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A state-space approach for ASAC simulation

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Abstract: The demands for improvement in sound quality and reduction of noise generated by vehicles are constantly increasing, as well as the penalties for space and weight of the control solutions. A promising approach to cope with this challenge is the use of active structural-acoustic control (ASAC). Usually, the low frequency noise is transmitted into the vehicle's cabin through structural paths, which raises the necessity of dealing with vibro-acoustic models. This kind of models should allow the inclusion of sensors and actuators models, if accurate performance indexes are to be accessed. The challenge thus resides in deriving reasonable sized models that integrate structural, acoustical, and even electrical components and the controller algorithm. The advantages of adequate ASAC simulation strategies relies on the cost and time reduction in the development phase. Therefore, the aim of this paper is to present a methodology for simulating vibro-acoustic systems including this coupled model in a closed loop control simulation that also takes into account the interaction between the system and the control actuators and sensors. It is shown that neglecting the sensor/actuator dynamics can lead to inaccurate performance predictions.

Keywords: vibro-acoustics, active-control, state-space

NOMENCLATURE

[b] = matrix composed by zeros and ones for state-space formulation
C = damping matrix
[c] = matrix composed by zeros and ones for state-space formulation
c = lumped damping
F = vector of loads
I = identity matrix
I = current
j = imaginary number
K = stiffness matrix
k = lumped stiffness
kf = shaker electric constant
M = mass matrix
m = lumped mass
N = number of modes
p = vector of acoustic pressure
q = modal amplitudes
u = vector of structural displacement
u = displacement
v = velocity
x = vector of states
y = vector of outputs

Subscripts

~ modal values
a acoustical
bemf back electro-magnetic force
c coupled
i inertial shaker
int interface
s structural
r mode index
L left
R right
T matrix transpose

Greek Symbols

Φ = mode shape
Γ = mass weighted damping
ρ = density
ω = frequency
Ω = diagonal matrix of natural frequencies

INTRODUCTION

The demands for improvement in sound quality and reduction of noise generated by vehicles are constantly increasing, as well as the penalties for space and weight of the passive control solutions. A promising approach to cope with this challenge is the use of active structural-acoustic control (ASAC). During the design stage, simulation plays an important role in predicting the performance and feasibility of active control solutions. As a result, the demands for more reliable simulation techniques are also ever-increasing. The advantages of adequate simulation strategies relies on the time and cost reduction in the development phase, enabling the engineer to try different strategies, sensors and actuators configuration and control strategies with minimum physical prototyping.

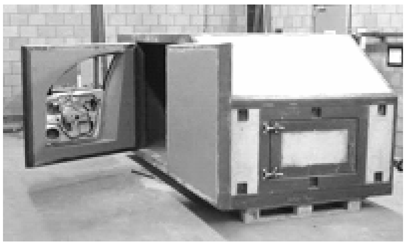
Usually, the low frequency noise is transmitted into the vehicle's cabin through structural paths, which raises the necessity of dealing with vibro-acoustic models. This kind of models allows the use of acoustic disturbance and even secondary sources. It also should allow the inclusion of sensors and actuators models, not only in the sense that their own dynamics may significantly change the original system's physical properties, but also that they can present frequency, phase or amplitude limitations to the control performance. The challenge thus resides in deriving reasonable sized models that integrate structural, acoustical, and even electrical components and the controller algorithm.

The aim of this paper is to present a methodology for simulating vibro-acoustic systems including this coupled model in a closed loop control simulation that also takes into account the interaction between the system and the control actuators and sensors. Such methodology consists of deriving a fully coupled finite element (FE) model of the vibro-acoustic system which is further reduced and exported as a state-space model into Matlab/Simulink, where models for sensors and actuators are included and the controller implemented.

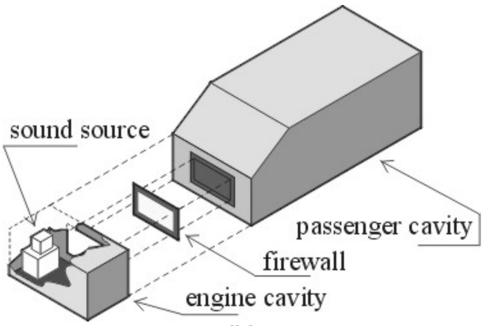
MODELLING PROCEDURE OVERVIEW

Bringing research results on intelligent materials to the level of industrial application requires the design processes of active systems to become part of the complete product development cycle. In other words, it is necessary that the product functional performance simulation models, which are the cornerstone of today's design process, must be capable of supporting the specific aspects of advanced materials, active systems, actuators, sensors and control algorithms. Moreover, it must be possible to integrate these models into system level virtual prototype models (Van der Auweraer *et al.*, 2006). This task involves developing modelling capabilities for the intelligent material systems, sensor and actuator components and control systems as well as for their integration in system-level application design. It is clear that no single integrated solution will be able to fulfil all requirements of the various material and control approaches; therefore the focus of the research is on supporting, as much as possible, the use, combination and extension of existing codes and tools.

