

A Procedure for Efficient Trimmed Body FE Simulations, Based on a Transfer Admittance Model of the Sound Package

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ABSTRACT

Porous materials are extensively used in the construction of automotive NVH parts. The sound package design during vehicle development requires simulation methods at vehicle level that can take into consideration the dynamical behavior of porous materials. This need has led to different numerical technologies based on Biot's equations. In particular, direct FE implementations of Biot's equations have been included into some commercial FE software programs. Such implementations, while giving good results, are time consuming and difficult to apply within the time constraints given by the timeline of vehicle development programs.

This paper presents an alternative methodology, thanks to which it is possible to build the coupled vibro-acoustic model of a trimmed vehicle without modeling physically the trim components. The trim is represented by means of transfer admittance matrices whose evaluation is based on an analytical space-domain procedure relying on a modified version of the classical transfer matrix method. A tool was developed to calculate these transfer matrices, starting from the construction and materials of the trim part. These matrices are stored and suitably included into the vibro-acoustical FE model of the vehicle body. The method allows an efficient simulation of the trim effect and as a consequence the fast prediction of sound package modifications during vehicle development. All the procedure was fully integrated within two different commercial software programs used by OEMs for virtual NVH vehicle development.

This paper presents the methodology, its validation on a test case and its application on a full-vehicle up to 500Hz, both for structure-borne and air-borne excitation.

1. INTRODUCTION

Due to their dissipative properties, porous materials are extensively used in automotive acoustics for the purpose of passive noise control; as a matter of fact these materials are, in different forms (fibers, felts etc.), present in almost all sound package parts. Along the years, the issue related to the optimization of their performance/weight/cost has led to the need of models suitable for the analysis of their dynamical and acoustical behavior and this has generated a considerable research effort [1,2,3,4,5,6,7,8,9,10] aimed at developing both analytical and numerical porous material models, all based on the system of equations initially developed by Biot [11,12].

Direct finite elements (FE) implementations of Biot's system of equations for the analysis of porous materials have been available in commercial software programs for more than 10 years now (e.g. [13,14]). All these implementations start from a weak integral form of Biot's equations and rely, as a consequence, on the physical FE modeling of the sound package components. Their application is generally limited to the mid-low frequency range (<400Hz), mainly for reasons related to computational efficiency. As a matter of fact, FE calculations involving porous materials are computationally very expensive (e.g. [15]); this is due to the fact that the FE matrices of porous materials are frequency-dependent and intrinsically complex and do not allow the use of any kind of modal superposition technique. As a consequence, direct



inversion techniques have to be used to solve the FE equations and this determines an important increase of the computational time. This in turn leads to limitations in the size of the model that can be solved and to limitations in the frequency range that can be covered. In this respect, it has also to be considered that the porous materials that are used in sound package components are generally rather soft ($E < 10^5$ Pa). As the frequency increases, the structural wavelengths of the dynamical phenomena that take place inside these materials will quickly become rather short and, as a consequence, a correct representation of such phenomena would require a very fine model, again difficult to manage from the computational point of view. As a last point, it has also to be taken into account that the typical shape of a sound package part like a dash insulator or a carpet insulator can be very complex and the meshing operations needed for a preparation of a (good) FE model can easily take days of work.

These computational issues have led some OEMs to believe that NVH simulation methodologies based on poroelastic finite elements, while giving accurate results (see all the results published in [10,15]), remain difficult to apply within the constraints given by the typical timeline of current vehicle development programs. For this reason, more efficient alternative procedures have been proposed in the last years ([16,17]), thanks to which it is possible to build the coupled vibro-acoustic model of a trimmed vehicle without modeling physically the sound package components. In the context of these procedures, the trim is represented by means of transfer admittance matrices whose evaluation is based on analytical space-domain numerical techniques relying on some version of the classical transfer matrix method [4]. These procedures, being analytical, are very efficient from the computational point of view and, in principle at least, should be also applicable in the mid-high frequency range. On the other hand it is clear that, being based on a 2-dimensional mathematical model, they bring with themselves some approximation and their application to real sound package parts is questionable.

