Virtual Prototyping for Sound Quality Design of Automobiles

Herman Van der Auweraer and **Karl Janssens**, LMS International, Leuven, Belgium
Leopoldo de Oliveira, **Maira Da Silva** and **Wim Desmet**, Katholieke Universiteit, Leuven, Belgium

Methodologies are presented to model smart structures, actuators and sensors to produce a sound synthesis model for sound quality optimization. A "concrete car" is used to validate the optimization procedure.

During the last few years, promising research results have been obtained for smart materials and active control concepts. In order to bring these research results to real-world applications, related design processes have to become a part of the complete product creation process. This requires that product functional performance simulation models, which are the cornerstone of today's design process, must support advanced materials, active systems, actuators, sensors and controls and integrate these into system level virtual prototype models. To go from acoustic design to sound quality design, the actual temporal and spectral signal structures from the controlled sound need to be optimized to meet sound quality targets.

Modeling Challenges

Shortening development cycles, reducing design costs and improving product performances requires that correct design decisions be made early in the design process. In recent years, major progress was made on the extensive use of a virtual prototyping before the first physical prototypes are available. Such approaches are based on performance simulation models which are derived from multi-attribute optimization schemes.[1]

It is important that the virtual model takes into account the human perception of product characteristics. For automotive applications, a Virtual Car Sound (VCS) synthesis environment[2] has been developed. This allows engineers and designers to listen to the sound in virtually assembled vehicle models including the effect of structural modifications. Such functionality allows fast 'what-if' studies on virtually assembled models, helping engineers tune the design properties toward an optimal result.

Design solutions that make use of smart systems technology, require use of simulation-based optimization approaches to identify optimal configuration and control strategies. Typical design choices include selection between structural or acoustical control, location and number of actuators and sensors, selection of the correct material and dimensional parameters and selection of controller algorithms and settings.

A schematic view of an actively controlled mechanical structure is shown in Figure 1. The basic parts that are to be included in the model are the structure itself (including appropriate acoustic cavity and vibro-acoustic effects), mechanical and electronic parts of the actuators and sensors and related circuits and the controller.

To be of practical use in solving industrial problems, the simulations must make use of standard available simulation tools such as major FE/BE (Finite Element/Boundary Element) and MultiBody Simulation (MBS) codes, 1-D control simulation tools, etc. The most challenging element, however, is to link the different worlds of 1-D control simulation and 3-D geometry-based structural/vibro-acoustic simulation.

When the 3-D structural model is a time domain model, such as a Multibody Simulation (MBS) model, the 1-D and 3-D models can be expressed in terms of state equations. The actual model integration and calculations can then be executed in co-simulation, or the system equations of one model can be embedded in those of the other model. This situation is characteristic for applications such as vehicle suspensions, vehicle dynamics in general, internal engine dynamics, etc.

Based on a paper presented at Inter-Noise 2006, International Congress on Noise Control Engineering, Honolulu, HI, December 2006.

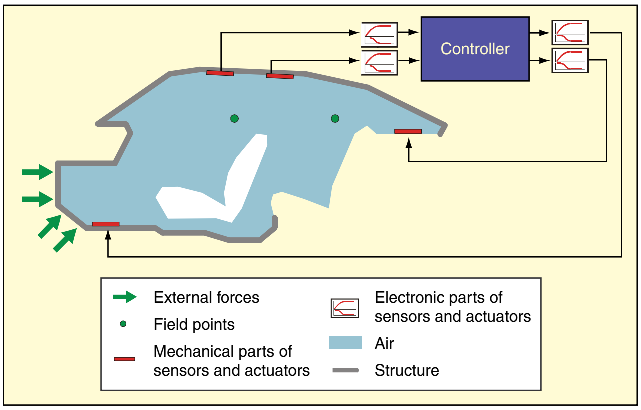

Figure 1. Components of a smart structures solution.

However, classical frequency-domain approaches used for simulating the structural and vibro-acoustic behavior, and which are based on the use of FEM and BEM, are not directly compatible with time-domain approaches to model, simulate and optimize control system performance. In general, the structural model is far too large to be directly transformed into an equivalent state space model and to serve as a basis for controller design or time/frequency response analyses for checking controller performance. Essentially, two approaches are followed to make such combined analysis feasible:

- Reduction of the structural model to an equivalent low-order state-space model. This can happen either through applying mathematical reduction techniques to the original model or by curve-fitting an approximate parametric model to the responses of the original model.
- Integration of control concepts inside the FE model formulation. This approach keeps the original complexity of the FE model but adds the actuator/sensor and control elements by additional degrees of freedom and constraints and the use of special circuit elements.

