ICSV13 - Vienna
The Thirteenth International Congress on Sound and Vibration
Vienna, Austria, July 2-6, 2006

SIMPLIFIED INTEGRAL ENERGY METHOD: APPLICATION TO PASS BY NOISE

B. ANDRO *[1], S. CHAIGNE [1], T. SCHMITT [2] and A. SHAH [3]

1 RENAULT Research Department, TCR AVA 1 63, 1 av. Golf, 78288 Guyancourt, FRANCE
bertrand.andro@renault.com
2 RENAULT NVH Department, Le Parc de Gaillon, 27940 Aubevoye Cedex, FRANCE
3 Currently at KTH, MS Student, Sound & Vibration, The Marcus Wallenberg Laboratory, SE 100, 44 Stockholm, SWEDEN.

Abstract

The pass-by noise measurements defined in a standard procedure constitute a legal test for every new vehicle. Nowadays, the improvements of the engineering process allow automotive manufacturers to reduce the vehicle development cycles. Consequently, the acoustic optimization of the vehicle applied to reduction of the exterior noise needs to be considered as soon as possible to avoid repeated road tests depending strongly on the environmental conditions.

At the early stage of the development process, Renault would like therefore to use an accurate tool which predicts the engine compartment contribution to pass-by noise. This model will give indications to answer technical issues like: the influence of acoustical materials or height of the vehicle on pass-by noise in the high frequency range.

In medium and high frequency domains, classical numerical methods such as the Finite Element Method (FEM) or the Boundary Element Method (BEM) are not well suited to predict the engine contribution because of the prohibitive computation time and memory occupation. Some energy methods such as statistical energy Analysis (SEA) will only give global values in each substructure and are not suited to outside airborne noise propagation.

In this paper, a simplified integral energy method is developed to predict the noise induced by the engine sources during the pass-by noise test in the medium and high frequency range. We will consider a local energy balance and solve an integral equation to predict the noise emitted by the engine in a short computation time. One of the main contributions of this paper is the calculation of the visibilities between elements to take into account the presence of the engine. Standard pass-by noise measurements done on a test track are compared with those obtained by the simplified energy method.

INTRODUCTION

Nowadays, improvements of the engineering process allow automotive manufacturers to reduce the vehicle development cycles. Consequently, acoustic optimization of the vehicle applied to reduction of the exterior noise needs to be considered as soon as possible to avoid repeated road tests.

The pass-by noise measurements are defined in standard procedure ISO 362 [1]: it determines the maximum SPL radiated by an accelerating vehicle under specific conditions. Microphones are placed at 7.5 m from the vehicle axis on both sides. The noise produced by the vehicle during a run up (full open throttle on the track) is measured simultaneously on both microphones as mentioned on figure 1.

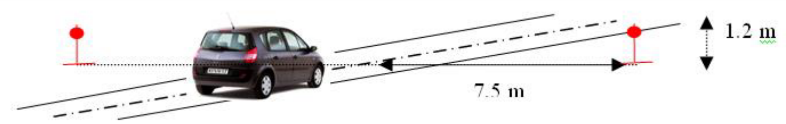


Figure 1 : Pass-by noise test

As mentioned above, a simple and accurate tool is needed to predict the engine contribution to pass-by noise in the early stages of the design process. This method should cover the outside airborne noise propagation of the engine sources in a wide frequency range: from 500 to 5 000 Hz. Classical deterministic methods such as the finite element or boundary element methods, which are based on the resolution of the Helmholtz equation, are well suited to solve problems in the low frequency range. For vehicle-sized problems however, this type of formulation cannot be extended to this kind of frequency range because of computation time and memory occupation issues. As a consequence, several type of energy variables-based methods have been developed to tackle this problem.

One of the most famous method called SEA (Statistical Energy Analysis) [2] consists of substructuring the whole domain and using a global energy balance between finite subsystems. No information will be given by SEA about the energy distribution in a domain, which is a significant drawback since an averaged energy value in each subdomain is not suitable in our case to make a reasonable acoustic optimization. In this paper, a local energy balance is considered to solve an integral equation in order to get information on spatial energy distribution in a short computation time. Some references [3][4][5][6][7] give in-depth formulations used in the context of local energy approach applications. The main goal of this paper is to compare computed results using the simplified integral energy method and standard pass-by noise measurements in an industrial context.