This paper presents a numerical technique that is basically a modification of those proposed in [16,17], with two further developments to make it applicable to real-life test cases. First of all, a tool was developed that, starting from the CAD of the sound package part, can generate the proper transfer admittance matrices needed to interface this same part with the vehicle structure and cavity. This involves the possibility to manage in a suitable way the complex shape and construction that typical sound package parts have. Secondly, the procedure was fully integrated within two commercial FE solvers used by OEMs for their virtual NVH developments. These developments were undertaken to allow the inclusion of the numerical procedure into the virtual NVH development process in a relatively smooth way: the elements involved in

the practical use of the procedure are the same that are already used by sound package designers (CAD drawings) and NVH analysts in their daily activity.

The paper is organized as follows. Section 2 summarizes the main theoretical aspects of the methodology, while section 3 reports the steps needed for its implementation and its integration into commercial FE software programs. Section 4 describes a first validation of the method on a plate/box test case and, eventually, section 5 describes an application to the simulation of a trimmed vehicle both for airborne and for structure borne excitations up to 500Hz.

2. REVIEW OF THE METHOD

The problem we intend to analyze (1) involves, from the physics point of view, three separate regions that we will call “domains”:

- An “elastic domain”, i.e. a structure that occupies a volume Ω^e and whose dynamics is governed by the equations of standard linear elasticity. The vibrational state of this structure will be described by a displacement field that we will indicate in what follows with $\mathbf{u}^e$. In automotive applications, the elastic domain is given in practice by the vehicle body and it can be then rather complex and inhomogeneous.
- An “acoustic domain”, i.e. a limited acoustic space that occupies a volume Ω^a and whose internal acoustic field is governed by the equations of standard linear acoustics. The dynamic state of this acoustic space will be described by an acoustic pressure field that we will indicate with p^a . In automotive applications, the acoustic domain is given in practice by the vehicle passenger compartment, eventually coupled with other cavities like the trunk compartment and/or the engine bay.
- A “trim domain”, i.e. a component that occupies a volume Ω^t and whose dynamics is governed by Biot's equations [11,12]. In what follows, to keep the notations as simple as possible, we will assume that the trim domain is homogeneous and it is constituted by a single porous material. The dynamic state of this domain will be described by the couple $(\mathbf{u}^t, p^t)$ representing the displacement field of its solid phase and the acoustic pressure field of its fluid phase. In automotive applications, the trim domain is given in practice by a sound package component (e.g. the dash insulator, or the floor insulator) and it can be rather complex and inhomogeneous.

In this situation, the problem we intend to tackle consists in calculating the dynamic state of the elastic domain Ω^e (represented by the displacement field $\mathbf{u}^e$) and the dynamic state of the acoustic domain Ω^a (represented by the pressure field p^a), taking into account all the effects due to the

presence of the trim domain, without having to model it in an explicit way. In doing this, we will assume that a FE discretization of the elastic and of the acoustic domain is given. The standard FE equations for these domains will then read:

$$\begin{pmatrix} K_e - \omega^2 M_e & C \\ C^T & K_a - \omega^2 M_a \end{pmatrix} \begin{pmatrix} u^e \\ p^a \end{pmatrix} = \begin{pmatrix} E^e \\ E^a \end{pmatrix} + \begin{pmatrix} T^e \\ T^a \end{pmatrix} \quad (1)$$

On the lhs of [equation \(1\)](#), K_e and M_e represent the stiffness and the mass matrix of the elastic domain, K_a and M_a represent the stiffness and the mass matrix of the acoustic domain and C represents the standard fluid-structural coupling matrix between these domains, accounting for 'direct' coupling regions not covered by the trim. On the right hand side of [equation \(1\)](#), E^e and E^a are two vectors defining the external excitations applied on the elastic and on the fluid domain respectively, while T^e and T^a represent the excitations on these domains related to the presence of the trim. More in detail: from the physical point of view the action of the trim on the elastic domain consists in a distributed surface force and gives rise, when discretized on the FE mesh of the coupling surface between the elastic and the trim domain Γ_{et} , to a vector of nodal forces, represented in [equation \(1\)](#) by T^e . On the other hand, the action of the trim on the acoustic domain consists in an imposed fluid acceleration and gives rise, when discretized on the FE mesh of the coupling surface between the acoustic and the trim domain Γ_{at} , to a vector of nodal volume accelerations, represented in [equation \(1\)](#) by T^a .

By means of techniques similar to the standard boundary reduction procedures used in structural analysis, it is possible to show [10] that vectors T^e and T^a can be expressed as a function of the trim component solid phase displacement on Γ_{et} and of the trim component fluid phase pressure on Γ_{at} :

$$\begin{pmatrix} T^e \\ T^a \end{pmatrix} = \begin{pmatrix} H_{ee} & H_{ea} \\ H_{ae} & H_{aa} \end{pmatrix} \begin{pmatrix} u^t|_{\Gamma_{et}} \\ p^t|_{\Gamma_{at}} \end{pmatrix} \quad (2)$$

where H_{ee} , H_{ea} , H_{ae} and H_{aa} are suitable matrices. From the physical point of view, [formula \(2\)](#) has to be interpreted in the following way. On surface Γ_{et} , the action of the elastic domain on the trim consists in an imposed displacement field. Similarly, on surface Γ_{at} , the action of the fluid domain on the trim consists in an imposed pressure field. Let us assume that a trim displacement field u_t on Γ_{et} and a trim pressure field on Γ_{at} are given. To 'force' these fields on the boundary of the trim it will be necessary to apply a certain force distribution on Γ_{et} and a certain volume acceleration distribution on Γ_{at} .

These latter distributions are given by T^a and T^e respectively and, the system being linear, they are linearly related to the input boundary fields. The [formula \(2\)](#) expresses such linear relationship.

Given the interface conditions:

$$u^e|_{\Gamma_{et}} = u^t|_{\Gamma_{et}} \quad p^a|_{\Gamma_{at}} = p^t|_{\Gamma_{at}} \quad (3)$$

and rearranging [equations \(1\)](#) and [\(2\)](#) one obtains:

$$\begin{pmatrix} K_e - \omega^2 M_e + H_{ee} & C + H_{ae} \\ C^T + H_{ea} & K_a - \omega^2 M_a + H_{aa} \end{pmatrix} \begin{pmatrix} u^e \\ p^a \end{pmatrix} = \begin{pmatrix} E^e \\ E^a \end{pmatrix} \quad (4)$$

i.e. a system of equations in the unknowns u^e and p^a where the trim variables do not appear explicitly and the effect of the trim is accounted for 'only' through matrices H_{ij} . In this way, the presence of the trim is considered without having to include it explicitly into the FE model of the elastic and acoustic domains.

As it is clear from [equations \(1,2,3,4\)](#), the core of the outlined procedure consists in the evaluation of matrices H_{ij} . In [10], [18] it is proposed to pre-calculate these matrices through a FE discretization of the trim component but, as explained in the introduction, this in general can lead to some computational issues. Alternative and procedures that are more efficient have been proposed in [16,17] for the approximate evaluation of these matrices by means of the classical transfer-matrix method. For the understanding of such methods, one has to assume that the multilayer under analysis is flat, infinite and with constant-thickness layers. In these conditions, in [16] the transfer matrix of the trim multilayer is pre-calculated in the following form:

$$\begin{pmatrix} \sigma^t|_{\Gamma_{et}} \\ w^t|_{\Gamma_{at}} \end{pmatrix} = \begin{pmatrix} h_{ee} & h_{ea} \\ h_{ae} & h_{aa} \end{pmatrix} \begin{pmatrix} u^t|_{\Gamma_{et}} \\ p^t|_{\Gamma_{at}} \end{pmatrix} \quad (5)$$

where σ^t is the normal component of the stress applied on the solid phase of the trim and w^t is the velocity of the fluid phase of the trim. Given [\(5\)](#), matrices H_{ij} appearing in [\(2\)](#) can be evaluated through a simple discretization process over the FE mesh of surfaces Γ_{et} and Γ_{at} . Implicitly, in this process, it is assumed that quantities σ^t , w^t , u^t and p^t appearing in [\(5\)](#) do not vary too strongly over the coupling surfaces.

The methodology proposed in [17] relies on the fact that in the 'continuous' space domain, the operators linking the

boundary fields u^t and p^t with the corresponding forcing dual variables can be expressed as:

$$u^t(x) = \int_{\Gamma_{et}} D_{ee}(\omega, |x-y|) \sigma^t(y) d\Gamma(y) + \int_{\Gamma_{at}} D_{ea}(\omega, |x-y|) w^t(y) d\Gamma(y) \quad x \in \Gamma_{et} \quad (6)$$

$$p^t(x) = \int_{\Gamma_{et}} D_{ea}(\omega, |x-y|) \sigma^t(y) d\Gamma(y) + \int_{\Gamma_{at}} D_{aa}(\omega, |x-y|) w^t(y) d\Gamma(y) \quad x \in \Gamma_{et} \quad (7)$$

Functions D_{ij} appearing on the rhs of (6) and (7) represent ‘Green’s functions’ for the flat multilayer under analysis, i.e. responses to unitary point excitations. As it is apparent from formula (6) and (7), such functions depend not only on frequency, but also on the distance between the excitation and the response point. These functions can be evaluated analytically; from the discretization of (6) and (7) over the FE mesh of surfaces Γ_{et} and Γ_{at} and a subsequent matrix inversion it is possible eventually to get the H_{ij} matrices appearing in (2).

