

A FE Based Procedure for Optimal Design of Damping Package, with Presence of the Insulation Trim

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ABSTRACT

Typically, in the automotive industry, the design of the body damping treatment package with respect to NVH targets is carried out in such a way to achieve panel mobility targets, within given weight and cost constraints. Vibration mobility reduction can be efficiently achieved thanks to dedicated CAE FE tools, which can take into account the properties of damping composites, and also, which can provide their optimal location on the body structure, for a minimal added mass and a maximized efficiency. This need has led to the development of different numerical design and optimization strategies, all based on the modeling of the damping composites by mean of equivalent shell representations, which is a versatile solution for the full vehicle simulation with various damping layouts. However, these approaches, which can estimate correctly the beneficial vibration effect of damping pads application on the vehicle body, address the body NVH target with no consideration of the impact that the presence of the insulation on body panels can have on the final vibration result. On the other hand, the efforts carried out in the last years for FE implementations of Biot's system of equations have led to simulation methods at vehicle level, which can take into consideration the dynamical behaviour of porous materials and which allow including in an efficient and flexible way sound package parts into vehicle FE models used for NVH analyses. This paper presents a FE-based procedure, thanks to which it is possible to design the optimal damping lay-out with respect of panel mobility targets, while taking into account the presence of the insulation part on body panels. In a first section a design methodology for damping layout is presented. This method, that is completely integrated in Nastran, is able to provides the ranking and vibration pattern of the vehicle panels with highest mobility

for a given frequency range and set of loads in order to maximize the effect of the damping treatment. Then the problem of the influence of the acoustic treatment on the panel vibrations has been addressed. The proposed solution is represented by an implementation in MSC/Nastran of the Biot-Allard theory for porous media. This procedure allows a smart coupling of structural FE model with a FE boundary representation of the acoustic part. In the last section, the benefits of the joined use of the two techniques are highlighted by mean of their application on a simple test case as well as on a full-vehicle.

INTRODUCTION

The acoustic assessment of a vehicle usually involves different aspects and problematic. Two of them are related to the main physical phenomena responsible for noise transmission within a vehicle, namely air-born and structural-born noise. Different solutions are adopted in order to control these phenomena, and thus, to reduce the noise transmission from the sources to the interior. The most common countermeasures applied in the automotive are the following: adding damping material pads in strategic areas of the metal panels of the vehicle as well as treating the areas facing the vehicle cavity with an insulating treatment (trim). The two different solutions usually coexist within vehicles.

However the design of the acoustic treatment and the damping package are carried out independently with most of the time independent targets. For this purpose specific tools are used in order to match the given targets.

The design of the trim is performed in order to match the acoustic and weight targets provided by the OEMs. Usually the materials pile-up and layers thickness are designed in

order to improve absorption/transmission loss characteristics of the carpet. Specific methodologies for mid-high frequency range/air-borne noise, like statistical energy analysis or transfer matrix method, are used for acoustic package design. The impact that the trim has on the vibration of vehicle panel in the low-mid frequency range [100Hz-500Hz] is not considered in this design process. In this paper, for the prediction of the trim, we refer to a FE methodology based on the Biot-Allard's theory of propagation of sound in porous media [1-2]. More in detail the mixed displacement-pressure formulation for poroelastic materials, developed by N. Atalla et al. [3, 4, 5, 6], has been implemented in MSC/Nastran by means of DMAP language. This method allows [7] including sound package parts into Nastran vehicle FE models used for NVH in an efficient and flexible way. It uses Biot's theory in order to be able to embed the sound package in the vehicle model. In particular, the following necessary steps are automatically carried out within Nastran, namely first the boundary reduction, then the computation of interface matrices between incompatibility mesh of the sound package part and vehicle model, and finally the embedding of the trim effect in the final assembly run.

On the other hand, the design of the damping package is carried out without taking into account the effect of the carpet on the body of the car. The optimization is performed on the body in white of the vehicle and the targets are given in terms of panel vibration level and package weight. In this paper, we refer to a smart damping layout design procedure, which has been developed in order to support the design process of the damping package [8]. The methodology is completely developed in DMAP Nastran Language and completely integrated in the standard MSC/Nastran solution 111. The mentioned technique is able to identify the areas of a given set of vehicle panel that show a high vibration level given the frequency range. Also it is able to identify the most critical load cases for each panel, which means the load cases that provide more energy to the considered panel.

The novelty of the approach described in what follows consists essentially in the fact that it is shown how it is possible to carry out a damping package study considering the effect of the trim on the vehicle body vibration. This is made possible thanks to the combined use of the two methodologies mentioned before. In fact they can be used within a standard MSC/Nastran solution providing the possibility of analyzing the metal panel vibration pattern considering the influence of the acoustic package. So the joined use of the two techniques allows studying the interaction of the damping package and the acoustic treatment in the low-mid frequency range [150Hz-500Hz] and therefore a more accurate final design.

The paper is organized as follows. The theoretical aspects and technical features of the two referred methodologies for respectively damping simulation and trim simulation are

summarized in the first section. Then, the two methods for the design of the panel vibration damping with presence of trim are combined. The proof of the methodology is given in the second section on a simple plate test case, while the third section presents a numerical application of the methodology on a vehicle.

