

# Efficient Simulation for Strongly Coupled Noise-Vibration-Harshness (NVH)

R. S. Puri, D. Morrey

School of Technology, Oxford Brookes University, Wheatley Campus,  
Wheatley, Oxford OX33 1HX, United Kingdom, email  
[s.puri@brookes.ac.uk](mailto:s.puri@brookes.ac.uk)

E. B. Rudnyi

CADFEM GmbH, Marktplatz 2, 85567 Grafing b. München, phone +49  
8092 7005 82, fax +49 8092 7005 66, email [erudnyi@cadfem.de](mailto:erudnyi@cadfem.de).

## Abstract

Noise, Vibration and Harshness (NVH) is a critical consideration in the design of automotive and aerospace vehicles for comfort, and fatigue of components arising from interior structural and acoustic pressure fluctuations due to external structural or acoustic loading. In the low to mid frequency range, the starting point for computational NVH investigation is the displacement/pressure based unsymmetric, coupled Eulerian Finite Element/Finite Element formulation. One of the major bottlenecks in such computational NVH investigation(s) is that the finite element model obtained along this way is of a very high dimension and therefore, the solution for the coupled dynamic response of a single design becomes computationally expensive - even with modern hardware.

In this work, we show that the approach based on model reduction via the Arnoldi processes allows us significantly to speed up harmonic simulation necessary for low frequency, interior acoustic, fully coupled NVH problems. We demonstrate this effect for a strongly coupled NVH benchmark. The computational results indicate that the required coupled states (displacements and pressures) from a direct harmonic simulation could be obtained for time comparable with that for just a few frequencies. Finally, we demonstrate how the approach based on moment matching leads to efficient fully coupled structural-acoustic optimization.

## 1. Introduction

NVH stands for Noise Vibration and Harshness and is an industry term associated with the prediction and active/passive treatment of vibration and audible sounds. Findings show that NVH not only causes annoyance and fatigue, but also affects the efficiency and health of people. NVH is very important from an engineering and customer satisfaction point of view

and thus a very significant issue in automobile and aerospace cabin interior design. The concerns have primarily been those of speech interference, crew fatigue, and passenger comfort, the latter arising primarily in for example private/business jets and first class accommodations on commercial airlines.

Though there are currently no regulations on interior noise levels, airline operators require guarantees from manufacturers on these noise levels. In automobile design, during earlier years, NVH was primarily investigated at the end of the production line and additional damping (vibration absorbing) materials were added to reduce noise and vibration based on user tests on road conditions. However, today, in a passenger car, the NVH performance is one of the most important parameters that determine the quality of passenger comfort in manufactured car. In fact, NVH is rated as among the top five priorities by automobile manufacturers. Due to the nature of the current automotive business, where competition is very high, manufacturers are today striving to bring NVH levels down as low as possible.

Even with the advent of modern computers and cheaper computational power, the simulation of a complex structure such as a vehicle/aerospace cabin often involves tedious and time-consuming modeling procedures leading to higher dimensional models. Today, it is common practice to investigate NVH behavior using models, which range up 50,000 to 1 Million degrees of freedom (DOFs) and require a frequency sweep solution, that is, a solution for the entire frequency range. As a result, the simulation of such higher dimensional models in many cases is prohibitive or even impossible, especially in the case when repeated analysis is required during an optimization to calculate the best design parameters for good NVH characteristics over the entire applicable frequency range.

The goal of the current work is to present a new method based on model reduction that allows us to speed up simulation of a frequency sweep of a NVH model tremendously. In Section 2, we review the current practices to simulate NVH, and then in Section 3 we introduce model reduction. After that in Section 4, we present an adhesive bonded joint model as benchmark and compare its full scale simulation in ANSYS with the approach based on model reduction. Further, in Section 5 we describe the use of model reduction during structural acoustic optimization.

It should be mentioned that the much more comprehensive description of topics presented in this paper is available in [1] that also contains an extensive bibliography on the NVH simulation.

## 2. Simulating Noise, Vibration and Harshness

Today, the NVH behavior of a typical automotive body is typically classified into three distinct descriptors, based primarily on their transmission phenomenon and frequency range (see Table 1). This distinction can be attributed to the fact that the vibro-acoustic behavior of the vehicle structure under investigation tends to be different in all three frequency ranges.

