Response Computation of Structures with Viscoelastic Damping Materials using a Modal Approach - Description of the Method and Application on a Car Door Model Treated with High Damping Foam

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Abstract:
Numerous passive damping technologies including viscoelastic materials are used today in the design of automotive car bodies.

Viscoelastic materials are characterized by properties that are frequency and temperature dependent. In many FE codes, their use in automotive models is limited in terms of capabilities and often results in excessive computation time.

In the case of MSC/NASTRAN, for instance, there is neither possibility to define more than one viscoelastic material nor to introduce independent variations for tensile and shear moduli. Moreover, the efficient modal approaches must be put aside in favor of a direct resolution whose results, albeit accurate, cannot justify the increased computation time, especially for models with several million degrees of freedom.

This paper presents a method, based on a modal approach and implemented in MSC/NASTRAN using the DMAP language, to overcome these limitations. It is followed by its application to a Peugeot 207 car door treated with a high damping foam.

Keywords:
Damping, MSC/NASTRAN, viscoelastic materials, frequency-dependence, FEM.

1. Introduction

Nowadays, the automotive industry introduces different passive damping technologies for their performance gains in terms of vehicle weight, cost and acoustics. The majority of these technologies incorporate viscoelastic materials introduced during construction of the car body in various forms (sprayable materials, constrained layers, foams, etc.) in order to provide maximum energy dissipation while satisfying stringent design requirements.

Unlike metal-based materials, viscoelastic material properties are frequency and temperature dependent. Assuming linear viscoelastic behavior, master curves may be obtained experimentally to characterize the storage and loss moduli of the material as a function of the reduced frequency.

From a simulation point of view, the use of viscoelastic materials in many FE codes is limited in terms of capabilities and often results in excessive computation time. In the case of MSC/NASTRAN, only one frequency-dependent viscoelastic material with isotropic moduli may be defined. This limitation is incompatible with today's structures comprising several different viscoelastic materials, and whose tensile and shear moduli often vary independently. Moreover, when using frequency-dependent materials, the highly efficient modal approaches available in MSC/NASTRAN (Lanczos, AMLS, ACMS…) are abandoned in favor of a direct resolution whose results, albeit accurate, cannot justify the increased computation time, especially for automotive car body models with several million degrees of freedom.

To overcome these limitations and extend the capabilities of MSC/NASTRAN response analysis of automotive structures with viscoelastic materials, a method is proposed offering the following advantages:

1. handling several viscoelastic materials
2. using a modal approach to maintain computational efficiency
3. introducing frequency-dependent properties for both tensile and shear store moduli
4. compatibility with the DMAP language.

A detailed description of the developed method is given in the first part of this paper.

The second part describes its application to a Peugeot 207 car door treated with a high damping foam (HDF technology developed by Henkel). This latter aimed to evaluate the efficiency of the HDF treatment as a replacement for anti-vibration solutions used in mass production (bitumen pads and antiflutter material).

2. Viscoelastic Material

2.1 Introduction

The complex modulus approach ([1]-chapter 2.4) is often used to represent the dynamic behavior of viscoelastic material for harmonic response computations. Indeed, the experimental data characterizing the viscoelastic material are obtained in the frequency domain and therefore may be used directly to provide values for the different moduli and loss factors as a function of frequency and temperature.

In finite element analysis, frequency responses of structures with viscoelastic material are computed by interpolating the values of complex moduli from the tabulated experimental data at each excitation frequency and then assembling and solving the corresponding dynamic stiffness matrix.

This approach is available in NASTRAN via the direct frequency response (SOL 108) solution sequence [2]. Unfortunately, the implementation of viscoelastic material in NASTRAN is somewhat awkward and has several limitations.

First, only one viscoelastic material may be defined - clearly inadequate for today's automotive structures with numerous and different damping treatments.

Secondly, the viscoelastic material must be isotropic. In other words the real Young's and shear moduli are assumed to be related by a constant Poisson's ratio according to classical elasticity theory. Unfortunately, experimental data obtained from extensional and shear tests performed on the same viscoelastic material do not always satisfy this relationship. Moreover, loss factor values may greatly differ between extensional and shear tests imposing the need to distinguish between the two types of deformation.

Finally, viscoelastic materials cannot be used with the more efficient modal frequency response (SOL 111) solution sequence available in NASTRAN. This limitation may appear justifiable at first sight since the eigenvalue problem, and in particular the stiffness matrix, is frequency dependant. However, by combining eigenvectors calculated at a chosen reference frequency with residual modes used to represent the influence of the viscoelastic material, an enriched basis of modes can be formulated to overcome this obstacle.

The above mentioned improvements to the complex modulus approach implemented using NASTRAN are detailed hereafter.

2.2 Stress-Strain Relationship

The complex moduli of viscoelastic material are obtained experimentally by special vibration tests allowing the specimen to be deformed either in extension or shear as depicted below.

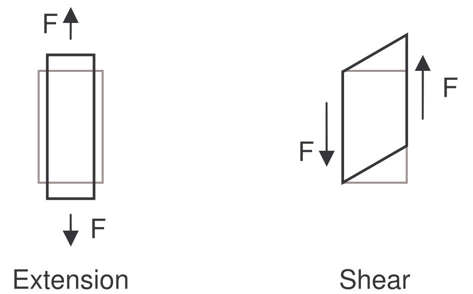

Fig. 1: Extension and Shear Deformation

From these tests, we obtain values for the Young's and shear complex moduli E^* and G^* as a function of frequency (or reduced frequency) which can also be expressed in terms of real moduli, *E* and *G*, and loss factors η_E and η_G as shown in the following stress-strain relationships.

$$\sigma_e = E^*(\omega)\varepsilon_e = E(\omega)\left[1 + i\eta_E(\omega)\right]\varepsilon_e \quad (1)$$

$$\tau_s = G^*(\omega)\gamma_s = G(\omega)\left[1 + i\eta_G(\omega)\right]\gamma_s \quad (2)$$

For a truly isotropic material, the Young's and shear moduli are related by the Poisson's ratio according to classical theory.

$$E = 2(1+\nu)\,G \quad (3)$$

Unfortunately, for many viscoelastic materials, the measured values of *E* and *G* do not satisfy the isotropic conditions of Eq. (3) using acceptable values of ν (close to 0.5). Moreover, the measured loss factors, η_E and η_G may also differ significantly. Therefore simple isotropic stress-strain relationships cannot be used to represent the viscoelastic material in the finite element model.

To overcome this difficulty, general *anisotropic* stress-strain relationships are used in order to uncouple the extensional and shear properties while maintaining *isotropic* behavior within each set of uncoupled properties.

For viscoelastic material modeled using plate elements, the stress-strain relationships (assuming plane stress conditions) are given by

$$\begin{bmatrix}\sigma_x\\\sigma_y\end{bmatrix}=\frac{E^*(\omega)}{1-\nu^2}\begin{bmatrix}1&\nu\\\nu&1\end{bmatrix}\begin{bmatrix}\varepsilon_x\\\varepsilon_y\end{bmatrix}\tag{4a}$$

$$\tau_{xy}=G^*(\omega)\,\gamma_{xy}\tag{4b}$$

with ν the constant real-valued Poisson's ratio used in the extensional stress-strain relationships only.

Similarly, for volume elements, the following 3D stress-strain relationships are used.

$$\begin{bmatrix}\sigma_x\\\sigma_y\\\sigma_z\end{bmatrix}=\frac{E^*(\omega)}{(1+\nu)(1-2\nu)}\begin{bmatrix}1-\nu&\nu&\nu\\\nu&1-\nu&\nu\\\nu&\nu&1-\nu\end{bmatrix}\begin{bmatrix}\varepsilon_x\\\varepsilon_y\\\varepsilon_z\end{bmatrix}\tag{5a}$$

$$\begin{bmatrix}\tau_{xy}\\\tau_{yz}\\\tau_{zx}\end{bmatrix}=G^*(\omega)\begin{bmatrix}1&0&0\\0&1&0\\0&0&1\end{bmatrix}\begin{bmatrix}\gamma_{xy}\\\gamma_{yz}\\\gamma_{zx}\end{bmatrix}\tag{5b}$$

2.3 Stiffness Matrix Assembly and Decomposition

Consider a finite element model containing one or several viscoelastic zones modeled using shell or volume elements. It is assumed that each viscoelastic zone corresponds to a single viscoelastic material with properties defined in Eq. (4) or (5).

Next, *reference* values, E_{ref} and G_{ref} are defined for the moduli corresponding to a reference frequency, ω_{ref}, usually selected within the range of excitation frequencies.

$$E_{ref}=E(\omega_{ref})\tag{6a}$$

$$G_{ref}=G(\omega_{ref})\tag{6b}$$

Using these reference moduli, the stiffness matrix for each viscoelastic zone may be assembled and then decomposed into axial and shear contributions, $\mathbf{K}^E_{ref}$ and $\mathbf{K}^G_{ref}$, by simple subtraction with a second set of stiffness matrices assembled after doubling the value of each of the reference moduli.

$$\mathbf{K}^E_{ref}=\mathbf{K}(2E_{ref},\nu,G_{ref})-\mathbf{K}(E_{ref},\nu,G_{ref})\tag{7a}$$

$$\mathbf{K}^G_{ref}=\mathbf{K}(E_{ref},\nu,2G_{ref})-\mathbf{K}(E_{ref},\nu,G_{ref})\tag{7b}$$

This "double assembly" procedure is a much simpler alternative to the "double meshing" technique employed to separate extensional and shear dampening effects. Indeed, no modification of the model's mesh is required.

Finally, the frequency-dependent complex stiffness matrices of each viscoelastic zone can be expressed as follows where the frequency-dependent moduli and loss factors are obtained from interpolation of the tabulated measurements.

$$\mathbf{K}^E(\omega)=\frac{E(\omega)}{E_{ref}}\mathbf{K}^E_{ref}(1+i\,\eta_E(\omega))\tag{8a}$$

$$\mathbf{K}^G(\omega)=\frac{G(\omega)}{G_{ref}}\mathbf{K}^G_{ref}(1+i\,\eta_G(\omega))\tag{8b}$$

2.4 Modal Condensation and Residual Modes

The viscoelastic stiffness contributions of Eq. (8) can be included in the system equations of motion expressed below.

$$(-\omega^2\,\mathbf{M}+i\,\omega\,\mathbf{C}+\mathbf{K}(\omega))\,\mathbf{u}=\mathbf{F}\tag{9}$$

The above system can be solved directly at each frequency step. Unfortunately, with large models comprising millions of degrees of freedom (DOF), the required computational time may be unacceptable, in which case a modal condensation is warranted.

The idea behind modal condensation is to project the physical displacements, $\mathbf{u}$, onto a Ritz basis comprising a truncated set of system normal modes, $\mathbf{\Phi}$, and associated generalized displacements, $\mathbf{q}$.

$$\mathbf{u}=\mathbf{\Phi}\,\mathbf{q}\tag{10}$$

Determining the normal modes of Eq. (10) can require a significant computational effort especially for large models with a large number of modes - even when using the highly efficient Lanczos algorithm.

To further reduce computation time, new algorithms based on substructuring techniques [3] and particularly well-adapted to parallel processing, have recently appeared in commercial finite element codes. Two of these algorithms, AMLS and ACMS are available in NASTRAN and may be used in conjunction with the present development.

The modal transformation of Eq. (10) is introduced in Eq. (9) to obtain the condensed equations of motion expressed in the modal subspace.

$$(-\omega^2\,\overline{\mathbf{M}}+i\,\omega\,\overline{\mathbf{C}}+\overline{\mathbf{K}}(\omega))\,\mathbf{q}=\mathbf{\Phi}^T\mathbf{F}\tag{11}$$

The condensed system of Eq. (11) may now be efficiently solved for the modal displacements $\mathbf{q}$, from which the physical displacements are obtained using the transformation of Eq. (10).

The risk with modal condensation is that the normal modes used in the projection are computed using constant material properties, whereas the viscoelastic ones are frequency-dependent. In addition, the normal modes *do not* account for the influence of the viscoelastic zones, and in particular the localized dissipative forces acting on the structure. As a result, the responses obtained from the condensed system of Eq. (11) will invariably suffer from truncation error. It is then necessary to increase the number of modes used in the projection in order to reduce the effects of truncation error [4]. By experience, this condition is not often enough and the truncation error can remain important.

To compensate for the modal truncation effects, residual modes whose purpose is to represent the influence of the viscoelastic material are added to the Ritz basis.

A residual mode is similar to a normal mode in that it satisfies the same orthogonality properties and also has an associated eigenvalue. However, it does *not* satisfy the eigenvalue problem since each residual mode is in fact a combination of all the truncated (superior) normal modes. Details on the formulation and use of residual modes can be found in [5-7].

To illustrate the computation of residual modes, consider a viscoelastic zone with extensional and shear stiffness matrices $\mathbf{K}_{ref}^{E}$ and $\mathbf{K}_{ref}^{G}$. A set of modal forces representing the combined influence of the extensional and shear behavior of the viscoelastic zone can be defined using

$$\mathbf{F} = (\mathbf{K}_{ref}^{E} + \mathbf{K}_{ref}^{G})\,\mathbf{\Phi} \qquad (12)$$

The modal forces of Eq. (12) are applied to the physical structure as a static load producing the static vectors, $\mathbf{\Psi}$.

$$\mathbf{F} = \mathbf{K}\,\mathbf{\Psi} \qquad (13)$$

The residual modes $\hat{\mathbf{\Phi}}$ are derived from the static vectors $\mathbf{\Psi}$ using the operation defined in Eq. (14) involving the normal modes, $\mathbf{\Phi}$, and the linear transformation matrices $\mathbf{C}$ and $\mathbf{V}$.

$$\hat{\mathbf{\Phi}} = (\mathbf{\Psi} - \mathbf{\Phi}\,\mathbf{C})\,\mathbf{V} \qquad (14)$$

The transformation matrix, $\mathbf{C}$, is used to subtract the contribution of the normal modes from the static vector according to the classical Gram-Schmidt orthonormalization procedure.

Next, the transformation matrix, $\mathbf{V}$, is used to render the residual modes orthogonal to each other, and is obtained via orthogonality conditions.

Finally the residual modes are simply appended to the normal modes in the Ritz basis as shown below resulting in limited truncation error when solving the condensed equations of motion in Eq. (11).

$$\mathbf{u} = \begin{bmatrix}\mathbf{\Phi} & \hat{\mathbf{\Phi}}\end{bmatrix}\,\mathbf{q} \qquad (15)$$

Similar procedures can be formulated to account for several viscoelastic zones or to distinguish between extensional and shear effects instead of combining their influence.

2.5 NASTRAN Implementation

The approach to viscoelastic modeling described above has been implemented in MSC/NASTRAN using a DMAP script based on the modal response solution SOL 111.

In addition, the following performance requirements and features were included in the DMAP in order to satisfy stringent industrial needs:

- Computational efficiency with large models
- Compatibility with ACMS and AMLS methods
- Compatibility with vibroacoustic analysis
- Minimum impact on user interface

The resulting development is schematized in the following flowchart indicating the three principal steps in the SOL 111 solution sequence and the modifications introduced by the DMAP script.

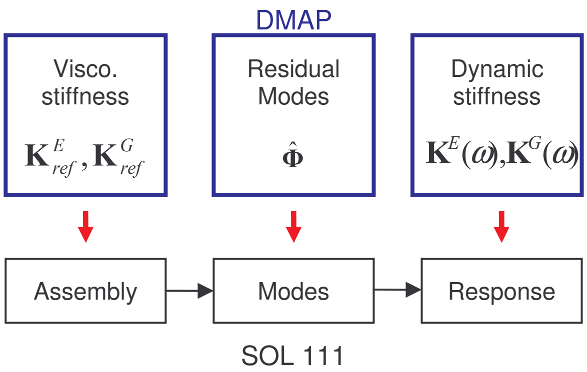

Fig. 2: Flowchart of New Modal Approach

In the assembly step, the viscoelastic stiffness matrices are assembled and decomposed into extensional and shear contributions. Next, the system normal modes are enriched with residual modes associated with the viscoelastic zones in order to minimize truncation errors. Finally the frequency responses are computed taking into account the frequency-dependent stiffness and damping properties of the viscoelastic materials.

Modifications of the user interface are limited to the material and table BULK entries used to define the viscoelastic material properties, and are described hereafter.

The reference moduli E_{ref} and G_{ref} are specified on MAT2 and MAT9 material property entries for plate and volume viscoelastic zones respectively. The different moduli are written according to Eq. (4) and (5). The viscoelastic materials are recognized by their material identification number (MID) whose starting value is defined by the PARAM VISCOMID.

For each viscoelastic material entry, 4 tables using the TABLED1 entries must be provided. These tables provide the measured moduli (*E*, *G*) and corresponding loss factors (η_E, η_G) for the viscoelastic material as a function of frequency. The tables are associated with each viscoelastic material by their identification number based on the corresponding MID.

3. Industrial Application

3.1 Introduction

The complex modulus approach described above and implemented in NASTRAN was applied to a Peugeot 207 car door in order to validate the modal approach with respect to the direct approach, and secondly to evaluate the efficiency of highly dissipative damping foam, HDF developed by HENKEL, as a replacement for anti-vibration solutions used in mass production (damping pads and antiflutter material). The HDF technology was presented at the SIA conference on Automobile & Railroad Comfort in 2006 [8].

A view of the car door is shown below. The currently used damping solutions are indicated schematically by the white squares and blue bars. The candidate zones for the highly dissipative foam (HDF) are outlined in red.

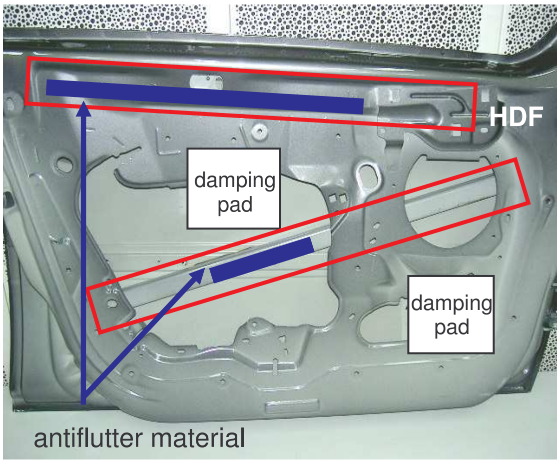

Fig. 3: Car Door with Damping Treatments

The production and HDF damping solutions were each integrated into a detailed finite element model with approximately 35 000 nodes. A view of the model comprising the door outer panel and the reinforcement beams is shown below.

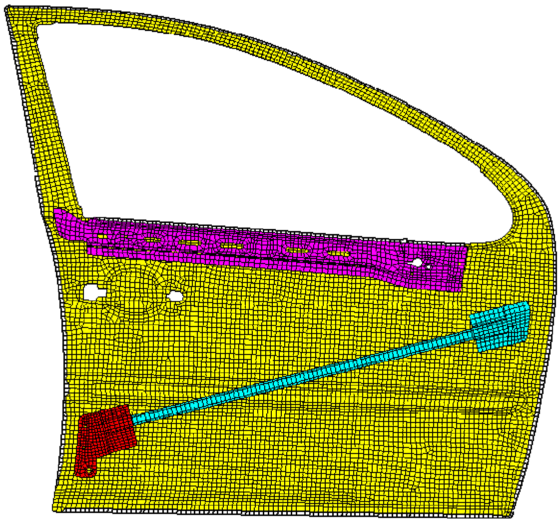

Fig. 4: Car Door Finite Element Model

3.2 Validation of Modal Approach

The car door model with HDF was chosen to validate the new method. To assess the capabilities and performance of the method compared to standard analysis techniques, three representations of the viscoelastic material listed below were considered.

- **Constant Isotropic**. The HDF is modeled using an isotropic material with a constant Young's modulus and loss factor obtained from the Young's complex moduli at 50 Hz.
- **Frequency-Dependent Isotropic**. The HDF is modeled using the frequency dependent complex Young's moduli. The shear modulus is related to the Young's modulus by a constant Poisson's ratio.
- **Frequency-Dependent Anisotropic**. The HDF is modeled using the frequency dependent complex Young's and shear moduli according to Eq. (4) and (5).

A series of frequency response calculations using NASTRAN was performed via direct (SOL 108) and modal (SOL 111) solutions involving the above viscoelastic material representations. Velocities were computed in response to the excitation of the door frame close to the latch position. The door was considered in free boundary conditions.

To simplify the comparison, a single mean quadratic velocity response was calculated on 36 points of the door outer panel over 500 frequency steps between 10 and 200 Hz.

All modal solutions were performed using ACMS to obtain a basis of 120 normal modes computed up to 400 Hz - twice the maximum excitation frequency - to minimize modal truncation effects related to the cut-off frequency. For solutions involving the new modal approach, approximately 80 residual modes were added to the normal modes to account for the influence of the viscoelastic material.

The first study involved the comparison between the classical direct method (SOL 108) and the new modal approach (SOL 111) using the frequency-dependent isotropic representation. The resulting responses are plotted in Figure 5 and indicate an excellent agreement between the two methods over the entire frequency range.

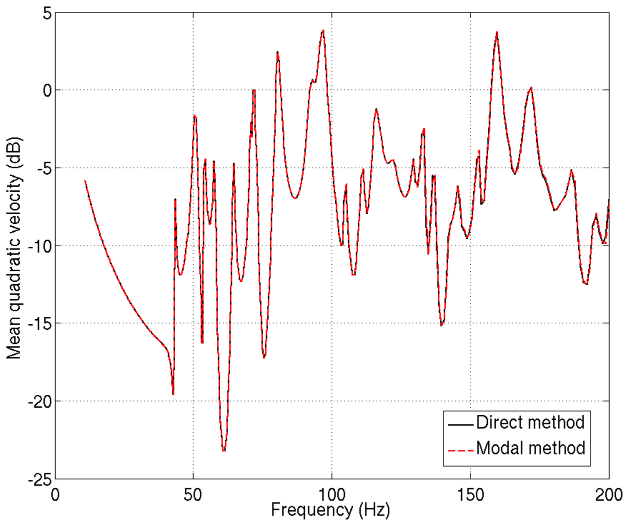

Fig. 5: Direct vs. Modal Responses

In terms of computational efficiency, the direct solution took 166 minutes whereas the modal approach required only 13 minutes including 9 minutes for computing the basis of normal and residual modes. This more than tenfold improvement in computation time can also be expected for larger models with a greater number of modes.

A second study was performed to evaluate the influence of the 80 residual modes added to the normal modes. Two response calculations were carried out using a constant isotropic representation of the viscoelastic material - one using the new modal method, and the second using the standard modal approach without residual modes. The two responses are plotted in Figure 6.

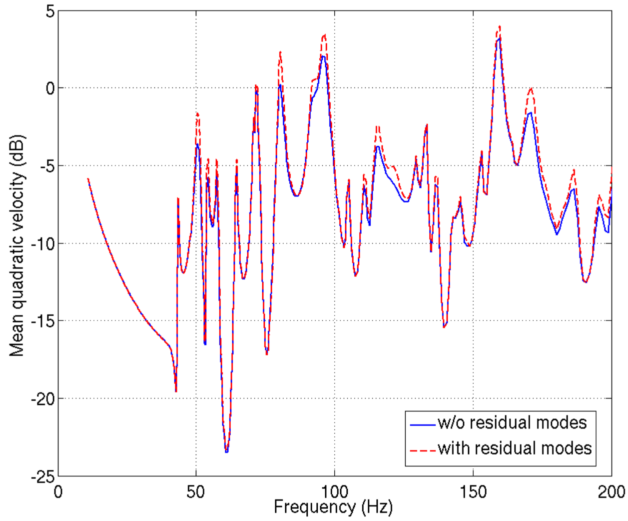

Fig. 6: Influence of Residual Modes

The influence of the residual modes is clearly visible by the large differences in amplitude at nearly all the resonant peaks. This illustrates the importance of the residual modes and their role in representing the strong and localized dissipative forces exerted by the viscoelastic material.

A third study was performed to assess the influence of the frequency-dependent properties of the viscoelastic material. The new modal approach was used to compute the frequency response using constant and frequency-dependent isotropic representations of the viscoelastic material. The resulting responses are plotted below in Figure 7.

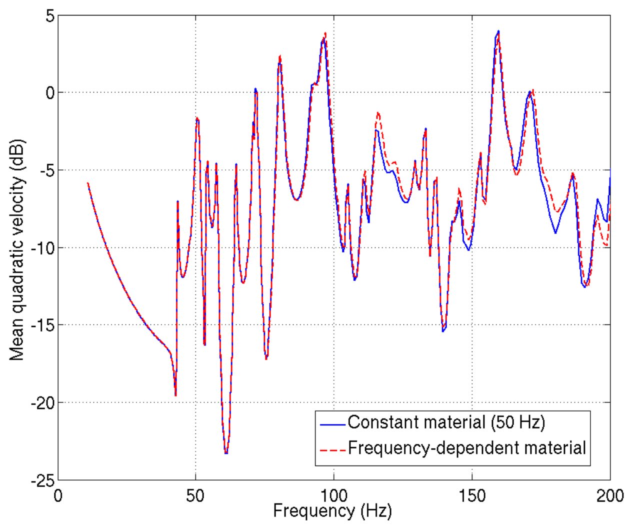

Fig. 7: Influence of Frequency-dependencies

Errors in both stiffness and damping are visible in Figure 7 as witnessed by shifts in the peaks' frequency and amplitude. Near the reference frequency of 50 Hz, the responses are similar -

whereas at higher frequencies the frequency-dependent properties are much more pronounced.

A final validation study was carried-out to investigate the influence of the anisotropic behavior of the viscoelastic material. A comparison was made using the frequency-dependent isotropic and anisotropic material representations via the modal approach. The corresponding frequency responses are plotted in Figure 8.

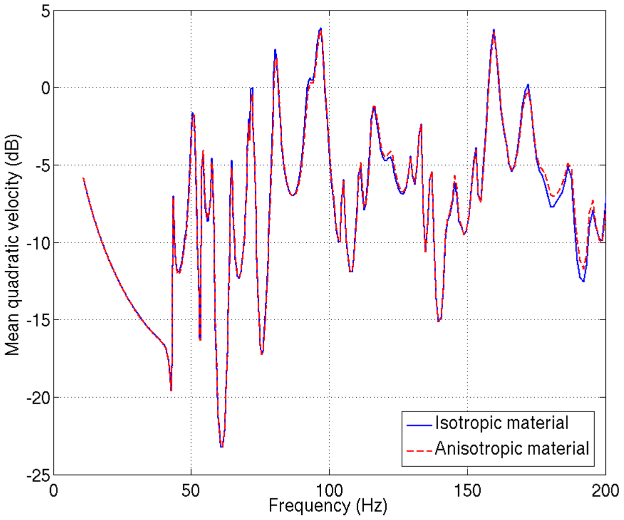

Fig. 8: Influence of Anisotropic Material

The influence of the anisotropic material is noticeable in the amplitudes of the peaks throughout the frequency range with differences up to approximately 1 dB. The differences are higher for the antiresonance at the end of the frequency range of interest. As the degree of anisotropy increases, its effect on the resulting responses will increase accordingly.

The new modal method comes at a small additional computational cost relative to the classical modal approach while offering the improved capabilities described above. As an illustration, consider the responses shown in Figure 9 and computed using the new modal method with anisotropic material compared with the classical modal approach using a constant isotropic material.

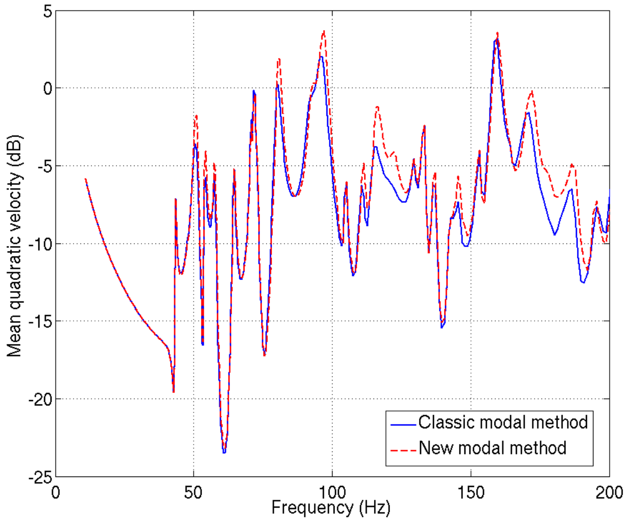

Fig. 9: New versus Classical Modal Approach

Due to the severe limitations of the classical modal approach (constant isotropic viscoelastic material and no residual mode compensation), the error in the response levels are significant - often exceeding 3 dB. In terms of computation time, the new approach required 13 minutes compared to 5 minutes for the classical approach and 166 minutes for the direct approach. Therefore the new modal approach provides the desired accuracy and full viscoelastic modeling capabilities while maintaining the numerical efficiency of a modal approach.

3.3 HDF Performance Study

The second part of the industrial application was aimed at evaluating the performance of the HDF dissipative foam as an alternative to the current production solution combining two bitumen pads and an antiflutter material.

Using the new modal approach two response calculations were performed up to 700 Hz - one with HDF represented as a frequency-dependent anisotropic material, and the second with frequency-dependent isotropic properties for the bitumen pads and constant isotropic properties for the antiflutter material. All products have been characterized using DMA equipment in order to use the measured data as inputs for the tables of the mechanical properties dependence. All have been tested in tension-compression. In addition, shear tests have been performed for the HDF material. It has been observed that the mechanical properties of the antiflutter material are close to be not frequency-dependent at room temperature. It explains why constant isotropic behavior can be used. Moreover, as it is well known that single layer bitumen pads are deformed in extension ([1]-chapter 6.3) it is not necessary to accurately model the shear effects. The results of the calculated responses are plotted in Figure 10.

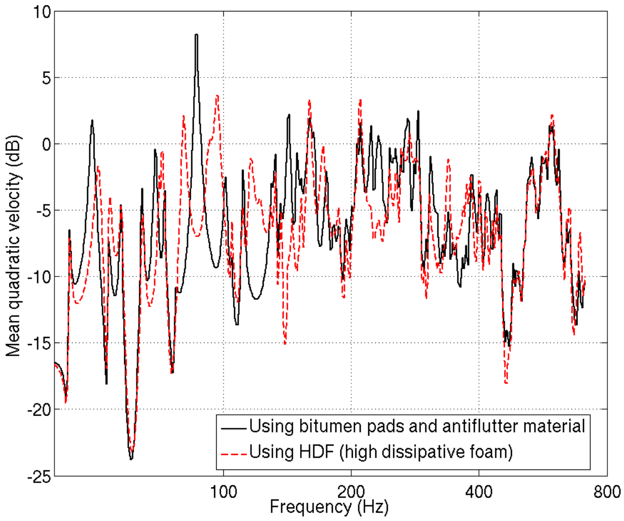

Fig. 10: Production vs. HDF Responses

The response with the production treatment shows a high peak just below 100 Hz and a second significant peak to the left at 50 Hz. Both of these peaks are effectively dissipated by up to 5 dB in the response with the HDF.

At higher frequencies, the peaks are shifted but the overall velocity level is comparable between the two responses.

Finally, these results demonstrate the ability of the HDF to replace the production treatment without any degradation of the vibratory level of the door panel in all the frequency range of interest. It leads to a weight saving of 175 g per door. In addition, the replacement of two products by a single product should decrease the total cost of the solution comprising:

- material costs
- manufacturing costs
- application costs in the door construction

Cost appraisals and acoustic validations must be done in partnership between HENKEL and PSA Peugeot Citroën in order to examine all the aspects of the proposed innovation.

4. Conclusion

An efficient method for frequency response calculations of structures with viscoelastic damping material has been developed and implemented using NASTRAN. The method uses an enriched modal approach combining normal modes and residual vectors in order to accurately account for the presence of the viscoelastic material.

Any number of viscoelastic materials may be represented using frequency-dependent isotropic or anisotropic properties to introduce separate tensile and shear complex moduli.

The method has been implemented in NASTRAN and produces the same results as the classical direct approach but more than 10 times faster.

Finally, a performance study was carried out using the method to demonstrate the efficiency of a highly dissipative damping treatment as a replacement for anti-vibration solutions used in mass production (damping pads and antiflutter material).

5. References

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6. Glossary

ACMS : Automated Component Mode Synthesis

AMLS : Automated Multi Level Substructuring

DMA : Dynamic Mechanical Analysis

DMAP : Direct Matrix Abstraction Programming

FE : Finite Element

FEM : Finite Element Model

HDF : High Dissipative Foam