SIXTH FRAMEWORK PROGRAMME
PRIORITY [4]
AERONAUTICS AND SPACE

SPECIFIC TARGETED RESEARCH PROJECT

Deliverable D3.1:

State of the art review on noise and vibrations

Project acronym: **SEAT**
Project full title: **S**mart t**E**chnologies for stress free **A**ir **T**ravel
Proposal/Contract no.: **AST5-CT-2006-030958**

Due date of deliverable: 20/12/2006
Actual submission date: 20/12/2006

Start date of project: 2006-09-01 Duration: 36 months

Organisation name of lead contractor for this deliverable: Acusttel

Project co-funded by the European Commission within the Sixth Framework Programme (2006-2009)		
Dissemination Level		
PU	Public	X
PP	Restricted to other programme participants (including the Commission Services)	
RE	Restricted to a group specified by the consortium (including the Commission Services)	
CO	Confidential, only for members of the consortium (including the Commission Services)	

Revision: A

Revision History

Date	Ver.	Author	Comments
20/12/2006	A	Acusttel	1st version sent to coordinator

List of Abbreviations

ABC	Acoustic Boundary Control
ANC	Active Noise Control
ANN	Artificial Neural Network
ANP	Active Noise Profiling
ASAC	Active structural Acoustic Control
ATVA	Adaptive Tunable Vibration Absorber
BEM	Boundary Element Method
CLMS	Complex LMS (algorithm)
DSP	Digital Signal Processor
FEM	Finite Element Method
FIR	Finite Impulse Response(filter)
FIRLMS	FIR LMS (algorithm)
FR	Forced Random (algorithm)
GS	Genetic Search (algorithm)
MEMO	Multiple-Error, Multiple-Output (algorithm)
MFA	Mean Field Annealing (algorithm)
PZT	Piezoceramic Zirconate Titanate
RAINBOW	Reduced And INternally Biased dome –shaped Oxide Wafer (transducer)
SA	Simulated Annealing (algorithm)
SEA	Statistical Energy Analysis
SI	Structural Intensity
SMD	Spring-Mass-Damper (system)
TBL	Turbulent Boundary Layer
THUNDER	THin-layer composite UNimorph ferroelectric Driver and sEnsoR (transducer)
TVA	Tuned Vibration Absorber

Table of Contents

Introduction

Passenger comfort is clearly one of the most influencing factors in user’s acceptance of transportation systems. In this context, modern aircrafts are very stable and more quite than their predecessors. However, noise and vibrations still affect the overall condition such as fatigue, that is one of the major causes of stress. Moreover, the vibrations we do not hear or feel, between 2 and 10 Hz generally, also affect our perception of comfort, since they tend to resonate with the natural frequencies of the different parts of our body. Thus, noise and vibrations remain major irritating factors for the passengers. That is why there is a real need for a novel approach to passenger comfort, accentuated by the fact that travel has become a global activity that is not confined to any geographic area, social or ethnic group.

The project focuses on upstream questions of an integrated system that will create a healthier and more comfortable cabin environment through, among other things, reduction of noise and vibrations. Concretely, this part of the project is focused on the development of a system for active and passive noise and vibration dampening incorporating smart technologies and textiles in particular. This system will have to be able to reduce noise and vibrations overall, as well as for each passenger, and to be included in the global “SEAT system prototype”, that will be an important stage of a development of e-cabin.

The project promotes a radically new concept where passenger comfort is taken to a new level. The SEAT system aims to develop smart responsive seats and interior environment, that will allow to analyse and adjust each annoying factor, and in special noise and vibrations. The entire approach is to create an environment that responds to the individual requirements and desires and is not centrally controlled or manually adjusted. On one hand, it is the adequate philosophy for the project in general, since the level of tolerance of the each individual is very different depending on age, temperamental characteristics, health status etc. On the other hand, an active noise control system, for the nature of this technique, can not be effective for the whole cabin. Moreover, to reduce vibrations, and noise, at each passenger level, first passive reduction measures could be taken for the whole aircraft, but as a last resort, the structure vibrations are mainly communicated to each passenger through his seat. Hence individualised flexible and intelligent active noise control is an appropriate strategy, with two main facets: tackling universally harmful vibrations and noises such as low frequency vibrations and engine noise and local attenuation of noise and vibrations through smart technologies. Finally, the system will be based on advanced technologies and systems developed by the partners as breakthrough research developments, that will make possible the system to work as a stand alone device.

In this context, the objective of this state of the art review is to resume the current technologies available in order to passively and actively control noise and vibrations in aircrafts. Concretely, the following aspects will be analysed:

- passive and active techniques and designs
- computational and test models
- use and integration of specific textiles and smart materials
- use of carpets and interaction between passenger feet and floor
- possibilities of integration
- vibroacoustic comfort

In the following sections, all these aspects will be detailed, and this state of this art analysis aims to establish the bases necessary to define the correct strategy to follow in the next steps of the project, in order to achieve the noise and vibrations reductions objectives.

1. Passive and active techniques and designs

Noise and vibrations mainly come from:

- propulsion system (including fans, jets, etc.) [78][194],
- airframe (including high lift devices, speed brakes, landing gear, etc.),
- other passengers (crying babies, loud music, activities such as frantic typing, etc.),
- other displacements of free surfaces or systems malfunctions.

Noise is transmitted into the cabin through several vibroacoustic paths. Engine vibrations are transmitted through the engine mounts into the wing structure, which then excites the whole aircraft body. Turbulence behind the engines excites the rear wing section and the vibrations propagate into the cabin structure, which radiates noise into the compartment [196][234]. Another important path is the vibroacoustic transmission through the fuselage in the plane. The resulting vibrations in the aircraft body radiate noise into the passenger cabin. The importance of the different transmission paths varies with frequency and flight conditions. At the engine fundamental frequencies, the sound field is usually excited throughout the whole cabin, while the harmonics tend to be excited primarily in the plane [72].

Such annoyances cannot be avoided, but some of them are constantly present and their effect can be minimised through appropriate passive design features, using appropriate smart textile materials for upholstery, wall covers or carpets. Other sources develop due to temporary faults or normal wear and must be tackled through active dampening and noise reduction systems.

Up to date, vibrations have been reduced through special wall panel design or through actuators on the aircraft structure. However, there are not reported results on passenger focused dampening and comfort assessment studies with this respect. Natural passive dampeners such as carpets do not seem to have been utilised.

Regarding to noise control, the aircraft fuselage is constructed as a light stiff wall with only a marginal low frequency transmission loss. Due to the low frequency content of the noise, typically below 400 Hz, and the restrictions that apply for acoustical and mechanical treatment in an aircraft, passive noise reduction methods are of limited practical use. One passive method that can significantly increase the fuselage transmission loss is the use of tuned dampers, or tuned vibration absorbers (TVAs) [78][196][234]. A tuned damper is a mechanical resonance system consisting of a mass supported by a spring with a fairly high mechanical loss factor. At the resonance frequency for the damper, the overall loss factor for the structure can be significantly increased. To cover all harmonics for all flight conditions, a fairly large number of TVAs must be used and they must be carefully distributed on the structure. One major disadvantage of the tuned TVAs is the added weight, which can be the equivalent of one passenger or more for a complete set of TVAs. This is significant for an aircraft designed for 20–30 passengers. An active noise control system (ANC) offers much more potential to the noise control engineer in the control of propulsion noise [195]. With proper system design, the overall attenuation of engine noise is generally higher than what can be achieved with passive methods. Since the controller is synchronised with the engines, the attenuation is maintained throughout the flight cycle, including cruise, climb and descent. If the controller is synchronised to both engines, the acoustic beating that appears as the engines become unsynchronised is also controlled.

For a complete flight cycle, some studies have shown that a properly designed twin-reference controller is better than or equal to a single reference controller [122]. Finally, even with many (more than 30) loudspeakers including cabinets, the active noise control system adds less weight to the aircraft than a normal set of TVAs [78].

In relation to noise control, active noise control, which uses loudspeakers placed in the cabin as control sources, has been demonstrated to be an effective method. ANC systems typically can be implemented to control sound generated by any combination of disturbance sources.

Also Active structural acoustic control (ASAC), which uses direct actuation on structural members to reduce their radiated sound, has also received much attention. Smart trim panels have been used as a secondary source for active noise control in aircrafts. The smart trim panel is a rectangular segment of aircraft trim panel which is suspended by a flexible support. This support converts the stiff trim panel into flexibly mounted pistons which can be driven by light-weight and low-profile force actuators.

Acoustic Boundary Control (ABC) is another active control strategy for quieter aircraft. ABC involves the distribution of light-weight, low-profile acoustic sources along the domain boundaries (i.e., the inside walls of an aircraft cabin). It has been shown that this approach provides advantages of both traditional loudspeaker-based systems and structural vibration control systems.

Although good performance is often obtained, localized sound attenuation control spill over and the inability to obtain high frequency sound reduction are some of the drawbacks associated with both the ANC and ASAC approaches. On the other hand, the use of passive materials such as damping viscoelastic material added mass and porous layer can be considered for reducing sound radiation. However, such a passive technique is not very efficient in the low-frequency region. The complementary nature of passive and active noise control techniques can be used to develop hybrid devices for control over extended frequency range. Indeed, in the last years, there has been an increased interest in the reduction of sound and/or vibrations by use of hybrid active/passive control techniques. The passive device usually carries the primary attenuation function, while the active component is used to enhance the passive system performance or overcome the limitations of the passive system.

1.1. Active noise control

During the past two decades, a large amount of works in the field of active sound and vibration control has been focused on the development of optimised algorithms [141][197][201].

In the case of the global active noise control of the whole enclosed sound field, a multiple-error, multiple-output (MEMO) control strategy is needed [122][142][177][201][205][206]. These MEMO algorithms are based on the Complex LMS algorithm [105][244] and so, have the virtue to converge rapidly.

On the contrary, under stationary conditions, the maximum achievable attenuation depends on the acoustic conditions (acoustic field, sources acoustic properties, spatial configuration of the sensors and actuators) rather than on the control algorithm [201]. During the preparation stage of the process, it is particularly important to define the optimal number and positions of actuators and control sensors, as well as the evaluation method of the system.

1.1.1. Characterisation of the excitation

The most favourable control situation occurs when the control source distribution exactly matches the noise source distribution for all frequencies. This would result in global control inside the whole volume, i.e. perfect control at any position. In most practical cases, the spatial source distribution of the primary noise sources extends over several confining surfaces. It is furthermore not unusual that the shape of the source and its position vary as a function of frequency. To perfectly match such a noise source with a number of discrete point sources is not practically solvable, since the point sources are in practice limited by number and by the available positions. The difficult task is therefore to find the positions for a given number of secondary sources that best match the primary source distribution for the given frequency range.

Considering that the sound field to be improved is a combination of modes, a pressure sensor placed at a pressure node is not useful for observing that particular mode. For the same reason, a source position placed at a pressure node is useless for controlling this mode. In a practical application with non–trivial acoustic mode shapes, this can be difficult to accomplish by mere intuition. In addition to this, the spread in magnitude between eigenfunction values

$$\sigma = \frac{\left|\Psi_n(r_k)\Psi_n(r_{q,k})\right|_{\max}}{\left|\Psi_n(r_l)\Psi_n(r_{q,l})\right|_{\min}}, \forall(k,l)$$
,

where Ψ_n is the eigenfunction related to the mode n shape, r_k and r_l are recording locations and $r_{q,k}$ and $r_{q,l}$ are source locations, has a major influence on the convergence properties of the active control system. This will be further commented in following sections, but for best performance, the spread should be as small as possible [201]. To fulfil this requirement under the practical circumstances that generally apply, some kind of optimisation technique must be used to find the most suitable positions for the control sources and sensors.

Consequently, it will be fundamental for the noise and vibration tasks of this project to define the typical excitations for each airplane of interest. Moreover, although the fundamental noise frequency varies slowly, several running conditions can be distinguished and are of importance (take-off, climb, different cruise condition, descent and landing), as well as the transitions between these steady state situations.

1.1.2. Prediction of the performance

To be able to predict and evaluate the performance, it is necessary to define the design goals, through some kind of system cost function. The quantities to be considered would be the total potential energy in the sound field or the mean power in the driving signals for the control sources, although other interesting cost terms could be the size or the weight of the complete control system and the cost or the fuel consumption for the complete design in which the active control system is incorporated.

The problem of minimising the potential energy in the sound field is that it is generally difficult to assess the true potential energy within a volume. However, an estimate can be made, based on a spatial sampling of the sound field, covering the volume of special interest. If the sound field is sampled in points, for each single frequency (or harmonic), a vector can be formed describing the primary sound field in these positions.

A secondary sound field is applied by using control sources, producing the output signals and the cost function to be minimised is the total potential energy as measured in the evaluation positions, obtained as the sum of the squared evaluation errors. The minimisation problem is further complicated by the circumstance that the signals from the evaluation error sensors – in most practical applications – are used only at the design stage. The reason for this is that the sensors cannot be placed exactly in those positions where the attenuation is needed, which is usually at the ear positions of people in the enclosure. Instead, a new set of control error positions is chosen, which describes the primary sound field sampled in these positions and the control error is found in analogy with the previous theoretical case. Finally, the problem of minimising the total potential energy in the sound field resides in finding, for a given number of control sources and a given number of control error sensors, the optimal configuration of control source positions and control error positions that minimises the sum of the squared errors measured in the evaluation positions.

1.1.2.1. Definition of optimal positions

In most practical applications, finding the optimal positions for the control sources and the control error sensors is a non–trivial task. If classical loudspeakers are used as control sources, the number of practically useful source positions is usually limited, since loudspeakers for low frequency generation with little distortion require a certain volume behind the speaker. If the Active Structural Acoustic Control (ASAC) is used, the source positions may be chosen with greater freedom since force actuators can be made fairly small and can often be hidden on the structure behind the trim.

For the active control of sound, according to Sjösten [201], pressure sensors, i.e. microphones, should be used as control sensors. Electret microphones are small and cheap and the practical upper limit for the number of control sensors is really given by the cost of the control system and the cost of wiring and installation of the sensors. Microphones do not need very much space and can be placed almost anywhere on the confining surfaces. It is usually advantageous to place these sensors so that extra cabling is minimised and to minimise the risk of exposure to dust, dirt and physical damage.

To find the optimal positions for the control sources and sensors – with the restriction that only certain positions are allowed – the first step would be to identify the positions that are acceptable from acoustical and design/installation points of view. This part of the process results in a set of conceivable source positions and conceivable control error positions. From these sets, the best configuration of control sources and control error sensors should be found.

The first version of a schematic programme to make an exhaustive search of all possible configurations may be:

- Choose a set of control source positions from the set of conceivable positions
- Choose a set of control error positions from the set of conceivable positions
- Calculate the optimal output signals
- Calculate the cost function
- Keep the present cost and configuration if it is better than in previous configurations
- Do this for all possible sets of control error positions
- Do this for all possible sets of control source positions

In some applications, it is also conceivable to use different configurations of sources and sensors for different running conditions.

1.1.2.2. Data collecting

A certain amount of input data is required to make the calculations in the listed programme above. Frequency response measurements have to be made in every conceivable control source and control error positions for the whole frequency range of interest with the proper frequency resolution determined by the nature of the application. Thus, for each frequency, frequency response matrixes can be formed.

In addition to the information given by the measured frequency responses, data describing the primary sound field in the control error and evaluation positions must also be available for the selection process. In effect, the sound pressures must be known in terms of both magnitude and phase. This can be assessed by recording the microphone signals during a specified running condition and, from this data, calculate the required quantities. The recordings should preferably be made in all microphone positions simultaneously but, as described below, it is also possible to record these signals in smaller sets provided that the running conditions can be repeated with some accuracy. In an aircraft application, there are different running conditions to be taken into account: take-off, climb, different cruise conditions, descent and landing. The fundamental noise frequency varies slowly in this case and only within a fairly small bandwidth, but the acoustic field can vary extensively for different running conditions. Then, it is necessary for the completion of the optimisation process to record the sound field data for each condition and additionally, it may also be of interest to record transitions between different running conditions. By using off-line evaluation, it is then possible to evaluate the transient behaviour of the control system.

To assess the complex elements in the sound field vectors, the phase is measured relative to a reference position, which may be a dedicated sensor within the configuration or a totally separate sensor, whichever is the most suitable for the situation. The amplitude and phase for sensor point can be obtained by estimating the cross spectrum between the reference sound pressure, and the sound pressure measured at a concrete point. Since the cross spectrum calculation involves an averaging operation, short–term stationary signal conditions are expected for the duration of the measurement. Using this procedure, it is possible to record the signals from the sensors in smaller sets provided that the reference sensor is included in each recording set. Another requirement is that the same running condition is kept for the complete session, including the measurements of all sensor sub-sets. Alternatively, it must be possible to repeat the same running condition with some accuracy to ensure that different sets are recorded under (close to) identical conditions. Then, to obtain the correct pressure magnitude, the cross spectrum must be normalized with the auto power spectrum measured at the reference point. Finally, for each frequency or harmonic, the sound field vector, contains the sound pressures in all conceivable control error positions and its elements describe the sound pressures with an rms magnitude and a relative phase.