Table 1: NVH description based on Phenomenon and Frequency Range

Descriptor	Phenomenon	Frequency Range
Low Frequency	Structure borne	0 – 100 Hz
Mid Frequency	Structure and air borne	100 – 250 Hz
High Frequency	Air born	> 250 Hz

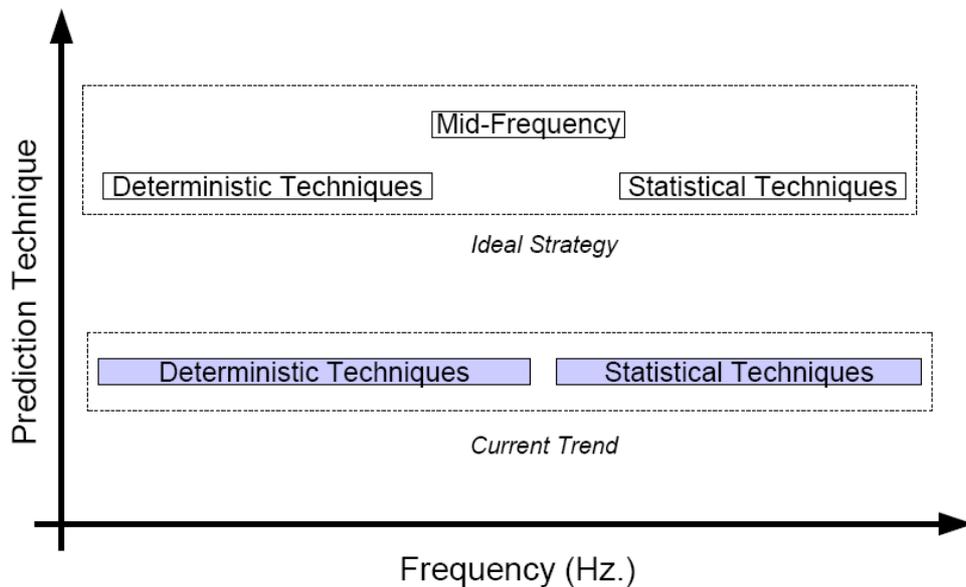


Fig 1: Ideal and current trend for NVH prediction

At low frequencies, an exact solution is sought for a set of governing equations arising from the discretization of the structural and acoustic fields into mass, stiffness and damping matrices in space, subject to a number of boundary conditions including enforcement of coupling between the fluid and structural domains. To counter the problem at high frequencies, researchers have developed energy-based methods, based on the principle of conservation of energy, to predict the statistically time/frequency averaged response, on the fluid or the structural domain. In the mid-frequency range, none of these techniques alone has been found to be adequate enough predict the response accurately. However, it has now become a common practice to investigate the mid-frequency range, by extending the

analysis frequency range of deterministic and statistical energy based methods in order to cover the mid-frequency gap. An illustration of techniques for interior NVH behavior prediction based on frequency range is shown in Fig 1. The steps involved in a broadband structural-acoustic optimization are shown in Fig 2.

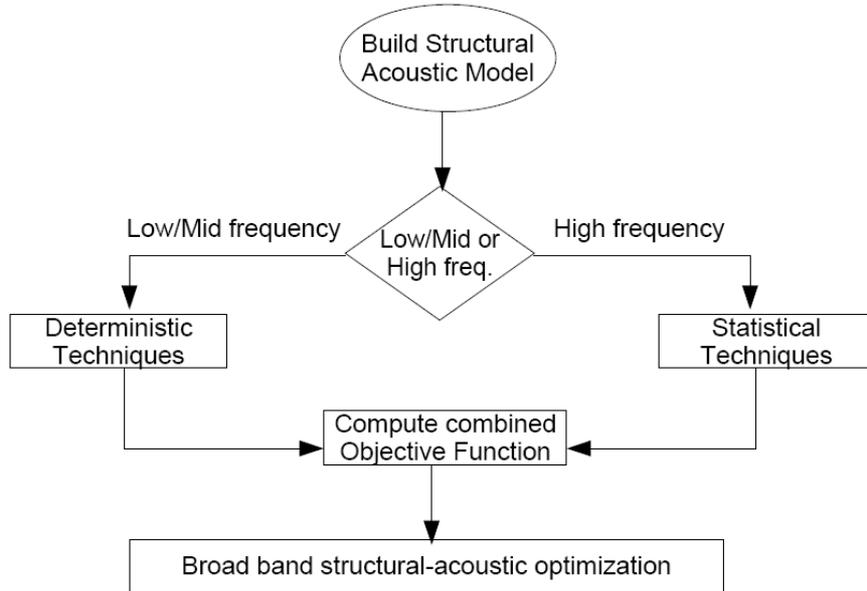


Fig 2: Strategy for broadband NVH optimization

The computational bottleneck lies with the deterministic method (statistical methods are computationally very efficient) and one of the big challenges in fully coupled vibro-acoustic modelling nowadays is the development of new deterministic prediction techniques and solution procedures, which provides accurate prediction results but with an enhanced computational efficiency. Most common way is the finite element discretization that under the combined Eulerian displacement – pressure (u/p) formulation, which leads to the coupled structural-acoustic model as follows:

$$\left( -\omega^2 \begin{bmatrix} M_s & 0 \\ M_{fs} & M_a \end{bmatrix} + j\omega \begin{bmatrix} C_s & 0 \\ 0 & C_a \end{bmatrix} + \begin{bmatrix} K_s & K_{fs} \\ 0 & K_a \end{bmatrix} \right) \begin{Bmatrix} u \\ p \end{Bmatrix} = \begin{Bmatrix} F_s \\ F_a \end{Bmatrix} \quad (1)$$

Equation above has a solid physical background, with an equivalent experimental counterpart (where displacement and pressures are the field variables) and often referred to as the Cragg's formation. At the same time, it demands high-end computational resources. The requirement that the element size should be smaller than wavelength leads to high

dimensionality of the state vector and the asymmetry of system matrices makes the solution of a system of linear equations even more complicated.

Our solution to this challenge is the use of model reduction. This way we can reach computational efficiency without any compromise from the physics side.

### 3. Model Order Reduction

Model order reduction is an area of mathematics that goal is to find a low dimensional approximation of a large-scale dynamic system [2]. It allows engineers to bridge a gap between device- and system level simulation (see Fig 3) in the case of linear dynamic systems.

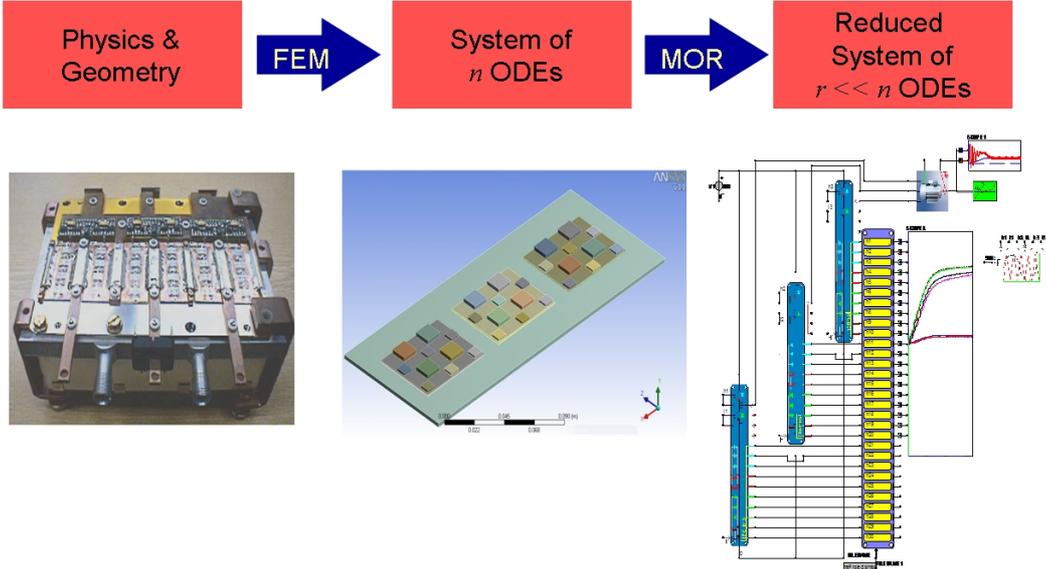


Fig 3: Model order reduction is an efficient means to enable a system-level simulation. Figure shows an example of using a compact thermal model for an IGBT block in a joint electro-thermal simulation [3].

Model reduction is based on an assumption that the movement of a high dimensional state vector can be well approximated by a small dimensional subspace (Fig 4 left). Provided this subspace is known the original system can be projected on it. This is illustrated for the system of the first order in Fig 4 and it can be generalized to the second order systems the same way.

The main question is how to find the low dimensional subspace that possesses good approximating properties. In structural mechanics, the subspace formed by the eigenvectors corresponding to the lowest frequencies enjoys widespread use. However, it happens that

there are much better choices based on Krylov subspaces [2]. It is actually faster to compute them as compared with the modal analysis and, at the same time, they possess better approximation properties. Another advantage is that the projection subspace remains real valued also for unsymmetric matrices.

The model reduction theory is based on the approximation of the transfer function. It has been proved that in the case of Krylov subspaces the reduced system matches moments of the original system for the expansion point. In other words, if we expand the transfer function around the expansion point, the first coefficients will be exactly the same, as for the original system.

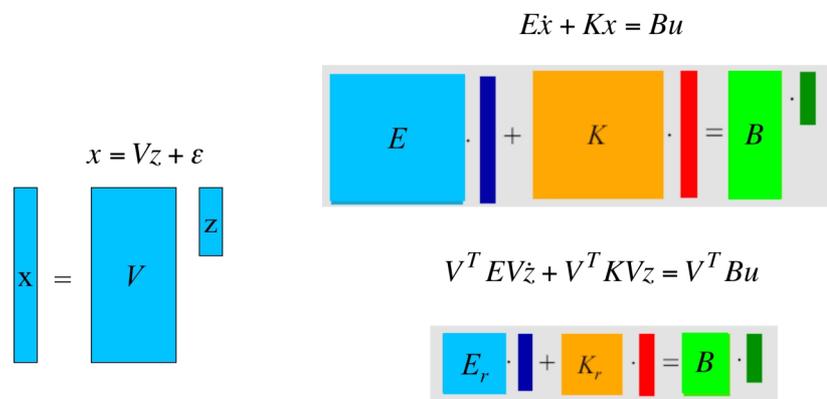


Fig 4: Model reduction as projection.

The Arnoldi and second order Arnoldi algorithms have been implemented in the software MOR for ANSYS [4]. The software allows us to apply model reduction directly to ANSYS models (see Fig 5). At the beginning, the software has been employed for thermal systems but then its use has been expanded to structural mechanics and multiphysics problems. On the Web-site <http://modelreduction.com/Applications/> there are numerous examples of using the software to electro-thermal MEMS, structural mechanics, piezoelectric actuators for control, pre-stressed small-signal analysis for RF-MEMS, and thermomechanical models. First application of model reduction to strongly coupled structural acoustic problems has been reported by us in [5].

The algorithms mentioned above are referred to as one-sided, as only one projection matrix is employed (see Fig 4). In [1] these algorithms have been generalized to the two-sided, when two different (left and right) projection matrices are generated during the model reduction process. This way it was possible to increase the number of matched moments and hence to enhance the approximating properties.

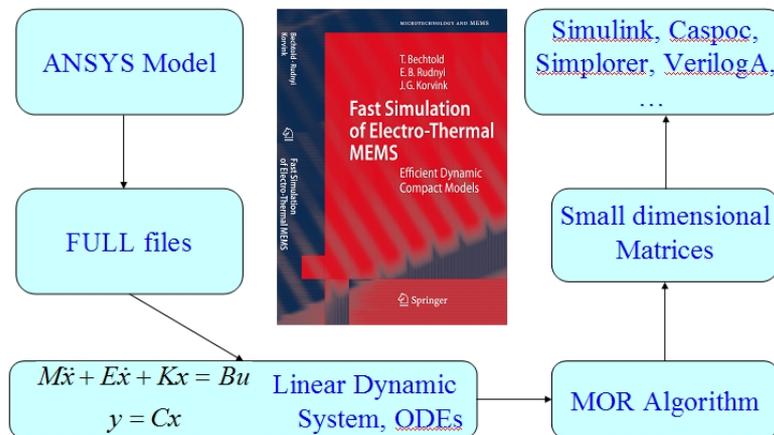


Fig 5: The structure of MOR for ANSYS [4]

The original idea of model reduction was to develop a compact model for system level simulation (see Fig 3). This is not exactly the case for an NVH model, as usually it is necessary to perform the frequency scan for a given model only once. Fortunately, the time for model reduction is roughly comparable to the time to evaluate a few frequencies in harmonic simulation of the original high dimensional system, that is, it is much faster than the frequency scan of the original system. As a result, it is possible to use model reduction as just a fast solver, in other words, as a tool that allows us to perform fast frequency scan. In this paper, we will consider model reduction only from this viewpoint.

#### 4. Adhesive Bonded Joint Benchmark

In [1] to illustrate the applicability and suitability of the reduction methods, the proposed methods have been applied to six different fully coupled structural-acoustic systems represent typical academic benchmarks. Both air and water have been considered as fluid medium in order to test the accuracy and computational efficiency for both weakly and strongly coupled problems. The model reduction based on the Krylov subspaces has been also compared with the mode superposition method. All finite element models have been implemented in ANSYS.

In this section, we describe the benchmark with an adhesive bonded joint. It is well known that the modeling of joints, gaps and sealants in a vehicle or an aerospace FE model leads to better vibro-acoustic prediction accuracy over the entire frequency range. However, one of the major challenges facing CAE modeling of joints is also efficiently to balance the requirement for accuracy with the need for maintaining reasonable vehicle FE model sizes

and computer run time. Therefore, in this test case, an adhesive bonded panel-frame structure is considered to validate the accuracy of Arnoldi based projection formulations.

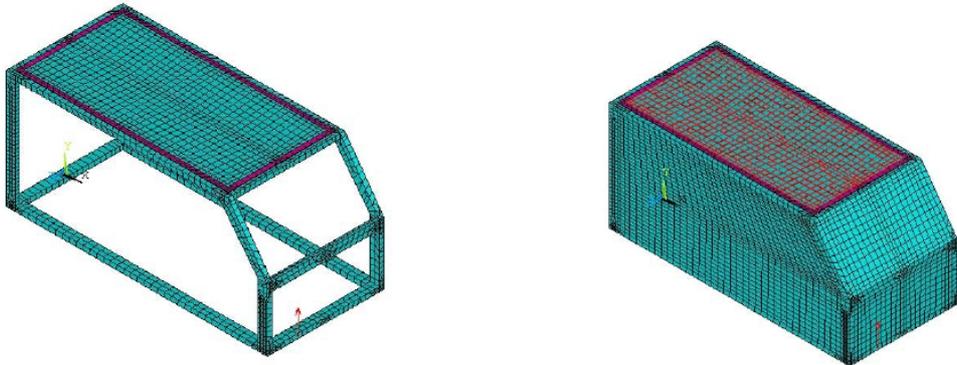


Fig 6: Structural model with adhesive bond (pink elements) and coupled structural-acoustic model.

For this test case (see Fig. 6), one of these areas, which corresponded to the vehicle roof, was meshed using four noded quadrilateral shell elements (ANSYS SHELL181), with six degrees of freedom at each node. The roof panel is bonded to the beams, using SOLID45 elements the solid-shell macro modeling approach. Two different adhesives, Epoxy Betamate 5318-4 and Polybutadiene Terostat 3218F have been considered for analysis. The distribution of elements in the finite element coupled model areas follows: 4472 structural (4258 SHELL 181 and 214 SOLID45 for the adhesive bond) elements, 740 acoustic interface elements (FLUID30) along the fluid-structure interface and 32742 acoustic elements (FLUID30) not in contact with the structure. The coupled model was excited using a unit point force excitation at  $(x, y, z)=(1.45\text{m},0, 0.15\text{m})$ . The structural and coupled models are shown in Fig 6.

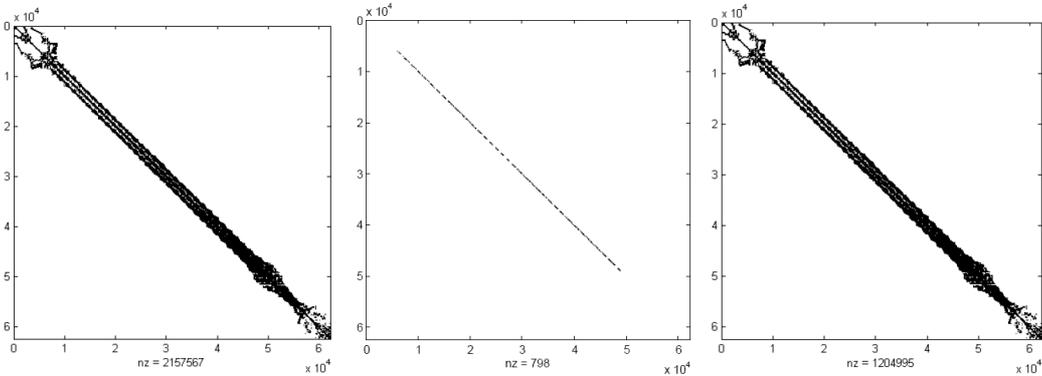


Fig 7: Sparsity plots for the mass, damping and stiffness matrices.

In Fig 7, there are sparsity plots for the mass, damping and stiffness matrices and Fig 8 shows the comparison between the noise transfer function computed in ANSYS and with the reduced model. One can see that the reduced model of dimension 90 reproduces the results of the ANSYS model (dimension 62353) with the error smaller than the line thickness.

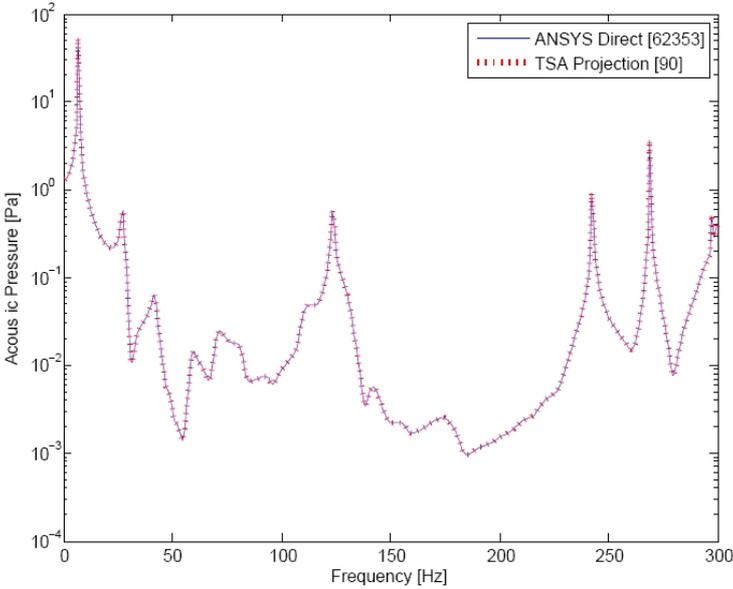


Fig 8: ANSYS Direct and double sided Second Order Arnoldi projection predicted Noise Transfer Function at fluid node (0.9m, 0.5m, 0.08m).

### 5. Structural Acoustic Optimization

The idea of using model reduction during optimization is displayed in Fig 9. As has been already mentioned, model reduction is used a tool for fast but accurate frequency scan and thus it allows us to reach the same results as with the full scale harmonic analysis in ANSYS (direct inversion in Fig 9) but considerably faster. The optimisation has been performed with the mesh adaptive direct search (MADS) method with the use of Latin hypercube sampling (LHS) to find the initial design [1].

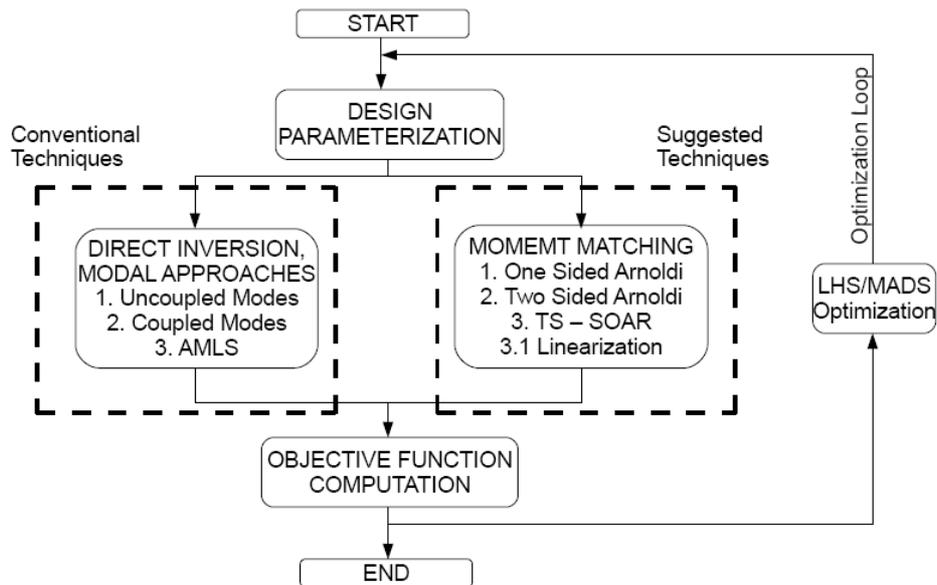


Fig 9: General framework for vibro-acoustic optimization via conventional and suggested techniques.

Over the recent years, different novel materials have been developed to control noise and vibration in vehicles and commercial aircraft. In particular, fiber reinforced composites have generated significant interest in the development of structural materials due to their low density, high stiffness and excellent damping characteristics. However, the coupled structural-acoustic problem is complicated if the flexible structural portion are built of thin laminated composite plates since they interact with the acoustic field in their own strange manners (Niyogi et al. 2000). Additionally, the orthotropic nature of such fiber reinforced composite materials implies that the directional stiffness depends on the orientation of fibers. Such flexibility can often be exploited to tailor the material to obtain the required structural acoustic performance.

A scale model structure is considered as the test case for fully coupled, vibro-acoustic optimization. One boundary surface of an acoustic cavity consists of a flat rectangular plate with fully clamped boundaries, while all other cavity boundary surfaces are perfectly rigid. The acoustic cavity is comprised in the volume of an enclosing rectangular prism (see Fig 10).

The first candidate material for the fully clamped plate is a glass fiber reinforcement in the form of uni-directional fabric (Owens-Corning - Standard E-Glass Fiberglass) and epoxy resin (Ciba Geigy XR-1553) with catalyst addition (HY - 956) as 246 matrix for the composite material. The second material is a Polypropylene glass composite, is made up of continuous

unidirectional glass reinforced polypropylene prepregs, produced by melt impregnation, in lay-ups for thermoplastic composite structures.

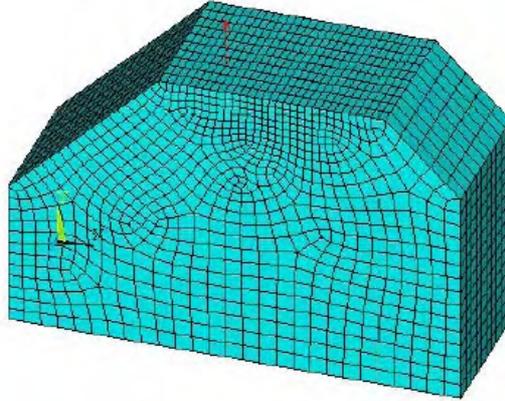


Fig 10: Coupled Structural-Acoustic model for vibro-acoustic optimization.

310 structural elements – ANSYS SHELL181, and 7310 acoustic fluid elements (including 330 interface elements) were used for the coupled model. SHELL181 is a 4-node shell element with six degrees of freedom at each node: translations in the global x, y, and z directions, and rotations about the x, y, and z-axes. The accuracy in modeling composite shells via SHELL181 is governed by the first order shear deformation theory (usually referred to as Mindlin-Reissner shell theory or FSDT). It is worth noting that for the composite elements, an alternate cartesian co-ordinate system is chosen and invoked, in order to align the strongest fiber along the length of the plate.

In Fig 11, we compare the response computed in ANSYS with the response of the reduced model of dimension 25. The difference between them is negligible while the computational time for the frequency scan is reduced by more than 50 times. Fig 12 compares the sound pressure level for designs with simple orientations of layers in the composite with the optimized slacking sequence. A general decrease in SPL levels for the considered fluid output is apparent over the entire frequency range.

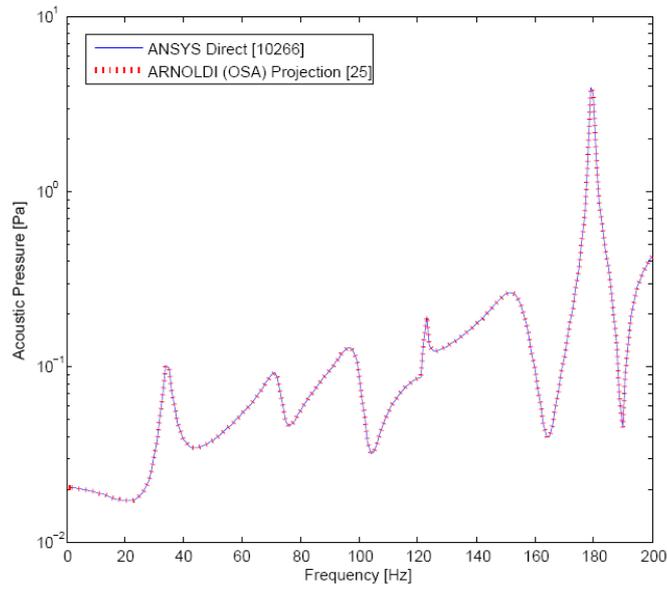


Fig 11: A comparison of noise transfer function obtained by ANSYS direct and One-sided Arnoldi (OSA) projection for optimized eight layer stacking sequence (with material PP-Glass Fiber composite).

