Damping Specification of Automotive Structural Components via Modal Projection

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Abstract

In the automotive industry damping has become an important and highly effective design parameter for reducing low-frequency vibroacoustic noise in the vehicle compartment. It is a fairly straightforward procedure to compute vibroacoustic frequency responses of a damped car body for a given excitation and to compare the resulting response levels to a prescribed specification over a frequency range. However, when specification levels are exceeded at particular frequencies, the engineer must be able to localize the relevant structural components to be damped and subsequently define the appropriate damping at the component level in order to satisfy the specifications at the vehicle level.

This problem of damping specification has been under study at PSA Peugeot Citroën over the past several years and has recently resulted in a new methodology and corresponding software tool which are presented in this article.

1 Introduction

Vehicle comfort as perceived by the driver and passengers has become a major consideration in the design of cars resulting in increasingly stringent acoustic and vibration (ACV) specifications which must be satisfied. The degree of comfort can be quantified in terms of both mechanical vibrations and acoustic noise in the vehicle compartment coming from multiple sources including engine, suspension, wheels and aerodynamic loads.

To satisfy design specifications, increased stiffness has often been considered to reduce response levels. However, this comes at the cost of increased mass which negatively impacts production costs and vehicle performance. As an alternative, damping is now being used as an efficient design parameter.

As a follow-up to a study carried out by PSA Peugeot Citroën [1], a software tool has been developed to demonstrate the feasibility of including damping as a design parameter during the specification phase of the design process. Based on a substructuring approach, the tool allows the engineer to identify the critical vehicle components to be damped and estimate the amount of damping to be introduced in order to satisfy design specifications.

The methodology relies on the computation of normal modes for the global structure (including acoustic cavities) as well as the modes for each of the considered substructures (components). Both unconstrained (free-free) and constrained boundary conditions may be used depending on the particular component (roof, suspension, floorboard, etc.).

By projecting the component modes onto the basis of global modes, a transformation may be derived in order to establish a relationship between modal damping values at the component level and the corresponding modal damping (generally coupled) in the global structure. Identifying the critical component modes to be damped is achieved by the use of modal participation factors based on modal strain energies.

The vibroacoustic frequency responses of the global structure may then be efficiently computed for any prescribed set of component damping values by adding the resulting modal damping matrix to the generalized dynamic stiffness matrix and then solving the system for a given excitation. Using this procedure, it is therefore possible to interactively adjust component damping values until specification levels are satisfied over all frequency bands.

The methodology has been implemented in a software tool written in MATLAB and interfaced with MSC/NASTRAN for the computation of the global structure and component modes. Several industrial requirements had to be satisfied including the following needs:

- handling of large models (millions of DOF and thousands of modes)
- use of a seamless interface with MSC/NASTRAN (located on a high-performance server)
- interactive response computation and damping specification

In this article the theoretical background of the underlying methods is presented. Next the architecture of the software is described as well as the different computational steps: importation of the global structure and component models, modal analysis, computation of frequency responses and modal participation factors, and damping specification. Finally an industrial application is provided to illustrate the interest of the methodology for vibroacoustic damping specification.

2 Nomenclature

2.1 Abbreviations

DOF	Degree-of-freedom
FRF	Frequency Response Function
MPF	Modal Participation Factor
GS	Global Structure
S, SS	Substructure(s)

2.2 Matrix conventions

In general, $\mathbf{X}_{ij}$ designates a matrix with n rows relative to the DOF *i*, and m columns relative to the DOF *j*.

This convention implies that the transpose of the matrix may be expressed by the relation $\mathbf{X}_{ji} = \mathbf{X}_{ij}^T$ which corresponds to permuting the row and column subscripts. Moreover the matrix, $\mathbf{X}_{ii}$ (which is not necessarily diagonal) is necessarily symmetric. These matrix properties assume that the reciprocity principle is respected. However in some situations such as coupled fluid-structure systems, some symmetries are lost, in which case special precautions are taken in the matrix notation to remove any possible ambiguity.

2.3 Scalars and matrices

A	Fluid-structure coupling matrix
E	Elastic energy
f	Frequency (Hz)

F	Force
g	Structural damping for substructure(s)
G, GFL	Structural damping for global structure and fluid
i	$\sqrt{-1}$
K, k	Stiffness (force/displacement)
M, m	Mass (force/acceleration)
p	Pressure
q	Modal displacement
u	Physical displacement
Y	Left eigenvector
Φ	Eigenvector, right eigenvector
Ψ	Junction mode (static constraint mode)
t	Fraction of elastic energy
w	Circular frequency $w = 2p\ f$

2.4 Subscripts

e	Element, excitation	s	Structure
f	Fluid	t	Truncated mode
i	Internal DOF	S	Substructure
j	Junction (fixed-interface) DOF		
k	Substructure mode		
l	Fluid (acoustic) mode		
m	Global structure mode (without fluid)		
n	Global structure mode (coupled with fluid)		
o	Observation (response) DOF		

3 Theoretical Background

3.1 Normal modes of a coupled fluid-structure system

The equations of motion governing the harmonic response of a coupled fluid-structure system comprising s structural degrees of freedom (DOF) and f fluid DOF are expressed below.

$$\left(-w^2 \begin{bmatrix} \mathbf{M}_{ss} & \mathbf{0}_{sf} \\ -\mathbf{A}_{fs} & \mathbf{M}_{ff} \end{bmatrix} + \begin{bmatrix} \mathbf{K}_{ss}(1+iG) & \mathbf{A}_{sf} \\ \mathbf{0}_{fs} & \mathbf{K}_{ff}(1+iGFL) \end{bmatrix} \right) \begin{bmatrix} \mathbf{u}_s \\ \mathbf{p}_f \end{bmatrix} = \begin{bmatrix} \mathbf{F}_s \\ \dot{\mathbf{Q}}_f \end{bmatrix} \tag{1}$$

$\mathbf{M}_{ss}$ Structure mass matrix (symmetric) $\mathbf{u}_s$ Vector of structural displacements

$\mathbf{K}_{ss}$ Structure stiffness matrix (symmetric)

$\mathbf{p}_f$ Vector of fluid pressures

$\mathbf{M}_{ff}$ Fluid "mass" matrix (symmetric)

$\mathbf{F}_s$ Vector off applied forces

$\mathbf{K}_{ff}$ Fluid "stiffness" matrix (symmetric)

$\mathbf{Q}_f$ Vector of acoustic sources ($\dot{\mathbf{Q}}_f = iw\,\mathbf{Q}_f$)

$\mathbf{A}_{fs}$ Coupling matrix ($\mathbf{A}_{sf} = \mathbf{A}_{fs}{}^T$)

G, *GFL* Structural damping for structure and fluid

Only structural (hysteretic) is considered in Eq. (1) for the sake of simplicity. In the presence of viscous damping, an additional viscous damping matrix proportional to *iw* would be added to the equations of motion.

