Acoustic Transfer Vectors for Numerical Modeling of Engine Noise

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This article presents the numerical modeling of noise radiated by an engine, using the so-called Acoustic Transfer Vectors and Modal Acoustic Transfer Vectors concept. Acoustic Transfer Vectors are input-output relations between the normal structural velocity of the radiating surface and the sound pressure level at a specific field point and can thus be interpreted as an ensemble of Acoustic Transfer Functions from the surface nodes to a single field point or microphone position. The modal counterpart establishes the same acoustic transfer expressed in modal coordinates of the radiating structure. The method is used to evaluate the noise radiated during an engine run-up in the frequency domain. The dynamics of the engine are described using a finite element model loaded with an RPM-dependent excitation. The effectiveness of the method in terms of calculation speed, compared with classical boundary element methods, is illustrated. The evaluation of the results, as a function of RPM and frequency, shows the potential of the approach during the engine development process.

In the automotive industry when designing an engine, engineers have to take into account several constraints, which are of completely different nature and sometimes even partially contradictory, e.g., performance, low weight, durability, cost and acceptable noise levels. Due to market pressures and competition car manufacturers are constantly working to shorten this development cycle.

The use of numerical modeling has become more and more important to help the engineer in this design process and to reduce the number of physical prototypes that will have to be built and tested. The availability of fast, simple and accurate modeling tools has thus become essential in automotive design and more specifically in the field of NVH.

The process of numerical evaluation of engine noise is summarized in Figure 1. Multibody simulation provides a method for estimating the forces acting on an engine during operation, taking into account various relevant effects such as combustion pressure and bearing loads. Together with a structural finite element model of the engine, they are used to evaluate the engine's structural response to operational conditions as a function of both RPM and frequency. The vibro-acoustic relationship between engine vibrations and the acoustic pressure field is then evaluated in order to calculate radiated engine noise.

This article illustrates a new approach, based on the Acoustic Transfer Vector concept, which is used to evaluate the vibro-acoustic response of a radiating structure in an efficient way enabling a faster engine development process.

Acoustic Transfer Vectors Methodology

The Acoustic Boundary Element method (BEM) is a well-established numerical method for solving acoustic radiation problems in unbounded domains.[1] In the classical approach, the acoustic response is calculated by solving the system of equations for each loading condition, preferably in a multiload case solution sequence. In general this is very time-consuming since the BEM system matrices (full, symmetric, complex and frequency dependent) need to be assembled and solved at each frequency and for each set of load cases.

The Acoustic Transfer Vector (ATV) concept consists of another organization of the calculation process whereby the objective is to reduce the computational effort to a minimum by reuse of calculated nominal Acoustic Transfer Functions. ATVs are indeed input-output relations between the normal structural velocity of the vibrating surface and the sound pressure level at a specific field point position.[4] ATVs, commonly also referred to as Contribution Vectors or Acoustic Sensitivities, can be interpreted as an ensemble of Acoustic Transfer Functions from the surface nodes to a single field point. They only depend on the configuration of the acoustic domain, i.e., the geometric shape of the vibrating body, the fluid properties (speed of sound and fluid density), the acoustic surface treatment modeled by local reacting acoustic impedance conditions, the frequency and the field point. They do not depend on the loading condition. The ATVs can be used in all applications where a one-way coupling is assumed between the structure and the acoustic field (vibrations of the structure not influenced by the fluid).

A highly efficient procedure for calculating ATVs has been implemented in LMS SYSNOISE Rev 5.5, limiting the computational cost of an ATV to the cost of a single frequency response calculation. The Acoustic Transfer Vectors from the radiating surface to specified field points are evaluated in a first step across the frequency range of interest at fixed frequency intervals. In a second step, the acoustic response in the field points is calculated for all loading conditions by combining the ATV with the normal structural velocity boundary condition vector at any frequency within the range. This ATV response calculation is a vector-vector product, given as,

$$p(\omega) = \{atv(\omega)\}^T \{v_n(\omega)\} \tag{1}$$

and involves negligible computation time.

An important observation is the fact that, in the case of sound wave propagation in an open space, the fluid domain around the radiating object exhibits hardly any resonant behavior. Therefore, ATVs are rather smooth functions of frequency. Coefficients can be accurately evaluated at any intermediate frequency, called slave frequency, using a mathematical interpolation scheme based on a discrete number of frequencies, called master frequencies. It is important to note that structural normal velocities cannot be similarly interpolated, since these directly depend upon the highly resonant dynamic behavior of the structure.

Another important advantage is that these frequency dependent ATVs can also be used for contribution analysis, i.e., by a 'partial' vector-vector product taking into account only the normal velocity boundary conditions on part of the radiating surface, i.e.,

$$p_c(\omega) = \sum_{e=1}^{n_p} \{atv^e(\omega)\}^T \{v_n^e(\omega)\} \tag{2}$$

whereby the superscript e denotes an element contribution. This way, the contribution of groups of elements corresponding to distinct panels of the structure can be derived, providing more insight into the noise generation mechanisms.

The engineering process to compute the structural normal velocity on a vibrating surface relies usually on the structural Finite Element Method and often on a modal superposition

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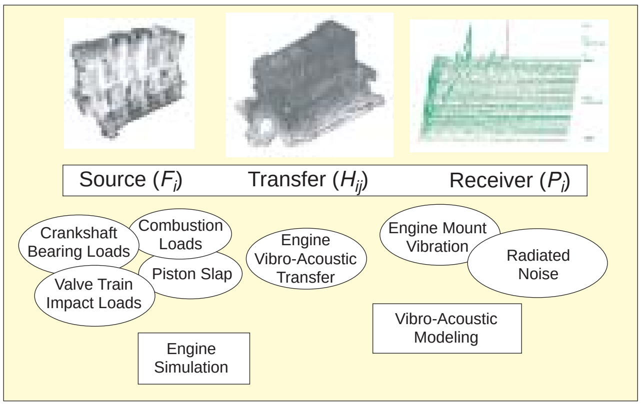

Figure 1. Engine noise modeling process.

approach, where the structural response is expressed as a linear combination of the mode shapes of the body as in the following relation

$$\{v_b(\omega)\} = i\omega[\Phi_n]\{mrsp(\omega)\} \quad (3)$$

where $[\Phi_n]$ is the matrix composed of the modal vectors, projected on the local normal direction of the boundary surface and $\{mrsp(\omega)\}$ is the modal response (vector of the modal participation factors) of the structural model at a given excitation frequency.

Combining (3) with Equation (1) leads to,

$$p(\omega) = j\omega\{atv(\omega)\}^T\,[\Phi_n]\{mrsp(\omega)\} \quad (4)$$

where

$$i\omega\{atv(\omega)\}^T\,[\Phi_n] = \{matv(\omega)\}^T \quad (5)$$

is called the Modal Acoustic Transfer Vector (MATV) which can be directly combined with the modal response vector to give the sound pressure at a field point:

$$p(\omega) = \{matv(\omega)\}^T\{mrsp(\omega)\} \quad (6)$$

MATVs are the modal counter part of ATVs. They express the acoustic transfer from the radiating structure to a field point in modal coordinates and therefore contain the acoustic contributions from each individual structural mode. The acoustic response in the field point is obtained by recombination of the MATV with the corresponding structural modal responses. Working in modal coordinates results in an important data reduction. It's clear however that MATVs are no longer independent of the structural model as they are linked to the structural modal basis. Whenever the structural modal basis changes, e.g., due to structural design modifications, the set of MATVs needs to be reevaluated. From Equation (5) it is clear however that, for a given structural mode set, the corresponding set of MATVs can easily be regenerated by projecting the ATVs, independent of the structural model, into the modal space. It's important to note that this quick generation of MATVs by projecting the ATVs into a new modal basis is only valid if the acoustic configuration has not been changed due to structural design modifications.

A wide range of different acoustic response calculation sequences based on ATVs and MATVs has been implemented in the software.[4] For the application at hand a multi-RPM MATV response solution sequence was chosen whereby the structural modal response calculated in MSC/NASTRAN v.70.5 for all load cases is imported in the acoustic model for evaluating a complete acoustic signature of the engine, as explained in the following sections.

