Predicting Car Body Dynamics at Higher Frequencies: Challenges and Possible Solutions

Jean-Louis Migeot, Jean-Pierre Coyette and Karl Meerbergen
Free Field Technologies SA
16 place de l'Université - B-1348 Louvain-la-Neuve - Belgium
Tel: +32 10 45 12 26
Fax: +32 10 45 46 26
jean-louis.migeot@fft.be

ABSTRACT

Finite element methods are well suited for the dynamic analysis of vehicles in the low frequency range (below 100 Hz). Similarly, SEA methods are well adapted to the higher part of the spectrum (above 500 Hz). A lot of research has gone over the past ten years in the development of methods dedicated to the mid-frequency range but none of them proved to be robust and flexible enough to be used in an engineering context. There is now a widespread agreement to consider that the mid-frequency range will be filled by : (1) enhancing finite element models to push their frequency limit up to, say, 200 Hz and (2) releasing some of the assumptions of SEA to make it a viable method from 200 Hz onward.

We basically see three challenges hampering the fulfilment of the first task :

- higher frequency means larger model and one must improve solver technologies to face the growing computational load ;
- it makes little sense to predict the response of a body-in-white above 100 Hz and one must improve the way damping is handled in dynamic models ;
- dispersion of results is already significant at these frequencies and parameter uncertainty must be taken into account in a stochastic framework if one wants to provide meaningful results to the engineering process.

Our paper presents three recent developments made by FFT to overcome each of the difficulties :

1. A new solver for the prediction of narrow-band frequency response functions has been developed. An approximation of the entire FRF is obtained by factorizing the matrix at a single frequency. The procedure is adaptive and provides results within a user specified tolerance. It could easily be integrated within MSC.Nastran. The cost of calculating the FRF with our new procedure is a tiny fraction of the cost of the standard frequency sweep and provides a much better frequency resolution.

2. Free Field Technologies develops the ACTRAN software for a consortium of automotive companies. ACTRAN is used to analyze the insulation, absorption and damping provided by acoustic trim. In terms of damping, ACTRAN results can be converted in equivalent frequency-dependent visco-elastic properties that can then be used to enhance the global body model. The MSC.Nastran body in white model can be used to predict the behavior of the trimmed body simply by applying the visco-elastic properties calculated by ACTRAN.

3. Different routes are currently being investigated for including data uncertainty (thickness, elastic constants, loads) in MSC.Nastran dynamic models without resorting to computationally expensive Monte Carlo procedures. The goal is to predict response intervals vs deterministic responses using the most recent developments in the stochastic finite element method. Preliminary results will be shown and development routes discussed.

Introduction and context

Finite element methods are well suited for the dynamic analysis of vehicles in the low frequency range (below 100 Hz) and MSC.Nastran ([1]) is the reference tool for such analysis. Similarly, SEA methods are well adapted to the higher part of the spectrum (above 500 Hz) and tools like SEAM([2]), SEADS ([3]) or AutoSEA ([4]) provide the required toolbox for modelling car bodies using this global technique based on spatial and spectral averaging energy concepts.

A lot of research has gone over the past ten years in the development of methods dedicated to the mid-frequency range. One can for instance think of energy flow analysis ([5,6]), asymptotic modal analysis ([7,8,9]), mobility energy flow analysis ([10]), energy flow analysis using classical finite elements ([11]), general energy formulation method ([12]), energy (phase) envelope model ([13]) and, for the acoustic media alone, ray tracing([15]). Recent work by Christian Soize ([14]) also fall within this category. None of these techniques however proved to be robust, powerful and flexible enough to be used in an engineering context and failed when applied outside the strict context within which they were developed.

There now seems to be a rather widespread agreement to consider that the mid-frequency gap can only be crossed by :

- enhancing finite element models to push their frequency limit up to, say, 200 Hz (our concern here are car body and frequency bounds are related to this specific application) ;
- releasing some of the assumptions of SEA (weak coupling) or mixing FEA and SEA (addition of FEM calculated stiffness multipliers in the formulation for instance) to make it a viable method from 200 Hz onward.