To demonstrate the proposed modelling procedure, the Sound Brick set-up at LMS.International was selected. It consists of a simplified car cavity, built-up in concrete to ensure acoustic boundary condition, thus avoiding uncertainties during the vibro-acoustic modelling phase. A previous version of the setup is shown in Fig.1a. The current configuration has a flexible firewall and a rigid engine compartment (EC) as in Fig.1b. A sound source placed in the EC works as a primary disturbance source. The firewall is assembled between the EC and the passenger cavity (PC) which allows a set of structural sensors and actuators to realize the control signals for increasing the transmission loss through these cavities. The PC main dimensions are about 3400 x 1560 x 1270mm; for the EC, 900 x 1100 x 750mm and the firewall 920 x 545 x 1.5mm. Collocated accelerometers and inertia-shakers are used as structural sensors/actuators pairs (SAPs).



(a)



(b)

Figure 1 – System under investigation (a) photo (b) schematic view

Due to the necessity of having acoustic excitation of the firewall from a source placed in the EC and the further radiation of the firewall into the PC, the modelling approach converged to a coupled vibro-acoustic FE/FE model. Another advantage of using a fully-coupled approach is the accuracy of the estimated performance, as an uncoupled analysis can overestimate the controller efficiency (Mohammed & Elliott, 2005). The different model types and software for each component are show in Tab.1. The full modelling procedure is illustrated in Fig.2.

The procedure is based on a modal model of the coupled system. It starts with the structural FE model, to which the structural influence of placing sensors and actuators (mass and/or stiffness loads) is taken into account before a modal base is extracted (Fig.2). This modal base is then used, together with the acoustic FE modal base, to calculate the coupled vibro acoustic modal base. In a further step, the vibroacoustic state-space model is derived from the coupled modal base. If the SAP electrical dynamics are considered to be neglectable, this modal based state-space model already allows the implementation of control systems with idealised inputs and outputs. Some preliminary results can be achieved with such models, although a more accurate access to the controller performance requires a more comprehensive model of the SAP, which in this case takes place in the controller design environment (Matlab/Simulink). The next sub-sections describe in more detail each one of the required steps.

Table 1 – Component model type and software

component	model type	Software
cavities	FEM	LMS.Sysnoise
firewall	FEM	Patran / Nastran
inertia-shaker	State-Space	Matlab / Simulink
controller	State-Space	Matlab / Simulink

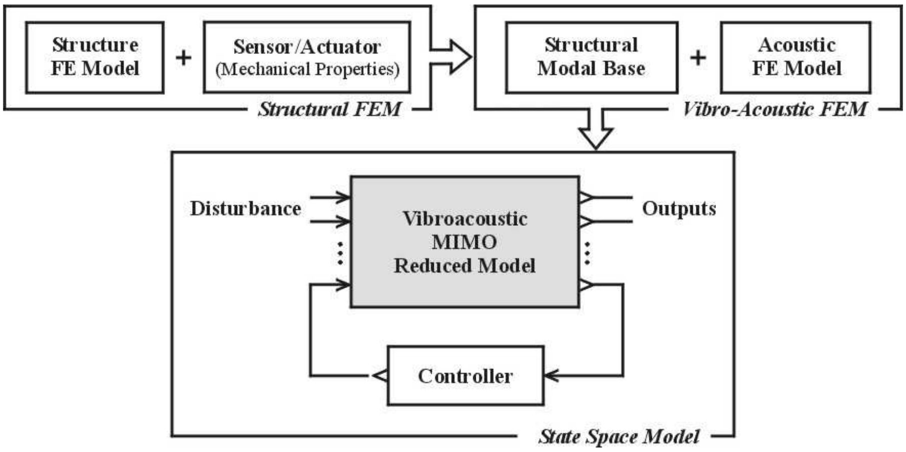
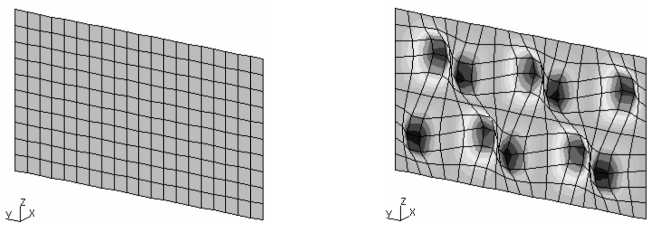


Figure 2 – Modelling procedure scheme

Vibro-acoustic modelling

The prediction of vehicles vibro-acoustic behaviour through FE modelling always requires the set-up of FE meshes for both, vehicle structure and interior cavity. Since the acoustic wavelengths are usually longer than structural ones, optimized acoustic meshes can be much coarser than the structural meshes. However, if the meshes are compatible, some intermediate numerical steps can be neglected resulting in a simplified procedure (Coyette & Dubois-Pèlerin, 1994). Therefore a trade-off choice for the size of structural and acoustic FE mesh was taken. In this way they present coinciding nodes without affecting the accuracy of the predicted results within the frequency range of interest (0 to 200Hz).