Instead of solving Eq. (1) directly, a modal approach is used offering several important advantages such as numerical efficiency (reduced computation time), and physically interpretable parameters (natural frequency, modal energies, etc) necessary for damping specification.

The normal modes of the coupled system may be computed from the eigenvalue problems shown below.

$$\left(-w_n^2\begin{bmatrix}\mathbf{M}_{ss} & \mathbf{0}_{sf}\\ -\mathbf{A}_{fs} & \mathbf{M}_{ff}\end{bmatrix}+\begin{bmatrix}\mathbf{K}_{ss} & \mathbf{A}_{sf}\\ \mathbf{0}_{fs} & \mathbf{K}_{ff}\end{bmatrix}\right)\begin{bmatrix}\mathbf{F}_{sn}\\ \mathbf{F}_{fn}\end{bmatrix}=\begin{bmatrix}\mathbf{0}_s\\ \mathbf{0}_f\end{bmatrix} \tag{2}$$

$$\begin{bmatrix}\mathbf{Y}_{ns} & \mathbf{Y}_{nf}\end{bmatrix}\left(-w_n^2\begin{bmatrix}\mathbf{M}_{ss} & \mathbf{0}_{sf}\\ -\mathbf{A}_{fs} & \mathbf{M}_{ff}\end{bmatrix}+\begin{bmatrix}\mathbf{K}_{ss} & \mathbf{A}_{sf}\\ \mathbf{0}_{fs} & \mathbf{K}_{ff}\end{bmatrix}\right)=\begin{bmatrix}\mathbf{0}_s & \mathbf{0}_f\end{bmatrix} \tag{3}$$

Although distinct left and right eigenvectors ($\mathbf{Y}$,$\mathbf{F}$) appear, they are related by the following expression due to the particular form of asymmetry introduced by the coupling matrix **A**.

$$\begin{bmatrix}\mathbf{Y}_{sn}\\ \mathbf{Y}_{fn}\end{bmatrix}=\begin{bmatrix}\mathbf{F}_{sn}\\ \mathbf{F}_{fn}/w_n^2\end{bmatrix} \tag{4}$$

Unfortunately, solving Eqs. (2-3) can be very time-consuming for large models due to the special form of the system. To reduce the computational effort, the system is first condensed using the *uncoupled* structure and fluid modes as defined below.

$$\left(-w_m^2\,\mathbf{M}_{ss}+\mathbf{K}_{ss}\right)\mathbf{F}_{sm}=\mathbf{0}_s \tag{5}$$

$$\left(-w_l^2\,\mathbf{M}_{ff}+\mathbf{K}_{ff}\right)\mathbf{F}_{fl}=\mathbf{0}_f \tag{6}$$

Eq. (5) provides *m* modes of the structure without fluid, whereas Eq. (6) provides *l* modes of the fluid assuming rigid cavity walls. Using the associated eigenvectors, $\mathbf{F}_{sm}$ and $\mathbf{F}_{fl}$, the physical system of Eqs. (2) and (3) can be condensed to the modal system shown below.

$$\left(-w_n^2\begin{bmatrix}\mathbf{m}_m & \mathbf{0}_{ml}\\ -\mathbf{A}_{lm} & \mathbf{m}_l\end{bmatrix}+\begin{bmatrix}\mathbf{k}_m & \mathbf{A}_{ml}\\ \mathbf{0}_{lm} & \mathbf{k}_l\end{bmatrix}\right)\begin{bmatrix}\mathbf{F}_{mn}\\ \mathbf{F}_{ln}\end{bmatrix}=\begin{bmatrix}\mathbf{0}_m\\ \mathbf{0}_l\end{bmatrix} \tag{7}$$

$$\begin{bmatrix}\mathbf{Y}_{nm} & \mathbf{Y}_{nl}\end{bmatrix}\left(-w_n^2\begin{bmatrix}\mathbf{m}_m & \mathbf{0}_{ml}\\ -\mathbf{A}_{lm} & \mathbf{m}_l\end{bmatrix}+\begin{bmatrix}\mathbf{k}_m & \mathbf{A}_{ml}\\ \mathbf{0}_{lm} & \mathbf{k}_l\end{bmatrix}\right)=\begin{bmatrix}\mathbf{0}_n & \mathbf{0}_m\end{bmatrix} \tag{8}$$

The terms ***m*** et ***k*** are diagonal mass and stiffness matrices associated with the structure and fluid modes. Eqs. (7) and (8) may now be efficiently solved to obtain the eigenvalues w_n^2 and generalized eigenvectors of the coupled system. The physical eigenvectors are obtained by applying the following back-transformation.

$$\begin{bmatrix} \mathbf{F}_{sn} \\ \mathbf{F}_{fn} \end{bmatrix} = \begin{bmatrix} \mathbf{F}_{sm} \mathbf{F}_{mn} \\ \mathbf{F}_{fl} \mathbf{F}_{ln} \end{bmatrix} \tag{9}$$

3.2 Residual modes

To minimize truncation errors, the uncoupled structure and fluid normal modes $\mathbf{F}_{sm}$ and $\mathbf{F}_{fl}$ obtained from Eqs. (5) and (6) must be enriched by a set of residual modes that provide information about the coupling effects across the structure-fluid boundaries.

A residual mode is similar to a normal mode in that it satisfies the same orthogonality properties and has an associated eigenvalue. However it does not satisfy the eigenvalue problem since each residual is in fact a particular linear combination of *all* the truncated (superior) normal modes.