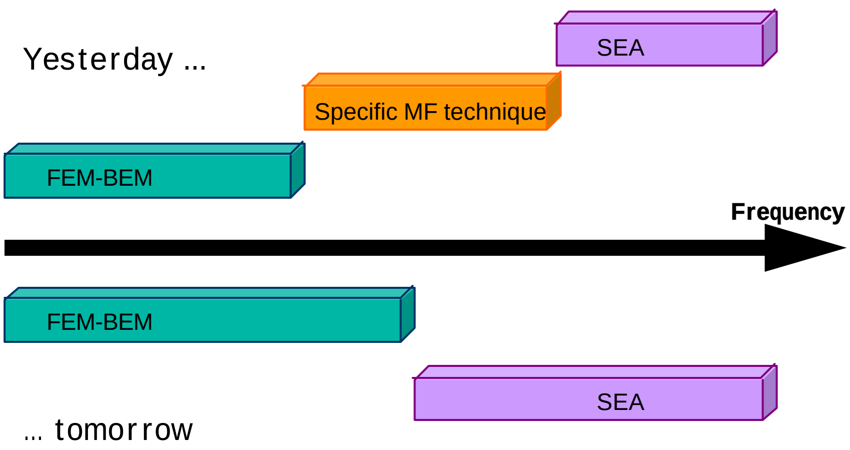

Figure 1 - FEA and SEA filling the mid-frequency gap

We see three fundamental challenges hampering the fulfilment of the first task i.e. the extension of classical FE model above the current threshold :

- **computing time** : higher frequency means larger model and one must improve solver technologies to face the growing computational load ;
- **damping modelling** : it makes little sense to predict the response of a body-in-white above 100 Hz and one must improve the way damping is handled in dynamic models ;
- **uncertainty and variability management** : dispersion of results is already significant at these frequencies and parameter uncertainty must be taken into account in a stochastic framework if one wants to provide meaningful results to the engineering process.

We will now briefly present some of Free Field Technologies' ideas and most recent development providing (partial) responses to each of these problems. The presentation will obviously be in connection to MSC.Nastran and the presented solutions compatible with its solution procedures.

A new solver for narrow-band FRF calculations

The cost of frequency response procedure is high for two reasons: (1) the cost of solving the system of equation at each frequency can be tremendous and increases significantly with the mesh density and the complexity of the problem and (2) all realistic problem requires the solution to be evaluated for many individual frequencies. When accurate frequency response functions must be obtained for large frequency intervals, the computational cost becomes so high that the calculation cannot realistically be made in an industrial framework.

Several approaches have been proposed to reduce the cost of calculating the response and, in the context of acoustic analysis, the Padé approximant technique ([16]) has been shown to give accurate results and a quite formidable efficiency gain ([17,18]). Developments are currently being done by Michel Rochette of CADOE SA (France) to extend the technique to damped dynamic systems and to implement it in MSC.Nastran ([19]).

We have chosen an alternative and yet unpublished approach based on an approximation of the solution in a low dimension Krylov subspace ([20]). The key advantages of the method are:

- a perfect encapsulation making it suitable for seamless integration within MSC.Nastran;
- its applicability to both damped and undamped system;
- its robustness: the method indeed predicts the FRF but also estimates the error (*a posteriori* analysis) and chooses a new central frequency if it is not able to compute the solution within the precision level defined *a priori* by the user.

The mathematics are quite elaborate and their presentation is not relevant in this context but some results are of high interest. Consider an aluminium plate of dimensions 0.5 m x 0.5 m. Young modulus is $7 10^{10}$ N/m^2, thickness is 1 mm, Poisson ration 0.33 and density 2700 kg/m^3, at this level we consider no structural damping. The plate was discretized by a grid of 16 x 16 solid shell elements. The plate is subjected to a unit point load at coordinate (0.125 m, 0.125 m). The purpose is to calculate the mobility at driving point in the [10, 110] Hz frequency range with a frequency step of 1 Hz. The results are presented on figure 2. In fact the approximation is so good that results obtained by the Krylov procedure and a classical frequency sweep are indiscernible. The Krylov procedure is however 10 times faster and the entire FRF is obtained for the cost of a few frequency lines by the standard procedure.

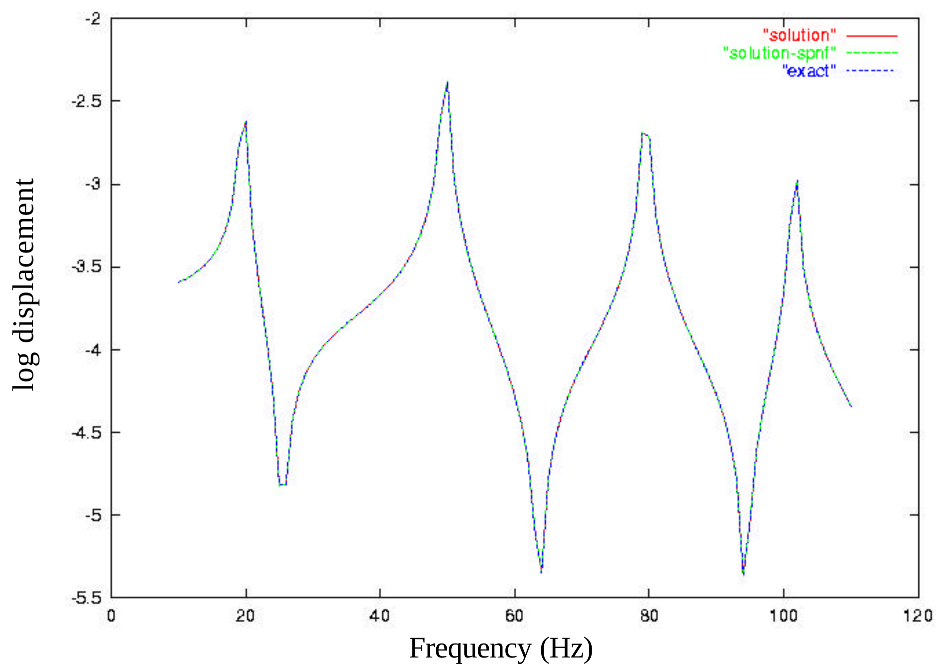

Figure 2 - The FRF curves predicted by the fast-Krylov technique and by a classical frequency sweep cannot be distinguished ...

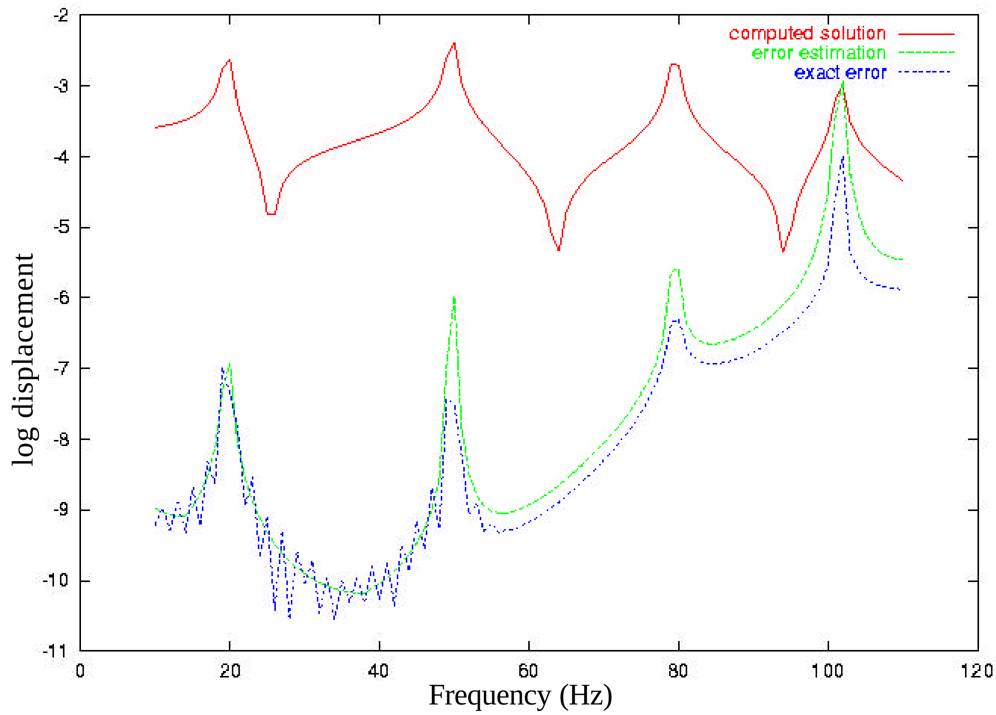

Figure 3 - Graph showing the calculated FRF, the "exact" error and the "estimated" error showing the robustness and accuracy of the procedure.