Numerical Procedure

Structural Model and Excitation. The dynamic FEA powertrain system model (Figure 2) which includes the cylinder block, the oil pan, the right and left cylinder head, the front cover and the right and left cam cover, was developed to perform the NVH assembly analyses, for a total of 124,145 elements and 160880 nodes. The engine dynamic loads used in the MSC/Nastran frequency response analysis (solution 111) are: the main bearing loads (Figure 3), the major and minor piston side thrust loads (Figure 4) and the combustion pressure force on

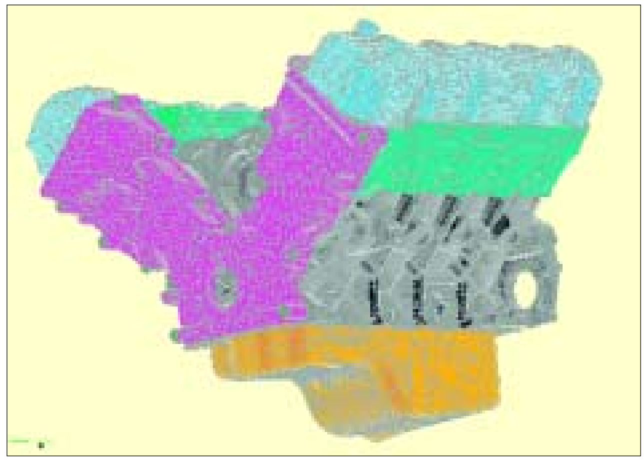

Figure 2. Finite element model of the baseline engine assembly.

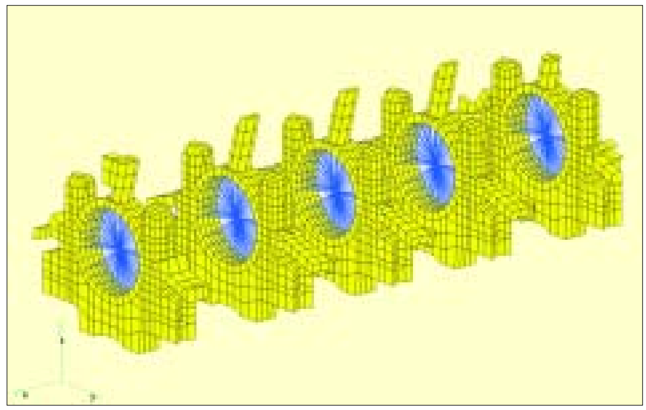

Figure 3. Main bearing loads.

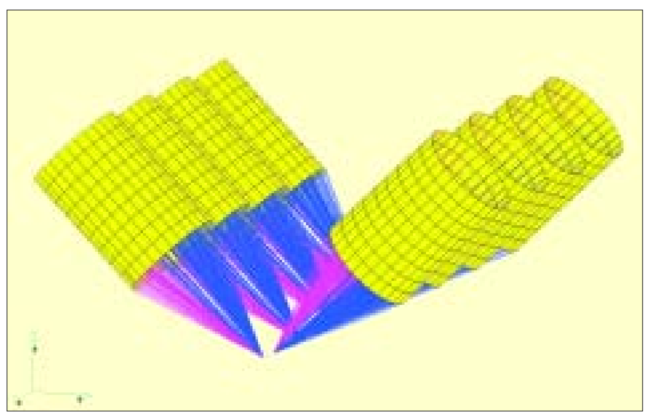

Figure 4. Piston side thrust loads.

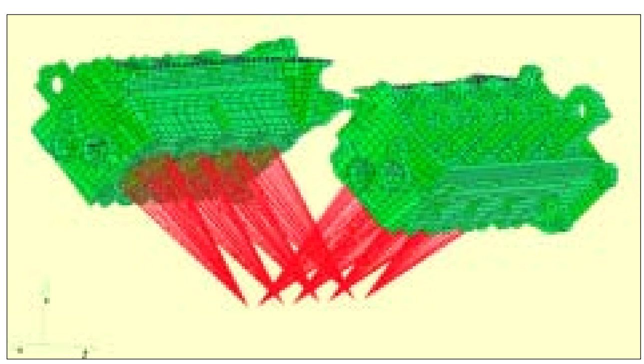

Figure 5. Combustion pressure loads.

