add sounds even in a purely reducing mode, thus ideally meet the conditions of realising arbitrarily designed sounds.

Even if this approach would be seen to be unacceptable as built-in solution for large product quantities, it opens the fascinating possibility of implementing arbitrary sounds in situ, thus enabling a fully functional test and evaluation environment to define and experience the most appropriate sound.

The technical concept of these approaches differs only in the target specification, which, for sound reduction, should be defined as zero while, for sound design, a desired signal should be defined which meets the characteristics of the sound to be aimed.

PHYSICAL INTERACTION IN ACTIVELY CONTROLLED SYSTEMS – A SYSTEMATIC APPROACH

The task of designing an electromechanical system which is able to control some target signals to desired values, shapes or parameters has to start by looking for the most effective way of interfering with those signals finally determining the target signals. As arbitrary changes of a wave-form require the potential of fully compensating it, the effectiveness of additional sources is limited by the capability of destructive interference: If a new generated sound shall dominate the sound characteristics, the previous, original sound has to be destroyed.

It is thus natural to follow the way and the propagation of the sound to be interfered on its complete propagating path, starting from the sound generating mechanism up to the point or points where useful target signals may be observed. Structuring this path in a systematic way comes up with the following scheme which has to be passed by any waveform. It is clear that in practice, depending on the specific configuration, some elements of this scheme may be skipped. However, this should not question the overall value in finding a bottleneck where the wavefield under consideration can be influenced most easily.

If divided into groups according to the dominating mechanism, the zones of interfering interaction can be characterised by the following principles:

- active control of sound generation
- active control of sound introduction
- active control of sound propagation in structures
- active control of sound radiation
- active control of sound propagation in fluids (and air)
- active control of sound immission

These principles shall be treated shortly within the following.

<u>Active Control of Sound Generation</u>

The most effective way of reducing sound perhaps is to avoid its generation. This can be particularly useful if a suitable interaction with the process of sound generation prevents the mechanism from being started at all. An early and impressive example was the control of sound wave reflections at the open end of a Rijke tube ([10]). Comparable results were obtained for the suppression of instabilities of Helmholtz resonators excited by air flow.

In both examples, active control succeeded in stabilising a self excited vibrating mechanism. Thus, minimal energy was needed to interact with the resonant vibration because this interaction had to cope with the low amplitudes of the stabilised vibration only.

In spite of the fascination of this concept and in spite of many attempts to solve problems like combustion or flutter instabilities, the many difficulties to be solved in practical applications prevented the approach from being converted to standard industrial implementations.

<u>Active Control of Sound Introduction</u>

In contrast to noise generating mechanisms, where the source itself is prevented from acting, active control of noise introduction has to cope with the effects of this source on a receiving structure or medium. Effective global cancellation of the excited wave field can be achieved most easily by applying a negative copy of the source, a so-called anti source, as close as possible to the primary source.

This approach is promising in practice if the dimensions and the directivity of the primary source allow it to be treated as a monopole source. Typical examples are the outlet of pipes and ducts or the excitation at points connecting some vibrational sources to a receiving structure. An early demonstration installation of such a system at the exhaust of an industrial gas turbine is described in ([11]). Similar concepts were further developed and are in limited use today in cases where the overall process environment makes classic solutions rather difficult.

An example of compensating the effect of point sources by applying active isolation mounts will be given below.

Active Control of Sound Propagation in Structures

First theoretical studies as well as early experimental realisations were based on a modal approach trying to excite and thus control specific modes of interest. This may be adequate if the modes to be influenced are known in advance and if any generation or amplification of uncontrolled modes (spillover) can be accepted. However, in many cases a wave oriented approach can be much more appropriate and effective. This is particularly true if the deactivation of transfer paths as well as the suppression of resonances is to be achieved.

In practice, this approach is limited to the one-dimensional case and to simple wave types like bending and longitudinal waves. But in spite of some successes in controlling the propagation of such wavetypes under laboratory conditions, a useful application within real technical installations still has to be implemented, apart from special constructions for large space structures.

However, to give an illustration of the capabilities of the approach, some typical laboratory results are reviewed below.

Active Control of Sound Radiation

In a naive approach, active suppression of sound radiation from a radiating structure would be aimed to suppress the vibrations of the structure. Although being sufficient with respect to the target, the criterion would exceed necessity by demanding more then radiation suppression requires. This is because a structure may vibrate in radiating as well as in non-radiating modes.