1. SIMPLIFIED INTEGRAL ENERGY METHOD

In order to describe the energy transfer in the domain, two energy variables are considered in the formulation: The first one is the total energy density W defined as the sum of the kinetic energy density and the potential energy density. The second one is the energy flow $\vec{I}$.

1.1 Main Assumptions

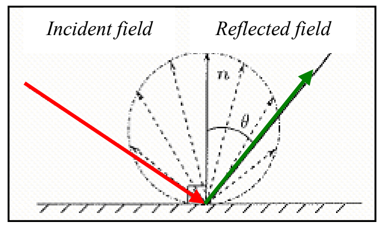
The following assumptions will be done to get the energy formulation:
(a) The system is supposed to be linear, isotropic, homogeneous in steady states conditions.
(b) A propagative approach will be used: only propagative waves are taken into account (no evanescent and near field waves effects). This assumption leads to the following fundamental formulation:

$$\vec{I} = cW\vec{u} \tag{1}$$

where $\vec{u}$ is the unit vector and c the wave velocity.
(c) The effects of interference between propagative waves will be neglected. As a consequence, energy quantities will be added: propagative waves are considered to be uncorrelated in the high frequency range (no phase consideration).
(d) Reflection on the surfaces is supposed to be diffuse and to follow Lambert's law:

$$\sigma(P,\theta_P) = \sigma(P)\cos\theta_P \tag{2}$$

where θ is the angle between the normal $\vec{n}$ and the propagation direction.



1.2 Green Functions

In order to determine $\vec{I}$ and W, it is necessary to calculate the expressions relative to the direct field called energetic Green kernels and noted G for the energy density kernel and $\vec{H}$ for the intensity kernel. Considering a source point S and an observation M, an energy balance relationship can be written as follows:

$$div_M\,\vec{H}(S,M) + mcG(S,M) = \delta_s(M) \tag{3}$$

Where: - The first term represents the outgoing power per volume
- The second term represents dissipated power density (m is the factor of atmospheric absorption and c the velocity of sound)
- The right hand side corresponds to the injected power

The (b) hypothesis allows us to write:

$$\vec{H}(S,M) = cG(S,M)\vec{u} \tag{4}$$

Where $\vec{u}$ is the unit vector from S to M.
Solutions of equations (3) and (4) in 3 dimensions then verify:

$$G(S,M) = \frac{1}{4\pi c}\frac{e^{-mSM}}{SM^2} \tag{5}$$

$$\vec{H}(S,M) = \frac{1}{4\pi}\frac{e^{-mSM}}{SM^2}\vec{u}_{SM} \tag{6}$$

Where SM is the distance between the source and the observation point.

1.3 Integral Formulation

To get the integral formulation we need to determine the expressions of W and $\vec{I}$ in a domain Ω with boundaries δΩ.

The following figure represents this domain containing a source S with an amplitude ρ and surface potentials can be seen as secondary sources with an amplitude σ on the boundaries of the domain.

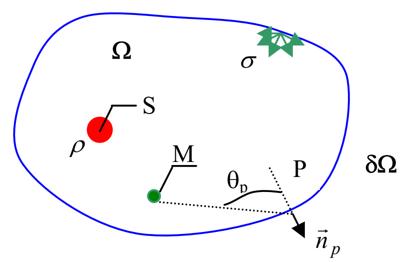


Figure 2 – Notation for the acoustic problem

Two principles can now be applied to build the energy distribution considering Green kernels. The hypothesis of uncorrelated waves (c) allows to apply the principle of linear superposition of energy quantities.