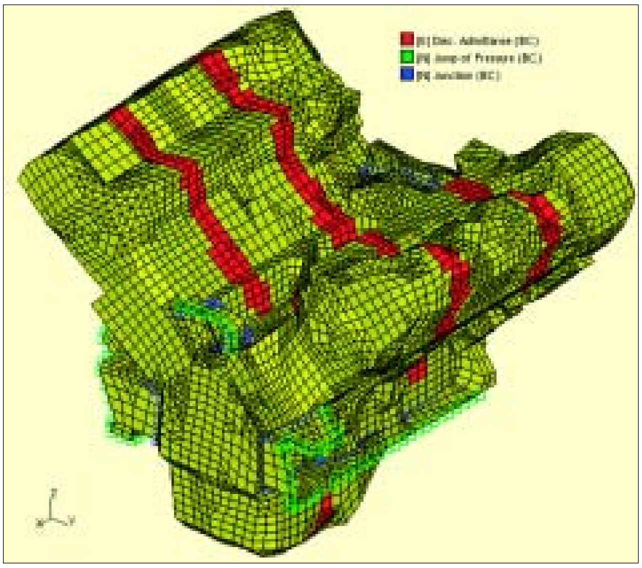

Figure 6. Acoustic Boundary Element model of the engine.

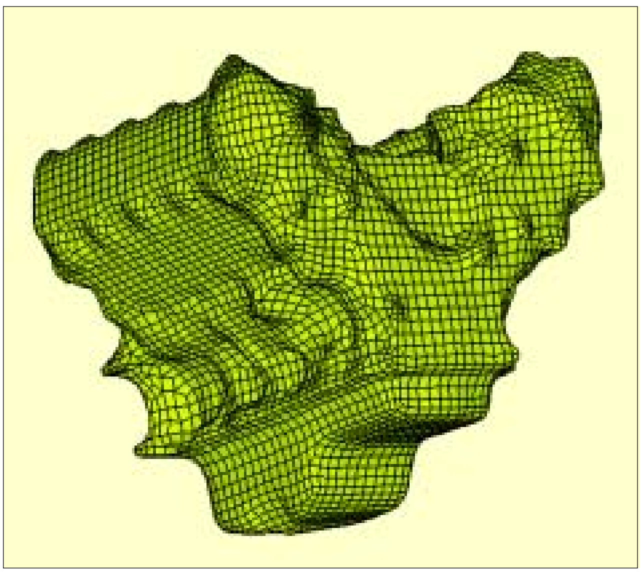

Figure 7. Optimized mesh obtained using Virtual.Lab Pre/Acoustics.

the heads (Figure 5). These forces are computed for each rpm-value. The outputs from the MSC/Nastran analysis are the natural frequencies, mode shapes and the modal responses.

Acoustic Boundary Element Model of the Engine. The acoustic model has been defined in LMS/SYSNOISE Rev 5.5. Figure 6 illustrates the acoustic Boundary Element model of the engine and consists of 11359 elements and 9034 nodes.

This reduced mesh is the result of a mesh coarsening procedure applied on the structural FE mesh with the help of the dedicated software LMS Pre/SYSNOISE Rev 5.4. The size of the elements is small enough to describe the acoustic wavelength corresponding to the highest frequency of interest. In this case it satisfies the classical 6 elements per wavelength criteria up to 1.5 kHz.

The acoustic mesh generation is time expensive (it usually takes a couple of weeks) and represents the major bottleneck of the proposed approach. Virtual.Lab Pre/Acoustics allows the user to generate an acoustic mesh within hours. Figure 7 represents a typical mesh obtained using this new tool. The mesh contains 7232 elements and 7224 nodes and is valid up to 3 kHz. This resulting mesh is optimized for acoustic BEM computations. More importantly, it is obtained in less than two hours. Unfortunately, Virtual.Lab Pre/Acoustic was not available at the time of this study and the results shown here were computed using the mesh shown in Figure 6. New results obtained using the optimized mesh will be presented in the future together with experimental validations.