The sound quality aspect can then be studied either through the integration of modified system characteristic (as structural and/or vibro-acoustic noise transfer FRFs) in the VCS scheme or by direct generation of time domain responses of the controlled response.

But even after adopting these schemes, the distance to real industrial applicability remains large. Each of the listed approaches requires that the exact location of the active control sensors and actuators is known and major design difficulties remain regarding pre-selection of the number and feasible locations of these sensors and actuators. This requires the adoption of a design engineering approach which extends beyond mere structural control simulation.

Modeling Approach

In the InMAR research project, a pragmatic approach was adopted, consisting of separating the modeling problem in different steps:

Identification of the Noise Transfer Paths. The first step towards a successful active control design is to obtain full understanding of the basic noise generation mechanisms, transfer paths and relative contributions. This is done by applying standard structural and vibro-acoustical simulation and testing methods to assess the most critical and sensible locations for control.

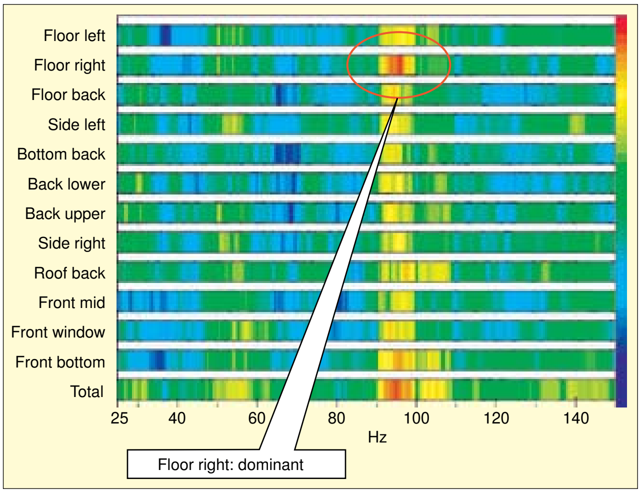

Figure 2. Panel contribution analysis result – contribution overview.

Possible tools include: Transfer Path Analysis (TPA) and Panel Contribution Analysis (PCA) approaches known from standard (passive) vehicle NVH engineering.[3-5] In the case of a structure borne engine noise comfort problem, the outcome can consist of an assessment whether to best pursue an active control on the engine mounts, the chassis frames or on selected cabin panels. Although they are generally overlooked, such fundamental concept decisions are by far the most influential in starting the design process from pre-defined assumptions. An example of a Panel Contribution Analysis is given in Figure 2, where the dominant noise generating panels of a vehicle cabin are identified.

Number and Location of Sensors and Actuators. The selection of the actual location and number of actuators and sensors can be done in a first approximation by using simplified or idealized actuation and control models through the use of additive "secondary sources" in the structural and vibro-acoustical simulation models.

Let $\{y\}_{prim}$ be the original acoustic/structural response field to be reduced then, for a given active control actuator configuration, a set of idealized active control actuator strengths $\{a\}$ can be derived such that the resulting secondary field $\{y\}_{sec}$ maximally reduces the original field.

$$\{e\} = \{y\}_{prim} + \{y\}_{sec} \Rightarrow \text{minimize } \|\{e\}\| \text{ at field response points}$$

$$\{y\}_{sec} = \left[\mathbf{H}_{ya}\right]\{a\} \Rightarrow \text{Response at field points, due to secondary sources, } \left[\mathbf{H}_{ya}\right] \text{ being the FRF matrix between actuators and field responses}$$

$$\{a\} = -\left[\mathbf{H}_{ya}\right]^{+}\{y\}_{prim} \Rightarrow \text{Secondary source strength vector } \{\mathbf{a}\} \text{ at identified locations by minimizing } \|\{e\}\|$$

For example, when considering an acoustic response field, pressure control sensors and active force control actuators, the resulting response field at: a) the control microphones (e.g. at head-rest position in a car); and b) target noise assessment locations different from the control locations (e.g. driver ear); can be estimated using the FRF matrices $[\mathbf{H}_{p2}]$ and $[\mathbf{H}_{p3}]$ respectively, the FRF between the actuator positions and the error microphone positions and the FRF between the actuator positions and the target microphone positions:

$$\{p\}_{error(nf)} = \{p\}_{prim\,(nf)} + \left[\mathbf{H}_{p2}\right]\{f\}_{actuator\,(na)} \Rightarrow \text{at field points of } \|\{e\}\|$$

$$\{p\}_{target(np)} = \{p\}_{prim\,(np)} + \left[\mathbf{H}_{p3}\right]\{f\}_{actuator\,(na)} \Rightarrow \text{at other (target) field points}$$

This approach is compatible with the classical vibro-acoustic simulation approaches, supporting the use of FEM/BEM models to calculate the FRFs $[\mathbf{H}_{ya}]$ and loads $\{\mathbf{a}\}$. An example flowchart is shown in Figure 3. Some case studies using experimental and/or numerical aircraft, car and railway models are discussed in References 6-9.

State Space Model Integration. For this given configuration, the detailed structural model can be reduced into a state-space representation for time domain simulation. The vibro-acoustic model is reduced to transfer vectors that link the state-space outputs to the acoustic responses (Figure 4). Adequate model reduction procedures are hence a key to this approach.

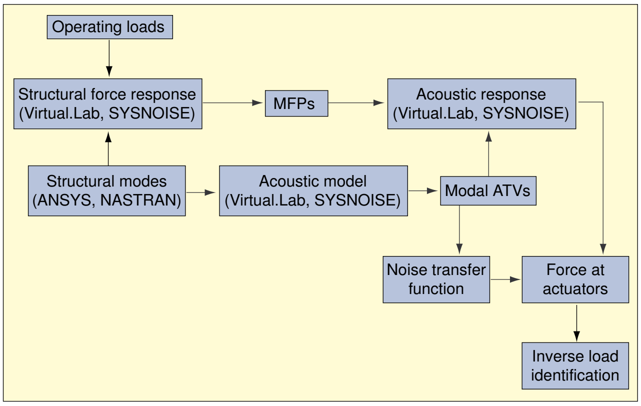

Figure 3. FE/BE calculation procedure for secondary source quantification.

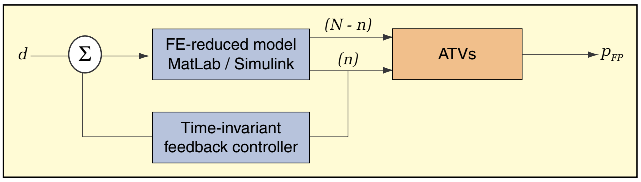

Figure 4. Control system integration through FE model reduction.

Optimization of Control Approach. Finally, the selection of control approach and algorithms, optimization of controller parameters, the evaluation of time domain performances and stability etc., can be done through the use of time domain system simulation models with detailed controller models and using the reduced structural representation.[10]

It is very important with this approach to include adequate parameterized models, not only for the control, but also for the actuators as these are the most critical element in the complete smart structure solution. These models can be incorporated in the structural FE model or can be directly included in the state-space model. An overview of some Simulink based actuator models developed at the Fraunhofer Institute LBF[11] are shown in Figure 5.

Application to a Vehicle Simulation Model

The "Concrete Car" System. The methodology was applied to a system consisting of a simplified car cavity, built with concrete to assure the acoustic boundary conditions (Figure 6). A flexible steel panel between the engine and passenger compartments represents a firewall. An acoustic source in the engine compartment represents the primary disturbance and a set of structural sensors and actuators on the firewall create measurements and realize the resulting control signals for increasing the transmission loss through the firewall. For this case, a simple type of colocated velocity feedback controller was chosen, due to its simple implementation and robustness.

Modeling Procedure. The modeling procedure to derive the state space model function starts with the structural FE model and features the possibility of incorporating sensors and actuator models to the FE/FE vibro-acoustic model. It includes the following steps using multiple software (Figure 7):

- Generate structural mesh and apply material properties (FE pre-processor: Patran)
- Add actuator and sensor mechanical models (FE pre-processor: Patran)
- Run a modal analysis (Nastran)
- Build the acoustic FE model and perform modal analysis (Virtual.Lab Acoustics)
- Import the modified structural model and couple it with the acoustic model (Virtual.Lab Acoustics)
- Export the mesh and coupled modal base
- Calculate actuator and sensor electro-mechanical coupling

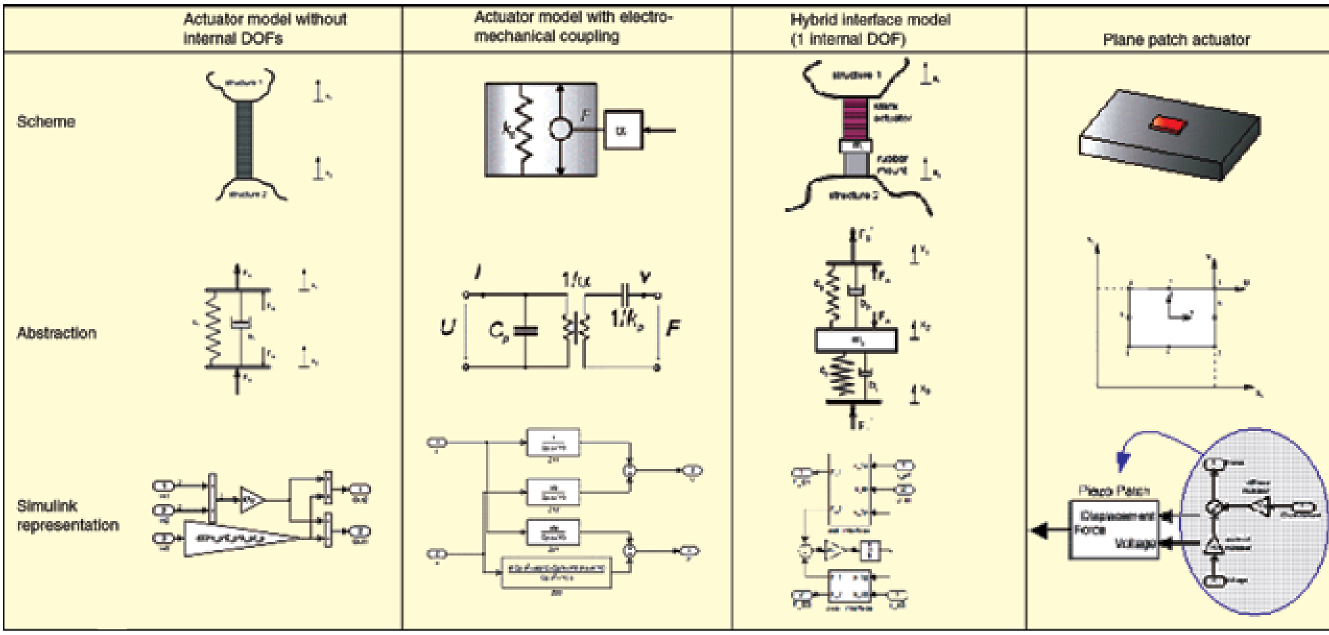

Figure 5. Active interface models.

(Samcef)
- Reduce and convert the FE model into a state-space model (Matlab/SDT)
- Given the coupled state-space model, implement the controller (Simulink)

Results of the Coupled Model Simulation. The coupling between acoustical and structural models is shown in Figure 8. After performing a coupled modal analysis, the desired degrees of freedom (DoFs) are taken to derive the State Space (SS) model. In this case, the SS model features 2 inputs (1 actuator on the firewall and a sound source in the engine compartment) and 4 outputs (3 pressures in the passenger compartment and one velocity on the firewall). The SS model derived from this coupled approach allows the implementation of any controller involving the pre-defined DoFs, and if the FE approach involves the systematic representation of the sensors and actuators, the resultant SS model is, in fact, a representation of the fully coupled electro-vibro-acoustic system, with any possible input/output relationships allowed by the chosen DoFs. More details of the various modeling steps can be found in Reference 12.

Concurrent Optimization. Using this model, an optimization procedure is performed using OPTIMUS as a simulation management and optimal search engine. OPTIMUS is able to manage the structural analysis using MSC.Nastran, the acoustic analysis for a coupled vibro-acoustic model made by Virtual.Lab Acoustics and, finally, a controller simulation using a state-space model. A crucial point is the efficient provision of reduced models for different parameter settings.

The cost function takes into account three parameters: the sound pressure level at the drivers head (performance); the input energy from the actuator (effort); and a penalty for the total mass (weight) of the structure, representing the financial cost impact. The variables are the firewall thickness and the gain of the velocity feedback controller. Initially, the position of the co-located sensor/actuator pair (SAP) is considered fixed, based on previous analysis;[13] in a further step, this parameter is also included in the optimization loop.

Optimization Results. The optimization starts by defining a firewall thickness. A FE model is generated in Nastran and the modal base exported to Virtual.Lab Acoustics to compute the coupled vibro-acoustic modes. The state-space model is built in Matlab, and the optimization engine derives feedback gain for the closed loop simulation. With these data, the mass, effort and closed loop performance and the cost function can be calculated.

The reduced model size allowed a simulation sequence to be defined for creating a response surface model (Figure 9), using a set of possible thicknesses and feedback gains and considering an arbitrary fixed position for the sensor/actuator pair (SAP). It can be seen that the problem is not a trivial convex optimization presenting lots of local minima. However, it clearly shows a global minimum to the chosen cost function – around 1.75 mm thickness and 100 N/m/sec feedback gain.

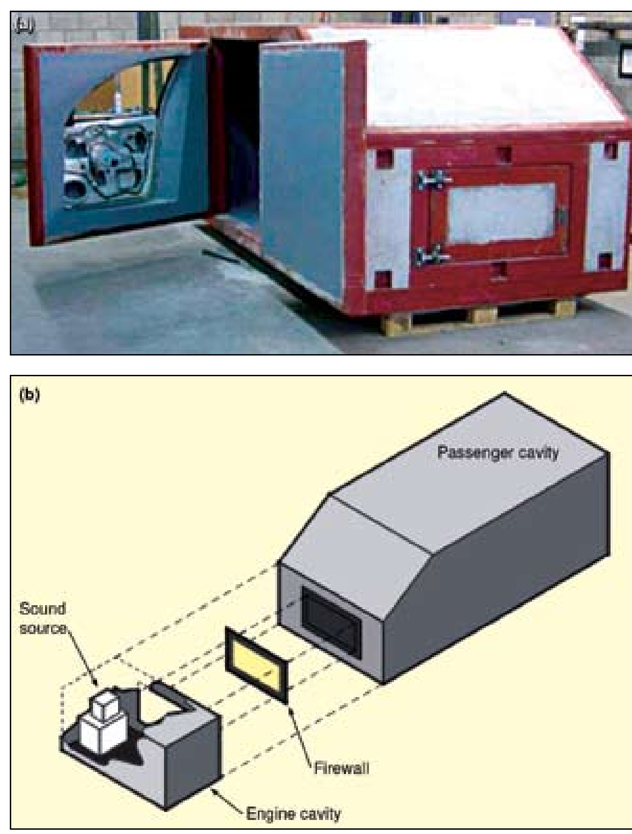

Figure 6. Concrete car system under investigation: (a) photo; (b) schematic view.

After this first approach, it was possible to delimit a subspace where a simpler optimization method could be applied, such as the Sequential Quadratic Programming method (SQP) based on Non-Linear Programming techniques using the design sensitivities (gradient of the objective and the constraint functions). The success of these methods is dependent on the starting parameters, which were made possible based on the previous analysis. The results of

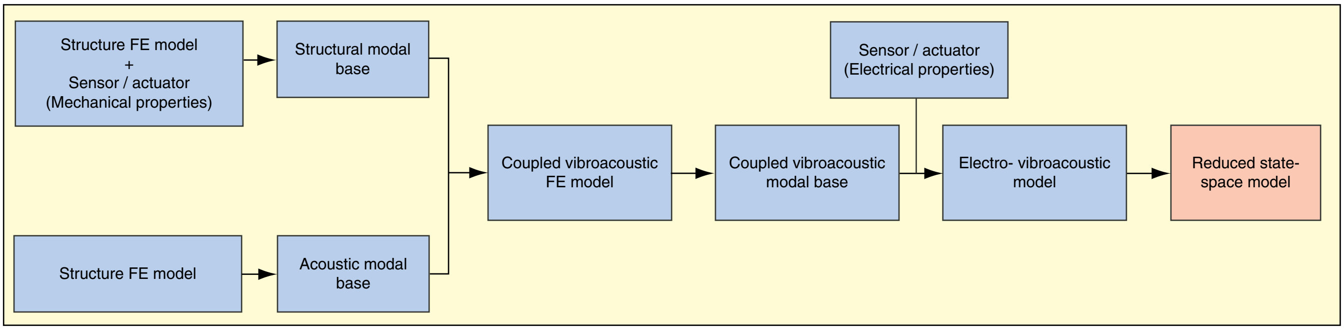

Figure 7. Simulation procedure.

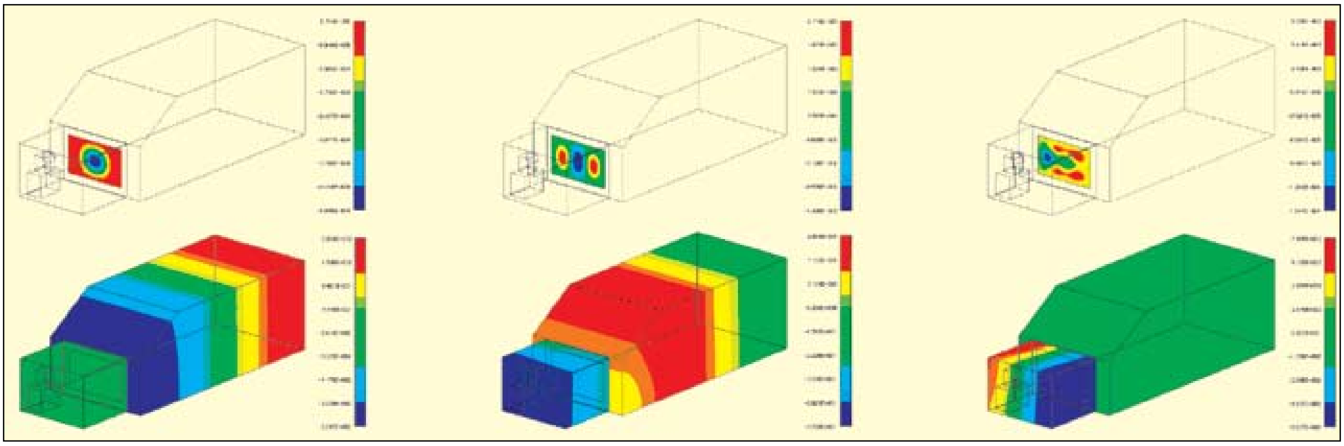

Figure 8. Vibro-acoustic mode shapes.

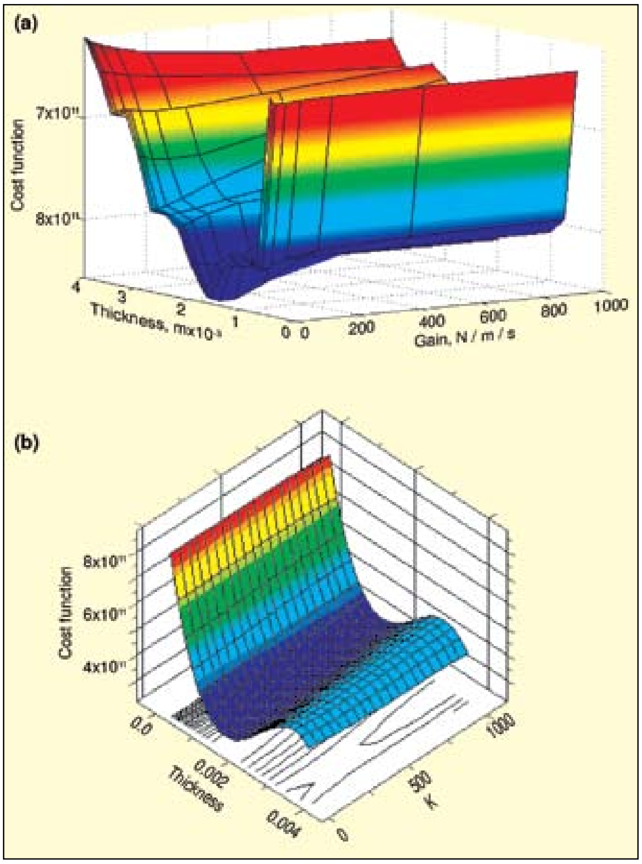

Figure 9. Response surfaces (upper) and several simulation experiments (lower) model surface.

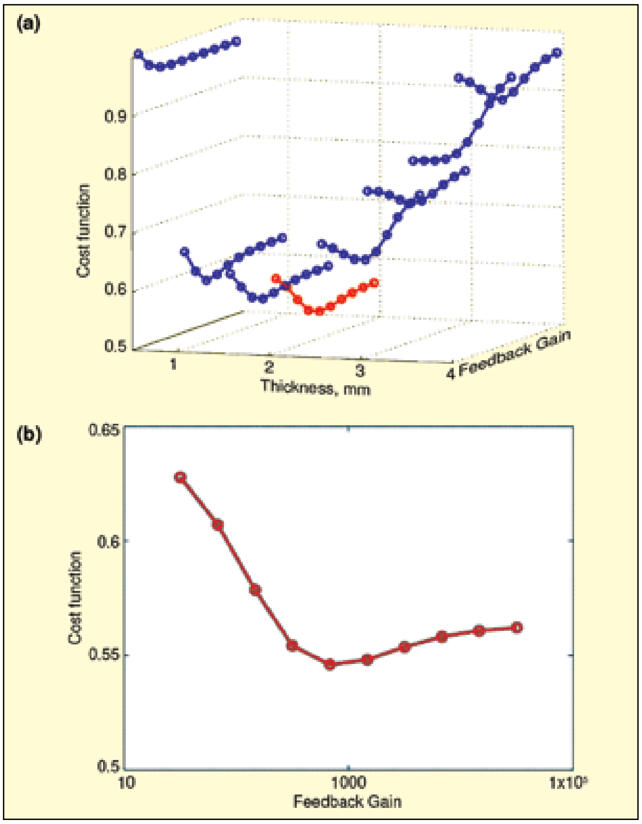

Figure 10. Cost function for each thickness and best SAP position.

the SQP pointed to a thickness of 1.745 mm and a feedback gain of 99.7 N/m/sec.

However, such active solutions are very sensitive to the placement of the sensor/actuator pair.[14] Therefore, this parameter was also included in the optimization loop. In this case a set of possible thicknesses (0.5, 1.0, 1.5, 2.0, 2.5, 3.0, 3.5, and 4.0 mm) was used to search for the best position and feedback gains. Again, the size of the reduced models allowed an extensive search, i.e., every possible position and feedback gain in the design range were tried.

Figure 10a shows the cost function for each thickness as a function of feedback gain, on the best SAP position for each case. Figure 10b shows again, the same plot, but now just for the global

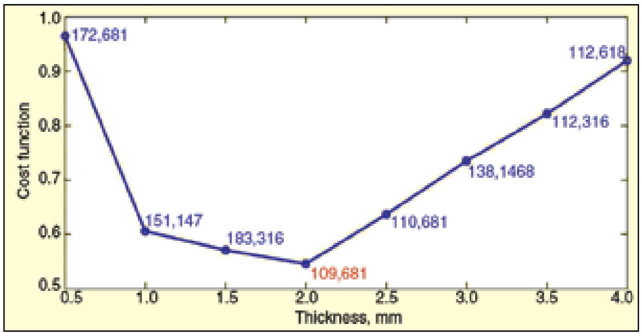

Figure 11. Best position and feedback gain for each firewall thickness

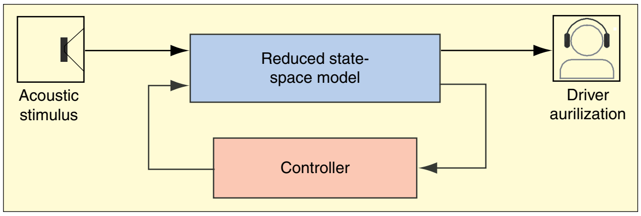

Figure 12. State space sound synthesis model.

optimum (2 mm firewall with SAP at node 109). From these plots, it can be seen that there is an optimum gain for each thickness and SAP position.

Figure 11 shows the best combination of SAP placement and feedback gain for each thickness. It is clear that the best position and feedback gain depend on the thickness, which indicates that the global optimum can only be achieved in such a concurrent design.

Sound Quality Analysis. To make the step to sound quality design, different approaches can be followed in the various phases. First, the acoustic targets can be expressed in terms of subjectively representative parameters (e.g. balance/level of response harmonics). These effects can be validated by spectral source or response modifications to the resynthesized Virtual Car Sound. In a second step, the structural/vibro-acoustical model can be represented by Frequency Response Functions, which are used by the VCS model. The impact of modifications to model parameters or directly to the FRFs (by passive or by active means) can then be evaluated through sound synthesis in a perception-representative way. The extension of the TPA model with "virtual secondary-sources," allows the response sound to be resynthesized for different sensor/actuator configurations. The final State Space then provides a real-time capable model for evaluating on-line the effect of control parameters. This is shown in Figure 12.

Conclusions

Virtual prototyping is an essential tool for optimizing product behavior. To allow smart structure solutions to make the step to industrial applications, the corresponding design and engineering processes must fully support specific aspects of these components. The "concrete car" set-up has been used as a validation tool for the optimization procedure. Automatic communication between FE (Nastran/SAMCEF) software and MATLAB supports the optimization engine (OPTIMUS).

Acknowledgements

The research presented in this article was developed as part of the Integrated Project InMAR, "Intelligent Materials for Active Noise Reduction," coordinated by Fraunhofer LBF and co-funded by the European Commission. www.inmar.info.

References

1. H. Van der Auweraer, M. Tournour, K. Wyckaert, K. De Langhe, "Vibro-Acoustic CAE from an Industrial Application Perspective," SAE paper 2005-26-050, Proceedings SIAT 2005, Symposium on International Automotive Technology, 2005.
2. K. Janssens, A. Vecchio, P. Mas, H. Van der Auweraer, P. Van de Ponseele, "Sound Quality Evaluation of Structural Design Changes in a Virtual Car Sound Environment," Proceedings Institute of Acoustics, 26(2) 2004.
3. D. De Vis, W. Hendricx, P.J.G. van der Linden, "Development and Integration of an Advanced Unified Approach to Structure-Borne Noise Analysis," Proceedings ATA Conference (1992), pp. 465-473, 1992.
4. P. van der Linden, "Quantifying the Noise Emission of Engine Oilsumps, Valve Covers, etc., Using Artificial Excitation," Proceedings European Conference on Noise and Vibration, IMechE, pp.311-320, 1998.
5. K. Wyckaert, W. Hendricx, "Transmission Path Analysis in View of Active Cancellation of Road Noise in Automotive Vehicles," Proceedings of 3-rd Int. Congress on Air and Structure Borne Sound and Vibration, pp.1437-1446, 1994.
6. H. Van der Auweraer, D. Otte, G. Venet, J Catalifaud, "Experimental Analysis of Primary and Secondary Sound and Vibration Fields in Aircraft Cabins," Proceedings Inter-Noise 93, pp. 71-78, 1993.
7. H. Van der Auweraer, T. Martens, E. Waterman, T. Olbrechts, "Optimisation Study for Structural Acoustical Control Configuration on the Fokker 100 Using Reciprocal Testing," Proceedings Noise-Con 96, pp. 177-180, 1996.
8. K. Wyckaert, H. Van der Auweraer, L. Hermans, "An Approach for the Simulation of Active Control Sensor and Actuator Configurations for Rolling Noise Reduction in Automotive Vehicles," Proceedings Active '95, pp. 747-754, 1995.
9. A. Vecchio, M. Rimondi, K. Janssens, P. Hovmand, F. Anders, "Hybrid Modelling Approach to Predict Engine Noise Reduction In Passenger Trains," Proceedings Euronoise 2006, Tampere (FI), May 2006, *Acta Acoustica* united with *Acustica*, Vol. 92 (2006) Suppl. 1, S14.
10. J. Mohring, A. Wirsen, "Robust Controller Design Based on Integrated Simulation of an ANVC-System," Proceedings Euronoise 2006, Tampere (FI), May 2006, *Acta Acoustica* united with *Acustica*, Vol. 92 (2006) Suppl. 1, S14.
11. S. Herold, H. Atzrodt, D. Mayer, M. Thomaier, "Integration of Different Approaches to Simulate Active Structures for Automotive Applications," Proceedings Forum Acusticum 2005, Budapest (HU), Aug. 29-Sep. 2, 2005.
12. L. P. R. Oliveira, A. Deraemaeker, J. Mohring, H. Van der Auweraer, P. Sas, Wim Desmet, "A CAE Modeling Approach for the Analysis of Vibro-Acoustic Systems with Distributed ASAC Control," Proceedings ISMA 2006, Leuven, Sept. 18-20, 2006.
13. L .P. R. Oliveira, M. M. Silva, G. Pinte, W. Desmet, P. Sas, P. Mas, H. Van der Auweraer, "Optimal Integrated Design Procedure for ASAC," Engineering Adaptive Structures, 2006, Maiori (IT), 2006.
14. L. P. R. Oliveira, B. Stallaert, W. Desmet, J. Swevers, P. Sas, "Optimisation Strategies for Decentralized ASAC," Proceedings Forum Acusticum 2005, Budapest (HU) Aug. 29-Sep. 2, pp. 875-880, 2005.

The author can be reached at: herman.vanderauweraer@lms.be.