1.1.2.3. Determination of the optimal configuration

By identifying a large number of conceivable source and control error positions it is more likely that the optimal configuration (or at least a very good one) exists among these. The problem in using this approach is that the number of possible combinations increases rapidly with both position numbers and finally, the optimisation may be unviable with an ordinary computer [201]. To obtain a sufficiently good solution to the optimisation problem in a reasonable time, some kind of search algorithm must be applied. The cost as a function of control source/sensor positions is non-convex and standard gradient search methods are thus not applicable, since they are unable to escape from local minima. A number of methods to search for a sub-optimal solution have been presented in the literature, which are briefly described here.

Forced Random Search Algorithm
A random search method with constraints has been presented in [14], called the Forced Random (FR) algorithm. The algorithm randomly adds or subtracts one control source or sensor, given certain requirements. The source or sensor chosen for removal is the one that has the least (bad) influence on the cost function. When a source or sensor is to be added, the change in the cost function is calculated for each of the possible candidates, and the most favourable candidate is chosen. The algorithm runs for a pre-determined number of random selections.

Cross-Power Algorithm
This algorithm is an attempt to directly calculate the optimal configuration on the basis of controllability and observability [98]. The general idea is to find the sources having the strongest coupling to the sound field. This is done by calculating the magnitude of the cross-power spectrum between all control sources and control error sensors. The source having the largest cross-power magnitude is selected and the optimal source output is calculated. The residual sound in each control sensor is recalculated – using the added source – and is used to determine the crosspower for the remaining control sources. Then, the best control source of the remaining sources is selected and the procedure is repeated until the desired number of control sources has been found. The virtue of this method is speed. According to the results presented in [14], this algorithm is approximately 200 times faster than the FR algorithm. On the downside is that the attenuation figures obtained with this method are slightly lower than those obtained with the FR algorithm. Furthermore it seems that this algorithm has only been used to find the optimal source positions, using all available microphones in the process.

Genetic Search Algorithms
Two powerful methods for searching the close-to-optimal configuration have been developed from natural prototypes. The Simulated Annealing (SA) algorithm is sprung from a mathematical model describing the fixation of atoms in a substance during the annealing process. An algorithm based on this method is described in some detail below. The Genetic Search (GS) algorithm is inspired by a Darwinistic evolutionary approach, where suitable parents are selected to produce the next generation. The GS algorithm is typically initiated by randomly creating a number of individuals to form a population. Each individual is described by a chromosome consisting of a number of genes. In applications for optimising positions for active noise control, the chromosome is usually a binary string with a length equal to the number of conceivable source and/or sensor positions [18][152][175]. The value for each gene is either one or zero, indicating that that position is either included or excluded in the configuration. A fitness value is calculated for each configuration, based on the cost function. Parents are then selected using some rule based on the fitness value and one or more offsprings are produced to create the next generation.

The algorithm runs for a predetermined number of generations. In the production of offsprings, certain bits in the chromosomes from the parents are exchanged to produce unique individuals, but with as many bits (genes) set to one as for the parents, i.e. the number of sources/senors is unaltered. A random element is introduced by involving a mutation operation, where two bits having opposite values are selected at random and whose values are inverted. This random operation provides an ability to escape from local minima in the cost function. In some implementations, the ability to mutate decreases as the algorithm progresses.

Simulated Annealing
Another search algorithm was developed by in order to find a sub-optimal solution to the optimisation problem in a reasonable time [177]. The method is based on a simulated annealing model [1][129][134], using the Metropolis sampler [245]. The virtue of the Metropolis sampler is its ability to escape from local minima in the cost function. As the search progresses, the process is annealing, which reduces the probability of accepting a configuration with a larger least squares error. Eventually the process will converge to a local minima that is a good estimate of the global minima, provided that the annealing process is sufficiently slow. New configurations are proposed by exchanging one or more sources and sensors in the configuration. The concept of source and sensor neighbourhoods is introduced as a collection of interchangeable positions, determined from acoustical considerations or from some other a priori knowledge about the system. New sources/sensors are chosen randomly from the neighbourhood, with uniform or non–uniform probability distributions. As the problem is formulated, it is necessary to optimise the positions of both the control sources and the control sensors. This problem can be approached in different ways: optimise the source locations first, then the sensor locations; optimise the sensor locations first, then the source locations; optimise the source locations and the sensor locations alternatingly; or optimise the source locations and the sensor locations simultaneously. The first procedure can be followed by using all control sensors in the process of finding the optimal source positions. The chosen source configuration is then used in the process of finding the optimal sensor positions. This gives satisfactory results, but it appears, however, that the combination of the two local minima obtained with the first two procedures above is not as good as searching for the combined minimum, as obtained the last two procedures.

The Mean Field Annealing algorithm
Mean Field Annealing (MFA) is an optimisation method that – as far as known – has not yet been used to find the optimal positions for control sources and sensors in active noise control applications [179][184]. Since MFA behaves approximately as the mean of the SA algorithm, it seems to have the potential to solve this class of problems. In contrast to SA, MFA is not based on random selection, but on direct calculation. The optimisation procedure is viewed in terms of statistical mechanics. A state vector describes which control sources and sensors that are excluded or included in a particular configuration.

1.1.2.4. Implementation of the control system

One commonly used objective for the adaptive controller is to minimise the sum of the squared sound pressures as measured in the control error positions. Assuming that the positions of the control sources and sensors have been properly optimised according to the procedures described above, this will also minimise the sound pressure in the evaluation positions, according to the system cost function. A number of low-complexity algorithms for multiple harmonic, multiple-error control can be developed, that are usually based on the Complex LMS algorithm [123][125].

One algorithm of special interest is the individually normalised algorithm, where the convergence properties of each control source are determined individually.

In a general control situation with control error inputs, control sources and references, where each reference has harmonics resulting in a total of harmonics to be controlled, each harmonic is a separate input signal to the algorithm, regardless of its origin (reference). Since the control is applied to a number of harmonics, rather than specific frequencies, this controller falls into the category of order–based controllers with a fixed sampling frequency. Finally, an instantaneous cost function, sensitive to the controller output power for the control algorithm, can be formulated for the complex, leaky narrowband MEMO control algorithm can be obtained.

According to Sjösten [201], the complex LMS algorithm (CLMS) has properties superior to those of the standard LMS algorithm using finite impulse response (FIR) filters (FIRLMS) in the control of harmonic sound fields. This implies that a MEMO control strategy is used. Some of the most interesting advantages are listed here:

Control properties
With the CLMS, each harmonic is controlled with a separate complex weight, and the control applied to one harmonic is totally independent of the control applied to other harmonics. This is not the case for the FIRLMS, where all harmonics are controlled by the same FIR filter and all adaptive filter weights are correlated. It is generally not feasible to use one FIR filter for each harmonic, since this would dramatically increase the calculation load. The complex weight has two components: one in–phase component and one quadrature component, with an internal phase difference of 90º. Thus, a minimum of control effort is required to adjust to any complex number [123]. To increase controller performance, it is usually necessary to increase the number of adaptive weights, which adds to the calculation burden of the Digital Siganl Processor (DSP).

Algorithm Normalisation
One of the greatest advantages of the CLMS as compared to the FIRLMS is the possibility to properly normalize the weight update algorithm. Normalisation generally implies compensation for the variation in signal power in the filtered input signal [103][141]. Normalisation is especially important in MEMO applications, where the magnitude of the different frequency responses can vary substantially. Without proper normalisation, the controllers for the different control sources would have very different convergence properties, usually resulting in a slower overall convergence and a reduced global attenuation. The CLMS opens a wider range of possibilities for spatial normalisation of the algorithm. The normalisations presented in [124] result in extremely efficient controllers, with properties close or equal to that of the Newton controller. Since all harmonics are controlled separately, the normalisation can be optimised for each harmonic. It is also possible to use different configurations of control sources and sensors for each harmonic. It is not possible to normalise the update algorithm with the FIRLMS as efficiently as with the CLMS at a reasonable computational cost. The applied normalisation is by necessity the same for all frequencies, and the signal power in all filtered reference signals must be estimated continuously during run–time. The individually normalised algorithm described above is based on an algorithm that uses Newton's method. With this algorithm, all frequency responses in the control configuration are used in the update of every adaptive weight. In an application with a large number of control sources and sensors, it is reasonable to assume that the acoustic coupling between the loudspeakers decreases as the distance between them increases and consequently, it can be demonstrated that this would result in a diagonally dominant matrix.

It should be mentioned that, as this convergence factor matrix is calculated off-line, it is perfectly possible to implement a normalisation according to Newton's algorithm. The matrix inversion is a risky operation, however, and special measures must be taken if the matrix is ill-conditioned.

It is evident from the discussion above that the frequency response functions play an important role, both in the definition of the optimal least squares error and in the convergence of the controller. Moreover, the eigenfunctions in the control source positions and in the control sensor positions determine the properties of the frequency response function and thereby also the least squares solution. Similarly, the attenuation obtained in the evaluation positions depends on the values of the eigenfunctions in these positions. A certain spread between different source/sensor pairs and for different frequencies can be compensated for in the controller by using the normalisation commented before. If the spread is too large, however, the wide dynamic range in the signal levels at the controller inputs and outputs will cause poor controller performance due to the limited resolution in the analogical to digital and digital to analogical conversion, as well as in the internal calculations. One parameter in the cost function could therefore be the spread in magnitude in the frequency response functions.

Implementation cost
A third major advantage is the simplicity of the CLMS algorithm in terms of implementation. Due to the clean structure of the algorithm, the administration overhead is kept to a minimum. Since all quantities are known a priori, most of the calculations can be made off-line and only a minimum of operations must be performed at run-time. If the signal power in the input signal is known (which is usually the case), the matrix elements are constants and can be evaluated before the algorithm is started. The run-time calculations for the scalar complex weight are then reduced. If the controller is supposed to operate over a specific frequency range, quantity must be evaluated for all frequencies within that range with the proper frequency resolution.

1.1.3. Verification of the performance

In the early days of active noise control, the performance of an active noise control set-up was often presented as the attenuation obtained at the control sensors. In a real implementation, it makes more sense to measure the attenuation where it is really needed, which generally results in lower attenuation figures than are obtained in the control positions. In most practical situations, the attenuation is needed at the ear level of people sitting (or moving around within the specified volume). By using the same set of evaluation sensors as defined before for the evaluation of the implemented control system, it is possible to compare the predicted attenuation from the optimisation procedure with the attenuation actually obtained. If the purpose of the data presented is to reliably illustrate the energy distribution within the controlled space, the sound field must be spatially sampled using a sufficiently dense measurement grid. It is usually recommended the grid size to be less than one-third of the wavelength for the highest frequency of interest. Using this criterion, an upper frequency limit of 300 Hz results in a grid size of about 0.3 m. This is somewhat more strict than is prescribed by the spatial-sampling theorem, which states that the grid size must be less than half a wavelength for the highest frequency of interest. If the evaluation positions are chosen with some planning, the results can be presented as a two-dimensional surface plot where colour or intensity illustrates the sound pressure levels.

1.1.4. Practical aspects

In this aircraft application, the costs of making the necessary measurements under in-flight conditions are very high. One obstacle is that all equipment must be certified for in-flight use. An alternative approach is to make the evaluation in a grounded aircraft. This is also a costly operation, and the results would not be directly comparable with in-flight results due to the influence of the ground on the sound field inside the aircraft. A third alternative and a compromise in the evaluation process is to use a section of a real aircraft, i.e. a mock-up, and use artificially generated noise in a laboratory environment [204]. The results obtained in a mock-up test probably correlate even less to the performance in an actual in-flight situation, primarily due to the difficulties involved in the generation of the sound field inside the cavity, which should be equivalent to that of a real in-flight situation. Another dissimilarity with the real application is the length of the mock-up, but a major advantage of using a mock-up is that the system performance and robustness can be investigated in the actual acoustical environment for a fraction of the cost of in-flight measurements. Another advantage is that the system can be tested for cases that are difficult to obtain during flight, such as different disturbances in the synchrophaser angle. The possibility of running different configurations and making repeated measurements is appealing, which is why the mock-up solution could be chosen for this study.

The use of structural control sensors to minimise an enclosed sound field is not recommended by some authors [204]. Imposing a constraint on the structure in certain positions leads to a change in the velocity distribution over the structure and does not necessarily reduce the radiated noise. It is the experience of the authors that microphones should be used as control sensors if the objective is to minimise the energy in the sound field.

1.1.5. Conclusions

In the active control of a steady state harmonic sound field, the maximum attenuation is completely determined by deterministic quantities. These quantities are the frequency response functions from the control sources to the control sensors and the primary sound field as measured in each control sensor position. The restrictions that apply regarding the positions of the control sources and sensors in a practical application emphasise the necessity of finding the most suitable compromise through some kind of optimisation procedure, for example based on simulated annealing using the Metropolis sampler. To compensate for the fact that the sound field cannot be controlled exactly in those positions where the attenuation is needed, a set of evaluation positions should be defined. The efficiency and performance of the control system should always be evaluated in the selected evaluation positions. The process of designing an active noise control system requires careful planning and a certain amount of work. The following is an example of steps that should be included in the process:

- Identify conceivable control source positions
- Identify conceivable control sensor positions
- Define the evaluation subspace
- Measure the frequency response functions
- Measure the primary sound field
- Find the optimal control configuration and predict the performance in the evaluation subspace
- Install the active noise control system
- Evaluate the system in the evaluation subspace

1.2. Hybrid active noise control

ANC is a viable method [103][173] for noise control in low frequency ranges and like other control system, ANC schemes may be implemented with the feedforward and/or feedback strategies. Feedforward ANCs depend on accurate secondary path models to generate destructive interferences at the sensor locations [53][79][141]. Feedback ANC schemes are able to introduce active damping to sound fields [52][251], if the feedback sensors are substantially collocated with the actuators. To combine the strengths of both strategies, hybrid feedforward and feedback ANC schemes are developed [18][72][80][223].

In many ANC systems, the errors sensors are placed substantially away from the actuators to avoid sensing the near field effects. Consequently, the feedforward part is not able to generate broadband destructive interference without a stable inverse of the secondary path and feedback damping is less effective for uncollocated sensor/actuator pairs. That is why an extra actuator is necessary to solve these problems [213][250]. The first advantage of using this additional actuator is to generate a broadband destructive interference without requiring stable inverse of the secondary paths [252]. Its second advantage is to introduce more degree-of-freedom in the ANC and improve feedback control [81][152][185][234]. Finally, it also reduces the ANC sensitivity to uncertainties in the path model.

More concretely, this ANC system consists of both feedback and feedforward parts, and a relatively easy approach consists of dividing the problem into two design steps. The first one is the design of feedback part with a focus on robust pole placement, and the second step is the design feedforward part with a focus on noise suppression in the error sensor.

1.3. Active structural acoustic control

In the past decade, an alternative method to using point velocity sources was presented [141]. In Active Structural Acoustic Control (ASAC), force actuators are placed on the confining structure, thereby effectively changing the source spatial distribution function, and consequently the amplitudes of the modes. The purpose is to reduce the radiated sound as much as possible. This is obtained by redistributing the structural modal amplitudes such that strongly radiating structural modes have low amplitudes while non-radiating (or weakly radiating) modes may have larger amplitudes.

ASAC involves secondary structural input either as mechanical shakers or piezoelectric ceramic actuators applied directly to the structures. The actuators are integrated in the structural surfaces or walls in such a way so as to modify and cancel the vibration of the panels and thus reduce the radiation/transmission. This has proven to be a very efficient strategy in several applications [72][141][195] and moreover, this active vibration noise cancellation approach can offer improved performance that can augment other methods to significantly reduce interior noise.

If ASAC is used, the source positions may be chosen with greater freedom since force actuators can be made fairly small and can often be hidden on the structure behind the trim.

1.4. Active control of the flow-induced noise transmitted through a panel

The turbulent boundary layer (TBL) developed over an aircraft fuselage can generate high levels of noise inside the cabin during cruising conditions and thus, directly affects the comforts of the passengers. Another point is that the flow-induced noise increases more rapidly, with respect to the vehicle velocity, than other noise sources [53]. The passive insulation of flow-noise transmission has been successful to a certain extent, but requires dissipative materials whose weight can be a major limitation [161]. Moreover, the passive approach is not very effective in reducing noise and vibration in the low-frequency domain [168].