The structural mesh is shown in Fig. 3(a) with a total of 231 nodes and 200 elements (4-noded shell elements). The steel firewall, 1.5mm tick, presents 13 modes from 0 to 200Hz and the higher order mode is the [5,2] well described by this number of elements as it can be seen in Fig.3(b). The inclusion of sensors and actuators mass (and/or stiffness) loading takes place in this step, by adding some concentrated mass to the nodes where the SAP is to be sited.



(a) **(b)**

Figure 3 – Firewall: (a) FE mesh and (b) mode [5,2]

The element type chosen for the acoustic mesh is the 8-noded brick (hexahedral), not only by the smaller number of elements needed, but also due to its higher accuracy in post processing pressure derived quantities (velocity and sound intensity) when compared to tetrahedral elements (El-Masri, *et al.*, 2002). The maximum frequency of interest (200Hz for this study case) was also taken into account, so that the acoustic model could have the minimum number of elements. The resulting mesh (around 30000 nodes and 25000 elements) and the highest mode shape in the frequency band are depicted in Fig.4. With respect to the elements size, this acoustic model is valid until 308Hz, considering 10 elements per wavelength (514Hz with 6 elem./wavelength), which is fairly suitable for this application.

Eventually, the structural and acoustic models are fully-coupled in this FE/FE approach. The effect of the interfacing fluid on the structure dynamics can be considered as a pressure load on the wet surface, thus turning the well known structural differential equation into the form of Eq. (1).

$$\left(\mathbf{K_s} + j\omega\mathbf{C_s} - \omega^2\mathbf{M_s}\right)\mathbf{u}(\omega) + \mathbf{K_c}\mathbf{p}(\omega) = \mathbf{F_s}(\omega) \tag{1}$$

where $\mathbf{K}_s$, $\mathbf{C}_s$ and $\mathbf{M}_s$ are respectively the structural stiffness, damping and mass matrices, $\mathbf{K}_c$ is the coupling matrix, $\mathbf{u}$ is the vector of structural displacements, $\mathbf{p}$ is the vector of acoustic pressures and $\mathbf{F}_s$ is the structural load vector.

In a similar way, the structural vibration works as an extra acoustic input and therefore must be taken into account as:

$$\left(\mathbf{K_a} + j\omega \mathbf{C_a} - \omega^2 \mathbf{M_a}\right)\mathbf{p}(\omega) - \omega^2 \mathbf{M_c}\mathbf{u}(\omega) = \mathbf{F_a}(\omega) \tag{2}$$

where $\mathbf{K_a}$, $\mathbf{C_a}$ and $\mathbf{M_a}$ are the acoustical stiffness, damping and mass matrices, $\mathbf{M_c}$ is the coupling matrix, and $\mathbf{F_a}$ is the acoustic load vector. For the sake of brevity any function '$h(\omega)$' is represented just as 'h' hereafter.

Regarding the special relation between $\mathbf{K_c}$ and $\mathbf{M_c}$ (Desmet & Vandepite, 2004), the combined system of equations, known as the Eulerian FE/FE model, yields:

$$\left(\begin{bmatrix} \mathbf{K_s} & \mathbf{K_c} \\ \mathbf{0} & \mathbf{K_a} \end{bmatrix} + j\omega \begin{bmatrix} \mathbf{C_s} & \mathbf{0} \\ \mathbf{0} & \mathbf{C_a} \end{bmatrix} - \omega^2 \begin{bmatrix} \mathbf{M_s} & \mathbf{0} \\ -\rho_a \mathbf{K_c}^T & \mathbf{M_a} \end{bmatrix}\right)\begin{Bmatrix} \mathbf{u} \\ \mathbf{p} \end{Bmatrix} = \begin{Bmatrix} \mathbf{F_s} \\ \mathbf{F_a} \end{Bmatrix} \tag{3}$$

Based on Eq.(3) it is clear that the resulting vibro-acoustic system is indeed coupled, though it is not symmetric. The main outcome of such non-symmetric nature is that the eigenproblem associated to this modified FE model is also non-symmetric, hence resulting in distinct left and right eigenvectors.

Since the damping matrix is uncoupled, a normal mode computation can handle this problem. Moreover, it has been indicated by Luo & Gea (1997) that, particularly for the Eulerian formulation (Eq.3), the left and right eigenvector, denoted here respectively by $\mathbf{\Phi_L}$ and $\mathbf{\Phi_R}$ can be related as:

$$\mathbf{\Phi_L} = \begin{Bmatrix} \mathbf{\Phi_{Ls}} \\ \mathbf{\Phi_{La}} \end{Bmatrix} = \begin{Bmatrix} \mathbf{\Phi_{Rs}} \mathbf{\Omega}^2 \\ \mathbf{\Phi_{Ra}} \end{Bmatrix} \tag{4}$$

where the sub-indexes **s** and **a** denote the structural and acoustical components.

When a coupled modal analysis is performed in LMS.Sysnoise, the resultant modal base is actually the right eigenvectors. Thanks to the relation in Eq.(4) it is possible to retrieve the left eigenvectors to build the complete modal model, which will be the base for the state space formulation described in more detail in a further section.

Model Reduction and State Space models

As mentioned before, due to the way fluid structure interaction is usually modelled in FE codes, the matrices are non-symmetric as shown in Eq.(3). Starting from the system generalized state space representation Eq.(5):

$$\dot{\mathbf{x}} = \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\mathbf{M}^{-1}\mathbf{K} & -\mathbf{M}^{-1}\mathbf{C} \end{bmatrix}\mathbf{x} + [b]\mathbf{F}, \qquad \mathbf{y} = [c]\mathbf{x} \tag{5}$$

where $\mathbf{M}$, $\mathbf{C}$ and $\mathbf{K}$ are respectively the mass, damping and stiffness matrices, $\mathbf{x}$ is the vector of states, $\mathbf{f}$ is the load vector, $\mathbf{y}$ is the output vector and $[b]$ and $[c]$ are rectangular matrices with ones on the desired DoFs positions and zeros everywhere else. It is possible to project Eq.(5) to the left and right eigenvectors from eigenvalue problem in Eq.(6) to find the reduced system in Eq.(7):

$$\mathbf{\Phi_L^T}\,\mathbf{M}\,\mathbf{\Phi_R} = \mathbf{I} \quad \text{and} \quad \mathbf{\Phi_L^T}\,\mathbf{K}\,\mathbf{\Phi_R} = \mathbf{\Omega}^2 \tag{6}$$

$$\begin{aligned} \begin{Bmatrix} \dot{\mathbf{q}} \\ \ddot{\mathbf{q}} \end{Bmatrix} &= \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\mathbf{\Omega}^2 & -\mathbf{\Gamma} \end{bmatrix}\begin{Bmatrix} \mathbf{q} \\ \dot{\mathbf{q}} \end{Bmatrix} + \begin{bmatrix} \mathbf{0} \\ \mathbf{\Phi_L^T}[b] \end{bmatrix}\mathbf{F} \\ \mathbf{y} &= \begin{bmatrix} [c]\mathbf{\Phi_R} & \mathbf{0} \end{bmatrix}\begin{Bmatrix} \mathbf{q} \\ \dot{\mathbf{q}} \end{Bmatrix} \end{aligned} \tag{7}$$

Equation (9) shows the modal formulation after the modal expansion in terms of the coupled right eigenvectors furnished by LMS.Sysnoise (Eq.8):

$$\begin{Bmatrix} \mathbf{u} \\ \mathbf{p} \end{Bmatrix} = \sum_{r=1}^{Nc} q_r \begin{Bmatrix} {}_r\mathbf{\Phi_{Rs}} \\ {}_r\mathbf{\Phi_{Ra}} \end{Bmatrix} = \mathbf{\Phi_R}\mathbf{q} \tag{8}$$

$$\left(\tilde{\mathbf{K}} + j\omega\tilde{\mathbf{C}} - \omega^2\tilde{\mathbf{M}}\right)\mathbf{q} = \tilde{\mathbf{F}} \tag{9}$$

where $\tilde{\mathbf{K}}$, $\tilde{\mathbf{C}}$ and $\tilde{\mathbf{M}}$ are the stiffness, damping and mass modal matrices, $\mathbf{q}$ is the vector of modal amplitudes and $\tilde{\mathbf{F}}$ is defined as:

$$\tilde{\mathbf{F}} = \mathbf{\Phi}_{\mathbf{L}}^T \begin{Bmatrix} \mathbf{F_s} \\ \frac{1}{\rho}\mathbf{F_a} \end{Bmatrix} \tag{10}$$

Since the left eigenvector can be calculated based on the right eigenvector (Eq.4), the modal formulation can be applied to the state space formulation in Eq. (7). In this way the second order system in Eq.(8) can be represented in the following first order state-space form:

$$\dot{\mathbf{x}} = \begin{bmatrix} \mathbf{0} & \mathbf{I} \\ -\tilde{\mathbf{M}}^{-1}\tilde{\mathbf{K}} & -\tilde{\mathbf{M}}^{-1}\tilde{\mathbf{C}} \end{bmatrix}\mathbf{x} + [b]\tilde{\mathbf{F}}, \qquad \mathbf{y} = [c]\,\mathbf{x} \tag{11}$$

where the state vector $\mathbf{x}$ is defined by the modal amplitudes:

$$\mathbf{x} = \begin{bmatrix} \dot{\mathbf{q}} \\ \mathbf{q} \end{bmatrix} \tag{12}$$

Actuators and Sensors modelling

In active control, sensors, actuators and structure are put together in such a way that the level of interaction between those elements turns any separately approach of the subsystems impossible (Preumont, 2002). Therefore, a unifying approach that takes into account the fully coupled system is needed. Among the present solutions are those based on the inclusion of sensor, actuators and control laws models into CAE software such as finite element (FE), multibody systems (MBS), etc. Another possible solution, when two time domain simulations are involved, is the co-simulation, usually applied to MBS models and a controller in Simulink. Finally, the methodology adopted in this work involves the inclusion of a reduced model of the mechanical systems into the control system design environment (Matlab/Simulink), where the interaction between structure and sensor/actuator is eventually taken into account.