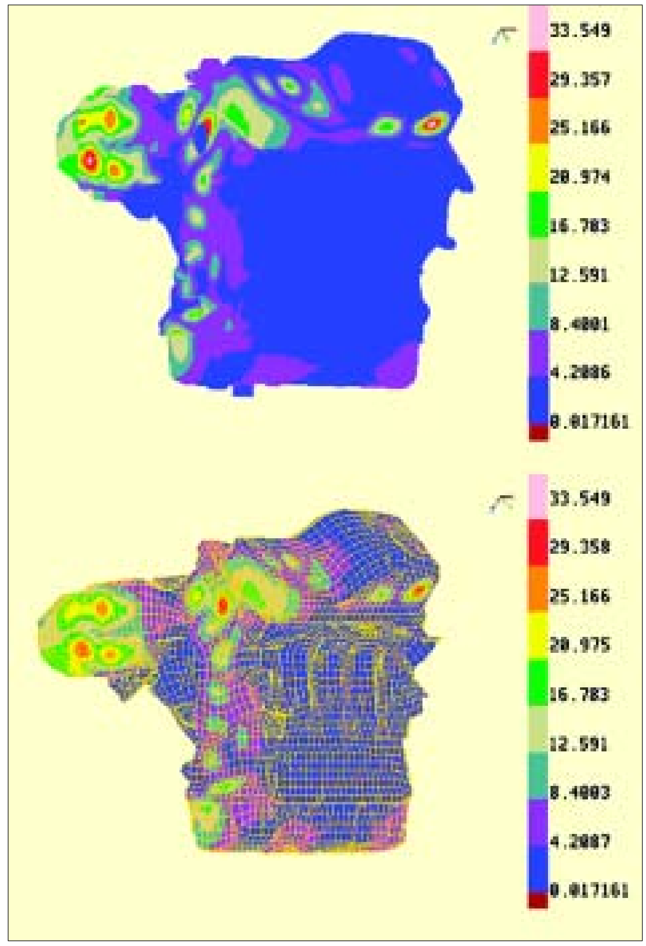

Figure 8. Mode shape comparison. Upper: full FE model; lower: reduced FE model.

A common numerical problem encountered in Boundary Element analysis is the so-called nonuniqueness problem at specific frequencies.[3] These irregular frequencies relate to resonance frequencies of the (imaginary) cavity enclosed by the BE mesh. A possible solution to solve this problem is to apply acoustic impedance boundary conditions on some elements at the interior side of the BE mesh. The characteristic impedance of air rc, where r is the mass density and c the speed of sound propagation, provides a sufficiently high amount of acoustic damping to alleviate this nonuniqueness problem.

Mesh Incompatibility Handling. For the purpose of compatibility of the structural and acoustic models involved in the vibro-acoustic coupling, a reduced structural FE model has to be defined which is compatible with the acoustic mesh. The mode shape data of the full FE model was transferred to the reduced model, using a geometric mapping scheme. The validity of the reduced model, often referred to as surrogate model for describing the highest frequency modes, as well as the quality of the mapping, was verified by a visual comparison, as illustrated in Figure 8.

Results of the Radiated Noise Evaluation

Process Flow. The acoustic evaluation procedure is summarized in Figure 9. In the acoustic model, the ATVs are computed across the frequency range of interest. They are then projected onto the space spanned by the normal component of the structural mode shapes to get the MATVs at different frequency intervals.