Although residual modes (sometimes known as residual vectors or pseudo-modes) have been in use for well over a decade, their application to coupled analysis is recent [2,3] and has been implemented in this study. The procedure for deriving the residual modes is summarized below.

For the structure, a set of *m* static modes $\mathbf{X}_{sm}$ is computed resulting from the forces exerted by the fluid modes across the fluid-structure boundary.

$$\mathbf{K}_{ss}\, \mathbf{X}_{sm} = \mathbf{C}_{sf}\, \mathbf{F}_{sm} \tag{10}$$

Similarly for the fluid, a set of *n* static modes $\mathbf{X}_{fn}$ is obtained using the pressure exerted by the structure modes across the same boundary but in the opposite sense.

$$\mathbf{K}_{ff}\, \mathbf{X}_{fn} = \mathbf{C}_{fs}\, \mathbf{F}_{fn} \tag{11}$$

Next, the static modes $\mathbf{X}_{sm}$ and $\mathbf{X}_{fn}$ are "filtered" or rendered orthogonal with respect to the normal modes $\mathbf{F}_{sn}$ and $\mathbf{F}_{fm}$, and then orthogonalized to form an orthonormal basis of residual modes $\hat{\mathbf{F}}_{sm}$ and $\hat{\mathbf{F}}_{fn}$ which are then appended to the normal modes to form the enriched modal bases shown below.

$$\mathbf{F}_{sn} \leftarrow [\mathbf{F}_{sn} \quad \hat{\mathbf{F}}_{sm}] \tag{12}$$

$$\mathbf{F}_{fm} \leftarrow [\mathbf{F}_{fm} \quad \hat{\mathbf{F}}_{fn}] \tag{13}$$

3.3 Frequency responses of the coupled system

The *n* coupled modes derived above may be used to condense the equations of motion of Eq. (1). The resulting generalized dynamic stiffness may be written as follows.

$$\mathbf{K}_{nn}(w) = -w^2\, \mathbf{m}_n + \mathbf{k}_n + i\, G\, \mathbf{K}_{nn,s} + i\, GFL\, \mathbf{K}_{nn,f} \tag{14}$$

Note that the hysteretic damping in the structure and the fluid is introduced by the matrices $\mathbf{K}_{nn,s}$ and $\mathbf{K}_{nn,f}$ which are symmetric but non-diagonal, and thus couple the system.

$$\mathbf{K}_{nn,s} = \mathbf{F}_{ns}\, \mathbf{K}_{ss}\, \mathbf{F}_{sn} \tag{15}$$

$$\mathbf{K}_{nn,f} = w_n^{-2}\, \mathbf{F}_{nf}\, \mathbf{K}_{ff}\, \mathbf{F}_{fn} \tag{16}$$

External forces, $\mathbf{F}_e(w)$ applied to the excitation DOF, *e*, are transformed to generalized forces by the following expression.

$$\mathbf{F}_n(w) = \mathbf{F}_{ne}\, \mathbf{F}_e(w) \tag{17}$$

In the case of excitation via acoustic sources, $\mathbf{Q}_e(w)$, the corresponding generalized excitation is written

$$\mathbf{F}_n(w) = (w^2 / w_n)\,\mathbf{F}_{ne}\,\mathbf{Q}_e(w) \tag{18}$$

The generalized responses, $\mathbf{q}_n(w)$, are obtained by solving the following linear system at each excitation frequency, w.

$$\mathbf{K}_{nn}(w)\,\mathbf{q}_n(w) = \mathbf{F}_n(w) \tag{19}$$

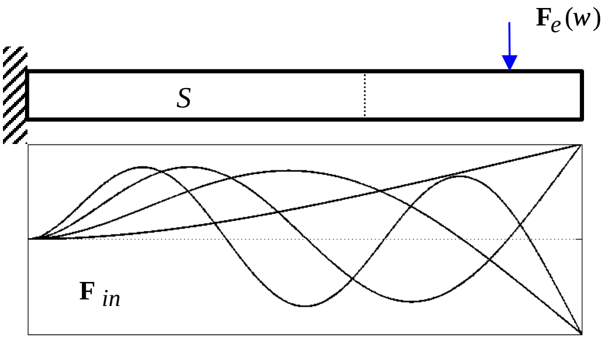
Finally, the physical responses $\mathbf{u}_o(w)$ are obtained by back-transformation using the eigenvectors partitioned on the corresponding observation DOF, *o*.

$$\mathbf{u}_o(w) = \mathbf{F}_{on}\,\mathbf{q}_n(w) \tag{20}$$

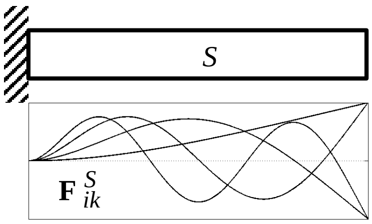
3.4 Projection of system modes on substructure modes

Since damping is to be specified with respect to the components or substructures, it is necessary to express the global structure's response in terms of the substructure's contributions. This is done by projection of the system modes onto the substructure modes.

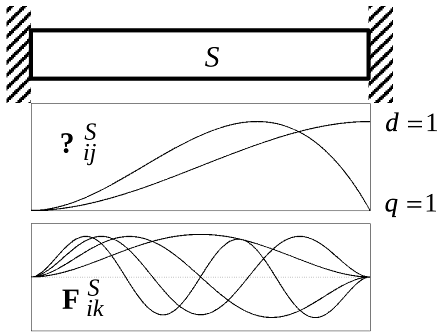
This projection is illustrated graphically in Figure 1 by considering the first four bending modes of a simple beam structure plotted in Figure 1a.



a) modes of the global structure



b) mode of the substructure *S* (free interface)



c) modes of the substructure S (fixed-interface)

Figure 1 - Modes of the Global Structure and Substructure

Consider a component of the structure designated by the substructure, *S*, whose modes will depend on the boundary conditions that we wish to impose at its interface.

For a free-interface, the *k* modes of the substructure illustrated in Figure 1b, are obtained from the following eigenvalue problem.

$$\left(-w_k^2\ \mathbf{M}_{ii}^S + \mathbf{K}_{ii}^S\right)\mathbf{F}_{ik}^S = \mathbf{0}_k \tag{21}$$

If one or more of the interface DOF are blocked (DOF *j*), a set of junction modes (static constraint modes), $\mathbf{?}_{ij}^S$ must be computed in addition to the fixed-interface normal modes. Illustrated in Figure 1c, the junction modes are obtained by successively imposing a unit displacement at each of the junction DOF, *j* as defined in the following expression.

$$\mathbf{K}_{ii}^S\ \mathbf{?}_{ij}^S = \mathbf{I}_{ij} \tag{22}$$

To determine the response as a function of the substructure modes, we start with the expression of the response on the substructure's DOF, *i*.

$$\mathbf{u}_i^S(w) = \mathbf{F}_{in}\ \mathbf{q}_n(w) \tag{23}$$

For the case of a free-interface substructure, we can express the response in terms of the substructure's modes comprising the *k* retained modes, $\mathbf{F}_{ik}^S$, and the *t* truncated modes, $\mathbf{F}_{it}^S$ resulting in the following expression.

$$\mathbf{u}_i^S(w) = \mathbf{F}_{in}\ \mathbf{q}_n(w) = \begin{bmatrix}\mathbf{F}_{ik}^S & \mathbf{F}_{it}^S\end{bmatrix}\begin{bmatrix}\mathbf{q}_k^S(w)\\ \mathbf{q}_t^S(w)\end{bmatrix} \tag{24}$$

Premultiplying Eq. (24) by $\mathbf{F}_{ki}^S\ \mathbf{M}_{ii}^S$ leads to:

$$\mathbf{F}_{ki}^S\ \mathbf{M}_{ii}^S\ \mathbf{F}_{in}\ \mathbf{q}_n(w) = \underbrace{\mathbf{F}_{ki}^S\ \mathbf{M}_{ii}^S\ \mathbf{F}_{ik}^S}_{\mathbf{m}_k^S}\ \mathbf{q}_k^S(w) + \underbrace{\mathbf{F}_{ki}^S\ \mathbf{M}_{ii}^S\ \mathbf{F}_{it}^S}_{0\text{ by orthogonality}}\ \mathbf{q}_t^S(w) \tag{25}$$

Finally, from Eq. (25) we can determine the contribution of the *k* substructure modes, to the *n* global modes via the modal projection matrix $\mathbf{T}_{kn}^S$.

$$\mathbf{q}_k^S(w) = \mathbf{T}_{kn}^S\ \mathbf{q}_n(w) \tag{26}$$

$$\text{with}\quad \mathbf{T}_{kn}^S = \mathbf{m}_k^{S^{-1}}\ \mathbf{F}_{ki}^S\ \mathbf{M}_{ii}^S\ \mathbf{F}_{in}$$

Similar expressions may be derived for the case of fixed-interface substructures.

3.5 Modal Participation Factors

Modal Participation Factors (MPF) are used to quantify the contribution or importance of a substructure or its modes with respect to the response of the global structure.

The MPF are derived from elastic modal energies in order to establish a direct link to the localization and specification of damping associated with the modes of substructure.

The total elastic energy, $\mathbf{E}(w)$, of a given response may be expressed in terms of the generalized displacements, $\mathbf{q}_n(w)$ and the associated generalized stiffnesses $\mathbf{k}_n$.

$$\mathbf{E}(w)=\frac{1}{2}\sum_{n}\mathbf{k}_n\left|\mathbf{q}_n(w)\right|^2 \tag{27}$$

The MPF provide the distribution of the total energy among the modes of the substructures expressed as a fraction of total energy as defined below.

$$\mathbf{MPF}_k^S(w)=\frac{\mathbf{E}_k^S(w)}{\mathbf{E}(w)} \tag{28}$$

with

$$\mathbf{E}_k^S(w)=\frac{1}{2}\mathbf{k}_k^S\left|\mathbf{q}_k^S(w)\right|^2=\frac{1}{2}\mathbf{k}_k^S\left|\mathbf{T}_{kn}^S\ \mathbf{q}_n(w)\right|^2 \tag{29}$$

By extension, MPF providing the contribution of each substructure may be defined using the substructure's stiffness matrix, $\mathbf{K}_{ii}^S$, according to the following expression.

$$\mathbf{MPF}^S(w)=\frac{\mathbf{E}^S(w)}{\mathbf{E}(w)} \tag{30}$$

with

$$\mathbf{E}^S(w)=\frac{1}{2}\mathbf{q}_n(w)^T\ \mathbf{K}_{nn}^S\ \mathbf{q}_n(w) \tag{31}$$

and

$$\mathbf{K}_{nn}^S=\mathbf{F}_{ni}\ \mathbf{K}_{ii}^S\ \mathbf{F}_{in} \tag{32}$$

And finally, MPF may be defined to determine the energy distribution between the structure and fluid using the matrices introduced in Eq. (15) and (16).

$$\mathbf{MPF}^s(w)=\frac{\mathbf{E}^s(w)}{\mathbf{E}(w)}\qquad \mathbf{MPF}^f(w)=\frac{\mathbf{E}^f(w)}{\mathbf{E}(w)} \tag{33}$$

with

$$\mathbf{E}^s(w)=\frac{1}{2}\mathbf{q}_n(w)^T\ \mathbf{K}_{nn,s}\ \mathbf{q}_n(w)\quad \text{and}\quad \mathbf{E}^f(w)=\frac{1}{2}\mathbf{q}_n(w)^T\ \mathbf{K}_{nn,f}\ \mathbf{q}_n(w) \tag{34}$$

3.6 Damping Specification

For the critical substructure mode k of the substructure S, identified by the MPF, damping may then be specified via structural damping factors $\mathbf{g}_k^S$. Using the modal projection matrix, $\mathbf{T}_{kn}^S$ of Eq. (26), the specified damping is expressed in terms of the global structure's modes by the matrix $\mathbf{D}_{nn}^S$ defined below.

$$\mathbf{D}_{nn}^S=\mathbf{T}_{nk}^S\left(\mathbf{g}_k^S\cdot\mathbf{k}_k^S\right)\mathbf{T}_{kn}^S \tag{35}$$

This additional damping is summed over all substructures and then added to the dynamic stiffness matrix of Eq. (14) resulting in the following expression.

$$\mathbf{K}_{nn}(w)=-w^2\ \mathbf{m}_n+\mathbf{k}_n+i\,G\,\mathbf{K}_{nn,s}+i\,GFL\,\mathbf{K}_{nn,f}+i\sum_{S}\mathbf{D}_{nn}^S \tag{36}$$

Using Eq. (36) the frequency responses are computed following the procedure described in §3.3.

4 Software Description

4.1 Introduction

The damping specification methodology has been integrated into an interactive software application named OSACA, using MATLAB as the principal development environment. OSACA was designed from the start to be used in an industrial context and therefore had to satisfy several functional requirements summarized below.

- Handling of large models (millions of DOF and thousands of modes)
- Interface with CAD tools used by PSA Peugeot Citroën
- Interface with MSC/NASTRAN (located on a high-performance server)
- Interactive damping specification and response calculations

4.2 Architecture

The overall architecture of OSACA includes five functional steps which are schematized below in Figure 2 and described hereafter.

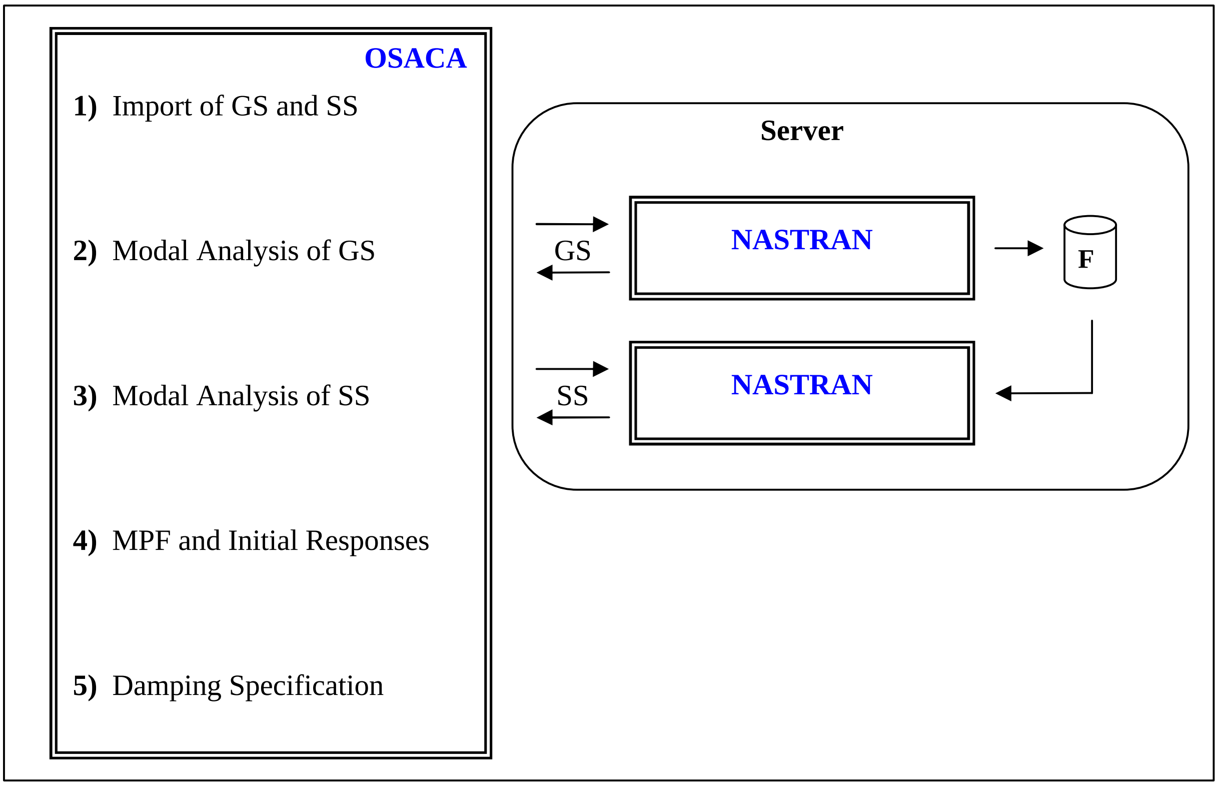


Figure 2 - OSACA Architecture

Import of GS and SS

The first step involves importing of the global structure (GS) and corresponding substructures (SS). The NASTRAN input files are read, checked and adapted in preparation for the subsequent analyses performed in NASTRAN and by OSACA. The definition of all load cases (excitation and response points, initial damping, excitation frequencies, etc.) is extracted automatically from the GS input file.

The mesh of each imported substructure must coincide with the corresponding part of the global structure's mesh. However, the GS and SS node and element IDs do *not* have to be the same since ID correspondence tables are computed automatically using geometrical and topological analysis.

In practice only the substructures for which damping is to be specified need to be imported. The boundary conditions as well as the definition of sub-objects are extracted automatically from the SS input file.

Modal Analysis of GS

The next step involves computing the normal modes of the global structure in NASTRAN. If fluid cavities are present, the modal projection technique described in chapter 3 using the uncoupled structure and fluid modes along with residual modes is performed to reduce computation time while maintaining accuracy. The modal projection technique is implemented into the NASTRAN using a specially developed DMAP script.

Output of the modal analysis is imported to OSACA and includes all the modal terms required to compute the initial frequency responses within OSACA. The global structure modes are also stored locally on the NASTRAN machine, and used as input for substructure analysis.

Modal Analysis of SS

The normal modes of each substructure are computed in NASTRAN and used along with the normal modes of the global structure to determine the modal projection matrix $\mathbf{T}_{kn}^{S}$ defined in Eq. (26). The normal modes and projection matrix for each SS are imported to OSACA and used for subsequent computation of MPF and damping specification.

MPF and Initial Responses

At this stage, initial responses according to Eq. (14) and the associated MPF defined in chapter 3.5 may be calculated with OSACA. The initial responses may be compared with prescribed specification levels to evaluate response levels over the entire frequency band. The MPF may then be computed at the critical frequencies where the response levels exceed the specification in order to identify the substructure modes for which damping allocation would be the most effective at reducing the response levels.

Damping Specification

The last step involves allocating damping to the substructure modes identified by the MPF and computing the corresponding damped responses according to chapter 3.6. Since response calculation is performed using a modal approach, the damping specification process is performed interactively. Moreover, the damped responses may be computed over any portion of the frequency band - allowing the user to efficiently examine local effects around one or two peaks, or broadband effects over the entire frequency range. An unlimited number of damping specification cases may be defined for any number of substructure modes.

4.3 Graphical User Interface

The OSACA graphical user interface is illustrated below in Figure 3. All functions are integrated into a single window combining menu and icon bars, control panels and graphical displays.

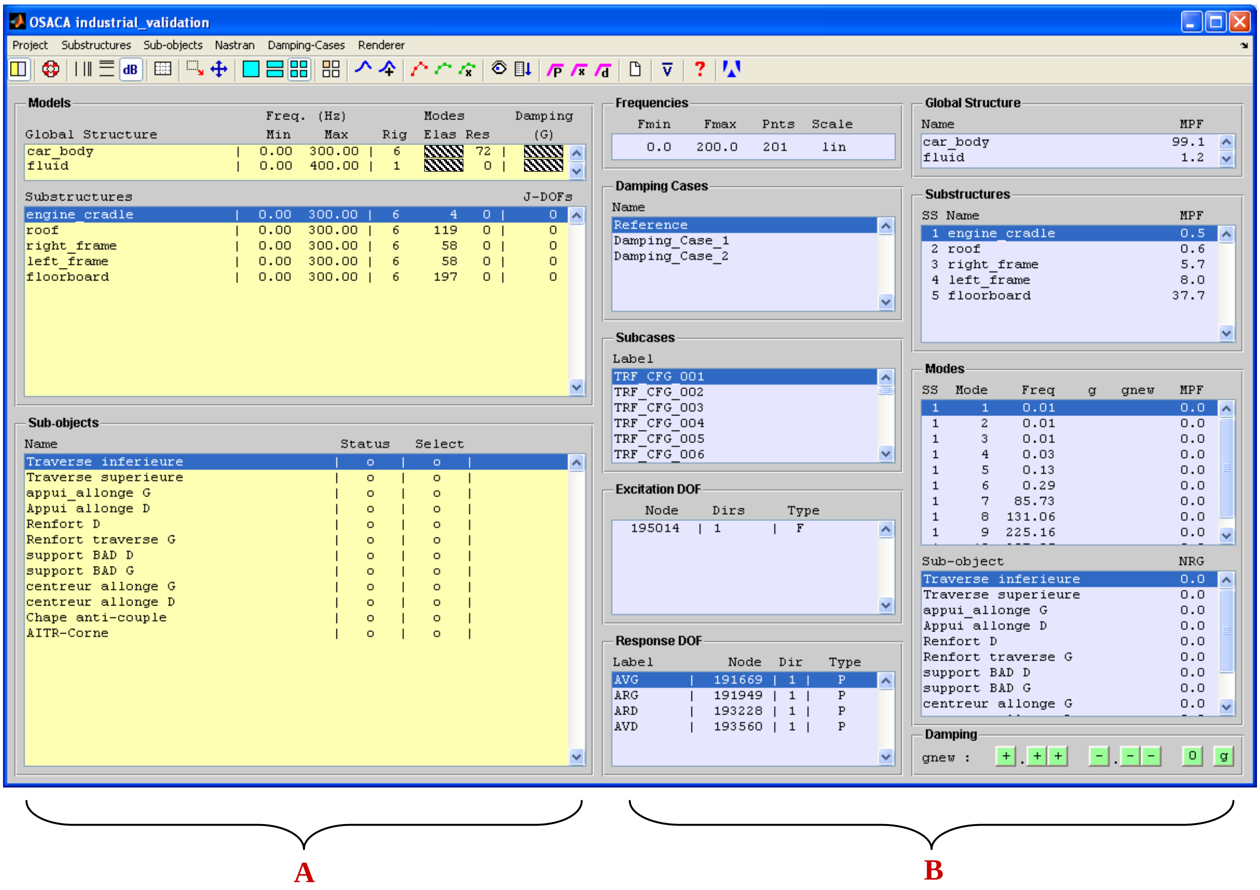


Figure 3 - OSACA Graphical User Interface

The user interface illustrated in Figure 3 is organized into 3 panels - each panel regrouping a number of related functions and displays.

The first panel (**A**) shown to the left in Figure 3 provides a list of the global structure and associated substructures along with a summary of the computed modes. Optional sub-objects representing structural zones of each substructure are also listed. This panel is used primary during initial model import and modal analysis in NASTRAN.

The second panel (**B**) shown to the right in Figure 3 contains the controls and displays relative to response calculations, MPF and damping specification.

A third panel (shown later on in Figure 6) is dedicated to response plots. From one to four response plots may be selected to display and compare responses along with specification profiles for any number of damping cases. Frequency bands may be selected graphically in this panel to perform response calculations locally around one or several peaks. An illustration of the plot panel is presented hereafter in the industrial application.

5 Industrial Application

5.1 Global Structure

The methods and software developed in the framework of the study were validated using the car body model illustrated in Figure 4 comprising the structural mesh shown at left and an acoustic cavity shown at right representing together the global structure.

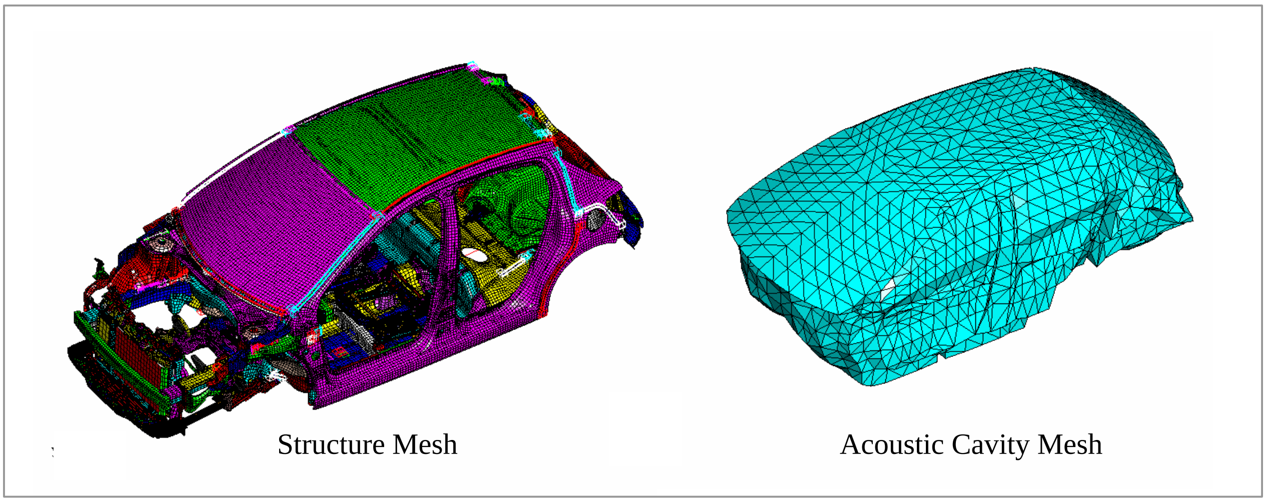


Figure 4 - Global Structure with Acoustic Cavity

For this particular industrial application OSACA was used to study and attenuate the acoustic pressure response levels computed at four points (fore and aft, left and right) within the passenger compartment. A total of 72 single point load cases located at 24 locations in the car body were used to excite the structure in the frequency range from 0 to 200 Hz.

Modes of the coupled fluid-structure were calculated using the uncoupled structure and fluid cavity modes comprising rigid body, elastic and residual modes up to 300 Hz for the structure and 400 Hz for the fluid cavity to minimize modal truncation effects. These results are displayed in the upper left of Figure 3 along with global damping values for the structure and for the acoustic cavity.

5.2 Substructures

To study the effects of damping allocation, the five following substructures of the car body were considered: floorboard, left frame, right frame, engine cradle and roof. The corresponding meshes are presented in Figure 5.

For each substructure the normal modes up to 300 Hz were calculated ranging in number from 4 elastic modes for the relatively stiff engine cradle to 197 elastic modes for the floorboard. The list of the substructures and corresponding modes is shown in Figure 3 below the global structure modes.

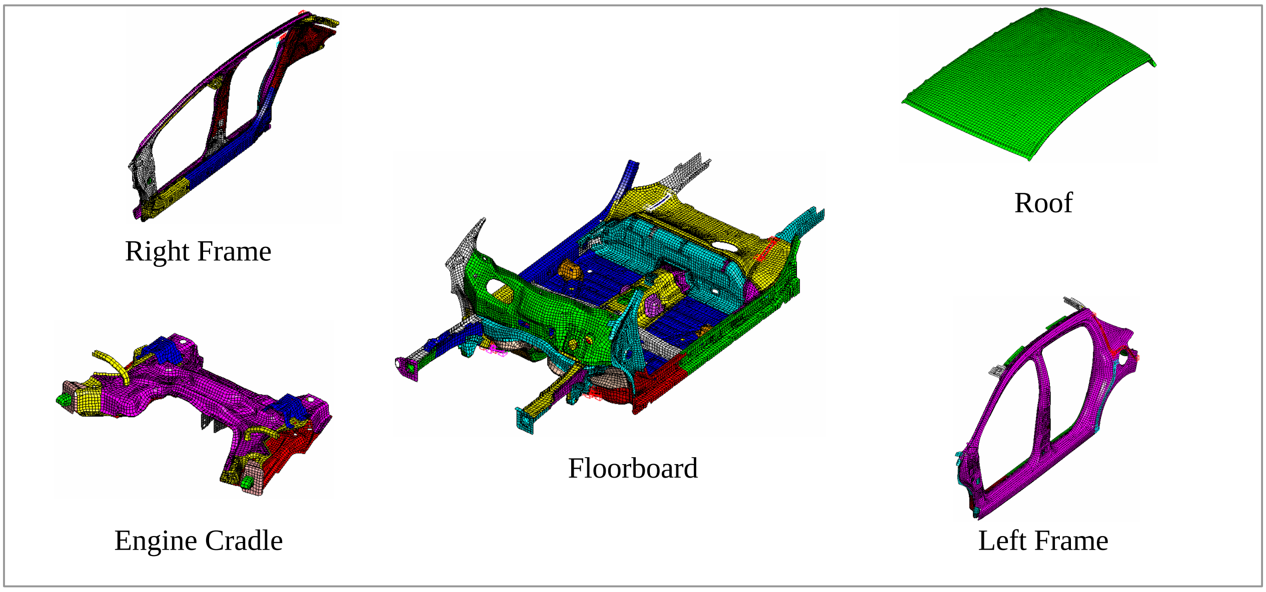


Figure 5 - Car Body Substructures

5.3 MPF and Mode Localization

Consider the acoustic response for a given load case and response point plotted in Figure 6. For this example we wish to attenuate the response level in the 120 -160 Hz frequency range encircled in red. By clicking on a peak (yellow cross at 148 Hz), the corresponding MPF are displayed allowing the user to readily identify the substructures and modes capable of effectively attenuating the selected peak. For the peak at 148 Hz we see that the floorboard mode 59 at 140.80 Hz contributes significantly (MPF = 3.4 %).