Huygens' principle then allows to build the total energy field as the superposition of the direct field components resulting from primary sources ρ inside the domain Ω and the reverberated field resulting from secondary sources σ of surfaces located on the boundary δΩ. One can thus write the following energy expressions:

$$W(M) = \int_{\Omega} \rho(S)G(S,M)dS + \int_{\delta\Omega} \sigma(P,\theta_P)G(P,M)dP \qquad (7)$$

$$\vec{I}(M) = \int_{\Omega} \rho(S)\overrightarrow{H}(S,M)dS + \int_{\delta\Omega} \sigma(P,\theta_P)\overrightarrow{H}(P,M)dP \qquad (8)$$

Primary sources are supposed to be known. The problem thus consists in calculating all the secondary sources located on the boundaries of the domain. Hypothesis (d) allows us to transform the secondary sources expression:

$$\sigma(P,\theta_P) = \sigma(P)\cos\theta_P \qquad (9)$$

1.4 Integral Resolution

The last step consists in writing a local energy balance at the point P by introducing the absorption coefficient α. The intensity balance is:

$$I_{reflected} = (1-\alpha)I_{incident} \qquad (10)$$

Formulation:

We can rewrite the corresponding power balance equation and recognize the Fredholm integral equation of the second kind on the potential σ:

$$\frac{\sigma(P)}{4} = (1-\alpha).[\int_{\Omega} \rho(S)\overrightarrow{H}(S,P)dS + \int_{\delta\Omega} \sigma(Q)\cos\theta_Q \overrightarrow{H}(Q,P)dQ\,].\vec{\mathrm{n}}_{\mathrm{P}} \qquad (11)$$

Equation (11) can now be rewritten (Id: identity operator):

$$\boxed{(Id - T)\sigma = g} \tag{12}$$

Where:

$$T.\sigma = \int_{\partial\Omega} \sigma(Q).K(Q,P)dQ \quad (13) \quad \text{and} \quad K(Q,P) = (1-\alpha).\frac{e^{-mPQ}}{\pi PQ^2}.\cos\theta_P \cos\theta_Q \tag{14}$$

$$g(P) = (1-\alpha).\int_{\Omega} \rho(S)\frac{e^{-mPQ}}{\pi.SP^2}.\cos\theta_P \, dS \tag{15}$$

Discretization:
We recognize primary sources in the term g. In order to determine the boundary unknowns σ (secondary sources), one just has to invert the system *(Id - T)*. Existence and uniqueness of the solution are investigated in [5].
In our case, the engine compartment was discretized and the collocation method (piecewise constant) was used to allow a fast computation.

Post processing:
Since we are using an integral formulation, one can now use equations (7) and (8) to calculate the energy density or the intensity at any specific point in the domain.

1.5 Visibility Calculations

It remains to mention that in the case of a concave domain boundary, obstacles can interfere between points P and Q. It is thus necessary to introduce a binary coefficient of visibility *v* (1 if no obstacle and 0 otherwise) between points P and Q in the expressions of T and g.
A simple algorithm was derived and figure 3 shows a typical result that one can get: one can observe the visibilities between an observation point located in the engine compartment side on the dash and the elements of the scene (inside engine compartment + ground). From the observation point it is not possible to see all the elements of the inside engine compartment because of the position of the engine. Some elements are visible on the ground through different apertures.

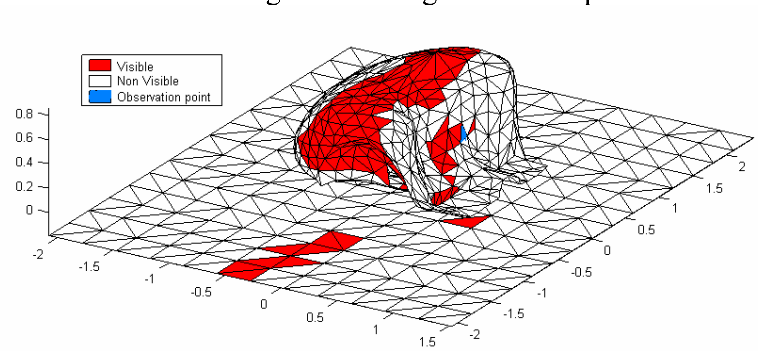


Figure 3 – Example of visibility coefficients

2. VALIDATION WITH EXPERIMENTS

In this second part, comparisons between experiments and computed results will be done. Measurements were made using a Renault Scenic minivan equipped with a 1.5 l direct injection Diesel engine rated at 100 bhp. Measurements were done on an ISO test track for operational condition testing considering a constant speed corresponding to 2500 rpm in 3rd gear. Various configurations of the engine compartment were tested: serial state, panel masking using septum-foam layers (leaving only the airborne path), additional absorbing treatments, removal of some absorbing treatments. These various configurations were done in order to assess the robustness of investigated methods as well as their ability to correctly represent absolute levels. During the test, exhaust and tyre noise were removed during post processing or masked during testing to keep the only contribution of the engine.