It may be possible to overcome these problems with active control systems having either acoustic or structural actuators. Indeed, both approaches have been successfully implemented for the control of tonal acoustic disturbances in propellers [196]. Also, low-frequency vibration of the fuselage structure of jet aircraft due to tonal disturbances (engine structure and airborne excitations) is often controlled by means of adaptive tunable vibration absorbers (ATVA) [85]. However, the airflow noise is a random phenomenon from which it is difficult to obtain a reasonable number of time-advanced reference signals and therefore, active control systems based on feedback strategies are generally required. Moreover, there is still a lack in understanding the detailed physics of the vibroacoustics phenomena involved, that is the excitation due to a TBL, the flow noise transmission through the structure and the noise radiation inside the cavity.

Maury et al. [161] restricted themselves to a consideration of collocated structural actuators and sensors, and so, even if a feedforward control arrangement were used with feedback cancellation, the block diagram would be identical to the internal model control architecture of a feedback controller [72]. If acoustic actuators had been used or we had considered double-panel systems, it may have been possible to obtain some time-advanced information from sensors closer to the primary disturbance than the secondary actuator, and a feedforward arrangement may have been advantageous [84][91]. The first objective was to investigate the excitation and sound radiation mechanism of a panel in contact with a turbulent airflow, and the second one, to provide guidelines about the effectiveness of various strategies of active structural acoustic control (ASAC).

To define a suitable control strategy, it is important to examine how the first structural modes or radiation modes of plate contribute to the vibroacoustic response by cancelling their participation in the modal expression for both kinetic energy and the sound power radiated. According to Maury et al., approximately the same level of attenuation was observed by cancelling the first radiation mode or structural mode of the plate [161]. When the higher-order modes are cancelled, however, sound power reductions are achieved more efficiently when suppressing the contribution of the radiation modes compared with the structural modes. However, the cancellation of the first radiation mode does not involve a significant reduction in the kinetic energy of the panel at very low frequencies compared with the cancellation of the first structural mode. Indeed, the cancellation of the volumetric or space-averaged velocity of the structure, which is a good approximation of the first radiation mode, does not necessarily guarantee the cancellation of the velocity of the structure. An important consequence for the passenger close to the aircraft panels is that, even when cancelling the volumetric contribution of the panel velocity, the reduction in the near-field pressure levels is not significant. However, the near-field levels do not increase at higher frequencies (not spillover effect). On the other hand, increasing the structural damping from 1 to 5% does not affect significantly the attenuation of the overall sound power radiated after cancellation of the first structural or radiation mode, although the performance of both strategies is slightly degraded when increasing the damping ratio.

Considering a feedback control with a uniform force actuator controlling a matched volume velocity sensor, the sound power radiated up to 1 kHz could be reduced by cancelling the net volume velocity, except that two extra peaks appeared [161]. They correspond to a minimum value for the sound power radiated by the plate when only excited by the uniform force actuator, that is, to minimum values of the plant transfer function. These extra peaks also appear in the plant response when they attempt to reduce the sound power radiated by a plate excited by a harmonic plane wave, but they are balanced by the plate antiresonance, which occurs at the same frequencies. Finally, the total sound power radiated up to 1 kHz can be reduced when cancelling the net volume velocity by a value very close to the attenuation expected when suppressing the contribution of the first radiation or structural mode.

Another important conclusion of this study is that the structural modes of each panel radiate sound independently and that a suitable strategy for the active structural acoustic control of the sound power inwardly radiated by the panels would be independent feedback control of each structural mode of each panel in the low frequency domain. Although in practice, with discrete actuators and sensors, spillover effects would appear and degrade the results, it has been shown in a concrete case, that this strategy provide attenuation up to 13 dB with six independent controllers.

1.5. Passive and active noise control using the structural intensity method

Understanding the vibroacoustic characteristic of thin-walled enclosures, such as aircraft cabins, is imperative for a satisfactory design. In an effort to reduce the levels of noise and vibration, many techniques have been developed, which can be broadly classified into active and passive control methods. Research on passive and active structural vibroacoustic control for suppressing cavity interior noise has been very active in the last few decades [84]0[91][113][116][117].

For passive and active vibration noise control, each one has its own advantages and disadvantages. The passive method is a simple approach which includes the use of absorbers, barriers, mufflers, dampers, etc. to reduce the interior noise. However, this only provides limited success as this method requires heavy damping materials which leads to significant weight penalties, offsetting the performance gains of the structures. Therefore, the factors have prompted research into applying active control methods. ANC and ASAC have been widely used to reduce low frequency sound transmission.

Over the last few decades, many works have been published concentrated on radiating noise reduction from simple structures such as beams or plates to complex structures. Regarding to passive control, the vibration and noise are reduced either by adding appropriate passive elements or by modifying the system. Fuller et al. [84]0 reported the reduction of interior noise pressure within the shells by changing its parameters and by using passive dynamic absorbers. Although the theory of using absorbers on the shell for eliminating the vibration and the interior noise has not been completely developed, adding dynamic absorbers on the shells of fuselage is a specific passive control technique in the aircraft industry [113][116][117]. Guigou and Fuller [91] presented a smart skin design, based on the passive absorption of an acoustic foam, for aircraft interior noise control. Rao [185] recently described the application of passive damping technology using viscoelastic materials to control noise and vibration in vehicles and commercial airplanes.

Regarding to active control, many active control techniques have been developed to reduce the vibration of the structures in order to attenuate the interior noise, decreasing the fuselage vibration [84]0 or using piezoelectric materials 0[87]. Thomas et al. [220][221] addressed the issue of optimising control force locations on the skin of cylindrical shells and showed that active control is most successful for structural resonance when the number of strongly excited modes is low. Nelson et al. [31][173] theoretically studied the feasibility of global control in a arbitrary enclosure excited at a single frequency under steady state condition. Snyder and Hansen [207] have derived the optimal control equation of the structural acoustic coupling system. On the other hand, the noise in the fuselage can be eliminated by including actively controlled secondary sound sources at the sound field. Bullmore et al. [36] presented the theoretical work of applying active sound sources. Simpson and Hansen [201] developed a genetic algorithm search technique for determining optimal actuator and sensor locations, although the implementation of the algorithm seems quite complex. Thus, it seems that the passive and active controls are successful approaches for controlling interior noise in enclosed spaces, although they still have some problems such as lack of global performance, instability or determination of the best locations for dampers, sensors or actuators.

It is clear that interior noise is mainly caused by vibrations of the panels of the structure and that this structural borne noise is due to the coupling between the structural vibration and the interior fluid. Thus, the acoustic mode shapes are dependent upon the design of the cabin but since it may not be easy to reduce the structural borne noise by modifying the acoustic mode shapes, the passive and active control of structural vibrations is an effective noise control approach [152]. A reduction in the sound power radiated form the vibrating surfaces can be achieved by vibration reduction or radiation efficiency reduction.

The concept of Structural Intensity (SI) was introduced in 1970s and extended the vector acoustics approach to energy flow in structures born sound fields [177][179][245]. Structural intensity is the power flow per unit cross-sectional area un elastic media and it is analogous to acoustic intensity in a fluid medium due to structural vibration. The structural intensity field indicates the magnitude and direction of vibrational energy flow at any point of a structure and it contains the velocity quantity and the structural intensity distribution can offer full information of energy transmission paths and positions of sources and sinks of mechanical energy. Hence, structural intensity may be used to overcome the problems which are linked to the passive and active vibration and noise control. According to Liu [152], the structural intensity approach and stream line presentation can overcome the problems or previous studies and are robust methods to identify the vibroacoustic interior noise source. From this study, two guidelines for passive and active interior noise control can be summarised as follows:

- For passive interior noise control, the dampers should be put at the locations which are nearby the energy power source position and cannot be attached to virtual sink locations where the interior level may be increased. In theses studies, Boundary and Finite Element Methods (BEM and FEM) have been used in order to determine the optimal positions.
- For active interior noise control, the active control forces should be placed at virtual sink positions which are effective control locations.

1.6. Vibration control using discretely distributed piezoelectric quasi-modal actuators/sensors

1.6.1. General literature review

In recent years smart materials, such as piezoelectric materials, have been used extensively as distributed sensor and actuator. Because the piezoelectric sensor and actuator are distributed on the host structure, an accurate control can be achieved. Smart structures have attracted significant attention in the field of control and dynamics, and many achievements have been made in the past 15 years. In 1985, Bailey and Hubbard [21] used polyvinylidene fluoride (PVDF) film as the distributed actuator to perform vibration control of cantilever beam. Since then, mathematical models for smart beams, plates and shells integrated with the piezoelectric actuators and sensors have been established [25][47][57][58][59][68][118][119][226][227][238]. Some finite element formulations for the piezoelectric plates and shells are also derived . Based on these models, much research on active vibration control, hybrid control, optimal placement, and sizing of the actuators, as well as experimental investigation, has been studied by many researchers [49][56][67][81][93][144][147][156][168][190][224][228][254].

Independent modal space control [163] is an effective method for vibration control of smart structures in which modal sensor and modal actuator are needed. Modal control for the smart plates can be performed using the piezoelectric modal sensor and modal actuator based on reshaping the piezoelectric layers with little control and observation spillover [93][144][147][224]. However, these modal sensor and actuator are sensitive to the change of the structures as well, and it is hard to suit these changes by software. An alternative modal sensor and modal actuator, which can overcome the preceding drawbacks, was presented based on segmentation of piezoelectric sensor and actuator layers [214][215]. It can suit the changes by software without changing the pattern of the piezoelectric transducers. In this method the modal sensor and modal actuator are designed based on segmenting the piezoelectric laminas into several continuous elements. However, the fully covered or the continuously distributed piezoelectric sensors and actuators are not practical in vibration control of large flexible structures. Therefore, other methods for vibration control of large flexible structures are needed in which a structure is not fully covered by the piezoelectric elements. Moreover, the optimal placement and sizing of the piezoelectric elements should also be considered in order to achieve better control results.

Sun et al. [213] investigated modal control of smart plates using discrete piezoelectric sensor and actuator elements. A novel approach for vibration control of smart plates is presented using optimised discretely distributed piezoelectric sensors and actuators. This method includes the approach to design quasi-modal sensor and quasi-modal actuator, the observation spillover and control spillover analysis, and the criteria for finding the optimal locations and sizes of the piezoelectric sensor and actuator elements. To optimise the locations and sizes of the actuators elements, both energy and control spillover must be considered and in addition, a compensator is used in each mode to perform a modal control of the smart plate independently, which can further suppress the observation spillover, so that the stability of the active control can be warranted. The optimal results obtained do not depend on the initial condition nor do they depend on the control gains.

1.6.2 FACE project

FACE (Friendly Aircraft Cabin Environment) is a European Project (5th PCRD) which focuses on environment comfort aircraft. Initially it was a project with aircraft manufacturers only, to understand and improve the main environmental behaviour linked to noise, vibration and air quality control in civil turbofan aircraft cabins. But Multimedia can not be ignored to improve perception of comfort, and these recommendations have been validated through different types of tests :

i) 3D simulation environment tests with passengers in a real aircraft cabin (questionnaires, observations and focus group) which allowed us to focus on few parameters for each test:

- PCU, Headsets, Keyboard, MMI
- High flexibility in configuration
- Comfort perception

ii) Experimentation (evaluation, technical test).

iii) Measurement.

At the end of FACE, we will be able to give obvious recommendations to aircraft manufacturers on what it shall be done in an aircraft to improve the comfort of the passengers

Noise levels inside modern aircraft are relatively low because engines, the main source of external noise in flight, have become much quieter and the insulation and sound proofing materials in the cabin wall generally work quite well in damping noise.

There is, however, still room for improvement. The distinct frequency peaks of the ‘buzz saw’ noise coming from the engine fans during take-off and climb can still be annoying for the passenger and the broadband noise from the engines’ exhaust and from the air streaming along the skin of the aircraft causes unpleasant sound levels during cruise. Noise from the outside is transmitted as vibrations through the cabin structure, which consists mainly of the outer skin and frames, a glass wool thermal insulation layer and the trim panels. These plastic linings radiate the sound waves into the cabin. While noise in the frequency range above 500 Hz is damped quite well, adding more sound proofing material to prevent the low-frequency buzz saw and broadband noise from reaching the passengers is not a viable option, because too much weight and bulk would be required.

The approach of the acoustics researchers at the EADS Corporate Research Centre (CRC) in Ottobrunn and Suresnes therefore is to convert the trim panels into anti-noise elements by taking advantage of the effects of destructive wave interference. Since all noise-induced vibrations are waves with peaks and valleys, noise can be suppressed by creating additional waves in an opposite phase. Peaks and valleys thus add up to zero and cancel each other out. Technically, this is done by sensing the structural vibrations with acceleration sensors and, at the same time, by counter-vibrating the panels with suitable actuators, which results in reduced sound radiation

Researchers at the CRC designed and built such actuators using the piezoelectric effect of certain ceramic materials. Just as bi-metallic switches bend when they are subjected to a change in temperature, piezo-ceramics bend when stimulated with an electrical current. While the resulting movement is small, piezo-actuators move instantaneously and fast and are thus able to create vibrations with the desired frequencies.

One solution to make the trim panels vibrate is to use active suspension elements, which replace the conventional, passive mounts attaching the panels to the fuselage structure. Every active element incorporates three actuator units and three acceleration sensors, an electrical power supply and the connection to a digital processor. Using the vibration data from the accelerometers in the suspension elements and on the trim panels themselves, the control unit calculates and sends the noise-canceling response to the actuators.

The other solution is to integrate several piezo-actuators inside the panels' honeycomb sandwich structure during manufacturing. With acceleration sensors on the panels and with the actuators driven by control electronics, the noise transmissions are minimized with the counter-vibrations induced by local flexing of the trim panels.

After extensive analysis and numerical simulations, "proof-of-concept" active suspensions and panels were built and tested in the CRC's acoustics laboratory in Ottobrunn. Specimens were fitted to a generic fuselage section in a window between the reverberation room (representing the outside), where powerful loudspeakers emit specific noise frequencies, and the anechoic room (representing the cabin side), where the remaining noise transmitted through the test panels was measured with an intensity probe. The effectiveness of reducing noise transmission by active panels and suspensions was so assessed, with both concepts providing good results. Reductions in radiated sound power of up to 20 dB (down to 1% of the initial level) for discrete frequencies and 10 dB (down to 10%) for broadband noises were measured.

Active panels and active suspensions will be combined in our upcoming tests with more realistic structures and this combination could well result in an even better noise reduction. Low costs, environmental, health and safety regulations requisites must be considered too.

Advancing from the proof-of-concept stage, researchers are presently in the process of assembling larger wall structure test sections made of carbon-fibre composites. Still within the FACE project, active suspension elements as well as curved active trim panels are going to be tested in the acoustics laboratory this year.

1.7. Vibrational neutralizer

In the last decade, the tunable vibration neutralizer has been the subject of extensive research for the purpose of global vibration control. The aim has been to find an alternative method to active control, for sound and vibration problems [27][31][34][47][96][98][116][129][135][143][145][160][190]. The most recent developments on the theoretical work has been the determination of the optimal tuning of a vibration neutralizer that minimises the kinetic energy of a host structure [31][32][65]. The method on how to determine this optimal tuning has been called the active-passive analogy and in another study, a procedure on how to apply multiple tuned tunable vibration neutralizers has been suggested [66].

Concretely, the optimisation of a tunable vibration neutralizer consists of the determination of the optimal tuning ratio in function of the stiffness and the mass of the neutralizer. In some cases, the performance can be further improved by active damping control [129]. According to Dayou and Brennan [64], the global reduction obtained by the optimised vibration neutralizer is comparable to that of active control.

The traditional dynamic vibration absorber, in general, is a spring-mass-damper system (SMD) capable of attenuating the vibration when the excitation frequency comes close to its natural frequency. This traditional SMD dynamic absorber could not deal with multiple frequencies excitation. The Dog Bone vibration absorber and the absorber designed by Hill and Snyder [108] prove to be effective to suppress the vibration at multiple frequencies, by tuning either the mass block or the length of the rod.

Wang and Chen [240] proposed an alternative method based on the impedance method, which consist of designing a multiple-tonal absorber by altering the thickness or the geometry, in general. The impedance method used in the analysis of dynamic coupling between a single PZT patch and a structure has been developed for years [150][257]. The advantage of this technique lies in that each component can be studied independently of the rest of the system. Therefore, the dynamic behaviour of the system can be easily examined even if one of its components has been modified. Taking the advantage of this characteristic, Cheng and Wang [49] formulated and investigated the dynamic couplings between multiple PZT actuators and the host structure. They later proposed a method of bonding additional structures, called structural patches, on a beam and then tuning the natural frequency of this beam to a designed value [241]. This technique of synthesizing natural frequency shows a great potential in the application of passive vibration control. However, the structural patch is required to be thin since its bending stiffness is further neglected. Wang and Cheng [240] also introduce a methodology of designing a multiple-tonal vibration absorber using additional structures bonded on a beam. However, this added structure does not need to be thin because its bending stiffness is included in the synthesis of the natural frequency of a beam and thus, the added structure is considered as the block mass. Nevertheless, the fundamental issues of the mass, the bending and stretching stiffness of the block mass on the natural frequencies can be examined in detail. Finally, the merit of this method is that it does not need any optimisation method that usually requires a time-consuming search process.