In this paper, sensors and actuators are respectively accelerometers and inertia-shakers. As far as the accelerometers' frequency range is observed, it can be considered as a lumped mass added to the FE nodes where a SAP is placed. However, for the inertia-shakers, a more detailed model for the electro-mechanic coupling within the actuator and its interaction with the structure must be taken into account. The causes and effects of the interaction between electrodynamic exciters and the structure under test (SUT) has been an issue for the experimentalists since the very beginning of modal analysis as in Unholtz (1961) and Tomlinson (1979) and still is a subject of research (McConnell, 1995; Olbrechts, *et al.* 1997; Lang, 2001 and Oliveira & Varoto, 2002).

Figure 4 shows the electro-mechanical model of an inertia shaker. Equations (13) and (14) describe the dynamics of the coupled electro-mechanical system with the amplifier set in voltage mode of operation. As it can be seen, the coupling occurs in Eq.(13) as the right-hand side expresses the magnetic force (acting on the inertia element m_i) proportionally to the current I flowing on the circuit; and in Eq.(14) in which a electric potential on the left-hand side (representing the E_{bemf}) is written in terms of the relative velocity between the SUT driving point and the shaker inertia. More detailed information about the shaker dynamics as well as the amplifier modes of operation can be found in McConnell (1995) and Oliveira & Varoto (2002).

$$m_i\ddot{u}_i + c_i\left(\dot{u}_i - \dot{u}_s\right) + k_i\left(u_i - u_s\right) = kf\ I \tag{13}$$

$$RI + L\dot{I} + kf\left(\dot{u}_i - \dot{u}_s\right) = E \tag{14}$$

As mentioned before, the objective of the proposed modelling procedure is to include the actuator (considering its interaction with the SUT) in the controller design environment. At that point the SUT is represented by a reduced modal model, from which the displacement of the driving point (with the accel. added mass) must be available. Since the shaker is rigidly connected to the structure, the movement of the base will be the same as the SUT driving point. In this way the inertia-shaker can be represented by the moving mass and the active interface. As proposed by Herold *et al.* (2005) for a hybrid isolation mount, the shaker active interface can also be modelled as a lumped impedance element that contains the passive and active parameters (Fig.5). Thus it is possible to include one or more inertia shakers, given the driving point displacements and the driving voltage E.

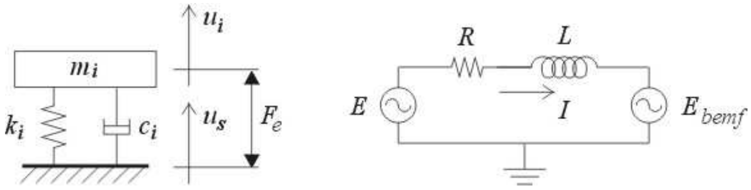


Figure 4 – Exciter's electromechanical model

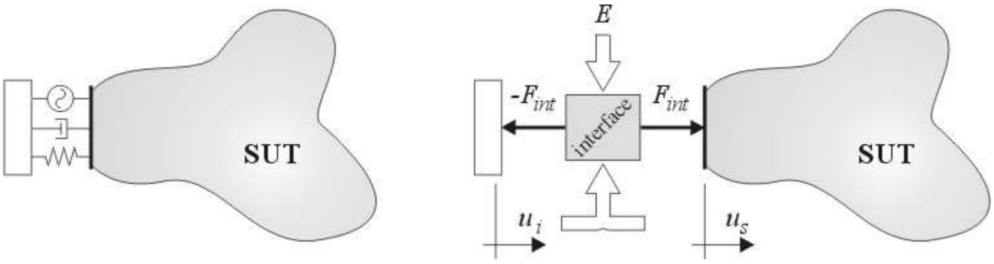


Figure 5 – inertia shaker active interface model

RESULTS

The next sections show the simulation results obtained using the proposed methodology. The first sub-section deals with the system open-loop simulation and the interaction between the firewall and the exciter, while the second shows how the modelling procedure can be applied to ASAC simulation where 2 SAPs are placed in decentralized velocity feedback controller.

Open-loop results

A 2-input/2-output state-space model has been built (Fig 6). The structural and acoustical inputs are respectively, force (F_s) and volume velocity (F_a); the outputs are the velocity (v) on the firewall driving point and pressure (p) at the driver's head position.