2.1 Absolute Values

First comparisons between experimental and numerical results will be done in the serial state configuration considering the Sound Pressure Level for different positions of the vehicle on the ISO track (left hand side).

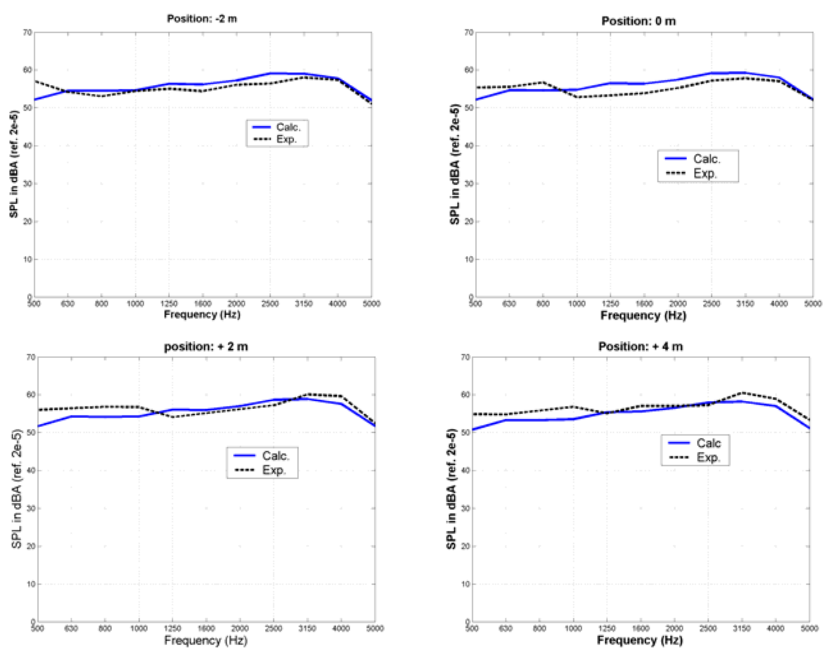


Figure 4 – SPL for the Serial State

Figure 4 shows a comparison between measured and predicted sound pressure levels between 500 and 5 000 Hz on the left hand side in 4 pass-by tests positions. The general trend of the frequency dependence of the SPL was well predicted. The main differences between numerical and experimental data arise in the lowest frequency range (until 1000 Hz) where the SPL is underestimated by numerical estimations by approximately 3.5 dBA. This type of conclusion was expected because the energy based method hypothesis are especially valid in the high frequency range.

2.2 Robustness Of The Method

This numerical tool has been developed in order to simulate various engine bay configurations, under various running conditions, associated with pass-by noise test. As a second step, we investigated the trends in terms of sensitivity to absorption and isolation under operational conditions to evaluate the method's potential for solutions design prediction.