1.8. Hybrid piezoelectric damping system

There are many studies on active and passive damping systems, such as active mass damper and oil damper to reduce the dynamic response due to external excitations. Passive damping utilises the own structural response in order to generate the control force without external power supply, whereas active damping achieves excellent vibration suppression performance with a large amount of external power supply. Therefore, the active/passive hybrid type of damping system can be combining the advantage of both passive and active damping systems [2].

The piezoelectric damping have been developed in various engineering fields: aeronautical, space, civil, railway, sports engineering, etc. The ability of piezoelectric material to transform mechanical into electrical energy and vice versa allows to design active and passive (using shunting circuit) damping [6][8][100][102][110][238]. Moreover, light weight and high frequency actuation capability are major advantages of the piezoelements for acoustic control application. Both numerical and experimental studies indicate that the optimally designed hybrid piezoelectric damping system can be successfully achieving excellent performance as compared to a purely active piezoelectric damping system [2]. It also means that the proper passive piezoelectric damping system is necessary to reduce active control effort under the equal vibration suppression performance condition [2][4]. Since the selection of the number and location of the piezoelements (tiles pairs) to be used for the passive and active damping quite affect the active control effort, these number and locations should be designed by numerical optimisation [2].

Ahmadian and DeGuilio [8] summarised the recent advances in using piezoceramic systems for vibration suppression. The piezoceramic systems are classified into three types: shunted, active and active-shunted piezoelectric absorbers. The configuration of the hybrid piezoelectric absorber usually integrates surface bonded or embedded piezoelements with an active voltage source and a shunting circuit in series. A hybrid piezoelectric absorber could achieve better damping performance with less active control effort as compared to a purely active piezoelectric absorber. Agnes [6] derived an analytical modal model for the active shunted hybrid piezoelectric absorber. The modal model derived can be predicted the structural damping effect of the hybrid piezoelectric absorber and significant reduction in the structural response to broadband disturbance was experimentally demonstrated. Tsai and Wang [222] also addressed the fundamental issues of the hybrid piezoelectric absorber and showed that the shunting circuit can enhance the active actuation for the structural vibration suppression and therefore demonstrated that the integrated active shunted hybrid piezoelectric absorber is more effective than the separated active shunted hybrid piezoelectric absorber. This reduction in the control effort by active control leads to the increase in the overall stability of the system [8]. However, the correlation between the active control effort and the damping performance of the hybrid piezoelectric absorber is still unclear.

Adachi et al. [2] showed that the active control effort of the hybrid piezoelectric absorber increases corresponding to increasing the required damping performance. The integrated hybrid piezoelectric absorber is sensitive as compared with the separated hybrid piezoelectric absorber for varying the damping performance. In the example presented by Adachi et al. [2], the separated hybrid piezoelectric absorber is superior to the integrated one at the required damping performance level below 14 dB for reducing the active control effort. On the contrary, the integrated hybrid piezoelectric absorber is superior to the separated one at the required damping performance level above 14 dB. Finally, it has been demonstrated that a new hybrid piezoelectric absorber based on a switching actuation is capable of the trade-off between the active control effort and the damping performance. This switching type of hybrid piezoelectric absorber can be installed by a simple combination of the integrated and separated hybrid piezoelectric absorbers and the switching actuation is operated by using a control map.

1.9. Active and passive suppression of panel flutter

Panel flutter is a self-excited aeroelastic phenomenon caused and maintained by the interactions between motions of structural panels and aerodynamic loads exerted on the panels. It occurs most frequently in high-speed, most often transonic, supersonic and hypersonic flow. The presence of high temperature by aerodynamic heating causes the panel stiffness to reduce, which enables flutter motions to occur at lower dynamic pressures than in the case without thermal influence.

Linear analysis on panel flutter provides information about airflow speed at which the panel becomes dynamically unstable and the amplitude of oscillation grows exponentially with time. In reality, the nonlinear effect of the panel appears as the amplitude grows to a certain level, and the vibration reaches a limited values (so-called limit cycle) [171].

Much research on the suppression of nonlinear panel flutter has been carried out by using piezoelectric material or shape memory alloy. Scott and Weissharr [198] performed active suppression research on increasing the flutter velocity using the piezoceramic and shape memory alloys with a linear plate memory and the Ritz method. Zhou et al. [254] used a Finite Element Method to suppress nonlinear panel flutter under uniform thermal loading by using a modal reduction scheme and linear quadratic regulator linear control. The feasibility of passively dissipating mechanical energy with electrical shunt circuits has been investigated. Hagood and Von Flotow [100] formulated the equations of the mechanical and electrical characteristics with piezoelectric material shunted with electrical circuits for the case of a resistor alone and an inductor-resistor resonant shunt to provide damping for the beam. Hollkamp [111] showed that multiple modes could be suppressed using a single piezoelectric patch connected to the multiple inductor-resistor-capacitor for a beam model. Tang et al. [217] showed that a structural vibration could be suppressed effectively using the active-passive hybrid piezoelectric network for a beam model. However, most of these papers on resonant-shunted piezoelectrics have the limitation in that the theories are applied to a uniaxial loading condition such as beams. For more complex structures and loading conditions, the theory should be modified to give reasonable solutions. Hollkamp and Gordon [110] investigated a suppression scheme for a two dimensional planar problem using a passive piezoelectric network. Lag mode suppression of hingeless helicopter rotor blade was investigated by passive piezoelectric damping [132].

The governing equations of the electromechanically coupled panel subjected to a thermal load can be derived for active and passive suppression [171]. They are based on the extended Hamilton's principle, on the classical laminated plate theory, and the von Karman plate model for the panel structure and quasi-steady first-order piston theory for the supersonic airflow. A finite element discretisation can be carried out and to achieve the best suppression effect, optimal shape and location of piezoceramic [lead zirconate titanate (PZT)] patches can be determined by using genetic algorithms [91]. In the development of the flutter suppression strategy, it is generally impractical to consider all modelled modes because of large degrees of freedom of the system equations of motion, so that modal transformation methods have to be used. A powerful piezoelectric force is necessary to suppress flutter effectively, and PZT satisfies this requirement.

First, in the active control method, the optimal controller based on the linear optimal control theory can be designed for flutter suppression of the panel. Second, a passive approach is suggested for suppression of the nonlinear panel flutter. A passive damping technology, which is believed to be a more robust suppression system in practical operation, and in which there are no need for power equipment, sensor systems, and controller, has been simulated by using one shunt circuit and two independent shunt circuits (multi L-R series shunt circuits) [171]. Optimal resistance and inductance can be determined by a pole placement method to suppress the vibration effectively.

With the use of Newmark-β method, Moon and Kim investigated the effects of active and passive suppression for composite panel flutter at various temperature and dynamic pressure conditions in time domain [171]. The limit-cycle motion at low temperature and certain dynamic pressure can be completely suppressed. However, as the temperature increased, limit-cycle motion cannot be completely suppressed with the limited piezoelectric force. The performance of passive damping is decreased at a higher temperature than that of active control and further studies at a higher temperature condition will be needed. However, there is no need for additional apparatuses such as sensor and power equipment in the passive damping scheme, and there is also no need for control instruments. Therefore, practically speaking, according to Moon and Kim [171], the passive damping suppression scheme can be preferable for the panel flutter.

1.10. Skyhook vibration control

This idea of control has been introduced for more than 30 years by Crosby and Karnopp [53][127] for vehicle suspension applications. Skyhook control indicates that when the absolute velocity of the sprung mass and the relative velocity across the suspension have the same sign, the damper is resisting the motion of the sprung mass and a damping force proportional to the sprung mass velocity is desired. Otherwise, the damping force does not resist the motion of the sprung mass and must be minimised. This logic emulates the ideal configuration of a passive damper “hooked” between the sprung mass and an imaginary “sky”, hence the name “skyhook”. Suspension systems with skyhook control draw a small amount of energy to operate a valve that adjusts the damping force and therefore, are referred to “semiactive” suspensions or dampers. The damper valve can be a mechanical element or a fluid valve, that takes the advantage of the rheological changes of the damper fluid [40][41].

The virtues of semiactive suspensions versus conventional passive suspension have been addressed in several [9][10][11][45][46][139][163]. Using various analytical methods, these studies have concluded that in nearly all cases of semiactive suspensions can more effectively reduce vibration transmission across the suspension and control the suspended body (or sprung mass), as compared to passive suspensions. Some of the studies have led to different control methods for semiactive suspensions, including several variations to skyhook control, which have been reviewed by Ivers and Miller [113]. Ahmadian also provided an analysis of the dynamic jerk that is caused by the change in damping force in skyhook systems, with two alternative formulations for eliminating dynamic jerk and comparative results of a series of dynamic tests that compare the alternative methods to the conventional skyhook control [7].

2. Computational and test models

Previous research on ANC to enclosures has included analytical, numerical, and experimental investigations. Some of more important works are discussed briefly next. Nelson et al. [173] carried out a theoretical investigation into the feasibility of global control in arbitrary enclosures excited at a single frequency under steady state conditions. They concluded that the reduction in acoustic potential energy in an environment of high modal velocity is only possible if the control source is located within half a wave length of the primary source. Bullmore et al. [37] showed that in an acoustical environment of low modal density, global attenuation of sound with a secondary source located further than half a wavelength from the primary source is achievable provided that the secondary source is placed at an antinode of the primary field. Placing sensors in enclosures corners and minimizing the responses there has the greatest effect on reducing acoustic potential energy in an enclosure when more than one mode dominates. However, for these cases to work, the primary excitation must be at an acoustic resonance. Elliot et al. [77] attempted an experimental verification of the above studies using a simple rectangular enclosure. As predicted by the computer simulation of Bullmore et al. [37], good reductions were indeed achieved only when the enclosure was excited on resonance. Elliot et al. reported very good correlations between the predicted and measured impedance transfer functions; however, correlation between the predicted and observed noise reduction were not good. They found that while the spatial distributions of the predicted and measured controlled sound fields in the enclosure were similar the amplitude of each were different. The average amplitude of the two fields was manually set equal to allow comparison. A prediction of the achievable acoustic potential energy reduction over a narrow frequency range showed that in most cases the acoustic potential energy would be reduced, but there was a possibility chance of increasing the energy at some frequencies.

Application of ANC to a BAe 784 twin turbo-prop aircraft fuselage was reported by Bullmore et al. [31]. They modelled the problem as a thin cylindrical shell and a cylindrical room (with floor). The damping of the aircraft cabin in this case was estimated by matching predicted and experimental results, and found to be approximately 30% across all modes. It is worth noting that it was not suggested that the dampening for the fuselage was 0.3, but that this was the parameter value that must be used in order to obtain reasonable agreement between experimental and theoretical results. The simulation results in [31] showed that for a control system comprised of 32 error sensors and 16 control sources, the achievable average sum of squared pressure reductions over a plane representation of the height of seated passenger was 14 dB for the first propeller blade passage frequency of 88 Hz, and 4 dB for the second harmonic. The validation of results in [31] were reported in [72][75]. Alternative configurations of control sources were tested, with improved control at the second and third harmonics obtained by concentrating the majority of the loudspeakers in the plane of the propellers. The reduction in the tone at the fundamental frequency was a maximum when a fully distributed control source arrangement throughout the cabin was used. When all three harmonics were controlled simultaneously, a less than optimal reduction was recorded at some microphones and an increase in sound level recorded at others. Another interesting aspect in [72] is the results reported from testing a two-microphone, two control source ANC system on a passenger seat. A grid was set up that enabled the sound pressure around the microphones to be measured. As one would expect, reduction down to the noise floor was achieved at the two microphones. Reasonable reduction were also recorded around the controlled microphones up to about 70 mm away. This was in agreement with results in [76] where it was shown that, on average, the result of driving the pressure to zero at an error sensor will be spherical zone of quite with diameter about one tenth of the wavelength of the excitation frequency. The sound level within this zone is expected to be at least 10 dB less than the primary level at that location.

A separate study on BAe 748 was preformed by Dorling et al. [64]. They predicted and verified the control of 88 Hz fundamental excitation frequency in the cabin using 72-microphones, 24-control source ANC system. Reduction of sound pressure at over 50 points in the cabin was measured and compared to reductions predicted using quadratic optimisation theory. Reasonable agreement between predicted and experimental data was achieved.

Thomas et al. [220] used a cylindrical shell model of an aircraft fuselage to study the effectiveness of using structural control sources to attenuate the noise level inside the cabin. The radial component of the kinetic energy in the structure was minimised for control and produced predictions of poor global attenuation inside the cavity. Consequently, Thomas et al [220] tried to use structural forces to minimise the acoustic potential energy of the cavity instead. In [221] it was shown that good global reduction of acoustic potential energy is possible at the two harmonics of interest using relatively few structural sources. Their experience seems to suggest that better reduction of acoustic potential energy was achieved using structural force inputs than acoustic sources.

Snyder et al. [207] reported a theoretical study for an ANC system in enclosed spaces. The importance of optimising both the physical and electronic parts of the control system was discussed. In [208] it was shown that the mechanism by which the acoustic potential energy is reduced in an enclosure with acoustic control sources is by sources unloading, and that in optimal conditions, the sources should never cause an increase in the acoustic potential energy. In [247] the effect of control source and error sensor configurations for a simplified light aircraft fuselage was reported. The level of acoustic potential energy reduction in the aircraft structure was much more a function of the locations of the control relative to the primary source [247]. Also, a higher reduction in acoustic potential energy was achieved when the primary excitation of the structure was at an acoustic resonance.

Application of the FEM to ANC was reported in [15]. Alvelid [15] investigated how a finite element model could be used for simulating the performance of an ANC system on an aircraft structure. He used the model to calculate the reduction possible by minimizing the sum of the quadratic pressures over the cabin using an optimisation algorithm. However, no detail was given as to how the model was verified. Later, Carletti et al. [39] developed a simple finite element model of an earth moving cabin and used it to predict the time averaged acoustic potential energy reduction possible at three frequencies related to engine and cooling system noise. They verified model by comparing the magnitude of sound level measurements taken in the cab with levels obtained by the model and achieved good agreement at two of the three frequencies. In [61] the importance of control source locations in an ANC system placed in an aircraft fuselage was investigated. Optimal passions were calculated to reduce the total acoustic potential energy in interior cavity at a single frequency. These locations were entered into a finite element model and sound pressure level results were calculated with and without control. The results were subsequently obtained with control sources located in random positions and compared to those obtained with the optimal source locations. Results showed that the optimal locations achieved greater global and local control than locations selected randomly. A modal coupling theory to finite element was reported in [43] and [222] for a curved panel with simple rectangular backing cavities. More recently, Lim et al. [143] reported on a finite element simulation of smart structures using an optimal output feedback controller for vibration and noise control. In their study, three dimensional finite elements was used to model the smart structure obtaining discrete piezoelectric sensors and actuators by use of combination of solid, transition and shell elements. Varadan et al. [231] developed three-dimensional finite element closed loop model to predict the effects of active passive damping on a vibrating structures.

There has also been interesting studies on modelling human body/seat system in a vibration environment [12][193]. In [112] the driving-point impedance and transmissibility techniques were used to evaluate the effects of military helicopter seat cushions on human body. The most relevant study can however be traced back to an EU project IDEA PACI project which was carried out under the Brite Euram funding. The aim of the project was to establish an aircraft passenger noise and vibration comfort index to relate subjective perception to mechanical and physical design properties. An artificial neural network was develop to simulate a virtual passenger, that is generic transfer function between external stimuli and human impressions. The main achievement of the project, as summarised in [212] and in the section 6. of this report, were as follows: a psycho-acoustic study was carried out to identify suitable descriptors and to develop a tool that allowed translating the environmental solicitation in subjective impressions. Then, experimental investigations both at ground and in flight were performed to produce a wide data-base for statistical investigations. On the bases of the acquired results, the architecture of the ANN was defined: it was successively trained and assessed on the experimental data coming from the main test campaign, so setting the aforementioned “virtual passenger”. As the last step, artificial neural network predicted comfort levels on both experimentally and finite element data.

FEM models can be used in order to predict the vibrations of each part in contact with the passenger. In the study of the complete model (with all the moving parts), a particular attention must be given to passenger critical zones, as head, back, etc. As the excitation and the boundary conditions are likely to be different according to the passenger position in the aircraft, the vibrations results will be different for each passenger too. Consequently, although FEM technique is a classical tool in aeronautics, specific models have to be developed.