METHODS

METHODOLOGY FOR DAMPING LAYOUT DESIGN

Introduction

During vehicle development, it is necessary to estimate correctly the beneficial effect of damping application on the vehicle body. Vibration mobility reduction can be efficiently achieved thanks to dedicated CAE FE tools, which can take into account the composite properties of damping composites, and also, which can provide their optimal location on the body structure, for a minimal added mass and a maximized efficiency. In this section, we present a simulation procedure based on an equivalent representation of the damping composite model, which is fully integrated in MSC/Nastran, and which provides the vibration map of vehicle's panel and the ranking of the vehicle's panels with highest mobility for a given frequency range and set of loads, in order to get the optimal effect of different damping treatments on vehicle body.

The method

The method consists in the following steps: the most important modes/Eigenfrequencies for each panel are identified given the load cases and the frequency range; the mobility (Velocity/Force) is averaged among the load cases and the most important modes, then the mobility is normalized in order to provide the percentage contribution mobility, which represents how much a single node of each panel contributes to its overall response.

In order to provide these results the methodology, developed in a Nastran DMAP, uses the structure of the standard solution 111 of Nastran just adding the necessary algebra. The method needs only the body FE model, the panel set definition, the damping model and the load case definition (figure 1). The simulation of the damped model exploits the technique presented by D. Ross et al. [9], in which the material properties of treated areas are substituted to the equivalent composite properties [10]. See such utilizations of the FE simulation in [11] for the optimization of damping package.

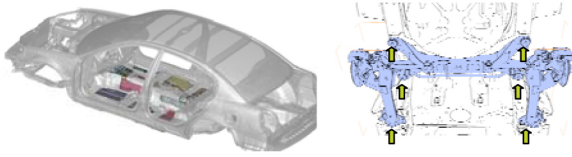


Figure 1. inputs for damping layout design - body FE model with panel node set definition, and load-cases definition.

The results of the customized solution 111 are organized in two files:

- the *.op2 file, which contains a “colormap” showing the mobility contribution of a given panel and frequency range, as illustrated [figure 2](#). For each panel, the colormap represents the contribution of each node to the total mobility. This contribution is an average value calculated by taking the average across the load cases and across the frequency spectrum.
- The *.aeso file, which contains the ranking of the load-cases, as illustrated [figure 3](#).

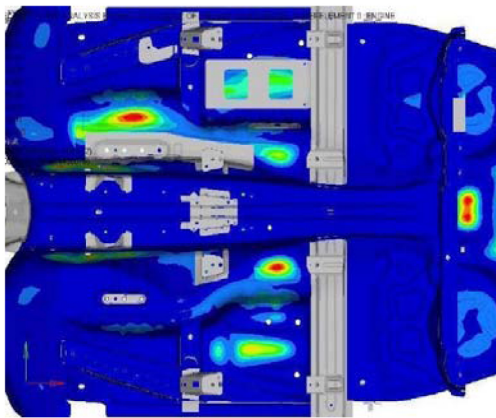


Figure 2. Percentage mobility contribution contour for one panel. For each panel, the colormap represents the contribution of each node to the total mobility.

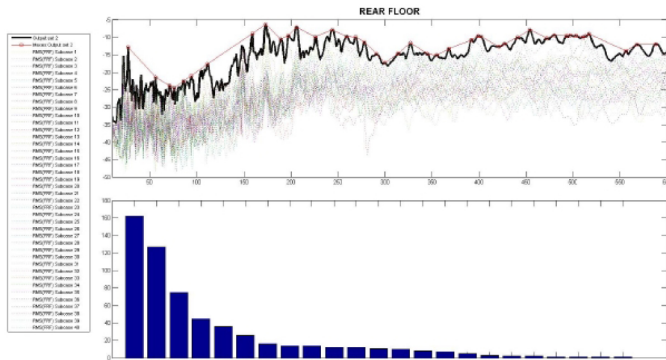


Figure 3. Upper: rms mobility FRF maximum envelope over the load-cases; lower: relative contribution of each load-case to the max envelop.

More in detail the method performs the following math in order to obtain the final results.

For each load case, the average mobility (rms) across the panel is calculated (1). For each load case (m), for each node (i), the contribution to the average mobility (rms) across the panel is calculated (2). A first average, the average percentage mobility contribution matrix AVGP, is then calculated across the M load cases for each panel (3).

$$\|\bar{v}\| = \sqrt{\sum_1^3 (\text{Re}(v_r)^2 + \text{Im}(v_r)^2)} \quad (1)$$

$$CP_{ij}^m = \frac{\|\bar{v}_i^m(f_j)\|^2}{\sum_n \|\bar{v}_i^m(f_j)\|^2} \quad (2)$$

$$AVCP_{ij} = \frac{\sum_m^M CP_{ij}^m}{M} \quad (3)$$

The $AVCP_{ij}$ matrix must be averaged on the frequency range in order to get a unique “contribution map” as on the [figure 2](#). For this purpose, the RMS mobility over all nodes of each panel is calculated (4), as well as its maximal envelope (5) over the M load-cases.