A new tool for including trim in dynamic models

Trim panels play three different roles:

- they isolate the passenger compartment from exterior noise (wind noise through the roof, engine noise through the firewall, rolling noise through the platform);
- they provide structural damping therefore reducing body vibration and radiated noise;
- they provide absorption to the sound field inside the passenger compartment.

At low frequency it is perfectly reasonable to neglect these elements and to focus the design work on the body-in-white. Above 100 Hz however the damped and undamped response bears little resemblance. At the same time the multiplication of local modes makes the definition of overall modal damping quite inaccurate. The correct modelling of trim materials therefore seems an unavoidable route.

The ACTRAN program ([21]) has been developed by Free Field Technologies since July 98 with this conclusion in mind. ACTRAN is already used by several car manufacturers and their trim suppliers to optimize the behaviour of individual trim components. ACTRAN is able to produce relevant indicators for each of the above effects:

- the transmission loss through a multi-layer panel can be calculated;
- the dynamic response of the trimmed and untrimmed panel can be compared;
- the surface impedance of the material can be calculated.

ACTRAN features all material types relevant to the modelling of such structures: visco-elastic, poro-elastic and incompressible. The sound field is modelled by a combination of acoustic finite elements and conjugated infinite elements ([22]). ACTRAN offers a lot of excitation models including diffuse sound fields and turbulent boundary layer models for estimating the transmission of aerodynamic noise inside the car.

The main target of car manufacturers is to use ACTRAN in conjunction with MSC.Nastran to calculated trimmed body dynamics up to 200 Hz. Two alternative techniques are currently investigated:

a. detailed ACTRAN model of a component are used to evaluate an average complex and frequency dependent bending and shear stiffness that is then applied to the relevant elements of the BIW model;

b. one component (the roof for instance) can be replaced by an impedance matrix produced by ACTRAN and including the effect of the trim.

These two strategies target the calculation of the dynamic response of the car i.e. try to assess the effect of the damping brought to the car by the trim. It is also possible to use surface impedance calculated by ACTRAN in the context of an MSC.Nastran / MSC.Akusmod coupled vibro-acoustic model.

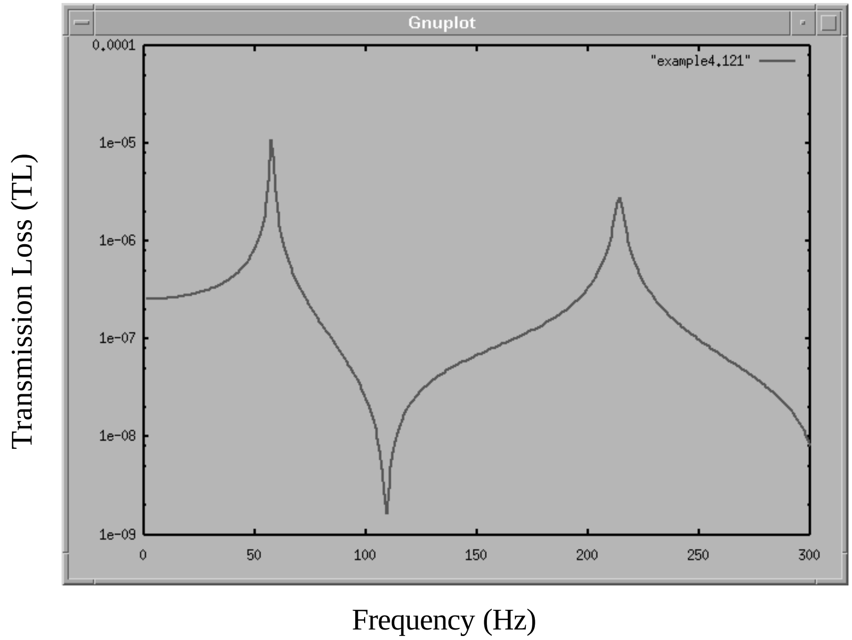

Figure 4 - Low-frequency acoustic sound transmission (TL) through a sandwich panel.

Variability and uncertainty management

The most accurate model of a structure is, at best, a good representation of a nominal structure ([23,25]). Several similar structures taken out of the production line will exhibit variations on all of these individual properties and this dispersion will reflect in the global performance of the structure ([24]). In all cases the scattering significantly increases with frequency.

This raises the problem of robust design : the structure must be designed so as to minimise the impact of the dispersion of individual parameters on the overall performance of the structure. An alternative formulation states that a useful model should be able to define acceptable tolerances on the individual parameter ensuring that the overall performance stays within pre-defined bounds.

The elementary approach to parametric analysis, known as 'Monte-Carlo' analysis ([26]), consists in studying not one but many different structures each with slightly different values of the input parameters. Even if optimised schemes based on the theory of design of experiments are used to select the variants, the number of variants to analyse increase exponentially with the number of parameters leading to an unacceptable calculation time.

Alternatives consist in using sensitivity analysis techniques ([27,28]) that not only predict the relevant index (point mobility for instance) but also its first (and second) derivative with respect to any of the uncertain parameters. A local perturbation model can be created around the nominal state using these sensitivities and can be used to analyse the effect of small parameter changes. On the other hand, the synthesis of the information given by the sensitivity analysis allows to estimate the variability (in terms of statistical estimators such as the mean value and the standard deviation) of local responses or global performances of the uncertain structure ([29]). Numerical tests show that this estimation can be reached at an acceptable precision and requiring a much lower computational effort than by using 'Monte-Carlo' techniques. This low cost is a direct consequence of the fact that sensitivity analysis is performed on the nominal structure, while 'Monte-Carlo' analysis bases on a very high number (the higher this number, the more accurate the solution) of different structures taken as samples.

Beside the computational method, one should also examine the basic modelling of the uncertain parameters, using discrete models (sets of random variables) or continuous models (random fields). Such continuous models, allowing the spatial representation of the random parameters, have however to be reduced to discrete models in order to be used in the sensitivity numerical analysis. Many discretization techniques are therefore available ([30]). Moreover, the random dimension of the problem can be reduced using particular decorrelation procedures. These modelling methods lead to different qualities and costs of the global uncertain analysis.

Free Field Technologies' current research work in the field of stochastic finite elements and in relationship with current features of MSC.Nastran will be presented in more details during the conference.

Conclusions

Three research and development areas to improve the relevance of higher frequency finite element models of car body have been briefly discussed:

- a new solver dramatically reducing the cost of calculating frequency response functions;
- a new finite element program for the modelling of trim panels;
- the need to take data uncertainty and response variability into account.

They are all fully compatible and can be integrated within MSC.Nastran. The authors would be happy to provide more information and detailed reports to interested readers.

References

(1) *MSC.Nastran User's Manual, Version 70.7,* The MacNeal-Schwendler Corporation, Los Angeles, CA , November 1999.

(2) *SEAM*, Cambridge Collaborative.

(3) *SEADS*, LMS International.

(4) *AutoSEA*, Vibro-Acoustics Science

(5) BELOV, V.D., RYBAK, S.A., TARTAKOVSKII, B.D., "Propagation of vibrational energy in absorbing structures" *Journal of Soviet Physics Acoustics.,* 23(2), 1977.

(6) NEFSKE, D.J., SUNG, S.H., "Power flow finite element analysis of dynamic systems: basic theory and application to beams", *Trans. of ASME Journal of Vibration and Acoustics, 111*, 1989.

(7) PERETTI, L.F., DOWELL, E.H., "Asymptotic modal analysis of a rectangular acoustic cavity excited by wall vibration", *AIAA Journal.,* 30(5), 1992

(8) KUBOTA, Y., DOWELL, E.H., "Experimental investigation of asymptotic modal analysis for a rectangular plate", *Journal of Sound and Vibration.,* 106(2), 1986.

(9) KUBOTA, Y., DIONNE, H.D., DOWELL, E.H., "Asymptotic modal analysis and statistical energy analysis of an acoustic cavity", *Journal of Vibration, Acoustic, Stress and Reliability in Design.,* 110(3), 1988.

(10) PINNINGTON, R.J., WHITE, R.G., "Power flow through machine insulator to resonant and non-resonant beams", *Journal of Sound and Vibration.,* 75(4), 1981.

(11) FREDÖ, C.L., "Derivation of energy flow with a finite element model", *Thesis, Chalmers University, Gothenburg, Sweden,* 1993.

(12) LEBOT, A., "Analyse énergétique de quelques problèmes vibratoires", *Report HP-62/93017, DER EDF,* 1993.

(13) CARCATERRA, A., SESTIERI, A., "An approximate power flow solution for one-dimensional dynamic structure", *Proceeedings of the first international conference on intensity techniques, Senlis, France, 1993.*

(14) SOIZE C., "Reduced models in the medium-frequency range for general external structural-acoustic systems", JASA, June 1998

(15) DE GEEST, E., PÄTZOLD, H., *"MOSART: A generalized, multi-domain ray tracing program with optimized solver"*, Proceedings of the Forum Acusticum Conf., Antwerpen, Belgium, 1996

(16) BREZINSKI, C., "*Padé type approximation and general orthogonal polynomials*", ISNM Vol. **50**, Birkhauser Verlag, Basel, 1980

(17) BLANCHE, J., COYETTE, J-P., LECOMTE, CH., MIGEOT, J-L., MIRKOVIC, D., ROCHETTE, M., "*Calculation of vibro-acoustic frequency response functions using a single frequency boundary element solution and a Padé expansion*", Acustica, 1999, 3

(18) MALHOTRA, M., PINSKY, P.M., "*Efficient computation of multi-frequency far field solutions of the Helmholtz equation using Padé approximation*", Journal of Computational Acoustics, 8(1), 2000

(19) ROCHETTE, M., "*Padé approximants for linear damped dynamic systems*", private communication, January 2000.

(20) MEERBERGEN, K., "*The solution of parametrized linear systems arising from mechanical systems*", Free Field Technologies Internal Report (available on request), July 2000.

(21) *ACTRAN Rev 1.1.*, Free Field Technologies, February 2000.

(22) ASTLEY, R.J., MACAULAY, G.J., COYETTE, J.P., CREMERS L., "*Three-dimensional wave envelope elements of variable order for acoustic radiation and scattering: Part I. Formulation in the frequency domain and Part II. Formulation in the time domain*", Journal of the Acoustical Society of America, Vol. 103, N° 1, 1998.

(23) PLUNT, J., "*Generic limitations of vibro-acoustic prediction methods for product noise*". Proc. Internoise Conference, 1996

(24) KOMPELLA, M.S., BERNHARD, R.J., "*Measurement of statistical variation of structural-acoustic characteristics of automotive vehicles*", Proc. of the SAE Noise and Vib. Conf., 1993

(25) MIGEOT, J.L., "*Méthodes numériques pour la prédiction du comportement vibro-acoustique des structures*", Thèse de doctorat, Université Libre de Bruxelles, 1997

(26) PAPADRAKAKIS M., KOTSOPOULOS A., "Parallel solution methods for stochastic finite element analysis using Monte-Carlo simulation", Comput. Meth. Appl. Mech. Engng., 168, 305-320, 1999

(27) HAUG E.J., CHOI K.K., KOMKOV V., "Design sensitivity analysis of structural systems", Academic Press, 1986

(28) ARORA J.S., LEE T.H., CARDOSO J.B., "Structural shape sensitivity analysis: relationship between material derivative and control volume approaches", AIAA Journal, 30 (6), 1638-1648, 1992

(29) KLEIBER M., HIEN T.D., "The stochastic finite element method : basic perturbation technique and computer implementation", Wiley Chichester, 1992

(30) ZELDIN B.A., SPANOS P.D., "On random field discretization in stochastic finite elements", J. Appl. Mech. (ASME), 65, 320-327, 1998