In the structural model, mode shape data are used together with the modal responses to get the deformation shapes of the structure at different excitation frequencies for all RPM load cases. The definition of a so-called weak fluid-structure link, i.e., one way vibro-acoustic coupling between those two mod-

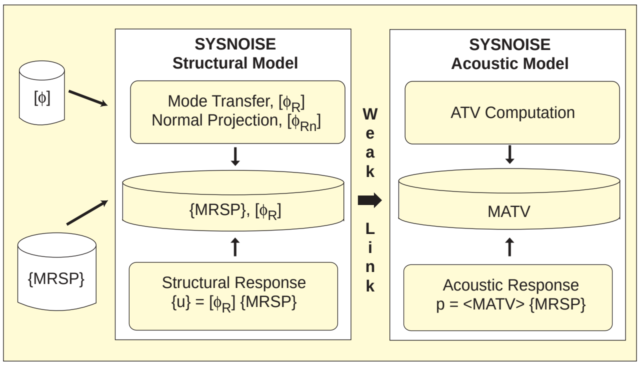

Figure 9. Acoustic evaluation procedure.

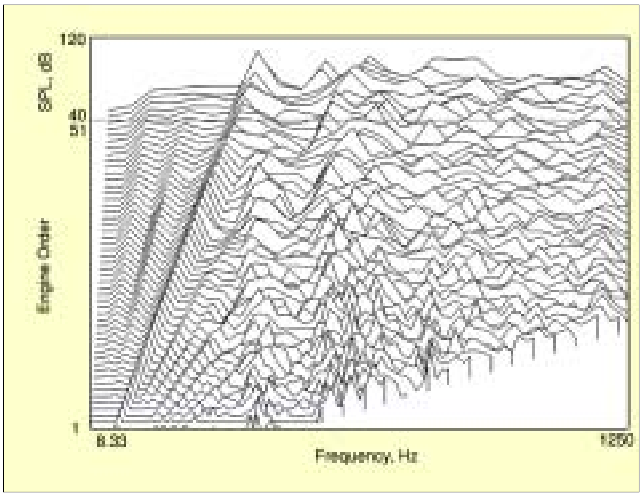

Figure 10. Waterfall diagram of sound pressure levels above the engine.

els, enables evaluation of the acoustic pressures at all predefined field points for a list of RPM load cases and excitation frequency values of interest.

The calculation parameters are as follows:

- ATVs are evaluated within a frequency range from 25 to 1250 Hz, with steps of 25 Hz (50 'master' frequencies only).
- A cubic spline interpolation scheme is used to estimate the ATV at all excitation frequencies in the range of interest. The requirements on the number of master frequencies, from which the ATV curves are obtained at any frequency, depends on the shape and smoothness of the frequency dependency of the ATV.
- Modal Responses are imported in the software, leading to an automatic creation of 52 RPM-related load cases.
- The multiload case calculation is done for the first 60 half-orders of crankshaft rotation. Note that calculation on the basis of orders of the rotation speed implies that the excitation frequencies are different for each RPM. Using the classical BE method where each frequency is processed separately, it would thus necessitate an enormous number of calculation sequences. In the MATV-response approach, the modal acoustic transfer relation is, when necessary, interpolated at the excitation frequency and recombined with the modal response in a highly efficient calculation.

As a result, the RPM- and frequency-dependent acoustic pressure are obtained and can be represented in the form of waterfall diagrams or as a color map on the field point mesh. As such, the (M)ATV approach does not enable the evaluation of acoustic power radiated by the system. However, this problem can be solved for a structure which, like an engine, radiates more or less uniformly in every direction through the use of the ISO3744-1994 procedure. This procedure relates to experimental evaluation of radiated acoustic power from the sound pressure level at a limited number of points (without phase information). Indeed, the use of the acoustic pressure values at 20 evenly distributed locations is sufficient to get a very accurate estimate of radiated acoustic power provided that the source does not radiate exclusively towards particular directions.

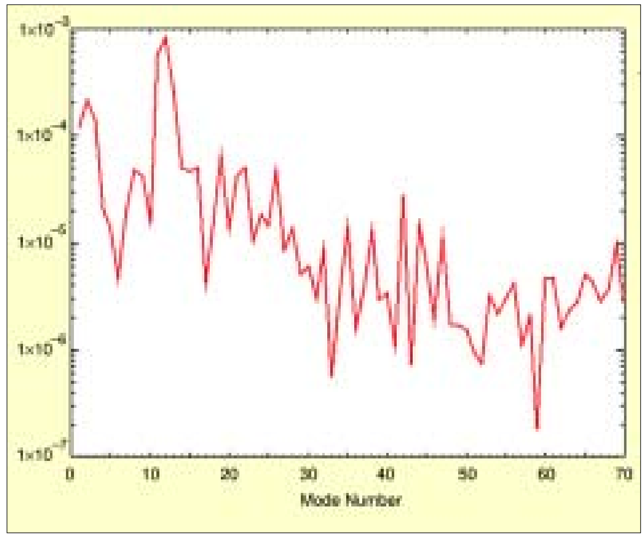

Figure 11. Modal participation factors (log magnitude scale vs. mode number).

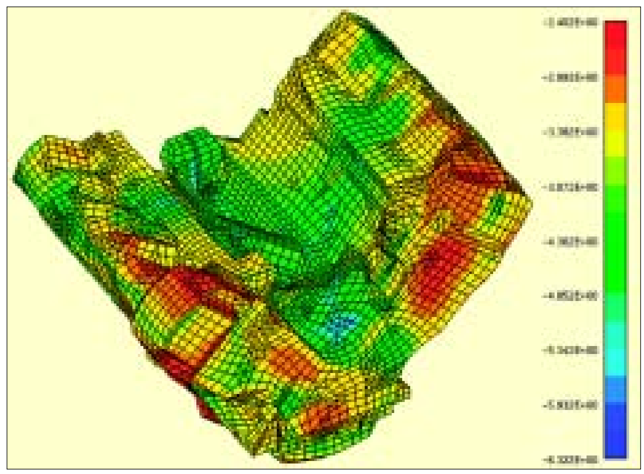

Figure 12. Deformation plot at 6000 RPM, 400 Hz.

Results of the Procedure. Figure 10 shows sound pressure levels, calculated 1 m above the engine, between 800 and 6000 RPM and for frequencies ranging from 8 to 1250 Hz. This diagram shows the orders from the engine excitation resulting in high levels at a frequency proportional to the RPM.

Relevant structural modes can be easily detected from the this graph as they result in high levels at a single frequency throughout the RPM sweep. This leads to finding the most critical RPM and frequencies. With this information, modal participation factors and deformation shapes at critical frequencies can be generated as illustrated in Figures 11 and 12. A subsequent analysis of the contribution of various engine panels to total acoustic excitation can then help the engineer understand the mechanisms of noise generation.

Computational Speed Comparison with Classical Approach. Direct evaluation of acoustic pressures with the Boundary Element method involves 2 steps: 1) the acoustic potentials on the surfaces defining the acoustic field are derived from the knowledge of velocity boundary conditions; 2) field point pressures are derived from potentials at any field point position of interest through an evaluation of the surface Helmholtz integral. The first phase is the most calculation intensive as it involves the solution of a dense, complex, symmetric system of equations.[1,2]

In the present case, we were interested in the results at 44 field points resulting in the following elapsed-time-statistics on a HP-C3000 workstation:

Classical Approach
1. Surface potentials – 30 min per frequency.
2. Field point pressures processing – 5 sec per frequency.

MATV Approach
1. ATV evaluation – 42 min per frequency for all field points.
2. MATV projection – 10 min for 70 structural modes at all frequencies.
3. Combination with MRSP – 7 sec per frequency.

In the present case, where the first 60 half-orders of 52 RPM cases are considered, the direct approach would therefore need approximately [(30+5/60)(60´52)]/60 = 1564 hours, i.e., >60 days. The MATV approach only needs [(50´42) + 10 + (7/60)(60´52)]/60 = 41 hours, i.e., <2 days

Conclusion

This article presents a new methodology for solving acoustic radiation problems in a more efficient way using the so-called Acoustic Transfer Vector concept. This approach is illustrated through the evaluation of the acoustic signature analysis of an engine during a run-up. The gains in terms of computation time, subsequent reduction in the amount of data to handle and improved efficiency of post-processing steps show the validity of the new method for solving large vibro-acoustics problems.

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