$$RMS(\bar{v}^m(f_j)) = \sqrt{\frac{\sum_i^N \|\bar{v}_i^m(f_j)\|^2}{N}} \quad (4)$$

$$ENV(RMS(\bar{v}(f_j))) = \max\left(\left\{RMS(\bar{v}^1(f_j)), \dots, RMS(\bar{v}^M(f_j))\right\}\right) \quad (5)$$

As shown on [figure 3](#), each curve corresponds to one load-case. This information is used to calculate and order the relative contribution of each load-case. The matrix $AVCP_{ij}$ is then sampled and averaged (6) by keeping only those frequencies corresponding to the “relevant” peaks of the rms envelope.

$$AVG_i(\underline{AVCP}) = \frac{\sum_j AVCP_{ij}}{N} \quad (6)$$

The post-processing can be performed via any FE post-processor and the visualization of the results directly on the FE model of the vehicle allows to sketch an optimized damping layout. The [figure 4](#) shows how the outcomes of the

analysis can be used to identify the optimal damping location in a real application.

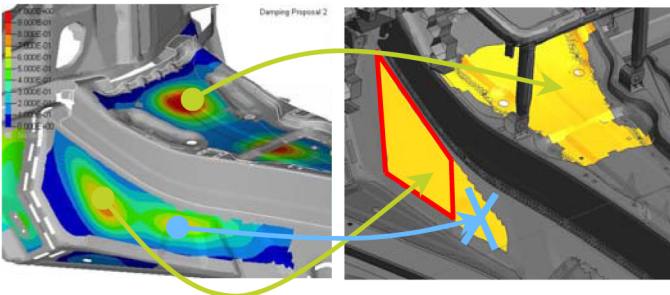


Figure 4. Post-processing. The mobility map is interpreted for optimal design of damping layout.

FE BASED METHODOLOGY FOR TRIM SIMULATION

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 (fibres, 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 behaviour and this has generated a considerable research effort devoted to the development of both analytical [12, 13, 14] and numerical [15-16] porous material models, all based on the system of equations initially developed by Biot [1-2]. The presented method implements the more recent formulations of the Biot equations presented by N.Atalla et al. [3, 4, 5, 6]. This formulation uses the displacement of the solid phase and the acoustic pressure of the fluid phase as field variables with a significant reduction in terms of degrees of freedom within respect to the original Biot formulation.

In the automotive industry, the main application related to the simulation of poroelastic materials consists in the analysis of the effect that one or more sound package parts can have on the vibro-acoustics of a vehicle body/passenger compartment coupled system.

Formulation of the problem

The Biot-Allard theory addresses the problem presented in figure 5. The problem involves, from the physical point of view, three separate regions that we will call “domains”. We can classify the different domains as:

1. “elastic domain”, i.e. a structure that occupies a volume Ω^e and whose dynamics is governed by the equations of standard linear elasticity;

2. “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 ;

3. “porous domain”, i.e. a component that occupies a volume Ω^p and whose dynamics is governed by Biot-Allard's equations. The dynamic state of this domain will be described by the couple (u^p, p^p) , the displacement field of its solid phase and the acoustic pressure field of its fluid phase.

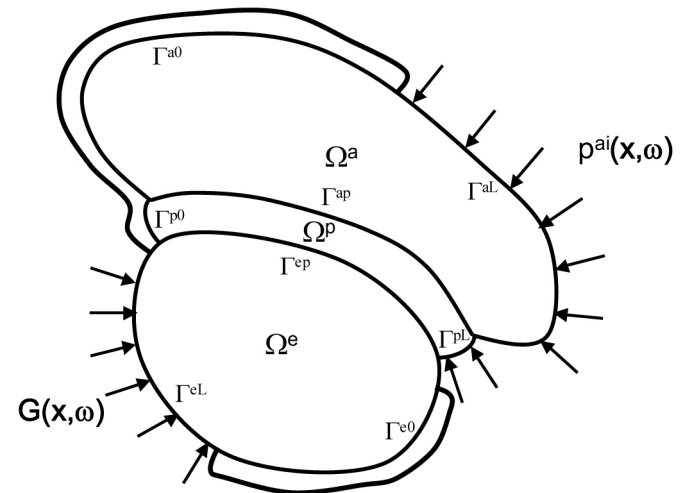


Figure 5. Schematic representation of the system under study including an elastic, a porous and an acoustic domain with various kinds of boundary conditions and excitations.

Each domain is represented by a different variational problem. Each variational problem is formulated within a suitable subspace of the Hilbert space of function $E_1(\Omega^i)$, where Ω^i can represents the elastic, acoustic or porous domain. The definition of the subspace of the admissible function for each variational problem defines also the boundary conditions of each domain.

Different coupling conditions are available within the method. Two are the available coupling conditions between the porous and the elastic domain:

1. glued, which means that the elastic-porous domain interface presents the same displacement;
2. sliding, which means that the on the elastic-porous domain interface the nodes of each domain have the same normal velocity but different tangent velocity.

Two are the coupling conditions between the porous and the acoustic domain: