Holistic Approach to Automotive Floor Design:

Considering Structural Construction, Beads, Damping Layers and Acoustic Trim Simultaneously to Improve Floor Design

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Abstract

In the automotive industry, it is common to have different departments designing different parts of the floor. The NVH department may define the overall structural stiffness and might request beading to be added to specific panels to reduce the vibration response locally. It may also request damping pads to be added on the remaining vibration hot spots of the floor. The acoustic group then needs to define the acoustic trim needed to meet the vehicle targets based on the constraints prescribed by the NVH department choices concerning structural stiffness, floor construction, beading, damping pads… In many cases, the NVH and acoustic groups are not communicating and some solutions proposed by one group are detrimental to the other group. For example, it has been shown that contrary to popular belief, adding beads to a structure can actually reduce noise at the first few modes of a plate while significantly increase noise radiation at higher frequency.

This paper presents an investigation of how the structure, the beading, the damping pads and acoustic trim can be integrated into a holistic design process to evaluate the effect of all these components on the floor vibration and sound pressure level (SPL) inside a vehicle. In this study only a floor panel is studied to isolate phenomena. A vehicle cavity with seats and appropriate damping is used to represent the interior of the vehicle. The floor structure, beading, damping pads, acoustic trim and acoustic cavity are modeled using finite elements (FEM). Biot parameters are used to represent the poro-elastic layer physical properties. Different configurations including classical steel and innovative composite laminated panels are compared and associated with different types of beading, damping treatments and acoustic trim to evaluate the effect of the full floor component on the vibration response of the floor and most importantly the SPL at the driver's ear.

Introduction

The work presented here is an extension of previous studies presented in [1,2,3] where the effect of beading on bare steel panels for an academic case and a real automobile floor panel was discussed. The main conclusion form this work is that stiffening a panel with beads is efficient at breaking the first few modes of a plate but might significantly increase the panel sound power radiated. It was also shown that the acoustic power radiated from a plate is directly proportional to the product of average panel velocity and radiation efficiency. Radiation efficiency is defined as "the acoustic power radiated by the plate into a half space, divided by the acoustic power that an infinite piston (all parts vibrating in phase) would radiate into the same half space if it were vibrating with the same RMS velocity as the plate" [3]. The radiation efficiency can be viewed as the ability of a panel to radiate noise (see equation 1).

$$\prod_{rad} = \sigma A \rho_0 C v_{rms}^2 \qquad (1)$$

where σ is radiation efficiency, Πrad is the power radiated by the panel, A is the radiating area of the panel, ρ_0C is the characteristic impedance of air and V^2rms is the average rms velocity of the panel.

It was also demonstrated that the so-called ERP (Equivalent Radiated Power) method is not able to properly predict the acoustics radiation of a panel and might lead to erroneous design decisions. Other acoustic methods such as FE acoustic, BEM (Boundary Element Method) or SEA (Statistical Energy Analysis) must be used to properly capture the physics behind panel radiation.

This paper discusses the different floor component designs used in the automotive industry. It compares the different construction types such as steel, MPM and CFRP, the effect of adding beads, the effect of adding damping patches (Alu-Butyl) and finally the effect of adding carpet to the base construction. It also discusses the conflicting objectives that structural and vibro-acoustics engineers are faced with. The paper finally suggests approaches to design more efficient floor components in terms of vibro-acoustics performance, weight and cost.

Description of Modeling Methods

This section describes the different modelling methods used to generate the results. Finite Element Methods (FEM) is used to model the floor structure and the acoustic volume. FEM is also used to represent the carpet using Biot parameters for the porous layer. A new modelling method is used to derive the equivalent properties of laminates such as Metal-Polymer-Metal (MPM). Finally, the effect of Alu-Butyl patches on steel and Carbon-Fiber-Reinforced Plastics (CFRP) floor is modelled using FEM and PCOMP cards.