For the noise evaluation, various approaches are possible, as BEM, SEA, TRIM or ray tracing methods. The precedent vibration results have to be taken into account, as well as all the other radiating parts of the aircraft. A particular attention must be paid to the noise levels perceived by each passenger, that is in his ears location. These studies are also more or less classical analysis in aeronautics, but more advanced and specific tools will be precisely defined.

Various annoyance models are available in function of noise levels or associate psychoacoustic parameters but the most adequate will be selected, validated and adjusted to various critical groups (people with heart conditions, pregnant women, passengers with mental instabilities, …). Moreover, annoyance models depending on vibration levels have to be developed.

Damping, insulation and absorbent systems are usual in aeronautics and noise active control solutions have been widely studied. Then, the best technique will be selected and adapted, in order to act on every passenger according to his critical group and the annoyance detected. The efficiency of the new system will be checked in several cases using the same computational models than before and further experimentally verified.

3. Use of specific textiles and smart materials

The noise and vibration levels could be measured using traditional microphones and accelerometers flushed in every seat, but for this project, new acquisition systems, based on piezoelectric materials integrated in textiles, for example, will be investigated. If possible, the smart textiles with sensing facilities defined in WP2 will be used. They should have been previously designed in order to get sound pressure and displacement data, among other feature. If not, they will be further developed for these functions.

During the last decade, fundamental research on smart structures using intelligent materials has raised industrial interest in applying these results to many problems found in commercial and civil life. Within InMAR project, the main objective of smart structure technology is noise and vibration reduction in civil engineering, machine tools, automobiles, trains, and aerospace engineering. Both strongly coupled phenomena limit the design of highly advanced and efficient light weight structures, where by nowadays noise is considered one of the worst forms of environmental pollution worldwide. That is why an approach based on light weight design and smart structure technology combined with traffic management concepts has to be pursued. Beyond their impact on noise, smart structure technology will for the first time allow for a concurrent light weight design that enables the efficient use of natural resources in the product itself (less fuel consumption, less exhaust emission, etc.). With the upcoming demand of highly efficient, emission-less light weight structures and increased standards for any type of emission, new intelligent materials systems are needed that allow for both highly damped and controllable as well as light but durable structures for any type of high-tech application.

The constantly growing demands on modern structure systems today increasingly cause conventional passive as well as active mechatronic approaches to reach their technical and economic feasibility limits. Adaptive structure technology (smart structures) based on intelligent material systems is an innovative, new cross sectional technology for the optimisation of structure systems. It is based on the integration of additional functionality by combining conventional structures with intelligent material systems, which extend the classic function of load-bearing and form-defining structure to sensing and acting capabilities. In connection with suitable adaptive controller systems, adaptive structure systems can optimally adapt to their respective operational environment. Intelligent material systems themselves are built up from of passive, conventional light weight materials and so-called transducer materials with acting and sensing properties. By using this approach, modern structures can be implemented easily and compactly as well as with low-vibration and low-noise and/or as dimensionally stable as well as with the potential for structure integrated monitoring. This leads to the protection of raw materials, lower environmental stress resulting from noise and emissions, lower system and operating costs and higher functionality and efficiency of systems. While mechatronics extends the functionality of the existing structure system pertaining to a design point mainly by adding components, the core goals of adaptive structure technology consist in the continuous interference into the structure -mechanical, mostly structure dynamic, characteristic of the overall system, and in the optimisation of the structure system by replacing structure components with intelligent material systems (effective in the sensor-actuator sense) to save mass and designed space. Adaptive or intelligent solutions are thus characterised by a function integration and a certain measure of function consolidation, i.e. a structure conformed integration of the sensor characteristics and actuator components, and an active system intervention while as suming mechanically load-bearing characteristics in the overall structure.

They fundamentally expand the structure characteristics and permit the fight against structure disturbances at their source, in the transmission paths and/or in sensitive areas of the system. Adaptive structure technology can be understood as a logical consequence of mechatronics, which does not lose their importance, however. The goal of adaptive structure technology is to influence structures. By so doing, a new technology is made available to the user to optimise his/her products. The main focus is in particular the vibration reduction, noise reduction and the structure integrated damage monitoring. Likewise of importance are shape control and the fine positioning. At the same time, further applications can be derived from the partial competences. The need for innovative adaptive measures is rising for many fields of application. A need for solutions usually exists in the vibration and noise reduction, shape control and the Structural Health Monitoring. By implementing adaptive structural m e as ure s , functionality, life span, comfort, security, production efficiency and quality are to be increased and stresses, weight and costs reduced.

3.1. Determination of textile optimal characteristics

3.2. Integration of sensors and actuators in textiles

3.3. Weight saving options

4. Use of carpets

4.1. Interaction between passenger feet and floor

5. Possibilities of integration

5.1. Active acoustic materials based on isodynamic transducer

Active control of the acoustic impedance of walls in rooms and auditoria have been on the market since the late 1960s. These systems were first aiming at increasing the reverberation time, but they are now capable of modifying (to a certain extent) other aspects [135][165]. In 1985 Guicking introduced the idea of active wall [91], that consists of an array of locally reacting cells, each cells comprising a sensor, an electronic circuit and an actuator. A single reciprocal transducer may in fact act both as a sensor and an actuator and the feedback is a linear combination of the velocity of the transducer diaphragm and the acoustic pressure at the diaphragm / air interface [61][163][166].

On one hand, if classical loudspeakers are used, a certain volume is generally required behind them for low frequency generation. On the other hand, Lissek and Meynial [152] showed that isodynamic transducers can provide an effective alternative to the conventional cone speaker. These devices use electrodynamic transduction and offer the main advantages of low moving mass (because the membrane is a flexible film and not a rigid cone), high $(Bl)^2/R_e$ factor and low electrical inductance. Another advantage of this type of transducer is that its typology is well suited for low cost manufacturing of large panels using rubber magnets. In future solutions, grooves could be involved in the magnets or no rectangular section magnets could be used. The membrane can be made with deposited aluminium or copper tracks and by means of FEM simulations, a design optimisation can be performed.

It could result difficult to directly apply this technology in the case of individual noise control in an aircraft cabin, but the use of this transducer for active materials with variable acoustic properties will have to be considered [154].

5.2. Solid-state ceramic actuator designs

Smart materials have the ability to perform both sensing and actuating functions and are, therefore, capable of adapting to changes in the environment. In this context, the need for high-accuracy displacement elements has led to renewed emphasis on displacement devices.

Displacement transducers can be classified into two main groups: conventional displacement transducers and solid-state actuators. There are three types of conventional displacement transducers: oil pressure cylinders, servo- or step motors, and voice coils. The oil pressure cylinder operates on hydraulic principles, where the displacement piston is driven by oil pressure from another piston. The principal disadvantages are large space requirements and long response times. The servo- or step motor converts the revolutions of an electromagnetic motor into linear movement by using a gear mechanism. Mechanical blacklash is a difficult problem for these displacement transducers. The voice coil (speaker) operates through a combination of electromagnetic coils and springs. The generative force of the voice coil is small compared to its input power, and its response time is slow. Each of these devices is capable of generating large displacements in the millimetre to the centimetre range, but they are inadequate for precise positioning range. A second category of displacement transducers consists of a number of different solid-state actuators. This group can be further classified into three categories based on the type of driving mechanism: shape memory effect (thermal drive), magnetostriction (magnetic drive) and piezostriction/electrostriction (electrical drive).

5.2.1. Solid-state actuators

5.2.1.1. Shape memory effect

The shape memory effect is a consequence of a crystallographically reversible martensitic phase transformation occurring in the solid state [69]. After deformation at low temperature, a shape memory alloy will regain its original shape when heated above the phase transmission temperature, and thermal cycling can be repeated indefinitely. Shape memory alloys can exhibit large coverable strains as large as 7% and therefore, they have been proposed as alternatives for solenoids, motors, and bimorph-type actuators, as well as for use in flexible robot structures. However, the large thermal energy requirements for martensic phase transformation together with large hysteresis and long response times are important drawbacks for shape memory alloys.

5.2.1.2. Magnetostriction

Magnetostriction is defined as the change in dimension of a magnetic material caused by a change in its magnetic state. Magnetostrictive materials can be used as both sensors and actuators. The Magnetostrictive strain arises from a reorientation of the atomic moments. High-power magnetostrictive actuators can produce forces exceeding 50 MPa with strains on the order of 0,6%. High reliability, ruggedness, and imperviousness to adverse environmental conditions are the advantages of magnetostrictive materials. However, magnetostrictive materials have the problem of requiring a coil to create the necessary magnetic field. This, in turn, induces noise into adjacent electronic circuits and devices [69].

5.2.1.3. Piezoelectricity

The direct effect of piezoelectricity is the electric polarisation generated in certain acentric crystals when they are subjected to mechanical stress. These materials also show a geometrical strain, which is proportional to the applied electric field, which is know as the converse effect. Natural crystals such as quartz, tourmaline and zincblende are the classical piezoelectric materials. For many years, these materials have served as transducers for converting mechanical energy into electrical energy and vice versa. In general, natural crystals have rather low piezoelectric coefficients. Ceramic piezoelectric materials were developed in the second half of the 20^{th} century and have been constantly improved since then. Modified lead zirconate titanate ceramics (PZT) are the leading materials for piezoelectric applications. The induced strain is nearly proportional to the applied field for low field levels. However, the strain curve deviates from this linear trend, and significant hysteresis is exhibited due to domain reorientation as the field becomes larger.

The magnitude of the piezoelectric coefficients depend markedly on dopants and defect structure because of their influence on domain wall motion. This in turn controls the nature of the hysteresis loop in the piezoelectric ceramic, and the interaction of the domains and defects leads to so-called soft and hard piezoelectric compositions. Piezoelectrically soft materials are characterised by high piezoelectric constants ands and high hysteresis as a result of relatively mobile domain wall. In hard piezoelectric ceramics, the domain wall motion is inhibited, resulting in lower piezoelectric constants and reduced hysteresis. Soft piezoelectric materials are preferred for most multilayer and bimorph actuator application because of their high strain [69]. For some actuator applications, which require nonhysteretic response, hard piezoelectric ceramic can be preferable.

Solid solutions of barium tin titanate are an interesting new family for actuator applications. Because of its small coercive field, the composition exhibits an unusual strain curve, in which domain reorientation occurs only at low fields. This is followed by a long linear range at higher fields.

5.2.1.4. Electrostriction

When an electric field is applied to a centrosymmetric dielectric material, it produces a strain proportional to the square of the field. This phenomenon is called electrostrictive effect. For high-permittivity solids, however, electrostative strain is proportional to the square of the polarisation. Because electrostriction is a result of the polarisation induced by the applied field, electrostriction may occur in all crystals whether or not the crystals have polarity. Electrostriction in PMN ceramics is extraordinarily large, with strains exceeding 0,1%, and an interesting feature of these materials is the absence of hysteresis. Moreover, displacements can be positive or negative depending on applied electric field. Electrostrictive oxide ceramics generally elongate parallel to the applied field and contract perpendicular to the field, but polymers such as PVDF contract in the field direction.

5.2.1.5. Phase-transition related strain

The phase transition from an antiferroelectric to a ferroelectric phase can induce larger strains than those found in either piezoelectric or electrostrictive materials. A unique characteristic of the phase transition-induced strain in the antiferroelectric phase is it isotropic volume expansion, similar to thermal expansion. With appropriate compositions, piezoelectric/electrostrictive materials can exhibit two kinds of phase transitions related to strain: a shape memory effect and digital displancement.

Once the ferroelectric phase has been induced, it is retained even at zero electric field. It can, however, be erased with the application of small reverse bias fields. This shape memory ceramic is used in energy-saving actuators. On the other hand, for the phase transition-related digital displacement, two well-defined (on/off) strain states can be achieved. The longitudinally induced strain approaches 0.1%, which is rather large when compared to the normal piezostrictors and electrostrictors. Finally, there are displacements of up to several tens of micrometers that can be controlled with a precision of 0,01 µm. Response speeds are on the order of 1-5 µs. Generative forces are as large as 3500 N and driving power is an order of magnitude smaller than electromagnetic motors.

5.2.2. Solid-state ceramic actuator designs

Standard ceramic actuators are classified into two groups based on their displacement mechanism: linear or bending. A linear exhibits bending displacement and reaches the highest value at the tip of the free end actuator. Multilayer actuators and single plates are linear displacement transducers. Monmorphs, unimorphs, bimorphs, and multimorphs are example of bending displacement transducers.

The most common examples of these two classes of actuators are multilayers and bimorphs. The multilayer actuator is composed of a number of ceramic layers alternating with internal electrodes. Individual internal electrodes are electrically connected in parallel with two external electrodes attached to the sides. When an electric field is applied to the element, the element expands along the longitudinal direction in accordance with the converse piezoelectric effect. Important features of multilayer ceramic actuators are low driving voltage (100 V), quick response (10 µs), high generative force (3 kN) and high electromagnetical coupling. However, they only exhibits displacements in the range of 10 µm, which may not be sufficient for some applications.

Bimorphs usually consist of two twin ceramic sheets bonded together with their poling directions opposed and normal to the interface. When an electric field is applied to a bimorph, one of the plates expands while the other contracts. This mechanism creates a bending displacements of several hundred micrometers, but the response time (1 ms) and the generative force (1 N) are low.

If three important characteristics (displacement, generative force and response time) are compared for multilayer and bimorph actuators, there is a sizeable gap between them with respect to performance [69]. Neither provides both a relatively large displacement coupled with an intermediate level of generative force, but recently there has been extensive effort to design actuators to fill this gap. The moonie, the cymbal, the reduced and internally biased dome –shaped oxide wafer (RAINBOW), the thin-layer composite unimorph ferroelectric driver and sensor (THUNDER), and the cerambow are new designs developed within the past 15 years. These new designs provide moderate displacements in conjunction with moderate generative force. Basically, all of the new design exhibit bending or flextensional displacements.

5.2.3. Flextensional transducers

Flextensional transducers are mechanical amplifiers that couple the longitudinal strains in a ceramic bar or disk to the radial flexure of the metal or ceramic shell. Originated in the 1920s for ship navigation, flextensional transducers can be classified into five different classes based on their shape and mode of operation. Class IV transducers are the most intensively studied, best understood and most widely used underwater transducers. Class V flextensional transducers contain two spherical cap shells joined to a radially vibrating ring or dish.

A simplified version of the flextensional or bending transducers emerged in the 1990s, with the basic idea to convert some of the lateral dilation in the planar direction to the longitudinal direction and thereby increase the longitudinal displacement. The moonie is a simple compact type, which with is unique properties, has been investigated for many potential applications. The cymbal is the improved version of the moonie transducer, with higher efficiency, displacement and generative force, whereas RAINBOW is another type of transducer, and flextensional designs referred to as cerambow, THUNDER and crescent are the latest generation transducers derived from RAINBOW structure.

5.2.4. Comparison of the solid-state actuator designs

It is rather difficult to compare different actuators because of different geometry and various operating conditions for specific applications. To make a fair comparison, similar dimensions for each actuator is considered. Displacement mechanisms of cymbal, moonie and RAINBOW class transducers are based on the same physical principle: converting the radial displacement to the axial displacement by flexing or bending the structure. In principle, cymbal, moonie and rainbows can exhibit comparatively high displacement if an appropriate diameter to thickness ratio is chosen. So far RAINBOW have achieved higher axial displacement than the moonie and cymbal because of differences in the diameter to thickness ratio [69]. Scaling analysis show that, being faire in comparison and choosing the same size, moonie and RAINBOW will present similar displacements. To exhibit a positive displacement, the applied field is in the opposite direction to the polarisation in the rainbows but in the same direction as the polarisation for the moonie and cymbal designs.

5.2.5. Applications and conclusions

Multilayer actuators and bimorphs are used extensively in many applications, such as active vibration control, electroacoustic transducer, etc. Moonie and cymbal actuators have great potential in the automotive and aviation industries, since they can be utilised as sensing and vibration suppression elements and as switching elements in valve designs. RAINBOW can be used for acoustic cancellation and pumps and as switches. The moonie and cymbal actuators can also be used as micropositioners for applications requiring small size and quick response time but only modes loads.

Finally, ceramic materials are not strong under tensile force due to their structure but are rather good under compressive forces. In the moonie and cymbal structures, electroactive ceramic elements are kept flat and sandwiched between two metal endcaps, which clamp the ceramic on the rim. Consequently, there are compressive forces over the bonding layer and small tensile forces under the cavity beneath the endcaps. On the other hand, RAINBOWS and THUNDER are stress-biased structures. Part of the structure is under high compressive forces, and others are under high tensile forces. RAINBOW and THUNDER are based on coefficient of thermal expansion between different parts of the actuator, and the displacement is highly temperature dependent. By double layering and using different ceramic composition, temperature-dependent behaviour can be eliminated to a certain extent.

