Predicting the Effect of Engine Structural Design Changes on Radiated Noise for Full Frequency Spectrum

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Abstract

In recent years, low and mid frequency analysis of engine has shown that many design decisions influenced negatively the acoustic performance of engines. To counteract this problem, design engineers have started to design engine covers that actually reduce the noise radiated before it gets too far from the engine. These methods basically modify the path the noise would take to get to a vehicle occupant's ear. Another approach is to modify the design of the engine construction itself in order to reduce the noise generated at the source. This paper presents a method to predict radiated noise from an engine for the full frequency domain of analysis (0-10000 Hz). This approach combines the Finite Element Method (FEM) to represent the structure and its load cases and the new Fast Multipole Boundary Element Methods (FMM-BEM)[1] to represent the fluid surrounding the engine. This approach allows engineers to add acoustics to their existing structural predictions and investigate the effect on radiated sound power of modifying the structure, adding beads, adding damping treatment or acoustic covers in the vicinity of an engine. Typical engine noise radiation results with these changes along with cases with and without acoustic covers are presented and discussed.

Introduction

Traditionally, design of automobile engine has been based primarily on the prediction of vibration levels of the outer skin. Controlling vibration levels and removing as much as possible vibration peaks provided a simple way of preventing noise issues at an early design stage. Today, new advances in radiated noise predictions make acoustic predictions also available at an early design stage for full frequency range of interest (0-10000 Hz). The global process to follow includes the definition of the loads, the representation of the vibro-acoustic paths and the post-processing of the acoustic results. The vibro-acoustics model can be described as a structural path and an acoustic path (Figure 1).

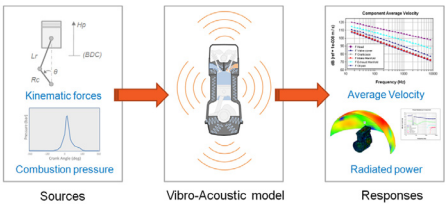

Figure 1: Modeling process from source to receiver

This paper focuses on the addition of acoustics to current engine design process. It assumes that the main forces acting on the structure are well understood and represented in a FEM structural model. Main excitations are forces at crankshaft bearing housings coming from internal moving parts such as pistons, cranks, etc. It also includes piston slap forces, valve seat and camshaft bearing forces and finally combustion pressure. In this study, a simple 1/3 octave force spectrum is applied on crankshaft bearing housing and valve seats to illustrate the modelling process (Figure 2).

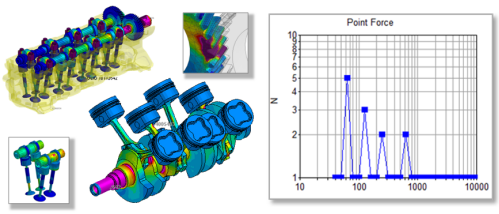

Figure 2: Left: Source modeling examples - moving parts modeled using FE. Right: Simplified force spectrum applied at crankshaft bearings and valve seats

Structural model

The FEM model of the structure is composed of five parts: valve cover, valve train, engine block, bulkheads and oil pan (Figure 3). To simplify illustration of concept, all parts are rigidly connected together. The FEM mesh resolution is sufficient to allow computation to 10 kHz. The structure's first mode is at 1169 Hz and the total number of modes up to 10 kHz is 263 modes

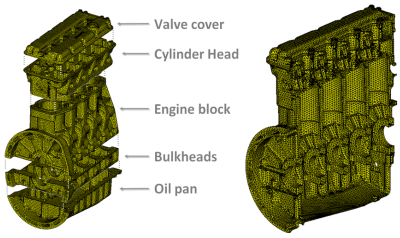

Figure 3: Engine FE model components

Design changes are introduced to study effect on radiated noise. From a nominal configuration, 3 modifications were made to the engine structure: the cylinder coolers were rebuilt into a round and square shape and beads were added to the oil pan (Figure 4). Vibration response is computed and only velocity at outer skin nodes is retrieved.

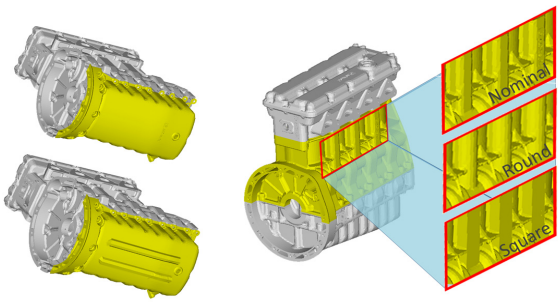

Figure 4: Design changes - Left: beading on oil pan. Right: Cylinder coolers shape modification

Acoustic model

To represent the fluid around the engine, the FMM-BEM method is used. This method provides a detailed description of the radiation and scattering of waves from a vibration body. A boundary mesh is automatically created using a shrink wrapper (Figure 5) and coarsened to fit maximum frequency required mesh density. A projection algorithm is used to handle incompatible mesh density between structural and acoustic meshes. Finally, a multicore aware FMM-BEM is used to compute sound powers radiated and SPL at any recovery mesh node [2].