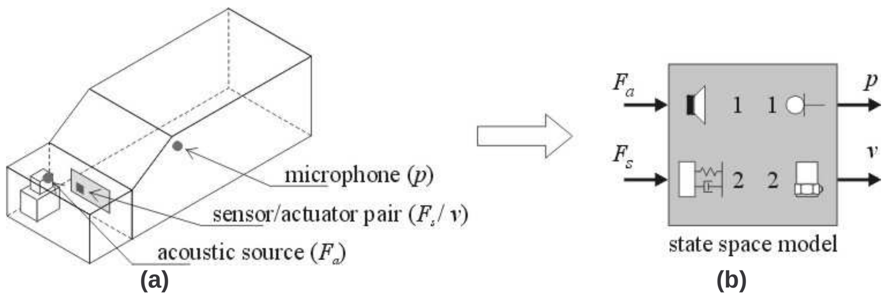


Figure 6 – Schematic view of the model (a) sensors and actuator location and (b) state space model

Figure 7 illustrates the state-space forced responses for both kinds of inputs: volume-velocity from the acoustic source and force at an arbitrary position on the firewall. The graphics show the pressure at the driver's head position and the velocity from the driving point on the firewall. It can be seen that the model is indeed coupled.

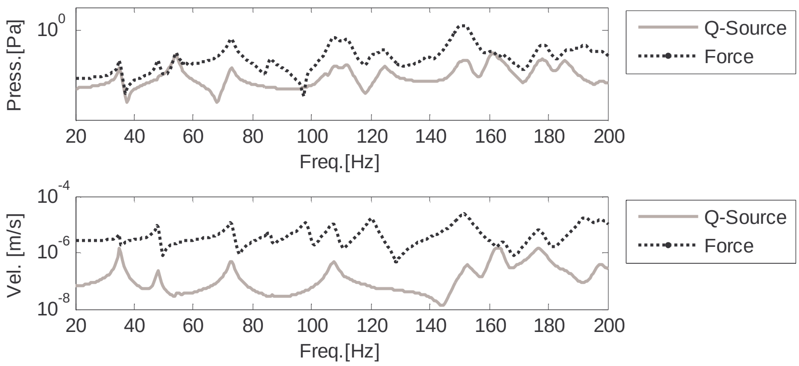


Figure 7 – State-space forced responses: Q-Source (1^{-6} m^3/s) and Force (1N)

The forced responses on Fig.7 came merely from the system transfer functions, *i.e.*, the input force is assumed ideal. However, if the model of the exciter is included in the simulation, it is possible to observe phenomena inherent to the use of such electrodynamic devices, *e.g.* force drop-off. Figure 8 shows, in the upper part, the structural FRF and in the lower part, a comparison of the idealized force input and the actual load provided by an inertia shaker. It can be seen that in the low frequency range and in the vicinities of structural resonances, the force level drops, as a result of shaker/structure interaction (Tomlinson, 1979; McConnell, 1995).

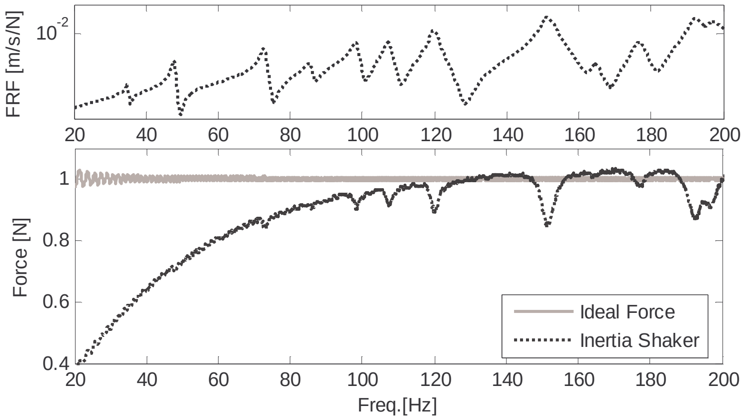


Figure 8 – System driving point FRF and input forces

The mass loading and drops in the exaltation force can lead to errors in the experimental FRFs (Olbrechts et al., 1997; Oliveira & Varoto, 2002) but mainly, when the active control system is concerned, can result in overestimated authority and performance.

ASAC Simulation

An initial configuration is proposed, where 2 collocated SAPs in a decentralized velocity-feedback control loop are applied to the firewall. The basic principle of this controller is to feed back the actuator with an amplified voltage proportional to the velocity of its collocated sensor (Fig.9).

The choice of using only structural sensors and actuators in this ASAC approach is based on the robustness of the control system. Since feedback is going to be used, an important aspect to the efficiency and stability of the control system is the transfer function between the sensor and actuator. The phase angle between these two signals should be bounded by +/- 90^{o}, otherwise the system can become unstable. Usually, the use of acoustic sensors and/or actuators presents a fast phase loss, which would impose severe limitations to the controller frequency range (Nelson & Elliott, 1992). However, since the SAP is collocated it can be proved that the control system acts in fact like a passive system and stability is always guaranteed, independent of the feedback gain (Preumont, 2002; Henrioulle, 2001).

