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An energy-based updated modal approach for the efficient analysis of large trimmed models

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Large trimmed models, widely encountered in the automotive industry, generally involve a body-in-white structure coupled to an acoustic cavity and covered by a set of trim components. Due to the large number of degrees of freedom of such models, the direct analysis in physical coordinates, though theoretically correct, is not feasible on current computers. The paper presents an alternative and efficient solution strategy in modal coordinates that relies on an update of the modal parameters of the car body and the acoustic cavity, based on the energetic behavior of the trim components. The description of the trim components in terms of an energetic database simplifies the exchange of data between the automotive manufactor and the trim provider. Furthermore, it enables a fast frequency analysis of various loadcases/trim scenarios and makes optimization possible. The updated modal approach implemented in Actran/Trim [4] is applied on a simplified car model on which various trim components are applied. The application shows how the stiffening, added mass and damping effects of each trim component can be derived from the energetic database and can guide an optimization process.

1 Introduction

Modal approaches are widely used in the automotive industry for analysing the frequency response of large vibroacoustic models. Generally, these models involve a body-in-white structure, represented by undamped *in-vacuo* modes, coupled to an acoustic cavity, also described by an undamped *rigid-wall* modal basis. The efficiency of the modal approach relies, on one hand, on a modal extraction for which efficient eigenvalue solvers are available, and on the other hand, on the fact that the problem in modal coordinates is of reduced size. The handling of trimmed configurations in the same modal context is however more problematic for the following reasons :

- the elimination or the over-simplification of the trim components is not appropriate;
- significant stiffness and mass variations may be introduced by the trim components;
- complex frequency-dependent dissipation mechanisms may occur in the various layers of the trim components.

Due to the involved model size, an approach in physical coordinates, though theoretically correct, is not possible.

Recently, an updated modal approach has been presented for circumventing the above limitations [1]. This approach mixes in a hybrid procedure the modal approach for both the structure and the cavity and a physical approach for the trim components. For each of these trim components, an energetic database is built that describes the component behavior when it is excited by the structural and fluid modes. This database is used to reduce the trim component to updates of the structure and cavity modal parameters so that a solution in modal coordinates can be performed. Various loadcases and trim scenarios are therefore analysed at a reduced cost. Also random excitations are supported by the method [2]. Finally, additional ingredients such as incompatible mesh handling, parallelization and modal filtering techniques can be integrated in order to enhance the performance and the applicability to industrial problems [3].

The present paper details the updated modal approach and shows how the update of the modal parameters can be obtained from energetic considerations. An application on a simplified car model involving two trim components is presented. The effect of both trim components on the car body displacement and acoustic pressure in the cavity is investigated. It is also shown how a post-processing of the energetic databases related to each trim component delivers additional information on the specific effect of each trim component.

2 Discrete vibro-acoustic problem statement

2.1 Modal representation

Let us consider the untrimmed and undamped vibroacoustic configuration, modeled as a linear elastic undamped structure S coupled to the undamped acoustic cavity F along the interface Γ_{SF}. A standard finite element discretization is assumed. If we denote by $U_S(\omega)$ the frequency-dependent response of the structure S, a modal representation of the response can be written as:

$$U_S(\omega) = \sum_{i=1}^{N_S} \alpha_{S,i}(\omega)\Phi_{S,i}, \tag{1}$$

where N_S is the number of structure *in-vacuo* modes $\Phi_{S,i}$ $(i = 1, \ldots, N_S)$ and $\alpha_{S,i}(\omega)$ are the associated frequency-dependent modal coordinates. The structure *in-vacuo* modes are identified by the solution of an associated eigenvalue problem :

$$\mathbf{K_S} \cdot \Phi_{S,i} = \omega_{S,i}^2 \mathbf{M_S} \cdot \Phi_{S,i}, \tag{2}$$

where $\mathbf{K_S}$ and $\mathbf{M_S}$ are the stiffness and mass matrices of the structure, respectively, and $\omega_{S,i}^2$, the associated eigenvalues. Assuming the normalization of the modes with respect to the mass, the following relations hold :

$$\Phi_{S,i}^T \cdot \mathbf{K_S} \cdot \Phi_{S,j} = \delta_{ij}\omega_{S,i}^2, \tag{3}$$

$$\Phi_{S,i}^T \cdot \mathbf{M_S} \cdot \Phi_{S,j} = \delta_{ij}. \tag{4}$$

A similar modal representation for the frequency response of the cavity F can be written as:

$$p_F(\omega) = \sum_{i=1}^{N_F} \alpha_{F,i}(\omega)\Phi_{F,i}, \tag{5}$$

where N_F is the number of cavity *rigid-wall* modes $\Phi_{F,i}$ $(i = 1, \ldots, N_F)$ and $\alpha_{F,i}(\omega)$ are the associated frequency-dependent modal coordinates. The fluid *rigid-wall* modes satisfy the relation :

$$\mathbf{K_F} \cdot \Phi_{F,i} = \omega_{F,i}^2 \mathbf{M_F} \cdot \Phi_{F,i}, \tag{6}$$

with the mass normalization convention :

$$\Phi_{F,i}^T \cdot \mathbf{K_F} \cdot \Phi_{F,j} = \delta_{ij}\omega_{F,i}^2, \quad (7)$$

$$\Phi_{F,i}^T \cdot \mathbf{M_F} \cdot \Phi_{F,j} = \delta_{ij}. \quad (8)$$

2.2 Projection into the modal space

The time-harmonic dynamic behavior of the structure and the cavity is described by the following system of algebraic equations:

$$\begin{pmatrix} \mathcal{Z}_{\mathcal{SS}} & \mathcal{Z}_{\mathcal{SF}} \\ \mathcal{Z}_{\mathcal{SF}}{}^T & \mathcal{Z}_{\mathcal{FF}} \end{pmatrix} \cdot \begin{pmatrix} \alpha_S(\omega) \\ \alpha_F(\omega) \end{pmatrix} = \begin{pmatrix} \mathbf{\Phi_S}{}^T \cdot F_S(\omega) \\ \mathbf{\Phi_F}{}^T \cdot \frac{F_F(\omega)}{\omega^2} \end{pmatrix} \quad (9)$$

with

$$\mathcal{Z}_{\mathcal{SS}} = \left(\delta_{ij}(\omega_{S,i}^2 - \omega^2)\right), \quad (10)$$

$$\mathcal{Z}_{\mathcal{FF}} = \frac{1}{\omega^2}\left(\delta_{ij}(\omega_{F,i}^2 - \omega^2)\right), \quad (11)$$

$$\mathcal{Z}_{\mathcal{SF}} = \mathcal{Z}_{\mathcal{SF}ij} = \int_{\Gamma_{SF}} (\Phi_{S,i} \cdot n)\Phi_{F,j} d\Gamma_{SF}. \quad (12)$$

In the above equations, F_S and $F_F = \left(i\omega \int_\Gamma v_n N_i d\Gamma\right)$ are the vectors of nodal excitations on the structure and the cavity, respectively.

3 Handling of the trim components

The handling of the trim components in the modal updated approach presented in this paper results from a two-step procedure :

- construction, for each trim component, of an energetic database describing the component behavior when it is excited by the trace of the structure and fluid modes;
- conversion of this energetic database into updates of the modal stiffnesses, modal masses and modal dampings, for both the fluid and structure parts, and update of the fluid/structure modal coupling matrix.

Note that the description of each trim component in terms of an energetic database results in several benefits for the proposed approach:

- due to their energetic nature, the contributions of the various trim components are additive. Once the energetic database is available for each trim component, the analysis of several trim configurations is straightforward. Sensitivity analysis as well as optimization procedures are therefore allowed.
- the energetic database is a stand-alone information package that can be shared by the car manufactor and the trim supplier. In this sense, the approach respects the distributed nature of the design process where the vehicle is under the responsibility of the OEM while the trim components are produced by multiple trim suppliers.

3.1 Energetic database

Let us consider a trim component T coupled to the structure S along an interface Γ_S and to the cavity along an interface Γ_F. For each structural mode $\Phi_{S,i}$ ($i = 1, \ldots, N_S$) and each cavity mode $\Phi_{F,j}$ ($j = 1, \ldots, N_F$), we denote by $\Phi_{\Gamma_S,i}$ and $\Phi_{\Gamma_F,j}$ the trace of these modes on the interfaces Γ_S and Γ_F, respectively. Given an appropriate finite element model of the trim component, it is possible to describe the energetic behavior of the trim component by evaluating the strain, kinetic and dissipated powers fo each of the following $N_S + N_F$ analyses :

- configurations $C_{S,i}$ ($i = 1, \ldots, N_S$) : finite element analysis of the trim component at the frequency $\omega_{S,i}$, subjected to a prescribed displacement $\Phi_{\Gamma_S,i}$ on Γ_S and to zero pressure on Γ_F;
- configurations $C_{F,j}$ ($j = 1, \ldots, N_F$) : finite element analysis of the trim component at the frequency $\omega_{F,j}$, subjected to a zero displacement on Γ_S and to a prescribed pressure $\Phi_{\Gamma_F,j}$ on Γ_F.

3.2 Update of the modal parameters

The update of the structure modal parameters is based on energetic considerations. In the absence of any trim component, the strain, kinetic and dissipated powers corresponding to a displacement field $U = \Phi_{S,i}$ at the frequency $\omega_{S,i}$ are given by :

$$V_S(\Phi_{S,i}) = V_{S,i} = \frac{\omega_{S,i}^3}{2},$$

$$T_S(\Phi_{S,i}) = T_{S,i} = \frac{\omega_{S,i}^3}{2},$$

$$W_S(\Phi_{S,i}) = W_{S,i} = \frac{\eta_{S,i}\omega_{S,i}^3}{2},$$

where $\eta_{S,i}$ denotes a damping factor for the i-th structure mode. These relations clearly relate the modal stiffness to the strain power, the modal mass to the kinetic power and the damping factor to the dissipated energy.

Now, if we consider the trimmed configuration and we assume that the mode shapes are not altered by the presence of the trim, the strain, kinetic and dissipated powers in the trimmed structure are increased according to:

$$V_{S,i} \rightarrow V_{S,i} + V_{T,S,i},$$

$$T_{S,i} \rightarrow T_{S,i} + T_{T,S,i},$$

$$W_{S,i} \rightarrow W_{S,i} + W_{T,S,i},$$

where $V_{T,S,i}$, $T_{T,S,i}$ and $W_{T,S,i}$ are the strain, kinetic and dissipated powers in configuration $C_{S,i}$, respectively. From an energetical point of view, the trimmed configuration can thus be replaced by an equivalent untrimmed structure having the following modal parameters :

$$\Phi_{S,i}^T \cdot \mathbf{K_S} \cdot \Phi_{S,i} = \omega_{S,i}^2 + \frac{2}{\omega_{S,i}} V_{T,S,i}, \quad (13)$$

$$\Phi_{S,i}^T \cdot \mathbf{M_S} \cdot \Phi_{S,i} = 1 + \frac{2}{\omega_{S,i}^3} T_{T,S,i}, \quad (14)$$

$$\eta_{S,i,\text{new}} = \eta_{S,i} + \frac{2}{\omega_{S,i}^3} W_{T,S,i}. \quad (15)$$

Similar considerations lead to the relations governing the update of the fluid modal parameters. Also the fluid/structure modal coupling is altered by the presence of the trim component. An amplification factor is consequently applied to the rows of the $\mathcal{Z}_{\mathcal{SF}}$ matrix :

$$A_{S,i} = \sqrt{\frac{< v^2 >_{\Gamma_F,i}}{< v^2 >_{\Gamma_S,i}}}, \qquad (16)$$

where $< v^2 >_{\Gamma_S,i}$ and $< v^2 >_{\Gamma_F,i}$ are the square velocities on the structure/trim and trim/cavity interfaces Γ_S and Γ_F in the configuration $C_{S,i}$, respectively.

4 Numerical application

The updated modal approach, implemented in the Actran/Trim software [4], is applied in this section on a simplified car body model coupled to an inner cavity. Figure 1 shows the finite element model and the two trim components (a dashboard insulator and a floor carpet). Both trim components consist in a two-layer foam and heavy septum assembly. Fluid-structure coupling occurs on the whole interface between the structure and the cavity. The structural modes in the frequency range $[0, 400]$ Hz (389 modes) and the fluid modes in the frequency range $[0, 800]$ Hz (288 modes) are considered. Global modal damping is assumed in both the structure ($\eta_S = 0.06$) and the cavity ($\eta_F = 0.08$). The model is excited by a normal unit point force at the engine mount point. The model response is observed at three structural locations (point 3785 is the engine mount point, point 4987 is on the right front floor panel and point 6170 is on the right dashboard insulator) and at the driver's ear location.

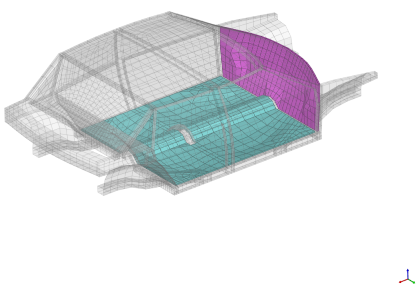

Figure 1: Simplified car application : finite element discrete model.

Figure 2 shows the normal displacement of the structure at the observation points when both trim components are considered simultaneously. A significant damping effect is introduced in the model due to the trim presence. In the figure 3, the acoustic pressure at the driver's ear is shown for different trim configurations. Depending on the frequency range, the dashboard insulator and the floor panel decrease or increase the pressure level in comparison with the untrimmed configuration. The curve related to the configuration in which both trim component are active lies between the curves related to one of the two trim components.

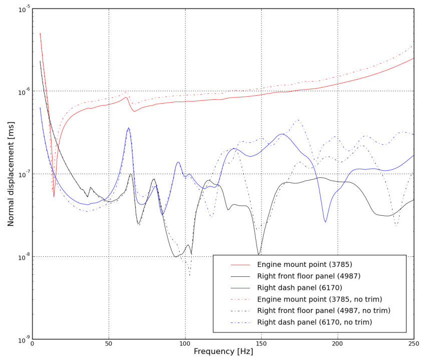

Figure 2: Normal displacement of the car body : comparison between trimmed and untrimmed configurations.

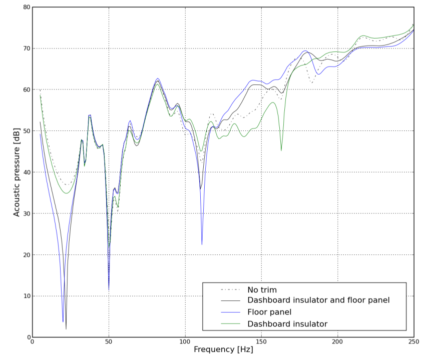

Figure 3: Acoustic pressure at the driver's ear : comparison between various trim scenarios.

In order to understand the effect of the trim components on the overall response of the model, the strain, kinetic and dissipated powers in each trim component can be analysed. The figures 4 and 5 show these values, directly extracted from the energetic database related to the floor panel component. Each point in the graph corresponds to an excitation by a structural mode (figure 4) or to a fluid mode (figure 5). The powers in the floor panel are compared to the modal powers in the structure and the cavity, respectively. As values for the strain and kinetic powers in the floor panel are low in comparison to the modal strain and kinetic powers in the structure and cavity, the floor panel does not add stiffening nor mass effect to the model. The values of the dissipated powers are however of the same order of magnitude as the powers dissipated in the structure and cavity by the global damping, which explains the damping effect of the floor panel.

An important point to understand the effect of the trim components on the cavity response is the modification of the fluid/structure coupling by the presence of the trim component. Figure 6 shows, for both the dashboard insulator and the floor panel, the value of the amplification ratio in Eq (16) involved in the update of the modal

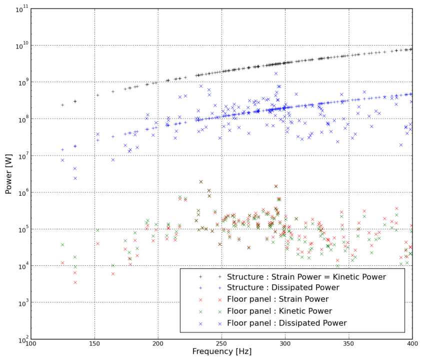

Figure 4: Strain, kinetic and dissipated powers in the floor panel submitted to the structural modes compared to the modal powers in the structure.

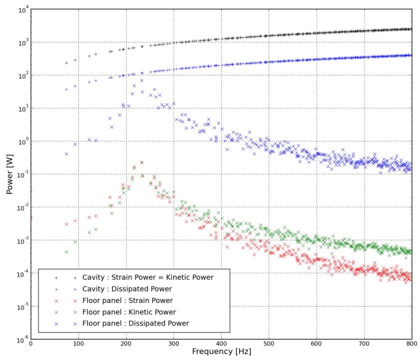

Figure 5: Strain, kinetic and dissipated powers in the floor panel submitted to the acoustic modes compared to the modal powers in the cavity.

fluid/structure coupling matrix. In the frequency range between 150 and 300 Hz, the amplification ratio significantly increases. This is related to a pumping resonance of the trim components and explains the increase of the pressure levels in the cavity.

5 Conclusion

The paper has presented an energy-based updated modal approach for the efficient analysis of large trimmed models. The approach accepts as input a modal representation of both the structure and the cavity as well as appropriate finite element models of the trim components. An energetic database for each trim component is constructed by successively exciting it by the structural and fluid modes. Based on energetic considerations, each trim component is finally reduced to (additive) perturbations of the modal parameters for the structure, the cavity and the fluid/structure coupling.

From a physical point of view, each trim component may introduce a stiffening effect, a mass effect or a damping effect on the structure and on the cavity. Also an amplification of the fluid/structure modal coupling may be introduced by the trim component. All these effects are taken into account in the presented updated modal approach. The approach has been applied on a simplified car model. In this application, the untrimmed configuration and three trim scenarios have been compared and the variations between the different predicted results have been explained by an analysis of the energetic behavior of the trim components when they are excited by the structural and fluid modes.

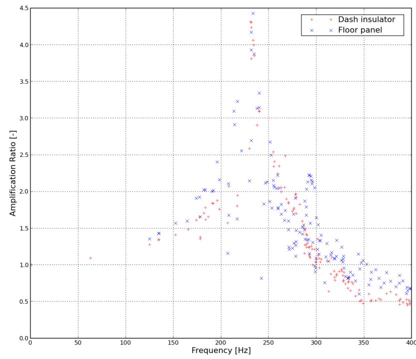

Figure 6: Amplification ratio used for the update of the fluid/structure modal coupling.

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