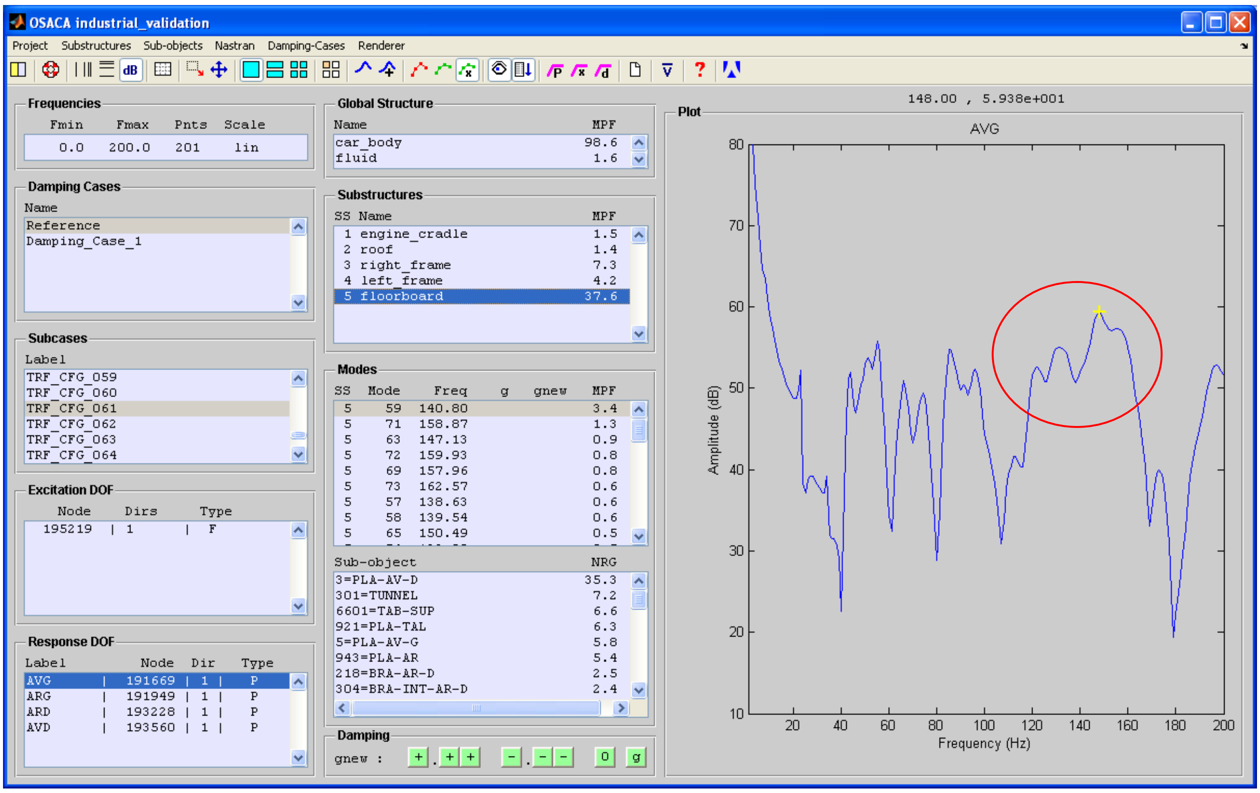


Figure 6 - Damping Localization using MPF

Following a similar examination of the MPF at the other peaks in the frequency range, we arrive at the list of identified substructure modes shown in Figure 7. Notice that the natural frequencies of the identified substructure modes are in general close to the frequency of the corresponding peaks.

Peak	Substructure	Mode	MPF
122 Hz	Left Frame	29 @ 154.61 Hz	2.1 %
131 Hz	Floorboard	58 @ 139.54 Hz	6.1 %
148 Hz	Floorboard	59 @ 140.80 Hz	3.4 %
156 Hz	Floorboard	69 @ 157.96 Hz	2.8 %

Figure 7 - Identified Substructure Modes

5.4 Damping Specification

Damping specification consists of assigning damping values to each of the identified substructure modes and determining the effect on the corresponding responses. For the industrial application, a damping value of g = 0.1 was assigned to all 4 substructure modes of Figure 7.

To evaluate the local influence of the specified damping, the response of Figure 6 was calculated in the frequency band of the selected peaks. The resulting damped response is plotted in green in Figure 8a and compared to the initial response plotted in red. As expected, all peaks are damped with attenuation consistent with the amount of prescribed damping and corresponding MPF. Notice that between peaks and in particular near antiresonances, the damped response amplitude may increase as illustrated below near 140 Hz.

Finally the response was computed over the entire frequency range in order to check the wideband effects of the specified damping as shown in Figure 8b. For this example, the influence of the specified damping remains concentrated on the selected peaks. However, in general substructure modes can have a wideband influence on the damped response.

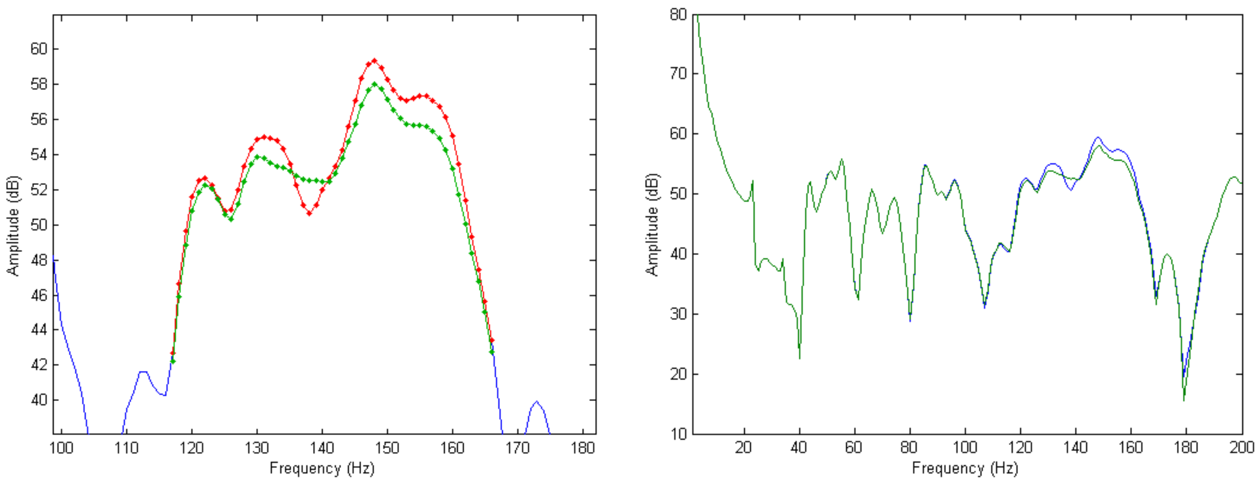


Figure 8 - Comparison of Initial and Damped Responses

6 Conclusions

A method for damping localization and specification in automotive structures based on modal projection has been developed and implemented in an industrial tool at PSA Peugeot Citroën. Using the modes of the global structural its substructures, the computation of responses, modal participation factors and damping specification may be efficiently performed in an interactive environment. Both structural and vibroacoustic analysis can be performed. The tool has been validated using several benchmark and industrial models and is now being used for the preliminary design of vehicles.

References

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