Also, the structural transfer functions are much less sensitive to typical changes in this kind of systems, such as open window or the placement of more people inside the vehicle. On the other hand, the control system just senses (directly) the structure. As a result, it is expected that only the predominantly structural resonances will be affected by the controller, while the predominantly acoustic ones may not be.

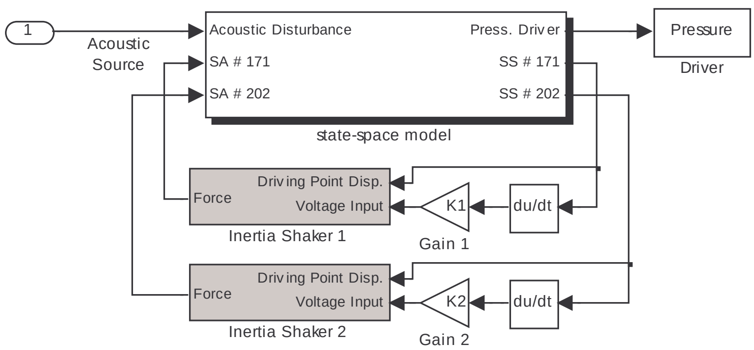


Figure 9 - Block diagram for ASAC simulation with 2 collocated SAPs

Therefore, given the fully coupled system described by the state-space formulation, it is possible to run time-domain analysis in closed loop simulation using Matlab/Simulink. As described before, the inertia-shaker model can be included in this step. The inertia shaker block is connected to the structural input/output ports of the state-space model in a velocity feedback configuration (Fig.9). Figure 10 shows in more detail the inertia-shaker block diagram, where the driving point (together with the moving mass) displacements and velocities are used to compute the interface force contribution, which together with the input voltage, compounds the force acting on the system.

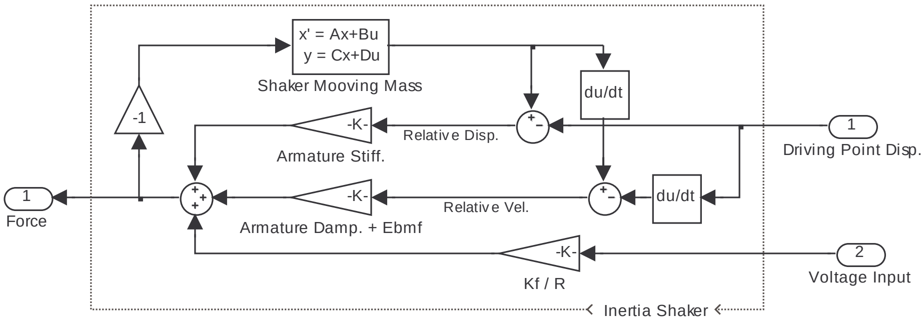


Figure 10 – Detailed block diagram for Inertia shaker

Equation 15 depicts the differential equation governing this block. It is based on Eqs.(13) and (14) where the coil inductance is neglected as suggested by Rao (1987) and Maia & Silva (1997). Since the system is in a velocity feedback configuration, the input voltage *E* will be proportional to the driving point velocity (Fig.9).

$$m_i \ddot{u}_i + \left(c_i + \frac{k_f^2}{R} \right) (\dot{u}_i - \dot{u}_s) + k_i (u_i - u_s) = \frac{kf}{R} E \tag{15}$$

Since the scope of this work is to present a simulation approach to ASAC, in spite of performing an optimization for choosing the SAPs' position, this choice was made upon previous experience. As demonstrated by Oliveira, *et al.* (2005), an arbitrary placement can lead to satisfactory results if just the feedback gains are optimized.

Figure 11 shows how the control system performance varies with respect to the feedback gains K_1 and K_2. The performance is considered as the sound pressure level (SPL) reduction in dB at the driver's ear. As it can be seen it is possible to access the optima gains for both SAPs based on the convex solution surface.

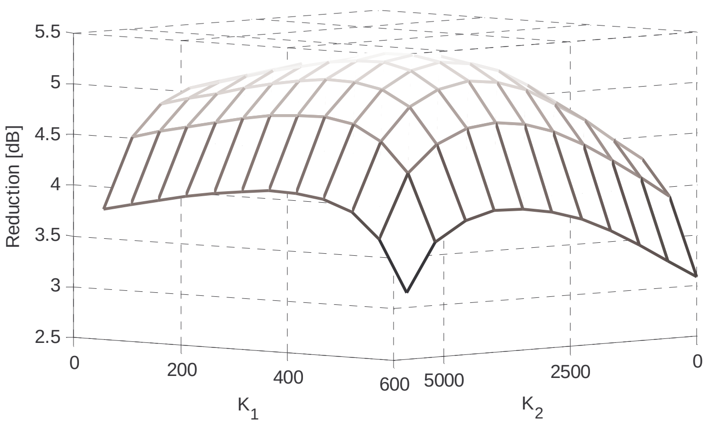


Figure 11 – Controller performance X feedback gains

Figure 12 shows, in the time and frequency domain, the comparison of the open loop response and the closed loop responses considering idealized force and the shaker model. As mentioned before it is possible to notice that the idealised force approach overestimate the control performance, mainly in the low frequency range. Also, it can be seen that for some resonances (primarily acoustic ones) the damping effect is not so pronounced, as the controller has mainly an structural effect rather than global.

As a result, the overall system performance is 5.3dB, with feedback gains K_1 and K_2 respectively at -285 and -3000 [N.s/m], as show in Fig.11. From Fig.12 it can be seen that the overestimate error from the idealized force controller can reach 6dB around the first resonance at 34Hz. It is also true that for higher frequencies this difference is much less pronounced.

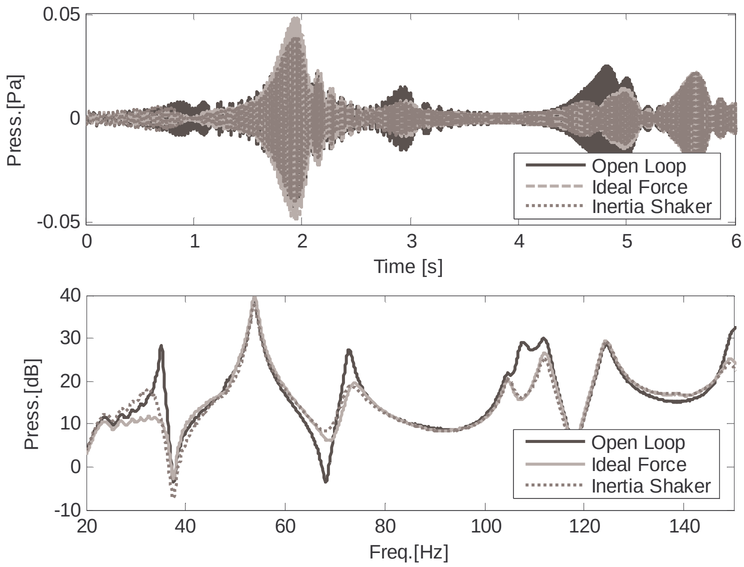


Figure 12 – Comparison of open and closed loop responses for pressure at driver's head

CONCLUSIONS

A modelling procedure for ASAC simulation, considering a fully coupled vibro-acoustic system has been presented. Structural FE models accounting for sensor/actuators placement are used to calculate a vibro-acoustic coupled modal base, which is eventually exported in a reduced form as a state-space model to Matlab/Simulink. Finally, the models for inertia-shakers are incorporated and the control system can be implemented.

The Eulerian modal base allows the representation of the vibroacoustic FE model in a state-space formulation featuring coupled structural and acoustical inputs and outputs.

The inertia-shaker was modelled as a lumped mass connected by an active interface to the structure. The results obtained through this time-domain procedure, *e.g.* the force drop-off phenomenon, are similar to those found in the literature for electrodynamic shaker. Hence, as it can be seen in Figs.7 and 8, the modelling procedure succeeded in representing such a coupled electro-vibro-acoustic system.

The ASAC simulation allows the inclusion of any kind of controller that uses structural or acoustical sensors and actuators. As an example, a decentralized velocity feedback with 2 SPA was presented. A Total reduction of 5.3dB was achieved. It is probable that the SAP placement is not optimum, therefore an increase in the achievable reduction can be reached, even if the current configuration (2 collocated SAPs) is kept. This solution could be accessed by an optimization routine that takes into account not only the feedback gain, but also the SAP placement. The amount of reduction could also be increased if more SAPs are placed simultaneously on the firewall. However, that measure can significantly increase the computational effort for the optimization procedure (Oliveira, *et al.*, 2005).

From Fig.12 it can be seen that the overestimate error from the idealized force controller can reach up to 6dB (70% error) around the first resonance at 34Hz. However, for higher frequencies, this difference is much less pronounced which already indicates an important aspect on the use of such exciters: the usable frequency range. Also in Fig.12 it can be noticed that besides all the advantages of a collocated controller (Preumont, 2002), and the fact that a structural controller would be robust against changes on the acoustic system, it presented a rather weak effect on damping the acoustic modes (Fig. 12).

As far as the primary source has a random nature, *e.g.* aerodynamic or tyre noise, the feedback approach is probably the most convenient solution. However, if the objective is to prevent the transmission of engine noise to the passenger compartment, and considering that a good reference signal is available (engine speed), a feedforward approach may lead to better results. In any case, the modelling approach proposed here fulfils the simulation requirements, providing a more accurate model of the plant which can include sensors and actuators dynamics.

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