Flextensional moonie, cymbal, RAINBOW and THUNDER actuators all give moderate generative forces and displacement values that fill the gap between multilayer and bimorph actuators. Each solid-state actuator design has attractive features. Advantages of the moonie and cymbal actuators are the easy fabrication and tailoring of the desired actuator properties by altering the cavity size and endcap dimensions.

For the RAINBOW actuator, a reduction step during the processing of the ceramic element at high temperature results in a semiconducting layer and stress bias, which is a potential problem for long-term usage [69]. The fatigue behaviour of the actuators depends markedly on the driving conditions. Dynamical studies under uniaxial prestress under a cycling electric field should be performed on all flexural transducers for aerospace and automobile applications. Thermal cycling tests under various humidity levels will also be very useful, particularly for aerospace and aeronautical applications.

6. Vibroacoustic comfort

6.1. Introduction

Air transport technology is very advanced in terms of performance, fuel consumption, velocity, noise and so on. Due to the increased use in highly industrialised areas within small distances, not only environmental impact has to be taken into account, but also internal comfort, that seems mainly related to the interior noise field. On the other hand, apart of the desirable noise reduction, subjective perception is an important criterion for passengers.

Recently, the attention of noise suppression methodologies moved towards active systems. These methods are complementary with the usual passive approaches, both showing qualities and faults. Through their application, certain thematic aspects were deepened, and unexpected results highlighted. The link between noise and vibration levels appeared ruled by complex relationships, and the results following the application of pure noise abatement systems were surprising: interviews on the passengers showed as environments with a low level of noise but unchanged vibrations, could result more uncomfortable than the original situation. It seems obvious that comfort is a subjective matter and therefore psychological studies play a fundamental role [259].

The objective of IDEA PACI [212], a Brite-EuRam project financed by the European Union, was to establish and model an aircraft passenger noise and vibration comfort index to relate subjective perception to mechanical and physical design properties; in other words to simulate a Virtual Passenger, i.e. the generic transfer function between external stimuli and human impressions. Dealing with people, Artificial Neural Network (ANN) based tools [104] seem the most suitable to develop a such Virtual Passenger. Though aimed at interior aircraft noise, the results enable other transportation means to improve general features of comparable construction and developing processes, avoiding or substituting expensive experimental tests by the use of ANN.

The main achievements of the research may be summarized as follows: first of all, a psycho-acoustic study was carried out to identify suitable descriptors and to develop a tool that allowed translating the environmental solicitation in subjective impressions. Then, experimental investigations both at ground and in flight were performed to produce a wide data-base for statistical investigations. On the bases of the acquired results, the architecture of the ANN was defined: it was successively trained and assessed on the experimental data coming from the main test campaign, so setting the aforementioned “Virtual Passenger”. As the last step, ANN predicted comfort levels on both experimental and numerical (FEM) data: the physical parameters that mainly affect the passengers' welfare during the flight were identified through proper simulations.

6.2. Psychoacoustics

In-flight passengers’ subjective comfort is determined by a large variety of parameters. Since interior noise and vibrations are among the known main components, the investigation was concentrated on these peculiar aspects of well-being, namely the sound pressure levels and the accelerations. In particular, the most suitable psycho-acoustic descriptors, used in the subjective evaluations, were identified and then, a comprehensive questionnaire, oriented to the passenger comfort inside aircraft cabins, has been designed by the University of Oldenburg.

Jet, propeller and helicopter aircraft were targeted. Both vibration and noise disturbance were taken into account. Standard methods of evaluating test-subjects answers (factor analysis, paired comparison analysis, etc.) yield a set of psycho-acoustic parameters being identified as the most important descriptors contributing to a comfort index [187]. As first, a qualitative analysis has been performed to relate the psycho-acoustic descriptors to noise and vibration fields; in fact, psycho-acoustic parameters were easily correlated with acoustical and signal-related quantities like spectral parameters, signal envelope and other time-dependent features. Then, the characteristics of the exterior disturbance signals were related to the subjective impression, directly, in a qualitative way. The problem was solved in two steps. Structural acoustics was concerned with the relationship between the exterior noise source and the produced interior noise field; in detail, its objective dealt with the reduction of the sound (and vibration) field in enclosures. Psychological acoustics was instead concerned with the effects of the interior sound field on the human comfort feeling; its objective dealt with the minimization of the noise annoyance. Both the experiences were combined, in order to address better the aircraft design “to noise”.

6.3. Sensitivity analysis

The importance of the application of the Virtual Passenger for a civil aircraft design is per se evident: by knowing the consequences of a variation in the design parameters it would be possible to predict the corresponding variation of the comfort level. Furthermore, by knowing the effects of the structure changes on the comfort level, it would be possible to design the aircraft in such a way to obtain the best possible comfort. These considerations gave the guidelines for the development of the last part of the project. The achieved goals were related to the analysis of the physical parameters that mostly affect the Comfort Index (CI) and the definition of the relationship between the CI boundaries and the input domain. In this way, the Virtual Passenger was integrated in a numerical tool aimed to the design to comfort. The first step was the calculation of the physical vibro-acoustic fields for each seat and for different values of structural characteristics using a FE model of one of the mock-up. The power spectra were computed up to 300 Hz because of the reliability limits of the calculation technique. For higher frequencies to adopt a different approach is mandatory, like the Statistical Energy Analysis (SEA). The results of this phase were the basis for the sensitivity analysis; here, the “quasi” optimal power spectrum was defined by pointing out the best set of structural parameters. In this way, a set of design indicators could be given for the reaching of the best comfort level.

6.4. Conclusions

The importance of the IDEA PACI project is evident from different points of view. First of all, it has been possible to collect a big amount of comfort-aimed experimental data. All the tests, that were carried out, allow to point out, in a detailed manner, the different parameters that contribute to define the comfort status of a civil aircraft passenger, at least from the point of view of vibro-acoustic solicitations. Of course, the fact that the aircraft tests were carried out in ground mock-ups is a limitation for their complete reliability. But, it is important to note that this approach is widely opened to many further improvements, taking into account other environmental parameters (humidity, temperature, ecc.). Regarding the definition and the assessment of the Virtual Passenger, the employed Neural Networks show a good effectiveness in “catching” the complex mapping between the physical data and the subjective comfort level. At the end, the sensibility study gives other important indications for the manufacturers, allowing to know the effect of some structural changes on the Comfort Index. Furthermore, it is also shown the way to define the structural design variations that are required to obtain good comfort levels.

References

[1] Aarts E and Korst J (1989)
Simulated annealing and Boltzmann machines
John Wiley and Sons Inc., 1989.

[2] Adachi K, Awakura Y and Iwatsubo T (2004)
Active control effort of hybrid piezoelectric absorber for structural control
Applied Acoustics 65, pp. 277-292

[3] Adachi K, Kitamura Y and Iwatsubo T (2004)
Integrated design of piezoelectric damping system for flexible structure
Applied Acoustics 65, pp. 293-310

[4] Adachi K, Park G and Inman DG (2002)
Passive damping augmentation using macrofiber composite actuators
Proc. ASME Int. Mech. Eng. Congress and Exposition 2002, New Orleans, Louisiana, USA, IMECE2002-39004

[5] Adachi K, Awakura Y and Iwatsubo T (2000)
Experimental investigation of hybrid damping for flexible structures by using surface bonded piezoelements
Smart structures and materials 2000: damping and isolation 2000, proc. SPIE 3989, pp. 312-321

[6] Agnes GS (1995)
Development of a modal model for simultaneous active and passive piezoelectric vibration suppression
J. of Intelligent Material Systems and Structures 6, pp. 482-487

[7] Ahmadian M, Song X and Southward SC (2004)
No-jerk skyhood control methods for semiactive suspensions
ASME J. of Vibration and Acoustics 126, pp. 580-584

[8] Ahmadian M, and DeGuilio AP (2001)
Recent advances in the use of piezoceramics for vibration suppression
The Shock and Vibration Digest 33(1), pp. 15-22

[9] Ahmadian M (1999)
On the isolation properties of semiactive dampers
J. Vib. Control 5(2), pp. 580-584

[10] Ahmadian M and Marjoram RH (1989)
Effects of passive and semiactive suspensions on body and wheelhop control
J. Commercial Veh. 98, pp. 596-604

[11] Ahmadian M and Marjoram RH (1989)
On the development of a simulation model for tractor semitrailer systems with semiactive suspensions
Proc. Special Joint Symposium on Advanced Technologies, ASME Winter Annual Meeting, San Francisco, DSC13, ASME, New York

[12] Ajovalasit M (2005)
Proc. IMechE, part D, 219 (2005), 499

[13] Alarcón Rovira G, Romeu Garbí J (2005)
Mecanismos de radiación: análisis de estrategias de cancelación
Proc. Tecniacustica Terrassa 2005

[14] Albihn J (1994)
Electroacoustical problems in active noise control for propeller-driven aircraft
Masters Thesis, Chalmers University of Technology, Applied Acoustics, Report S94-01, 1994.

[15] Alvelid M (1993)
Proc. Inter Noise 93, pp. 65-70
[16] Antila M (2006)
Networking local active noise control electronics
Proc. Euronoise Tampere 2006
[17] Aoki Y, Gardonio P and Elliott SJ (2006)
Stability of piezoelectric patch-accelerometer active damping control system in smart panel
Proc. Euronoise Tampere 2006
[18] Asami T and Nishihara O (2003)
Closed-form exact solution to H_∞ optimization of dynamic vibration absorbers (application to different transfer functions and damping systems)
ASME J. of Vibration and Acoustics 125, pp. 398-400
[19] Bai MR and Lai J (2003)
Broadband spatially feedforward active noise control algorithms using a Comb filter
ASME J. of Vibration and Acoustics 125, pp. 18-23
[20] Bai MR and Lin HH (1997)
Comparison of active noise control structures in the presence of acoustical feedback by using H_∞ synthesis tehnique
J. of Sound and Vibration 206, pp. 453–471
[21] Bailey T and Hubbard JE (1985)
Distributed piezoelectric-polymer active vibration control of a cantilever beam
J. of Guidance, Control and Dynamics 8(5), pp. 605-611
[22] Baek KH and Elliott SJ (1995)
Natural algorithms for choosing source locations in active control systems
J. Sound and Vibration 186, pp. 245–267
[23] Barton CK and Mixson JS (1981)
Noise transmission and control for a light twin-engine aircraft
Journal of Aircraft 18(7), pp. 570-575
[24] Bauer R, Lotton P, Hamery P and Bruneau AM (2003)
An active noise reduction earplug using a piezoelectric laterally radiating loudspeaker
Applied Acoustics 64, pp. 591-609
[25] Baz A and Poh S (1988)
Performance of an active control system with piezoelectric actuators
J. Sound and Vibration 126(2), pp. 327-343
[26] Berkhoff AP and Wesselink JM (2006)
Centralised and decentralised configurations for panels with piezoelectric actuators
Proc. Euronoise Tampere 2006
[27] Bernhard RJ, Hall HR and Jones JD (1992)
Adaptive passive noise control
Internoise Toronto 1992
[28] Bisnette JB, Smith AK, Vipperman JS and Budny DD (2006)
Active noise control using phase-compensated, damped resonant filters
ASME J. of Vibration and Acoustics 128, pp. 148-155
[29] Bonnot M, Romeu J, Capdevila R and Sánchez A (2005)
Aplicación del control activo a la reducción de ruido en la cabina de un avión ligero
Proc. Tecniacustica Terrassa 2005
[30] Bravo T and Cobo P (2002)
A demonstration of active noise reduction in a cabin van
Acta Acustica united with Acustica Vol. 88, pp. 493-499
[31] Brennan MJ and Dayou J (2001)
Global control of vibration using a tunable vibration neutralizer
Noise and Vibration Worldwide 32(5), pp. 16-23

[32] Brennan MJ and Dayou J (2000)
Global control of vibration using a tunable vibration neutralizer
J. of Sound and Vibration 232(3), pp. 585-600
[33] Brennan MJ (1998)
Control of flexural waves on a beam using a tunable vibration neutralizer
J. of Sound and Vibration 222(3), pp. 389-407
[34] Brennan MJ (1997)
Vibration control using a tunable vibration neutralizer
Proc. IMechE Part C: J. of Mechanical Engineering Science 211, pp. 91-108
[35] Brevart BJ and Fuller CR (1993)
Active control of coupled wave propagation in fluid filled elastic cylindrical shells
J. of Acoustical Society of America 94, pp. 1467-1475
[36] Bullmore AJ, Nelson PA and Elliot SJ (1990)
Theoretical studies of the active control of propeller induced cabin noise
J. of Sound and Vibration 140, pp. 191-217
[37] Bullmore AJ et al. (1987)
J. of Sound and Vibration 117, pp. 15-33
[38] Carbaugh DC (2001)
Airplane vibration and flight crew response
Boeing Co. International Air Safety Seminar Proc., pp. 149-156
[39] Carletti E et al. (1996)
Proc. Inter Noise 96, pp. 1183-1186
[40] Carlsson JD, Catanzarite DM and St Clair KA (1995)
Commercial magneto rheological fluid devices
Int. Conf. on Electrorheological, Magnetorheological Suspensions and Associated Technology, Sheffield
[41] Carlsson JD and Chrzan MJ (1994)
Magnetorheological fluid dampers
Patent No. 5277281
[42] Carme C, Mellin V and Kronast M (2006)
Sound profiling for active noise control system
Proc. Euronoise Tampere 2006
[43] Cazzolato BS (1998)
J. of Acoustical Society of America 104 (1998), pp. 2878-2889
[44] Collet M, David P and Berthillier M (2006)
Active acoustical skin using distributed electrodynamical transducers
Proc. Euronoise Tampere 2006
[45] Chalasani RM (1986)
Ride performance potential of active suspension systems – part I: simplified analysis based on a quarter-car model
Proc. ASME Winter Annual Meeting, Los Angeles, ASME, New York
[46] Chalasani RM (1986)
Ride performance potential of active suspension systems – part II: comprehensive analysis based on a full-car model
Proc. ASME Winter Annual Meeting, Los Angeles, ASME, New York
[47] Chandrashekhara K and Agarwal AN (1993)
Active vibration control of laminated composite plates using piezoelectric devices: a Finite Element approach
J. of Intelligent Material Systems and Structures 4(4), pp. 496-508
[48] Charette F, Fuller CR and Carneal JP (1996)
Control of sound radiation from plates using globally detuned absorbers
J. of Acoustical Society of America 100(4):2781

[49] Cheng CC and Wang PW (2001)
Applications of the impedance method on multiple piezoelectric actuators driven structures
ASME J. of Vibration and Acoustics 123(2), pp. 262-268
[50] Choi SB and Hong SR (2004)
An electrorheological fluid-based plate for noise reduction in a cabin: experimental results
ASME J. of Vibration and Acoustics 125, pp. 537-541
[51] Choi SB, Seo JW and Kim JH (2001)
An electrorheological fluid-based plate for noise reduction in a cabin: experimental results
J. of Sound and Vibration 239(1), pp. 178-185
[52] Clark R and Cole DG (1995)
Active damping of enclosed sound fields through direct rate feedback control
J. of Acoustical Society of America 97(3), pp. 1710-1717
[53] Corr LR and Clark WW (2003)
A novel semi-active multi-modal vibration control law for a piezoceramic actuator
ASME J. of Vibration and Acoustics 125, pp. 214-222
[54] Cousin G (1998)
Sound from TBL-induced vibrations
AIAA Journal 98-2216
[55] Crawford DH and Stewart RW (1997)
Adaptive IIR filtered-v algorithm for active noise control
J. of Acoustical Society of America 101(4), pp. 2097-2103
[56] Crawley EF (1994)
Intelligent structures for aerospace: a technology overview and assessment
AIAA Journal 32(8), pp. 1689-1699
[57] Crawley EF and Lazarus KB (1991)
Induced strain actuation of isotropic and anisotropic plates
AIAA Journal 29(6), pp. 944-951
[58] Crawley EF and Anderson EH (1990)
Detailed models of piezoelectric isotropic and anisotropic plates
J. of Intelligent Material Systems and Structures 1(1), pp. 4-25
[59] Crawley EF and De Luis J (1987)
Use of piezoelectric actuators as elements of intelligent structures
AIAA Journal 25(10), pp. 1373-1385
[60] Crosby MJ and Karnopp DC (1973)
The active damper
Shock Vib. Bull. 43
[61] Cugueró MA et al. (2005)
Comparación de algoritmos feed-forward adaptivos para el control activo de ruido en conducto
Proc. Tecniacustica Terrassa 2005
[62] Darlington P (1987)
Loudspeaker circuit with means for monitoring the pressure at the speaker diaphragm, means for monitoring the velocity of the speaker diaphragm and a feedback circuit
Int. Patent No. PCT/WO97/03536
[63] Dai Y and Fuller C (1995)
Proc. Inter Noise 95, pp 533-536
[64] Dayou J and Brennan MJ (2003)
Experimental verification of the optimal tuning of a tunable vibration neutralizer for global vibration control
Applied Acoustics 64, pp. 311-323

[65] Dayou J and Brennan MJ (2001)
Optimal tuning for tunable vibration neutralizer
Proc. IMechE Part C: J. of Mechanical Engineering Science 215, pp. 933-942
[66] Dayou J (1999)
Global control of flexural vibration of a one dimensional structure using tunable vibration neutralizers
PhD degree thesis, University of Southampton
[67] Devasia S, Meressi T and Paden B (1993)
Piezoelectric actuator design for vibration suppression: placement and sizing
J. of Guidance, Control and Dynamics 16(5), pp. 859-864
[68] Dimitriadis EK, Fuller CR and Rogers CA (1991)
Piezoelectric actuators for distributed vibration excitation of thin plates
ASME J. of Vibration and Acoustics 113(1), pp. 100-107
[69] Dogan A, Tressler J and Newnham RE (2001)
Solid-state ceramic actuator designs
AIAA Journal 39(7), pp. 1354-1362
[70] Dorling CM et al. (1989)
J. of Sound and Vibration 128 (1989), pp. 358-360
[71] Dreyer A, Halfmann T, Brotz J, Kataja J and Antila M (2006)
Simulation system of an active noise control system
Proc. Euronoise Tampere 2006
[72] Elliott SJ (2001)
Signal processing for active control
Academic press, London, pp. 289-295
[73] Elliott SJ and Sutton TJ (1996)
Performance of feedforward and feedback systems for active control
IEEE Transactions on Speech and Audio Processing 4, pp. 214-223
[74] Elliott SJ, Nelson PA, Stothers IM and Boucher CC (1990)
In-flight experiments on the active control of propeller induced cabin noise
J. of Sound and Vibration 140 (1990), pp. 219–238
[75] Elliot SJ (1989)
J. of Sound and Vibration 128 (1989), pp. 355-357
[76] Elliot SJ et al. (1988)
J. of Sound and Vibration 120 (1988), pp. 183-189
[77] Elliot SJ et al. (1987)
J. of Sound and Vibration 117 (1987), pp. 35-58
[78] Emborg U (1998)
Cabin noise control in the Saab 2000 highspeed turboprop aircraft
Proc. ISMA 23 (1998), pp. 13–25
[79] Eriksson LJ (1998)
Development of the filtered-U algorithm for active noise control
J. of Acoustical Society of America 89, pp. 257-265
[80] Esmailzadeh E, Alasty A and Ohadi AR (2002)
Hybrid active noise control of a one-dimensional acoustic duct
ASME Transactions ASME J. of Vibration and Acoustics 124(1), pp. 10-18
[81] Fanson JL and Caughey TK (1990)
Positive position feedback control for large space structures
AIAA Journal 28(4), pp. 717-724
[82] Francis BA (1987)
A course in H_∞ control theory, in Lecture notes in control and information sciences
Thoma M, Wyner A, New York, Springer-Verlag 1987

[83] Friot E, Guillermin M and Winninger M (2006)
Active control of scattered acoustic radiation: a real-time implementation for a three-dimensional object
Acta Acustica united with Acustica Vol. 92, pp. 278-288
[84] Fuller CR, Johnson ME and Griffin J (2000)
Active-passive control of aircraft interior boundary-layer using smart foam
AIAA paper 2000-2041
[85] Fuller CR, Maillard JP, Mercadal M and Von Flotow AH (1997)
Control of aircraft interior noise using globally mistuned vibration absorbers
J. of Sound and Vibration 203(5), pp. 745-761
[86] Fuller CR, Hansen CH and Snyder SD (1991)
Experiments on active control of sound radiation form a panel using a piezoceramic actuator
J. of Sound and Vibration 150, pp. 179-190
[87] Fuller CR and Jones JD (1987)
Experiments on reduction of propeller induced interior noise by active control of cylinder vibration
J. of Sound and Vibration 112, pp. 389-395
[88] Fuller CR (1985)
Mechanisms of transmission and control of low frequency sound in aircraft interior
AIAA paper 85-0879
[89] Gao JX and Liao WH (2005)
Damping characteristics of beams with enhanced self-sensing active constrained layer treatments under various boundary conditions
ASME J. of Vibration and Acoustics 127, pp. 173-187
[90] Gerner C, Sachau D and Breitbach H (2004)
Aircraft interior ANC with flat panel speakers
Proc. SPIE Vol. 5388, n 1, pp. 266-75
[91] Gibbs G, Cabell R and Juang J (2000)
Controller complexity for active control of TBL-induced sound radiation from panels
AIAA paper 2000-2043
[92] Goldberg DE (1989)
Genetic algorithms in search, optimization and machine learning
Addison Wesley Longman, Reading, MA
[93] Gu Y, Clark RL and Fuller CR (1994)
Experiments on active control of plate vibration using piezoelectric actuators and polyvinylidene fluoride (PVDF) modal sensors
ASME J. of Vibration and Acoustics 116(3), pp. 303-308
[94] Guickling D, Karcher K and Rollage M (1985)
Coherent active method for application in room acoustics
J. of Acoustical Society of America 78(4), pp. 1426-1434
[95] Guigou C and Fuller CR (1999)
Control of aircraft interior broadband noise with foam-PVDF smart skin
J. of Sound and Vibration 220, pp. 541-557
[96] Guigou C, Maillard JP and Fuller CR (1996)
Study of globally detuned absorbers for controlling aircraft interior noise
4th Int. Conf. on Sound and Vibration St Petersburg 1996
[97] Gulyás K (2006)
Numerical modelling of panphonics G1 flat loudspeaker
Proc. Euronoise Tampere 2006
[98] Gurgose M (1984)
A note on the vibrations of restrained beams and rods with point masses
J. of Sound and Vibration 96, pp. 461-468

[99] Gustavsson M (1991)
Optimisation to determine the number and positions of secondary sources and sensors in SAAB 340
ASANCA I, 1991
[100] Hagiwara I, Wang DW, Shi QZ and Rao RS (2003)
Reduction of noise inside a cavity by piezoelectric actuators
ASME J. of Vibration and Acoustics 125, pp. 12-17
[101] Hagood NW and Von Flotow A (1991)
Damping of structural vibrations with piezoelectric materials and passive electrical networks
J. of Sound and Vibration 146(2), pp. 243-268
[102] Hagood NW, Chung WH and Von Flotow A (1990)
Modelling a piezoelectric actuator dynamics for active structural control
J. of Intelligent Material Systems and Structures 1, pp. 327-354
[103] Hansen CH and Snyder SD (1997)
Active control of noise and vibration
E&FN SPON, London, 1997
[104] Haykin S (1994)
Neural Networks, a Comprehensive Foundation
Prentice-Hall, Upper Saddle River (NJ-USA)
[105] Haykin S (1991)
Adaptive filter theory
Prentice-Hall Inc., 1991
[106] Heertjes M and Van de Wouw N (2006)
Nonlinear dynamics and control of a pneumatic vibration isolator
ASME J. of Vibration and Acoustics 128, pp. 439-448
[107] Heertjes M, De Graaff K and Van der Toorn JG (2005)
Active vibration isolation of metrology frames; a modal decoupled control design
ASME J. of Vibration and Acoustics 127, pp. 223-233
[108] Hill SG and Snyder SD (2002)
Design of an adaptive vibration absorber to reduce electrical transformer structural vibration
ASME J. of Vibration and Acoustics 124, pp. 606-611
[109] Hirsch SM, Jayachandran V and Sun JQ (1998)
Structural-acoustic control for quieter aircraft interior – Smart Trim Technology
Composite Structures 42, pp. 189-202
[110] Hollkamp JJ and Gordon RW (1995)
An experimental comparison of piezoelectric and constrained layer damping
Smart Structures and Integrated Systems, Proc. SPIE Smarte Structures and Materials 2445, Society of Photo-Optical Instrumentation Engineers, Bellingham, WA, pp. 123-133
[111] Hollkamp JJ (1994)
Multimodal passive vibration suppression with piezoelectric materials and resonant shunts
J. of Intelligent Material Systems and Structures 5, pp. 49-57
[112] Hopcropt R
DSTO-TR-1756, DODa-AR-013-471
[113] Hou H and Yang JH (2004)
An electroacoustics impedance error criterion and its application to active noise control
Applied Acoustics 65, pp. 485-499
[114] Hu Q and Ma G (2006)
Spacecraft vibration suppression using variable structure output feedback control and smart materials
ASME J. of Vibration and Acoustics 128, pp. 221-230

[115] Huang YM and Chen CC (2002)
Optimal design of dynamic absorbers on vibration and noise control of the fuselage
Comput. Struct. 76, pp. 691-702
[116] Huang YM and Fuller CR (1998)
Vibration and noise control of the fuselage via dynamic absorbers
ASME J. of Vibration and Acoustics 120, pp. 496-502
[117] Huang YM and Fuller CR (1997)
The effects of dynamic absorbers on the forced vibration of a cylindrical shell and its coupled interior sound field
J. of Sound and Vibration 200, pp. 401-418
[118] Hwang WS and Park HC (1993)
Hwang W vibration control of a laminated plate with piezoelectric sensor/actuator: Finite Element formulation and modal analysis
J. of Intelligent Material Systems and Structures 4(3), pp. 317-329
[119] Im S and Atluri SN (1989)
Effects of piezoactuator on a finitely-deformed beam subjected to general loading
AIAA Journal 27(12), pp. 1801-1807
[120] Ivers DE and Miller LR (1991)
Semiactive suspension technology: an evolutionary view
ASME Advanced Automotive Technologies, DE-40, Book No. H00719-1991, ASME, New York, pp. 327-346
[121] Jakob A, Stütz M, Feldman J and Möser M (2006)
Experiments on active vibration control of a scale model rail wheel
Proc. Euronoise Tampere 2006
[122] Johansson S, Sjösten P, Nordebo S and Claesson I (2000)
Comparison of multiple- and single-reference MIMO active noise control approaches using data measured in a Dornier 328 aircraft
Int. J. Acoustics and Vibrations 5 (2000), pp. 77–88
[123] Johansson S, Nordebo S and Claesson I (2000)
Convergence analysis of a twin-reference complex least-mean-squares algorithm
IEEE Transactions on Speech and Audio Processing 10, pp. 213–221
[124] Johansson S, Nordebo S, Sjösten P and Claesson I (2000)
A novel multiple-reference algorithm for active control of propeller induced noise in aircraft cabins
Research report no. 16/00, Blekinge Institute of Technology, department of Telecommunications and Signal Processing, Sweden, 2000.
[125] Johansson S, Claesson I, Nordebo S and Sjösten P (1999)
Evaluation of multiple reference active noise control algorithms on Dornier 328 aircraft data
IEEE Transactions on Speech and Audio Processing 7, pp. 473–477.
[126] Johansson et al. (1999)
Proc. Active 99, pp. 741-752
[127] Karnopp DC and Cosby MJ (1974)
System for controlling the transmission of energy between spaced members
US patent 3807678
[128] Kataja J and Mäkelä MM (2006)
Minimization of waterberd effect using a frequency-domain algorithm
Proc. Euronoise Tampere 2006
[129] Keltie RF and Cheng CC (1995)
Vibration reduction of a mass-loaded beam
J. of Sound and Vibration 187(2), pp. 213-228

[130] Kidner MRF and Brennan MJ (1999)
Improving the performance of a vibration neutralizer by actively removing damping
J. of Sound and Vibration 221(4), pp. 587-606
[131] Kidner MRF (1999)
An active vibration neutralizer
PhD degree thesis, University of Southampton
[132] Kim SJ, Han CH and Yun CY (1999)
Improvement of aeroelastic stability of hingeless helicopter rotor blade by passive piezoelectric damping
Smart Structures and Integrated Systems, Proc. SPIE Smarte Structures and Materials 3672, Society of Photo-Optical Instrumentation Engineers, Bellingham, WA, pp. 131-141
[133] Kirkpatrick S, Gelatt JCD and Vecchi MP
Optimization by simulated annealing
IBM Research Report RC 9355
[134] Kirkpatrick S, Gelatt JCD and Vecchi MP
Optimization by simulated annealing
Science 220, pp. 671–680
[135] Kitagawa Y and Midorikawa M (1998)
Seismic isolation and passive response control buildings in Japan
Smart Mat. Struct. 7(5), pp. 581-587
[136] Kleiner M and Svensson P (1995)
Review of active systems in room acoustics and electroacoustics
Proc. Active 95, Newport Beach, CA, pp. 39-54
[137] Kohgo S, Fujihara H and Seto K (2000)
Vibration control of coupled flexible thin structure with same property using filtered LQ control theory
Proc. 5th Int. Conf. on Motion and Vibration control Sydney 2000, pp. 873-878
[138] Konstanzer P, Grünewald M, Jänker P and Storm S (2006)
Piezo tuneable vibration absorber system for aircraft interior noise reduction
Proc. Euronoise Tampere 2006
[139] Krasnicki EJ (1980)
Comparison of analytical and experimental results for a semiactive vibration isolator
Shock Vib. Bull. 50
[140] Kronast M, Mellin V and Carme C (2006)
A sound quality active noise profiling system for a passenger test vehicle
Proc. Euronoise Tampere 2006
[141] Kuo SM and Morgan DR (1996)
Active noise control systems – algorithms and DSP implementations
John Wiley and Sons Inc., New York, 1996
[142] Lagö TL, Winberg M and Johansson S (1997)
AVIIS, active vibration isolation in ships: source identification and ANVC approach
Proc. Int. Congress on Sound and Vibration Adelaide 1997
[143] Lang MA, Lorch DR, May DN and Simpson MA (1992)
MD-80 aircraft cabin noise control: a case history
NASA/SAE/DLR 4th Aircraft Interior Noise Workshop 1992, pp. 13-33
[144] Langote PK and Seshu P (2005)
Experimental studies on active vibration control of a beam using hybrid active/passive constrained layer damping treatments
ASME J. of Vibration and Acoustics 127, pp. 515-518
[145] Laura PAA, Pombo JL and Susemihl EA (1975)
A note on the dynamics of an elastically restrained-free beam with a mass at the free end
J. of Sound and Vibration 41, pp. 397-405

[146] Lee CK, Chiang WW and O'Sullivan TC (1991)
Piezoelectric modal sensor/actuator pairs of critical active damping vibration control
J. of Acoustical Society of America 90(1), pp. 374-384
[147] Lee CK and Moon FC (1990)
Modal sensors and actuators
J. of Applied Mechanics 57(4), pp. 434-441
[148] Lee JH and Kim KJ (2005)
An efficient technique for design of hydraulic engine mount via design variable-embedded damping modeling
ASME J. of Vibration and Acoustics 127, pp. 93-99
[149] Lee YY and Yao J (2003)
Structural vibration suppression using piezoelectric sensors and actuators
ASME J. of Vibration and Acoustics 125, pp. 109-113
[150] Liang C, Sun FP and Rogers CA (1994)
An impedance method for dynamic analysis of active material system
ASME J. of Vibration and Acoustics 116, pp. 121-128
[151] Lim YU et al. (1999)
Smart Material Structures 8 (1999), pp. 324-337
[152] Lin JY and Luo ZL (2000)
Internal model-based LQC/H_∞ design of robust active noise controllers for acoustical duct systems
IEEE Transactions on Control Systems Tech. 8, pp. 864-872
[153] Lissec H and Meynial X (2003)
A preliminary study of an isodynamic transducer for use in active acoustic materials
Applied Acoustics 64, pp. 917-930
[154] Lissec H (2002)
Les matériaux actifs à propriétés acoustiques variables
PhD thesis, Université du Maine
[155] Liu ZS, Lee HP and Lu C (2006)
Passive and active interior noise control of box structures using the structural intensity method
Applied Acoustics 67, pp. 112-134
[156] Liu ZS et al. (2004)
Vibration analysis of non-classically damped linear systems
ASME J. of Vibration and Acoustics 126, pp. 456-458
[157] Loewy RG (1997)
Recent developments in smart structures with aeronautical applications
Smart Materials and Structures 6(5), pp. R41-R42
[158] Luo J and Gea HC (2003)
Optimal stiffener design for interior sound reduction using a topology optimization based approach
ASME J. of Vibration and Acoustics 125, pp. 267-273
[159] Martin T and Roure A (1998)
Active noise control of acoustic sources using spherical harmonics expansion and a genetic algorithm: Simulation and experiment
J. Sound and Vibration 212 (1998), pp. 511–523
[160] Manikanahally DN and Crocker MJ (1988)
Vibration analysis of hysteretically damped mass-loaded beams
J. of Sound and Vibration 132, pp. 177-197
[161] Maury C, Gardonio P and Elliot SJ (2001)
Active control of the flow induced noise transmitted through a panel
AIAA Journal 39(10), pp. 1860-1867

[162] Mayer D et al. (2006)
Modeling of an active engine mount system for automotive applications
Proc. Euronoise Tampere 2006
[163] Meirovtch L (1990)
Dynamics and control of structures
Wiley Interscience, New York
[164] Meynial X and Lissek H (1999)
Active reflectors for room acoustics
Proc. IOA 1999, 21(6), pp. 99-107
[165] Meynial X and Vian JP (1998)
Active walls for room acoustics
Proc. Int. Convention on Sound Design Karlsruhe 1998, pp. 134-141
[166] Meynial X (1996)
Active materials for application in room acoustics
Proc. 3rd ICIM/ECSSM Lyon 96, pp. 968-973
[167] Miller LR (1988)
An approach to semiactive control of multiple degree-of-freedom systems
PhD thesis, North Carolina State University, Raleigh, NC
[168] Miller SE, Oshman Y and Avramorich H (1996)
Modal control of piezolaminated anisotropic rectangular plates. Part 1: Modal transducer theory
AIAA Journal 34(9), pp. 1868-1875
[169] Mixson JS and Wilby JF (1995)
Interior noise, in Aeroacoustic of flight vehicles, theory and practice
H.H. Hubard, NASA reference publication 1258, pp. 271-355
[170] Mohring J and Wirsen A (2006)
Robust controller design based on a integrated simulation of an ANVC-system
Proc. Euronoise Tampere 2006
[171] Moon SH and Kim SJ (2001)
Active and passive suppressions of non linear panel flutter using Finite Element Method
AIAA Journal 39(11), pp. 2042-2050
[172] Mouzakitis S and Provatidis C (2006)
Use of the Finite Element Method for the optimum selection of ANC sensors and actuators locations
Proc. Euronoise Tampere 2006
[173] Nelson PA and Elliot SJ (1992)
Active control of sound
Academic Press
[174] Nelson PA, Curtis ARD, Elliot SJ and Bullmore AJ (1987)
The active minimisation of harmonic enclosed sound fields, Part I: theory
J. of Sound and Vibration 117, pp. 1-13
[175] Newbury KM and Leo DJ (2001)
Structural dynamics of stiffened plates with piezoceramic sensors and actuators
AIAA Journal 39(5), pp. 942-950
[176] Ng SC, Leung SH, Chung CY, Luk A and Lau WH (1996)
The genetic search approach
IEEE Signal Processing Magazine (1996)
[177] Noiseux DU (1970)
Measurement of power flow in uniform beams and plates
J. of Acoustical Society of America 47, pp. 238-247

[178] Nordebo S, Claesson I, Lagö T, Lennerstad H and Persson P (1997)
Appendix to ANC configuration design for the SAAB 340 ANC verification test
Deliverable 68/2, ASANCA II, the BRITE/EURAM research programme, 1997
[179] Ozer MB and Royston TJ (2005)
Extending Den Hartog's vibration absorber technique to multi-degree-of-freedom systems
ASME J. of Vibration and Acoustics 127, pp. 341-350
[180] Park G, Farrar CR, Rutherford AC and Robertson AN (2006)
Piezoelectric active sensor self-diagnostics using electrical admittance measurements
ASME J. of Vibration and Acoustics 128, pp. 469-476
[181] Pavic G (1976)
Measurement of structure borne wave intensity, part I: formulation of the methods
J. of Sound and Vibration 49, pp. 221-230
[182] Persson P, Nordebo S and Claesson I (2001)
Hardware efficient digital filter design by multi-mode mean field annealing
IEEE Signal Processing Letters 8 (2001)
[183] Persson P, Lagö TL and Claesson I (1999)
Active vibration reduction in a light weight high speed train boogie
Proc. IFAC99 Beijing 1999
[184] Peterson C and Söderberg B (1989)
A new method for mapping optimization problems onto neural networks
Int. J. of Neural Systems 1 (1989), pp. 3–22
[185] Philen MK and Wang KW (2005)
Active stiffeners for vibration control of a circular plate structure: analytical and experimental investigations
ASME J. of Vibration and Acoustics 127, pp. 441-450
[186] Poulin KC and Vaicaitis R (2004)
Vibrations of stiffened composite panels with smart materials
ASME J. of Vibration and Acoustics 126, pp. 370-379
[187] Quehl J, Schick A, Mellert V and H Remmers H (2001)
Interaction of sound and vibration on comfort evaluation of aircraft flight situations
Proc. DAGA 2001, Hamburg
[188] Rafaely B and Elliot SJ (1999)
H_2/H_∞ active control of sound in a headset: design and implementation
IEEE Transactions on Control Systems Tech. 7, pp. 79-84
[189] Rao MD (2003)
Recent applications of viscoelastic damping for noise control in automobiles and commercial airplanes
J. of Sound and Vibration 262, pp. 457-474
[190] Rao SS (1995)
Mechanical vibration, 3rd ed.
Addison-Wesley, New York, pp. 527, 603-607
[191] Rao SS and Sunar M (1994)
Piezoelectricity and its use in disturbance sensing and control of flexible structures: a survey
Applied Mechanics Review 47(4), pp. 113-123
[192] Reichert BA (1997)
Application of magnetorheological dampers for vehicle seat suspensions
MS thesis, Virginia Polytechnic Institute and State University, Blacksburg, VA
[193] Rosen J (2003)
J. of Biomechanical Engineering 125 (2003), pp. 223-231
[194] Ross C (1999)
Active noise control in aircraft
Proc. Sixth Int. Congress on Sound and Vibration (1999), pp. 1611–1618

[195] Ross C (1998)
A comparison of techniques for the active control of noise and vibration in aircraft
Proc. ISMA 23 (1998), pp. 831–835
[196] Ross CF and Purver MRJ (1997)
Active cabin noise control
Proc. Active 97 Budapest, pp. 39-46
[197] Saravanos DA (2000)
Passively damped laminated piezoelectric shell structures with integrated electric networks
AIAA Journal 38(7), pp. 1260-1268
[198] Scott RC and Weisshaar TA (1994)
Controlling panel flutter using adaptive materials
Journal of Aircraft 31(1), pp. 213-222
[199] Seto K, Ooshima A and Fujii H (1998)
Active vibration control of a bridge tower structure using filtered LQ control technique
Proc. 4th Int. Conf. on Motion and Vibration Control Zürich 1998, pp. 545-551
[200] Shynk JJ (1992)
Frequency-domain and multirate adaptive filters
IEEE SP Magazine (January 1992), pp. 14–37
[201] Simpson MT and Hansen CH (1996)
Use of genetic algorithms to optimize vibration actuator placement for active control of harmonic interior noise in a cylinder with floor
Noise Control Engineering Journal 44, pp. 169-184
[202] Sipahi R and Olgac N (2003)
Active vibration suppression with time delayed feedback
ASME J. of Vibration and Acoustics 125, pp. 384-388
[203] Sjösten P, Johansson S, Persson P, Claesson I and Nordebo S (2003)
Considerations on large applications of active noise control. Part I: Theory
Acta Acustica united with Acustica Vol. 89, pp. 822-833
[204] Sjösten P, Johansson S, Persson P and Claesson I (2003)
Considerations on large applications of active noise control. Part II: Experimental results
Acta Acustica united with Acustica Vol. 89, pp. 834-843
[205] Sjösten P et al. (1997)
Active vibration control on a light high speed train: Preliminary results
Proc. Active 97 Budapest
[206] Sjösten P, Johansson S and Lagö TL (1996)
Active noise control in a twin–engine patrol boat
Proc. Inter–Noise Liverpool 1996
[207] Snyder SD and Hansen CH (1994)
The design of systems to control actively periodic sound transmission into enclosed spaces, Part I: analytical models
J. of Sound and Vibration 170, pp. 433-449
[208] Snyder SD et al. (1994)
J. of Sound and Vibration 170, pp. 451-472
[209] Sodano HA, Inman DJ and Belvin WK (2006)
Development of a new passive-active magnetic damper for vibration suppression
J. of Sound and Vibration 128, pp. 318-327
[210] Song X, Ahmadian M, Southward S and Miller LR (2005)
An adaptive semiactive control algorithm for mangetorheological suspension systems
ASME J. of Vibration and Acoustics 127, pp. 493-502
[211] Song X (1999)
Design of adaptive vibration control systems with application to magnetorheological dampers
PhD dissertation, Virginia Polytechnic Institute and State University, Blacksburg, VA

[212] Sorrentino A and Concilio A (2003)
Identification of a vibro-acoustic Comfort Index for aircraft: the IDEA PACI project
Acta Acustica 2003, S11, Proc. Euronoise Naples 2003, paper 538
[213] Sun D, Tong L and Wang D (2001)
Vibration control of plates using discretely distributed piezoelectric quasi-modal actuators/sensors
AIAA Journal 39(9), pp. 1766-1772
[214] Sun DC, Wang DJ and Xu ZL (1999)
Distributed piezoelectric element method for vibration control of smart plates
AIAA Journal 37(11), pp. 1459-1463
[215] Sun DC, Wang DJ and Xu ZL (1997)
Distributed piezoelectric segment method for vibration control of smart beams
AIAA Journal 35(3), pp. 583-584
[216] Swingbanks MA (1973)
The active control of sound propagating in long ducts
J. of Sound and Vibration 27, pp. 411-437
[217] Tang J and Wang KW (2004)
Vibration confinement via optimal eigenvector assignment and piezoelectric networks
ASME J. of Vibration and Acoustics 126, pp. 27-36
[218] Tang J, Wang KW and Philen M (1999)
Sliding mode control of structural vibrations via active-passove hybrid piezoelectric network
Smart Structures and Integrated Systems, Proc. SPIE Smarte Structures and Materials 3668, Society of Photo-Optical Instrumentation Engineers, Bellingham, WA, pp. 543-553
[219] Tewes S, Maier R and Peiffer A (2006)
Active control of sound transmission through aircraft structures – modelling and system design
Proc. Euronoise Tampere 2006
[220] Thomas DR, Nelson PA and Elliot SJ (1993)
Active control of transmission of sound through a thin cylindrical shell. Part I: the minimization of vibration energy
J. of Sound and Vibration 167, pp. 91-111
[221] Thomas DR, Nelson PA and Elliot SJ (1993)
Active control of transmission of sound through a thin cylindrical shell. Part II: the minimization of acoustic potential energy
J. of Sound and Vibration 167, pp. 112-128.
[222] Tsai MS and Wang KW (1999)
On the structural damping characteristics of active piezoelectric actuators with passive shunt
J. of Sound and Vibration 221(1), pp. 1-22
[223] Tseng WK, Rafaely B and Elliot SJ (1998)
Combined feedback-feedforward active control of sound in a room
J. of Acoustical Society of America 10(4), pp. 3417-3425
[224] Tzou HS, Chai WK and Arnold SM (2006)
Structonics and actuation of hybrid electrostrictive/piezoelectric thin shells
J. Vibration and Acoustics 128, pp. 79-87
[225] Tzou HS, Zhong JP and Hollkamp JJ (1994)
Spatially distributed orthogonal piezoelectric shell actuators: theory and applications
J. of Sound and Vibration 174(3), pp. 363-378
[226] Tzou HS (1992)
Active piezoelectric shell continua, in Intelligent structural systems
H.S. Tzou and G.L. Anderson, Kluwer Academic, Norwell, MA, pp. 9-74

[227] Tzou HS and Tseng CI (1991)
Distributed modal identification and vibration control of continua: theory, application and analysis
J. of Dynamic Systems, Measurements and Control 113(3), pp. 494-499
[228] Tzou HS and Gadre M (1989)
Theoretical analysis of a multi-layered thin shell coupled with piezoelectric shell actuators for distributed vibration controls
J. of Sound and Vibration 132(2), pp. 433-450
[229] Unruth JF (1988)
J. of Vibration, Acoustics, Stress and Reliability in Design 110 (1988), pp. 226-233.
[230] Van der Auweraer et al. (2006)
CAE approach for the design of smart structures applications
Proc. Euronoise Tampere 2006
[231] Varadan VV et al. (1996)
Smart Materials and Structures 5 (1996), pp. 685-694
[232] Västfjäll D, Kleiner M and Gärling T (2003)
Affective reactions to interior aircraft sounds
Acta Acustica united with Acustica Vol. 89, pp. 693-701
[233] Vecchio A, Rimondi M, Janssens K, Hovmand P and Anders F (2006)
Hybrid modelling approach to predict engine noise reduction in passenger trains
Proc. Euronoise Tampere 2006
[234] Vel SS and Baillargeon BP (2005)
Analysis of static deformation, vibration and active damping of cylindrical composite shells with piezoelectric shear actuators
ASME J. of Vibration and Acoustics 127, pp. 395-407
[235] Vel SS and Batra RC (2000)
Cylindrical bending of laminated plates with distributed and segmented piezoelectric actuators/sensors
AIAA Journal 38(5), pp. 857-867
[236] Vidyasagar M (1985)
Control system synthesis: a factorisation approach
Cambridge, MA, MIT Press
[237] Von Flotow A and Mercadal M (1995)
The measurement of noise and vibration transmitted into aircraft cabins
Sound and Vibration (1995), pp. 16–19.
[238] Wang BT and Rogers CA (1991)
Modelling of finite length spatially-distributed induced strain actuators for laminated beams and plates
J. of Intelligent Material Systems and Structures 2(1), pp. 38-58
[239] Wang KW and Kahn SP (1995)
Structural vibration controls via active-passive piezoelectrical networks – a hybrid approach
Proc. Int. Conf. On Structural Dynamics, Vibration, Noise and Control, Hong Kong, pp. 1059-1064
[240] Wang PW and Cheng CC (2006)
Design of vibration absorbers for structures subject to multiple-tonal excitations
ASME J. of Vibration and Acoustics 128, pp. 106-114
[241] Wang PW and Cheng CC (2005)
Natural frequency tuning using structural patches
ASME J. of Vibration and Acoustics 127, pp. 28-35

[242] Wang X and Mills JK (2005)
FEM dynamic model for active vibration control of flexible linkages and its application to a planar parallel manipulator
Applied Acoustics 66, pp. 1151-1161
[243] Wesselink JM and Berkhoff AP (2006)
Optimization of the reconstruction and anti-aliasing filter in a Wiener filter system
Proc. Euronoise Tampere 2006
[244] Widrow B, McCool J and Ball M (1975)
The complex LMS algorithm
Proc. IEEE (1975), pp 719–720
[245] Williams EG (1991)
Structural intensity in thin cylindrical shells
J. of Acoustical Society of America 89, pp. 1615-1622
[246] Winkler G (1995)
Image analysis, random fields and dynamic Monte Carlo methods
Springer–Verlag, 1995
[247] Yang S and Ngoi B (2000)
Shape control of beams by piezoelectric actuators
AIAA Journal 38(12), pp. 2292-2298
[248] Yuan J (2006)
Active resonators for noise absorption
ASME J. of Vibration and Acoustics 128, pp. 115-121
[249] Yuan J (2005)
Active suppression of narrowband noises in 3D fields
Applied Acoustics 66, pp. 89-103
[250] Yuan J (2004)
A hybrid active noise controller for finite ducts
Applied Acoustics 65, pp. 45-57
[251] Yuan J (2002)
Global damping of noise or vibration fields with locally synthesized controllers
J. of Acoustical Society of America 111(4), pp. 1726-1733
[252] Yuan J (2000)
Relaxed condition for ‘perfect’ cancellation of broadband noise in 3D enclosures
J. of Acoustical Society of America 107(6), pp. 3235-3244
[253] Zander AC (1994)
PhD
University of Adelaide
[254] Zhang X and Erdman AG (2006)
Optimal placement of piezoelectric sensors and actuators for controlled flexible linkage mechanisms
ASME J. of Vibration and Acoustics 128, pp. 256-260
[255] Zhao C, Chen L and Chen D (2006)
Semi-active static output feedback variable structure control for two-stage vibration isolation system
ASME J. of Vibration and Acoustics 128, pp. 627-634
[256] Zhou RC, Lai Z, Xue DY, Hauang JK and Mei C (1994)
Suppression of nonlinear panel flutter with piezoelectric actuators using Finite Element
AIAA Journal 33(6), pp. 1098-1105
[257] Zhou SS, Liang C and Rogers CA (1996)
An impedance-based system modelling approach for induced strain actuator-driven structures
ASME J. of Vibration and Acoustics 123(2), pp. 262-268

[258] Zuo L and Nayfeh SA (2005)
Optimisation of the individual stiffness and damping parameters in multiple-tuned-mass-damper-systems
ASME J. of Vibration and Acoustics 127, pp. 77-83
[259] Zwicker E and Fast H (1999)
Psychoacoustics
Springer-Verlag, Berlin