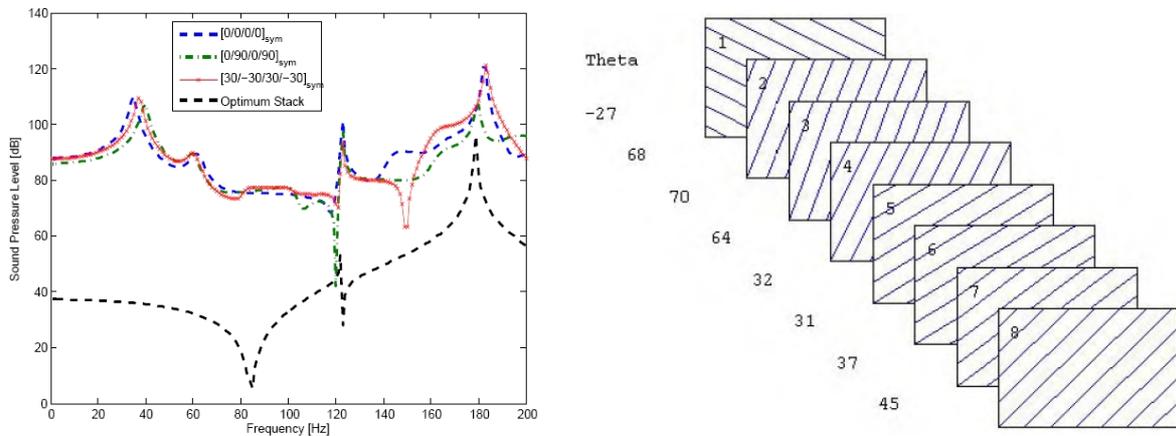


Fig 12: A comparison of Arnoldi predicted fluid pressure for composite stacking sequences:  $[0/0/0/0]_{sym}$ ,  $[0/90/0/90]_{sym}$ ,  $[30/-30/30/-30]_{sym}$  and optimum stacking sequence  $[153/68/70/64/32/31/37/45]$  obtained by LHS/MADS optimization. On the right is the optimal stacking sequence.

## 6. Conclusion

We have shown that model reduction allows us to speed up the frequency scan significantly. The speedup factor depends on how many Arnoldi vectors will be enough to approximate the frequency response in the given frequency range and how many frequencies should be

evaluated in the frequency scan. In our benchmarks, the speedup was between 30 and 100. This opens new perspective to simulate strongly coupled structural acoustic problems and makes it possible to perform structural acoustic optimization.

No doubts, the presented benchmarks were of academic nature. It is necessary yet to demonstrate the approach with industry benchmarks and this is the next natural step. So far this has been done by only Voith Siemens Hydro Power Generation with a FSI model to study the dynamic excitation of turbine rotors by rotating pressure field caused by rotor-stator interaction. This is not related directly to NVH, but the coupled physics is exactly the same as described by Eq (1). This gives us a hope that the method will also work for the industry NVH models.

## 6. References

- [1] R. S. Puri. Krylov Subspace Based Direct Projection Techniques for Low Frequency, Fully Coupled, Structural Acoustic Analysis and Optimization. PhD Thesis, 2008, Oxford Brookes University. Available online at <http://modelreduction.com/Applications/Acoustics.html>
- [2] A. C. Antoulas, Approximation of Large-Scale Dynamical Systems. Society for Industrial and Applied Mathematic, 2005, ISBN: 0898715296.
- [3] A. Dehbi, W. Wondrak, E. B. Rudnyi, U. Killat, P. van Duijsen. Efficient Electrothermal Simulation of Power Electronics for Hybrid Electric Vehicle. Proceedings of Eurosim 2008, p. 412-418.
- [4] E. B. Rudnyi and J. G. Korvink. Model Order Reduction for Large Scale Engineering Models Developed in ANSYS. Lecture Notes in Computer Science, v. 3732, pp. 349-356, 2006, <http://ModelReduction.com>
- [5] R. S. Puri, D. Morrey, A. Bell, J. F Durodola, E. B. Rudnyi and J. G. Korvink. Compact structural-acoustic coupled models via model order reduction. ICSV13, Thirteenth International Congress on Sound and Vibration (ICSV13), July 2 to 6, 2006, Vienna, Austria.
- [6] B. Hübner, J. Koutnik, Anwendung von Krylov-Unterraum basierter Modellreduktion auf harmonische Analysen gekoppelter Struktur-Akustik-Systeme, GMA-Fachausschuss 1.30 „Modellbildung, Identifikation und Simulation in der Automatisierungstechnik“, 2007, Salzburg.