As it is clear from their formulation, both methods outlined here above are directly applicable only to planar geometries and to constant-thickness layers. As a matter of fact, all published applications of these methodologies refer to simple constant-thickness trim geometries. To make them applicable to real sound package parts it is necessary to modify them, in such a way to make them able to manage the thickness variations and the in-homogeneities that are typical of real sound package parts.

The idea proposed here consists in ‘patching’ the coupling surface Γ_{at} in such a way that, within each patch, the trim multilayer construction is homogeneous (this is a must) and its thickness distribution is reasonably constant (this is a ‘nice to have’). The patches must cover all the coupling surface Γ_{at} ; for the method relying on formula (5) the patches have to be non-overlapping, while for the method relying on formulas (6) and (7) some overlap between the patches is required. It is worth mentioning here that the idea of ‘patching’ the trim is proposed in [15] also for poroelastic finite elements. In any case, for each patch the matrices H_{ij} can be evaluated using one of the methods described here above and suitably averaging over the thickness distributions of the different trim layers. After this, the H_{ij} matrices corresponding to the different patches are assembled according to the standard FE numerical procedures. It is important to remark here that experience has shown how it can be important to take into account the thickness variation of the trim layers (even within each single patch), particularly in those cases in which

materials with a strong ‘structural’ behavior are involved (e.g. viscous PUR foams). Furthermore, special care has to be taken in the management of the interface conditions between the trim domain and the elastic/acoustic domain, as well as of the interface conditions between the different trim layers.

3. IMPLEMENTATION AND INTEGRATION INTO COMMERCIAL FE SOLVERS

The procedure outlined in the last part of section 2 has been implemented, automated and fully integrated into two commercial FE solvers. This step was considered crucial to make the methodology really accessible to sound package designers and NVH analysts. Figure 1 shows in schematic form the ‘strategy’ followed for these developments and integration to commercial software, resulting into a new methodology.

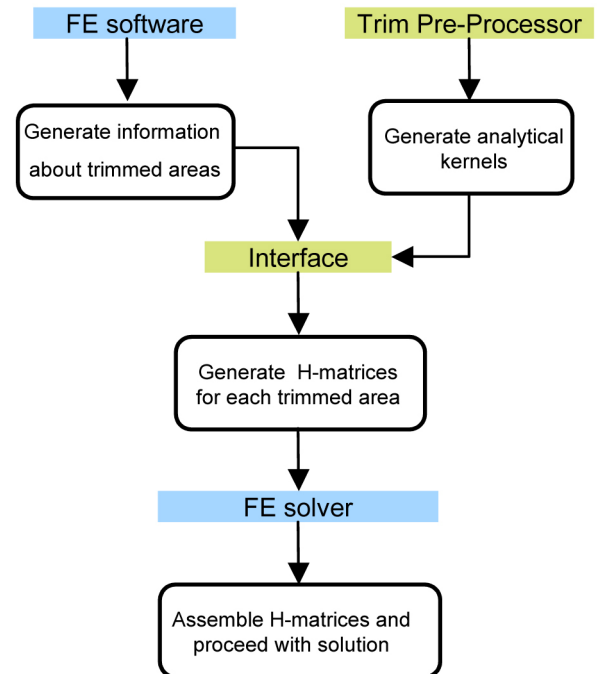


Figure 1. Scheme of the implementation process followed to integrate trim simulation into FE solver.

First of all, a ‘trim pre-processor’ tool was implemented. Starting from the CAD drawing of the sound package part suitably subdivided into patches (as explained at the end of the previous section), the tool evaluates the matrix coefficients h_{ij} appearing in (5) (or the kernels D_{ij} appearing in (6-7)) for each patch, taking into account the thickness variation of the trim layers. Obvious input data for this pre-processor are the construction of the trim multilayer (in terms of layers’ sequence) and, for each layer, the parameters that characterize the corresponding material. As an output, the