With respect to sound radiation it would thus be much more efficient to concentrate only on radiating vibrations and to find ways of selectively detecting and generating such radiating vibrations. This approach was established within the last 10 to 15 years as Active Structural Acoustic Control together with thorough investigations and experimental laboratory verifications of the concepts behind. However, practical industrial applications seem to suffer from physical and financial constraints which were not able to be overcome yet.

Active Control of Sound Propagation in Fluids (and Air)

The active control of one-dimensional sound waves in ducts was - besides the silencing of small zones - one of the earliest example of active control efforts in theory and in practice. This was due to the fact, that below the cut-off frequency of a duct only plane waves are able to propagate. For this reason, low frequency duct systems were characterised by only a few, often only by a single degree of freedom.

The classical paper of Swinbanks started an era of practical investigations and implementations differing mostly in the physical principle involved (absorbing of reflecting incoming waves, e.g.) and in the arrangement of sensors and actuators. Laboratory realisations were soon accomplished by attempts of installing such systems in practice and these were often intended to be more then just demonstrators.

Although the principal realisability as well as the stability could be proven as reliable in the past, practical applications mostly are confined to special situations by now, where existing constraints or extreme demands are able to overcome the drawback of higher costs compared with classical passive solutions.

However, this partial success story only holds for one-dimensional applications. In the two- or three-dimensional case, early conceptual dreams could not be fulfilled by reality because the task of coping with spatial sound fields and sound propagation is much too ambitious for justifiable efforts. This holds for all applications including such spectacular examples like the sound isolation of open windows or the improvement of barriers. However, the latter example shows that the success of active approaches may highly depend on the degree of sophistication of the underlying concept.

This can be impressively demonstrated for noise barriers, where the angle of diffraction depends on the impedance at the diffracting edge. In this case, the possibilities of realising an optimally shielding impedance can be substantially improved by allowing for active realisations of this impedance.

Active Control of Sound Propagation in Fluids (and Air)

If all interactions between source and receiver fail or tend to be too complicated, the last remaining approach is to concentrate control measures on some zone around one or several receiving points. Together with active control of one-dimensional waves, this approach has attracted high interest in the early days of active control of sound fields.

This again was due to the relatively simple nature of the problem for small volumes like ear protectors including headsets or small passenger compartments. It is interesting that these examples still today represent some of the most promising or even well established applications of active noise and vibration control.

Active headsets and active ear protectors may be purchased by everybody and active solutions to improve the interior sound of cars are more or less ready to be applied, especially for periodic low frequency engine noise. The approach could even be extended to propeller driven aircrafts to essentially reduce the high level tones at the blade passenger frequency.

However, until today it was not possible to overcome all the doubts about reliability, long term robustness and economy.

After these short remarks on possible physical concepts and the state of the art related to these concepts, some technical examples will be described and discussed.

ACTIVE CONTROL OF SOUND INTRODUCTION (ACTIVE VIBRATION ISOLATION)

The first application example will deal with the reduction of sound generation by active means and it will demonstrate the potential of the approach in the case of structure borne sound excitation where a suitably driven force actively supports a passive vibration isolation system.

There is a contradiction between the need to fix an aggregate within its environment while isolating this environment from the operational vibrations of this aggregate. For example, an engine driving a vehicle has to be carried with that vehicle. In addition, any oscillations with high amplitudes have to be limited to guarantee a stable coupling and continuous excitation of the driving path without any damage.

However, as to higher frequencies, it would be desirable not to have any coupling between aggregate and receiving structure, thus avoiding any excitation of the environment by allowing free vibrations of the aggregate. So, rigid connections for static and low frequency requirements are not consistent with the request for weak connections at mid and high frequencies.

Such an ideally selective frequency dependence cannot be realised by passive means. Using an active system to improve the isolation of a receiving structure from any vibrating device enables the assignment of nearly arbitrary stiffnesses to given frequency ranges. This was proven by various implementations in laboratory as well as under real life conditions ([12],[13],[14]).

It can be shown that any active connection between two structures fulfilling the above requirements has to contain an element with spring characteristics. It then makes sense to build the active system around a passive spring, completing it by a suitable actuating device. Among different possibilities to incorporate actuators into the connecting elements of an isolating mount, parallel arrangements of springs and actuators were preferred in the past, mainly because they kept away any static forces from the actuator.