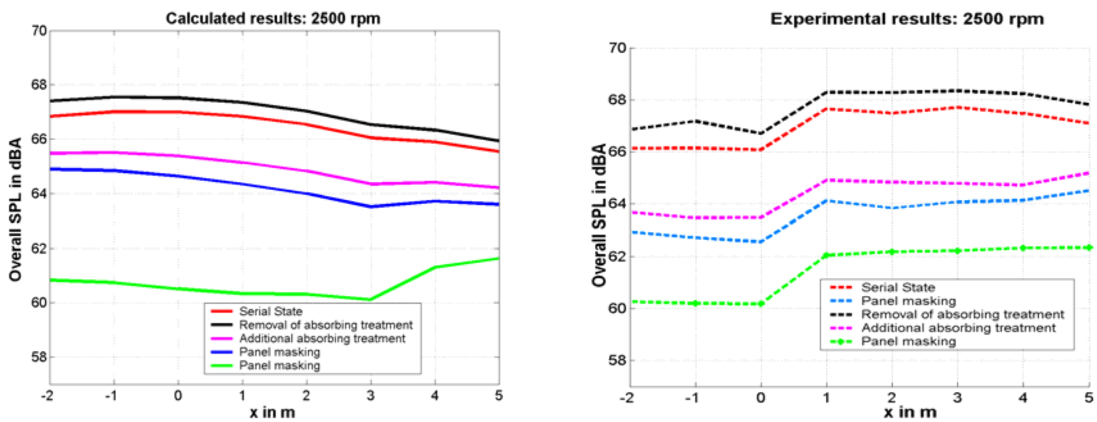


Figure 5 – Overall SPL for different configurations

Figure 5 shows the overall SPL (500 – 5000 Hz) in dBA on the left hand side for different configurations. We can see that the absolute values are very well represented for all the configurations. The differences never exceed 2 dBA. Consequently, the trends in term of absorption and isolation are well evaluated: Although the absolute values of SPL obtained from the code are slightly different from those obtained experimentally, the nature of relative changes is well represented. It should be recalled that this was the primary goal behind developing the code i.e. to get an estimate of the benefit or deterioration that is expected after a certain change in the compartment parameters has been made.

2.3 Practical Use

Post processing capabilities constitute another advantage of this numerical approach: it is possible to visualize the intensity field or the secondary sources distribution on the boundaries of the domain. Therefore, this tool allows the user to know easily the most impacted areas of the engine compartment according to the frequency but also to represent the effects of new technical solutions on the distribution of energy within the engine compartment and on the distribution around the engine compartment on the ground. The following figures give the distribution of the secondary sources on the third octave band 3150 Hz with and without absorbing material on the underhood: one can see higher secondary sources values on the ground and on the hood when one removes the underhood absorbing material.

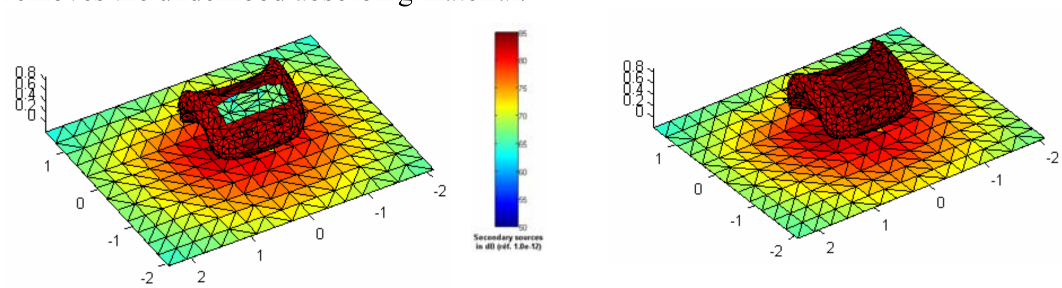


Figure 6 – Secondary sources distribution (3150 Hz) with and without absorbing treatment

CONCLUSIONS

In this paper, we have presented a method based on the resolution of an integral energy equation after having considered a local energy balance on the boundaries of the domain. Comparisons between pass-by noise tests and numerical results are encouraging in an industrial context: small computation time, good absolute SPL values and robustness of the method. Moreover, the post processing capabilities could reduce the optimization loops during the design process. Including apertures diffraction could be studied in the future to improve the absolute values.

REFERENCES

[1] ISO 362, Acoustics, Measurement of noise emitted by accelerating road vehicles, Engineering method, International Organization for Standardization.

[2| Lyon R.H., “Statistical energy analysis of dynamical systems: theory and application”, Cambridge, Massachusetts, MIT press, 1975.

[3] Le Bot A., “Equations énergétiques en mécanique vibratoire : application au domaine des moyennes et hautes fréquences“, PhD thesis, Ecole Centrale de Lyon, 1994.

[4] Ichchou M. N., “Formulation énergétique pour l’étude moyenne et hautes fréquences des systèmes : théorie et applications“, PhD thesis, Ecole Centrale de Lyon, 1998.

[5] Le Bot A., Bocquillet A.,“Comparison of an integral equation on energy and the ray tracing technique in room acoustics“, Journal of Acoustical Society of America (108), p 1732-1740, 2000.

[6] Schmitt T., “Diffraction and ground effects in simplified energy methods“, ICSV, 2002.

[7] Schmitt T., “Modélisation des transferts acoustiques en moyennes et hautes fréquences par méthode énergétique : application à l’encapsulage des compartiments moteur“, PhD thesis, Ecole Centrale de Lyon, 2004.