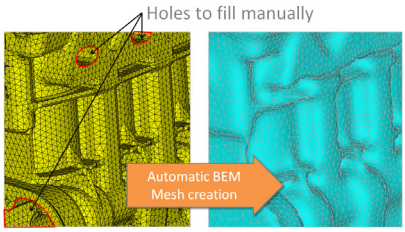

Figure 5: Creating acoustic mesh using shrinkwrap

Computation process

Computation process is illustrated in figure 6. Sources can be defined as forces on the FEM structural model, as modal participation factors or complex velocity field on engine outer skin nodes. Using the precomputed modal basis, the velocity on the outer skin of engine can be computed. Note that the modal basis needs to be computed only once.

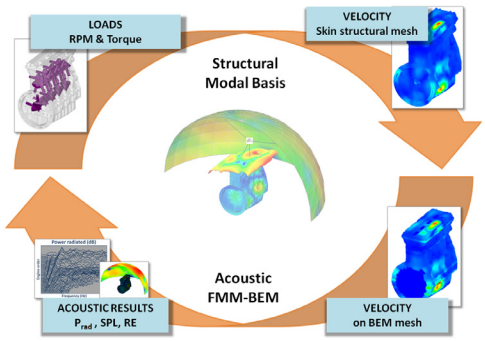

Figure 6: Computation process

Once the velocity response is computed, it is automatically projected onto the acoustic BEM mesh. Using FMM-BEM the sound radiated power can be computed directly.

Results

Sound power radiated and panel contribution are presented in figure 7. It has been shown that sound power radiated by a vibrating surface is proportional to the product of the average panel quadratic velocity and radiation efficiency [3].

$$\Pi_{rad} = \sigma A \rho_0 C v_{rms}^2 \qquad (1)$$

Radiation efficiency can be computed from (1) and compared with the so-called ERP (Equivalent Radiated Power) radiation efficiency which assumes a constant radiation efficiency value of 1 (one) over the whole frequency domain. Considering only velocity as an indicator of sound power radiated might lead to large errors. Below 1000 Hz the velocity is fairly high but on the contrary radiation efficiency is fairly low, therefore assuming that sound radiated would be solely proportional to velocity as in the so-called ERP (Equivalent Radiated Power) method, one would dramatically overestimate the radiated power.

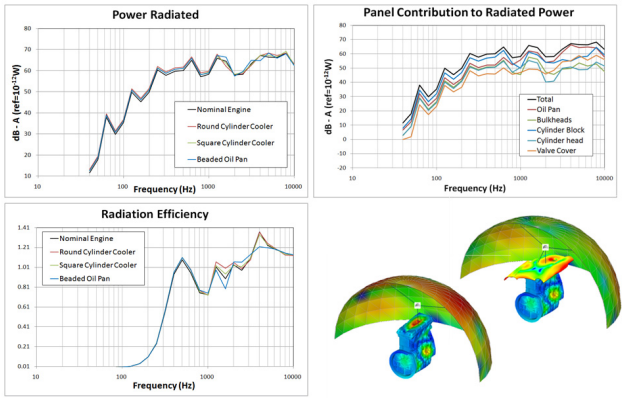

Figure 7: Typical results available

The process described in this paper also permits the insertion of covers between the engine and any recovery points. Figure 7 presents effect of cover on the SPL distribution around engine. Results presented take less than one hour on a 4 processors 64 bit workstation. Effect of beading can also be detrimental to sound radiated [4]. Detailed results can be found in [5].

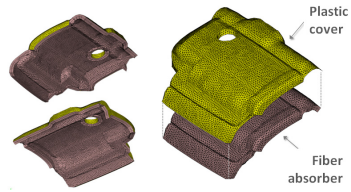

Figure 8: FE model of plastic cover and fiber layer

Conclusion

FMM-BEM combined with FEM structural model permits computation of sound power radiated by an engine up to 10 kHz in less than an hour. Thanks to AC Tech for providing ESI with a CAD model of the engine used in this paper.

Bibliography

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