One impressive example was realised by four active mounts connecting a small electric motor to a 2 m by 3 m stiffened plate structure ([12]). The mounts themselves were realised by three cylindrical rubber struts in parallel to an electrodynamic shaker. Constructive details were chosen to come up with a substantially lower stiffness in horizontal than in vertical direction.

For constant rotational speeds of the motor, the forces generated at the basic rotational frequency and its harmonics (characterised by ˅) could be compensated down to the surrounding noise level. As an example, Fig. 3 gives the result for one of the transmitted forces with (b) and without (a) active control. All harmonics could be attenuated to the noise level, which is uncorrelated with the periodic machine vibrations and thus prevented from being compensated. Note that the difference in broadband noise level is only caused by different resolutions of the measurement equipment. Depending on the signal to noise ratio, drastic noise reductions of up to some 80 dB could be achieved.

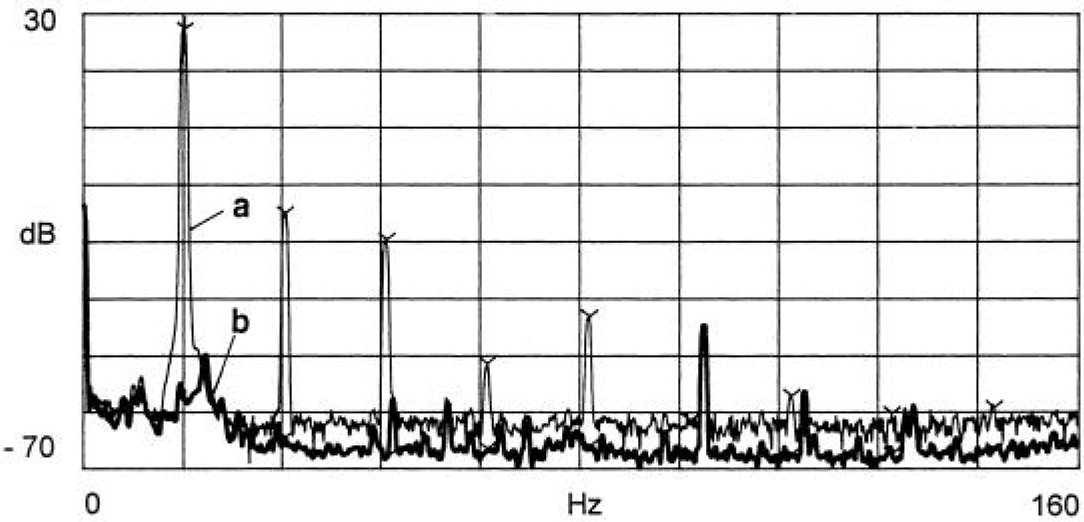

Fig. 3 Spectrum of the force transmitted to the foundation with (b) and without (a) active vibration isolation.

Further experiments showed that neither the minimisation of transmitted forces nor the minimisation of velocities in the contact points resulted in the minimal mean square velocity of the plate structure. The resulting question of what would be the best error criterium for vibration isolating arrangements was investigated recently and came up with the proposal to minimise the weighted sum of squared forces and velocities below the mounts ([15]). For the application reported here, however, these differences were minimal and not explored in more detail.

Note that active compensation of the forces exciting a structure may result in a global reduction of the vibrational field. This was verified for the stiffened plate for various distributions of velocity measurement points.

To demonstrate the operational robustness of the active system, it was installed to isolate the vibrations of a converter from the foundation connecting it to a ship structure. It could be shown that under normal operating conditions of the ship it was possible to attenuate the mean square vibration level on the target structure by up to 20 dB.

The high level reductions reported above require the predictability of sinusoidal disturbances. In the case of broadband excitations, a priori information which allows for predictive estimations of the disturbance is not available. Active systems then have to rely on feedback control. Although this limits the frequency range, some solutions for low frequencies could be implemented successfully to protect highly sensitive desks from environmental vibrations ([16]) or buildings from earthquakes ([17],[18]).

There are many further aspects like design rules, benefits, disadvantages, difficulties and application examples for active isolators. Comprehensive reviews together with extensive references may be found in ([7],[9]).