Poro-Elastic Material (PEM)

The propagation of elastic and acoustic harmonic waves, with an e-iωt time dependency in porous elastic media is governed by the modified Biot's equations. Refer to [4,5,6] for a complete theoretical derivations of these equations. As described in [7], the FEM trim of a full vehicle analysis can be added to the classical FEM structure/fluid coupled linear system as a trim impedance matrix. The dynamic equation of the trimmed vehicle can be written in the following form:

$$\left[\begin{pmatrix} Z_s & C_{sc} \\ C_{sc}^t & A_c \end{pmatrix} + \begin{pmatrix} \tilde{Y}_{ss} & \tilde{Y}_{sc} \\ \tilde{Y}_{sc}^t & \tilde{Y}_{cc} \end{pmatrix}\right] \begin{bmatrix} U \\ P \end{bmatrix} = \begin{bmatrix} F \\ Q \end{bmatrix} \tag{2}$$

Where Zs is the mechanical impedance of the master structure (car body in white), Ac is the acoustic admittance of the internal cavity. Csc is the surface coupling operator between the untrimmed master-structure surfaces directly in contact with the internal acoustic cavity. U is the displacement field vector of the master-structure, P the pressure field of the internal cavity; F the external force field applied to the master-structure, and Q represents internal acoustic sources. The matrix Y R'YR is the transferred impedance matrix of the porous component where R is the transfer operator relating the degrees of freedom of the porous component to the degrees of freedom of the master structure and of the internal cavity. Linear system in equation (2) is solved using structural and acoustic normal modes. This has the advantage of keeping the linear system to be solved for the trimmed body the same size as the initial BIW linear system (No additional DOF). This method has been extensively validated in [8]

Sandwich structure (MPM)

In this section, a modeling approach used for sandwich panel represented as a thin layer of viscoelastic material sandwiched between two elastic face layers (Metal/Polymer/Metal). This kind of structures has appeared recently as a viable alternative to classical add-on or spray-on treatments. These materials are widely used to reduce noise and vibration and to improve interior sound quality. It has been shown that this class of materials enables manufacturers to cut weight and cost while providing noise, vibration and harshness performance [9, 10]. Detailed states of the art have been presented by Nashif et al. [11], Sun and Lu [12], Beranek [13], Allen [14] and Vincent et al. [15]. Although initially confined to the aerospace field, sandwich structures are now applied in almost all industrial fields. This motivated the development of prediction methods for their vibration and acoustic performance. Existing methods based on finite elements necessitate the use of plate-solid-plate models which are computationally expensive. Recently, a new sandwich finite element for laminated steels has been applied successfully to the vibro-acoustic analysis of damped laminated steels [16]. It concerns the development of two efficient finite element sandwich plates: refined rectangular and triangular elements having four- and three-corner nodes, respectively. For more details, see reference [17]. Comparisons of the presented element versus both experimental and classical FE modeling prove its accuracy and computational efficiency for the modeling of the vibro-acoustic behaviour of the studied laminated steels. The element is based on the following assumptions: (i) the cross-section of each layer remains plane after deformation, (ii) the core contributes only by transversal shear stresses, (iii) transversal shear stresses are neglected in the skins, but the rotational influence of the transversal shearing in the core is accounted for and (iv) the core and the skins are assumed incompressible throughout the thickness. The latter assumption prohibits the use of the element for thick core sandwich panels however the latter is not the case of MPM panels. These elements are used here to calculate the equivalent properties of the sandwich MPM panel.

Carbon-Fiber-Reinforced Plastic (CFRP)

Another composite material used in this study is a carbon-fiber-reinforced plastic. This material is composed of plies. Each ply consists of fibers within a polymer matrix with addition of additives. It is a very strong and light fiber-reinforced polymer which contains carbon fibers. The composite may contain other fibers such as Kevlar and aluminum. It has many applications in aerospace and automotive fields, where its high strength-to-weight ratio and good rigidity is of importance. In this study, the PCOMP property card of Nastran is used to model these materials. A typical automotive CFRP is used and associated physical properties are presented in table 2.

Description of models used

The floor panel is modelled in FE (Finite Elements) and the panel is connected to a FE cavity representing the car geometry (see Figure 1c). The boundary condition used for the floor panel is free-free.

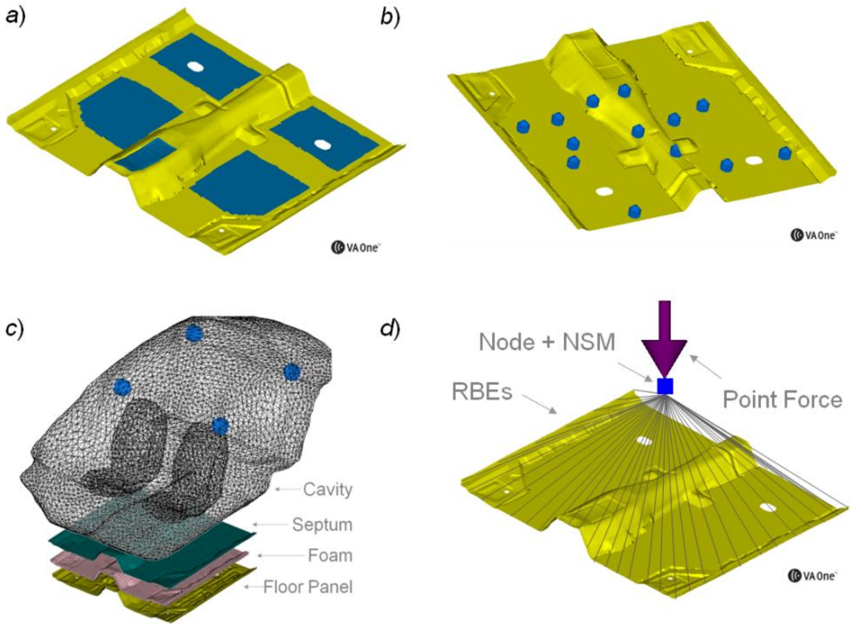

Figure 1: Description of model: a) Floor structure (yellow) and damping patches (blue), b) Accelerometers for average acceleration computation, c) Structure, carpet, cavity and interior microphones, d) Excitation at edge of floor panel using large mass method.

The structural FE model was built using 6 shell elements per wavelength and is valid up to 800 Hz. Normal modes are computed to 1040Hz to avoid truncation of modes. The structure used is 0.8mm steel with 4% structural damping. The large mass method is used to constrain acceleration at the edge nodes of the floor panel.

Material type	Physical Properties									
	Thick. (mm)	ρ (kg/m^{3})	E (Pa)	ν (-)	DLF (%)	σ (N.s/m^{4})	Φ (-)	α_∞	Λ (10^{-6}m)	Λ’ (10^{-6}m)
Steel	0.8	7820	2e11	0.29	4	-	-	-	-	-
Foam	20.0	63.8	4.2e4	0.30	17	1.9e4	0.93	1.7	40	120
Septum	2.0	2500	-	-	-	-	-	-	-	-
Butyl	1.6	1945	3.83e9	0.35	Table 1	-	-	-	-	-
Aluminium	0.2	2700	7.1e10	0.33		-	-	-	-	-
MPM-Steel	0.38	7850	2.05e11	0.30	1	-	-	-	-	-
MPM-Visco	0.04	1010	2.e6	0.49		-	-	-	-	-

Table 1: Properties of steel and MPM base panel, Al-Butyl damping patches and carpet

Thickness (mm)	ρ (kg/m3)	E1 (Pa)	E2 (Pa)	ν (-)	G12 (Pa)	G13 (Pa)	G23 (Pa)	Orientation	Number of Layers
2.0	1500	1.1e11	9.e9	0.3	4.e9	4.e9	4.e9	+45/-45/0/0/0 as mirror-	10

Table 2 CFRP typical material properties

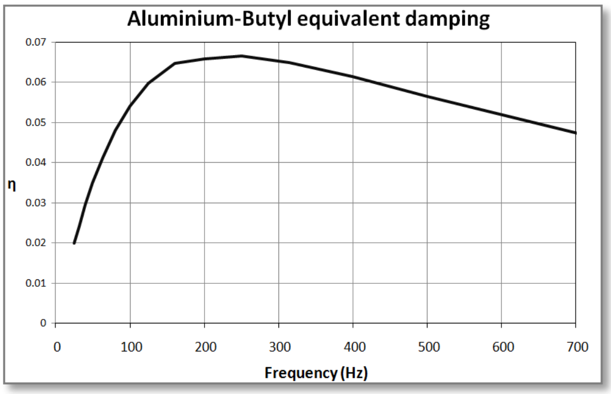

Figure 2: Aluminium-Butyl damping patches equivalent damping spectrum

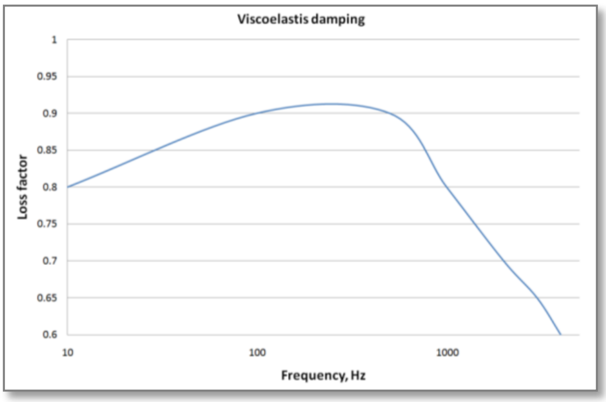

Figure 3: Structural damping loss factor of MPM viscoelastic layer

A mass of one metric ton is excited by a force of 10N in each 1 Hz bandwidth. The mass is connected to the floor panel edge nodes using RBEs. An acoustic damping representative of a production vehicle is used in this FE cavity and kept constant for all iterations. This acoustic damping is set to 7%. The coupled response has been computed every 2 Hz. Table 1 and Table 2 describe the material used in this study.

Analysis of results

Basic results presented in this work are the average velocity on the floor structure, the radiation efficiency of floor structure, the sound power radiated by the floor structure inside the acoustic cavity and finally the sound pressure level (SPL) at the outer driver’s ear.

Effect of construction type

Figure 4 and Figure 5 shows average acceleration and driver’s outer ear SPL for various typical automotive floor panel constructions such as steel, beaded steel, MPM and CFRP panels.

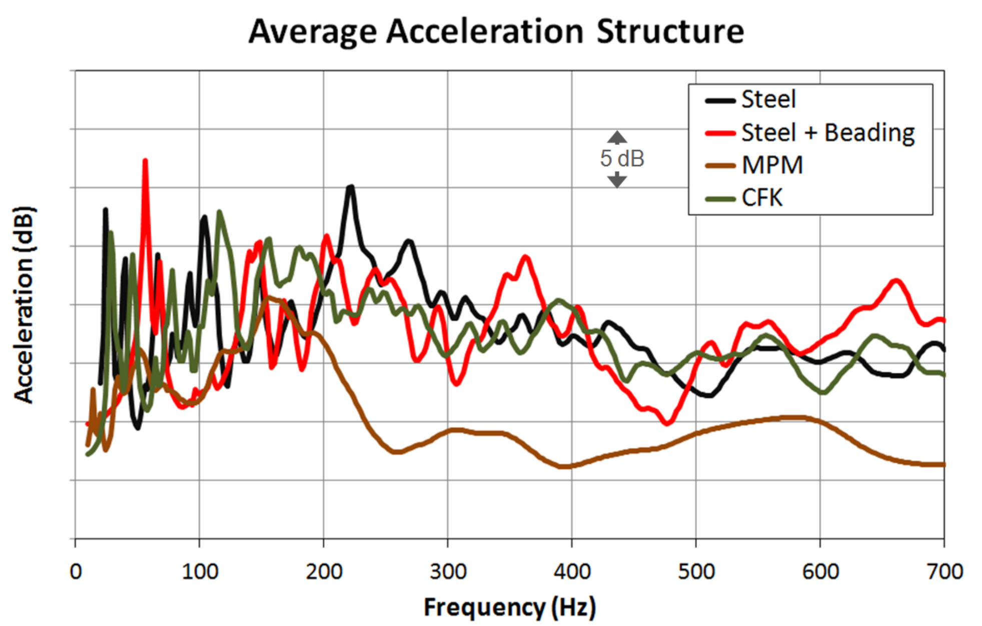

Figure 4: Effect of various panel constructions on average acceleration

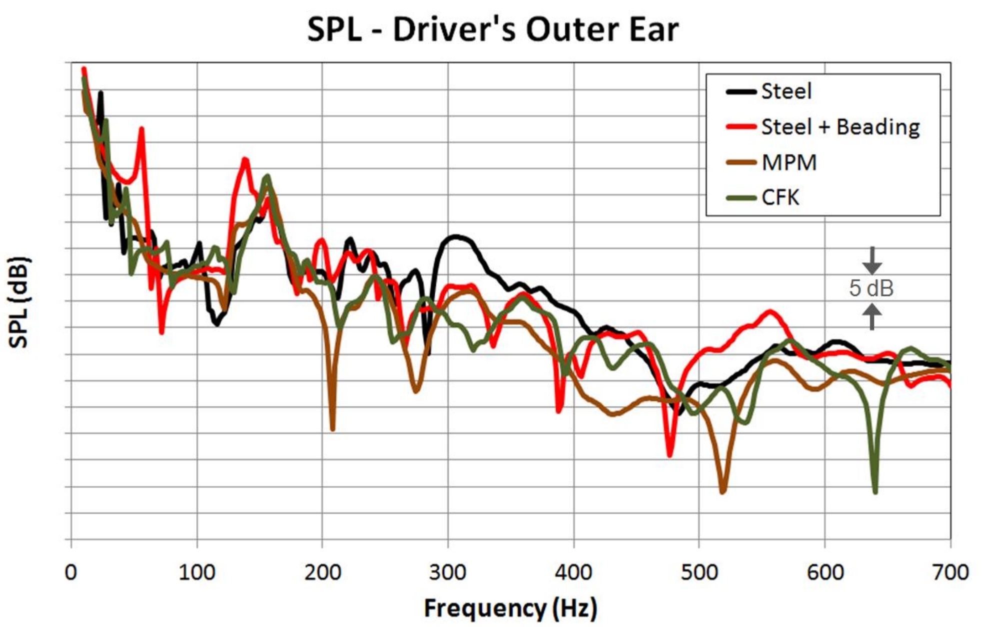

Figure 5: Effect of various panel constructions on driver's ear SPL

The results show typical effect of the floor construction on vibration and SPL. As expected MPM with its high damping yields very low vibration levels although not all this gain in vibration translate into a SPL gain. The CFRP used is very similar to the performance of steel sheets or beaded steel. These will serve as the basis for the rest of the paper.

Effect of beading

The effect of beading have been presented in earlier work [2,3] and reproduced in Figure 6. The result graphs have been placed on top of each other for better visualization of the results. On the average velocity graph (top), the benefits of the beads can be seen in the frequency range between 0 and 120 Hz except for a single peak at 58 Hz. In the rest of the frequency domain, the average velocity is slightly decreased except at some frequencies that have been circled in green. In the radiation efficiency graph (middle), most of the beaded curve lies above the non-beaded panel. The increase in bending stiffness contributes to the increase in radiation efficiency. The green circles are located at the same frequencies as for the average velocity graph. These circles indicate where the increase in velocity matches an increase in radiation efficiency. At these frequencies, the sound power radiated should also increase for the beaded case. The sound power radiated graph (bottom) confirms that for the beaded case, the sound radiated by the floor is higher at the frequencies where green circles can be found. The beading reduces the sound radiated power mostly at low frequency around the first few panel modes. It is interesting to note that for the beaded floor the highest level of sound power is 68dB as opposed to only 61dB for the non-beaded case. The beaded floor radiates more than the non-beaded floor by more than 3 dB for the frequency range of 10 to 800 Hz. Larger effect are to be expected on a wider frequency domain.