ACTIVE CONTROL OF SOUND PROPAGATION

Again, the illustration of active control of wave propagation within this text shall be based on a structural example. 15 to 20 years ago, there was great confidence that the early successes with active control of plane sound waves in ducts could be transferred to structural waves ([19],[20]). At least for simple wavetypes under favourable conditions it was expected that reflection and/or absorption of incoming waves could be achieved by suitable actuators.

The principal realisability of this expectation could be shown in the late 1980's for bending waves in beams in various laboratory experiments ([20],[21]). Fig. 4 gives the result obtained for a broadband active reflecting device attached to a beam.

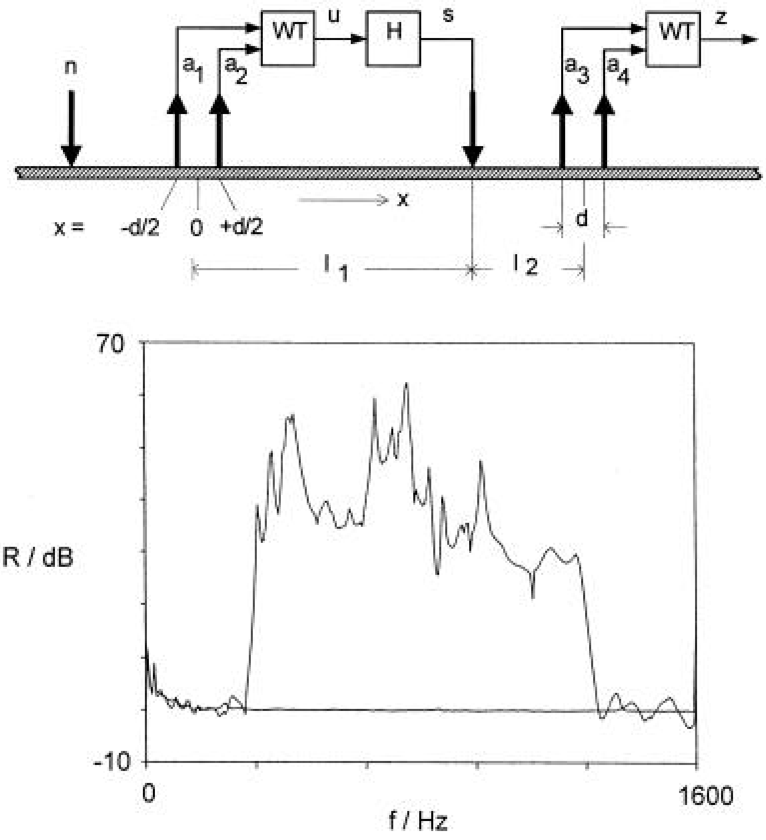

Fig. 4 Principal arrangement and transmission loss of an active reflector

The bending waves generated by a broadband noise source n were directionally registered by a two point measurement including a digital unit WT to separate incoming waves from waves with opposite direction. The signal processing unit then determines the optimal signal s to cause the force transducer to reflect this incoming bending wave field. Optimality was obtained by minimising the error signal z which again was derived by directional evaluation of a two point measurement.

As shown in Fig. 4, the transmission loss characterising the active reflector has a mean value of some 35 dB which means that approximately 99.97 % of the incoming power was rejected by the active device. In the experiment, all dimensions were chosen such that nearfields could be neglected. If this assumption doesn't hold, more then two sensors have to be applied ([22]).

Although flexural waves are the most important wavetypes in acoustics because of their radiation efficiency, other wave types like longitudinal and torsional waves may be equally important by supplying these radiators with energy. Assuming only knowledge of the wavenumber and its frequency dependence, the basic considerations of ([22]) are principally independent of the considered wavetype. For this reason, the approach was considered to be applicable to other wave types too. Related theoretical investigations may be found in ([23],[24]), results for a practical application in ([25],[26]). There, a strut was equipped with integrated actuators to reduce the interior noise in helicopters by controlling longitudinal and bending waves propagating along the strut.

Further extensions dealt with effects of actuator and sensor types and locations and with various end conditions for finite beams ([9]). Also, direct control of power flows instead of wave amplitudes was investigated carefully ([23]). However, as shown in ([27]) already, power flow must be treated with caution. This is mainly because any application of secondary sources acts back on the overall energy distribution. Applying an additional force to a structure may cause additional power to be supplied to the structure by previously existing forces. This may lead to the astonishing fact that actively absorbed energy can be exceeded by additionally introduced primary power, thus achieving the opposite of the original intention.

All these approaches may be applied to two - or even three-dimensional media. However, without improved proposals for wave-specific and handsome actuator arrangements, such generalisations tend to be limited to academic treatments.

In summary, it may be stated that in spite of having understood the basic mechanisms of potential realisations, practical implementations were not able yet to fulfil the expectations of laboratory results.

ACTIVE CONTROL OF SOUND RADIATION

Active Vibration Control

Active control of vibrational fields in predefined target areas may have different aims: the limitation of vibrational characteristics itself or the reduction of that part of a sound field which is radiated by the controlled structure. Although differing in the formulation of error criteria, these tasks have great formal similarity.

Many attempts were made in the past to implement both of these strategies. Reducing the amplitudes of given vibration patterns was one of the early tasks in active control research. Starting from the vision of controlling the vibrational behaviour of extremely light weight structures, a thorough understanding of the theoretical possibilities and limits was developed (see literature review in ([27]).

Unlike measures based on propagating waves, the minimisation of structural vibrations in target areas was generally approached by modal concepts. In addition, if no predictive signals can be derived from sensors detecting incoming waves or from narrowband assumptions of the primary fields, one has to rely on feedback strategies. Thus, this working area was predominantly established as modal feedback control.
Today, as a research area, this approach has gained great maturity and some practical experience. Besides many fundamental and experimental work, special emphasis was given to the problem of observability and controllability which is immediately linked to the important practical question of where to place how many sensors and actuators.

Actuator arrangements vary from simple forces acting against inertial masses (active vibration absorbers) to sophisticated distributed actuators which may be integrated within the structure (smart structures). To emphasise the benefits of these devices it may be helpful to interpret them in terms of virtual structural modifications like attached masses or dampers.

Basic texts on all relevant aspects as well as state of the art reviews may be found in ([7],[9],[27]).

Active Structural Acoustic Control

Active Structural Acoustic Control (ASAC) describes any attempt of using control inputs directly applied to a structure in order to reduce the sound radiation from that structure. Of course this changes the vibration distribution too, but only to serve another purpose. It thus may use the same strategies as active vibration control but is aimed to error signals directly correlated to radiated noise instead of vibration amplitudes.

Again, modal approaches are very appropriate to suppress acoustic radiation. By selectively sensing radiating modes and by concentrating control efforts to these modes, an efficient reduction of the radiated sound power may be achieved.

Of course, the radiation of sound may be reduced equally by reducing the overall vibration level. However, taking into account that different modes are characterised by different wavelengths and thus couple with different strength to the surrounding fluid, it is obvious that the modes with the highest radiation efficiency should be tackled particularly.

Instead of a modal transform, any spatial vibration distribution may be equally decomposed into its wavenumber components by spatial Fourier transform. Essentially, control of radiated sound means control of the part of the wavenumber spectrum where the structural wavelength is larger then the wavelength in the surrounding medium. Allowing for unconstrained vibration levels for non radiating wavenumbers, ASAC essentially is based on minimising radiating parts of the wavenumber spectrum or even on shifting vibrational energy from radiating to non radiating parts. Decreasing sound thus may be accompanied by an increase of structural vibration levels.

These basic facts were explored intensively in the 80's ([28],[29]) and meanwhile summarised in various papers and textbooks ([7],[9]). They are of specific benefit for the design and arrangement of sensors and actuators which determine whether sound radiation is observable and controllable. There are promising prospects that the integration of actuators into smart structures may open up many possibilities of elegant and effective sound radiation control. By adding additional passive foam layers, "active foams" ([30]) are obtained which might improve the bandwidth of solely active or passive measures.

To control sound radiation, the selection of actuators is not confined to any of the mechanisms described above. Having used the concepts described there to design an optimal arrangement of actuators, suitable sensors (and algorithms) will automatically force the system to minimise the radiated sound power if this is sufficiently correlated to the error signals. It is irrelevant then whether the structure is controlled at its excitation points, along its wave propagation paths or at radiating substructures. These differences were only taken into account to find an optimal arrangement with minimal overall efforts.

An example for controlling sound radiation by compensating the excitation of structures was realised with active mounts as mentioned above. The system was installed to isolate the vibrations of a transformer from the supporting structure on a vessel. By driving the actuators appropriately, an essential reduction could be observed in the surrounding water.

Finally, the effectivity of controlling sound fields by applying forces at suitable structural locations shall now be demonstrated by results for a high speed railway vehicle, where active means were considered to reduce low frequency noise components. Excited by multiples of the rolling wheel's rotational frequency (wheel harmonics) and reinforced by coinciding resonances of track and secondary bogie springs, High speed trains may suffer from raised vibrations and noise around 90 Hz.

Although direct sound cancellation by a set of loudspeakers was proven to successfully compensate the tonal noise in the compartments, a thorough analysis of the transfer path from vibration generation to sound radiation showed that applying forces to the vehicle body close to the secondary springs would be the most efficient way of reducing the interior noise.

After a successful case study ([31]), the concept was tested for a vehicle running on the rolling rig of the German Railway Corporation. Fig. 5 shows the result for four piezoceramic actuators (two on each side) being arranged in parallel to the secondary springs of a bogie at a speed of 200 km/h. The actuators were driven by a reference signal derived from the wheel's rotation and containing the frequencies to be considered.

It can be seen from the spatial distribution of Fig. 5 that the sound field generated by the bogie on the left side of the figure was essentially reduced in the whole volume of the compartment.

ACTIVE CONTROL OF SOUND IMMISSION AND ACTIVE SOUND DESIGN

For suitable, “fitting” applications, active control of noise and vibration was shown to be an efficient tool to reduce undesired field components to an acceptable level. Examples for successful test applications may be found in vehicles including ships and aircraft, the latter even with a series implementation. However, although having been a subject of considerable research and development efforts, active noise control in cars and trains has not succeeded yet in being implemented on a large scale. This is only partially due to technical shortcomings like insufficient actuator capabilities or long-term robustness and reliability. More relevant seems to be that the related cost could not be reduced to an acceptable level.

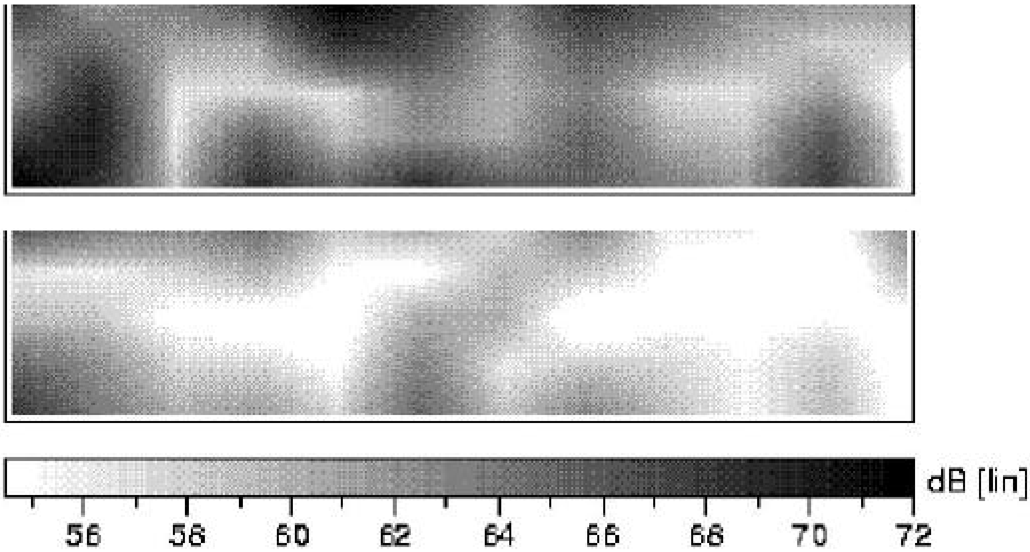

Fig. 5 Spatial distribution of the sound pressure level in the compartment of an ICE high speed coach with (below) and without (above) active control

Nevertheless, the related interest definitely will be intensified by the attractive possibility of rather changing the sound characteristics instead of just reducing or cancelling them.

Apart from some special cases, active sound control means generating additional sound interfering with the original sound pattern. This concept includes the possibility of deliberate sound alterations by allowing non interfering superposition of new sound components. By combining these newly generated field elements with cancelling components, almost arbitrary modifications of the sound field can be achieved. This approach extents the possibilities of traditional Active Noise Control (ANC) techniques and thus may be described appropriately by Active Sound Design (ASD).

Psychoacoustic quantities may be used to characterise subjective assessments of arbitrary sounds which - of course – may include sounds generated by active control measures too. This immediately creates the reverse idea of using active control to approximate sounds with prescribed, aurally adequate perception characteristics. In this case, the common goal of minimising some mean square error function would be replaced by an error function using the relevant sound parameters and characteristics.

However, non-linear and thus irreversible computation rules prevent a unique assignment of temporal sound characteristics to perception related sound descriptors. For repetitive components of engine and rolling noise a more pragmatic approach may be used instead. By prescribing well defined amplitude values of a given set of harmonics (engine orders or wheel harmonics), subsequent approximation of these order-related amplitudes will vary the resulting noise together with the associated subjective parameters ([32]).

Practical measures of active control shall be represented here by an example demonstrating the state-of-the-art potential for cars. This example will prove the feasibility of Active Sound Design in a small size 4 cylinder car (new beetle). To demonstrate the capabilities of such a system, it was used to realise the sounds of different engines from other cars ([32],[33]). The targeted sounds were obtained from measurements in these cars and then stored for use in the signal processing device. Based on the error signals from six error microphones and appropriate information on revolution speed and engine load, six secondary loudspeakers were fed such that the resulting sound field was an optimal approximation of the prescribed sounds.

Fig. 6 gives some typical examples by showing the time-dependent frequency spectrum at the driver's ear if the car is consecutively run up (to 6000 rpm) and down with four different sound implementations. The first run-up (lowest part of Fig. 6) shows the typical spectrum of a 12 cylinder sports car, then followed by an 8 cylinder sports car, a six cylinder sports car and, finally, by the original 4 cylinder engine sound (system switched off).

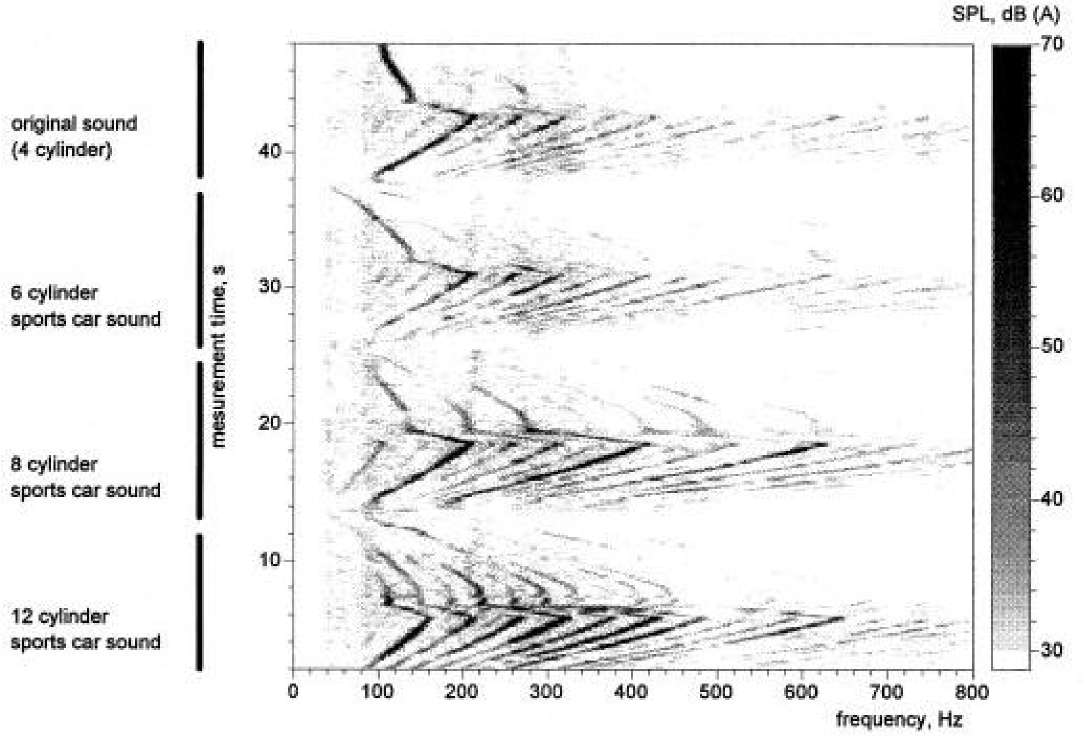

Fig. 6 A weighted sound pressure spectra at the driver's ear for four consecutive run-ups with different engine sounds.

The effect was - objectively (by measurement) and subjectively (by ear) – striking. The various sounds were immediately able to generate the acoustic illusion of driving a different car. Thus, the system might be more than just a toy: it could equally be used - and should turn out to be extremely useful - as an experimental tool to determine and test optimal sounds for given cars.

Finally, it should be mentioned that Active Sound Design recently was used to reduce and/or modify the noise of air intake systems of car engines. Experimental results from test set-ups ([34]) were quite promising and seem to have found serious interest of experts. It thus can be stated that - in spite of some uncertainty about the acceptance of sounds forged by electroacoustic means - the prospect of deliberately designed sound realisations is highly attractive.

The results of previous work finally led to the implementation of a system to demonstrate active interior sound design in a New Beetle. Besides reducing the engine noise in the vehicle interior, this system allows the creation of arbitrary interior sounds on the basis of rpm-dependent orders of the engine. At the push of a button, it is possible to modify the typical sound of a four cylinder engine to give the acoustical impression of a six cylinder engine in the vehicle interior. This sound design could just as well reproduce the sporty tone of a roadster or the acoustic comfort of a limousine.

Fig. 7 shows typical order spectra of the sound level at the drivers position for different sound scenarios: the original sound spectrum in the above diagram, the spectrum for the sound reduction mode below and the spectrum for a typical 6-cylinder engine at the bottom. Although the black and white representation makes a clear interpretation of the data rather difficult, it can be seen that the second order of the initial state for values of 5000 to 6000 rpm could be compensated in the reduction mode. Also, in the 6 cylinder mode, it can be seen that the 6^{th} and 9^{th} order are much more dominant then in the other cases which indicates that the virtual engine definitely has changed its characteristics.

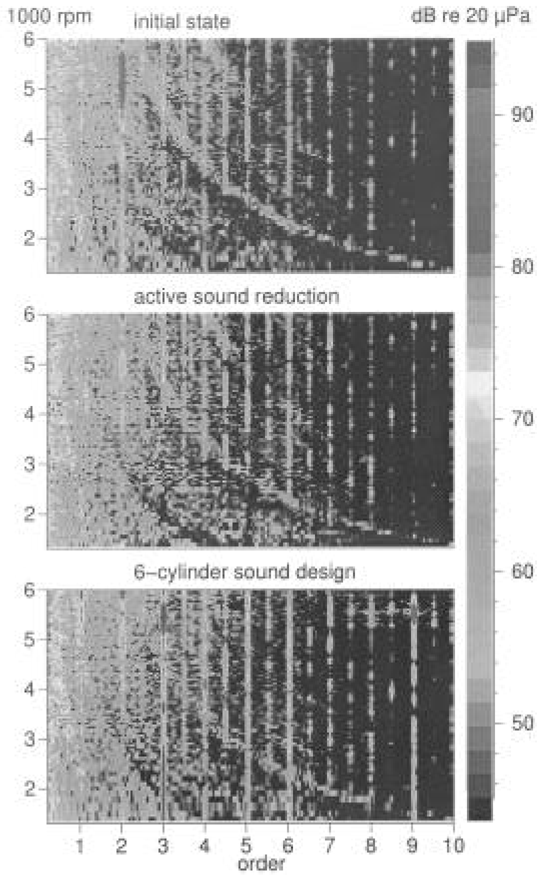

Fig. 7 Order spectra of the sound level at the drivers position during a slow run-up on the road

This feature qualifies the system to strongly support engineering work by enabling in-situ tests of sounds under fully operational conditions. It thus can help to demonstrate the goals as well as the effects of acoustic measures for both approaches: active and passive ones.

CONCLUSIONS

After the consecutive periods of enthusiasm and disillusionment in active noise and vibration control, a thorough and conscious recollection of basic physical concepts may help in finding out what applications have the potential to successfully overcome the reluctance of practitioners. Together with further progress, especially with respect to advanced actuators, this finally should lead to the technique's well-deserved breakthrough in the end. In any case, active sound control implementations open the possibility of experiencing sounds in-situ, i.e. outside the laboratory under real life and real time conditions. This can be foreseen to extent auditory tests and assessments to realistic environments – a natural and consequent improvement of classical auditory tests.

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