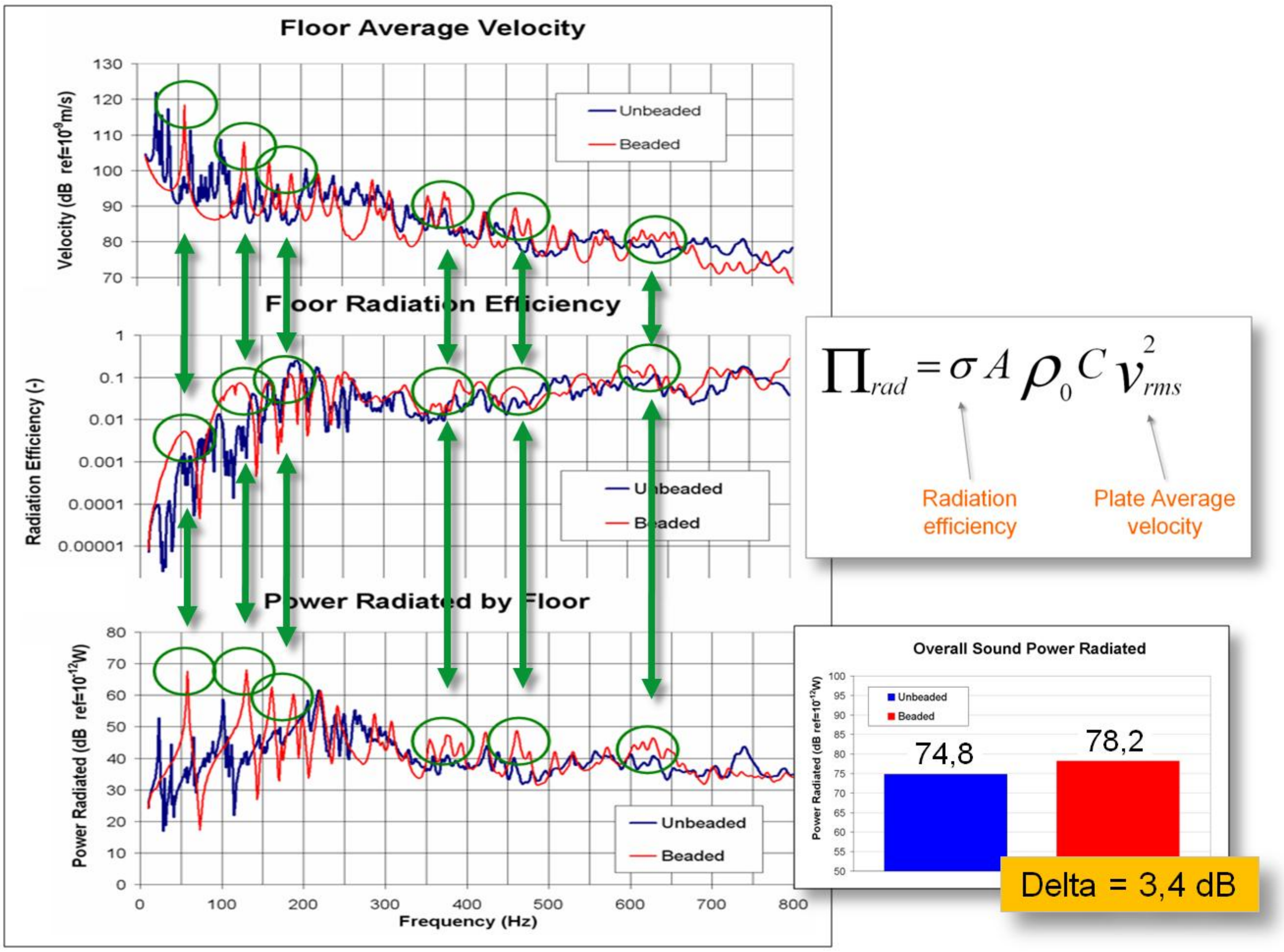

Figure 6: Effect of beading on velocity, radiation efficiency and sound power radiated by the floor

Effect of local damping patches

Another technique in the industry to reduce vibration of panels is to glue damping patches at strategically chosen locations in order to damp modes. These patches will add mass, damping and stiffness to the panel. In this study, the influence of damping patches on the unbeaded floor is studied. Patches are located on the large flat area of the floor to modify the vibration of this region and on the tunnel where a contributing mode has been detected. (See **Fehler! Verweisquelle konnte nicht gefunden werden.** for the exact damping patches location). Damping patches are glued on the panel and made of 1.6mm of butyl visco-elastic material and 0.2mm aluminium sheet which is typical for the automotive industry.

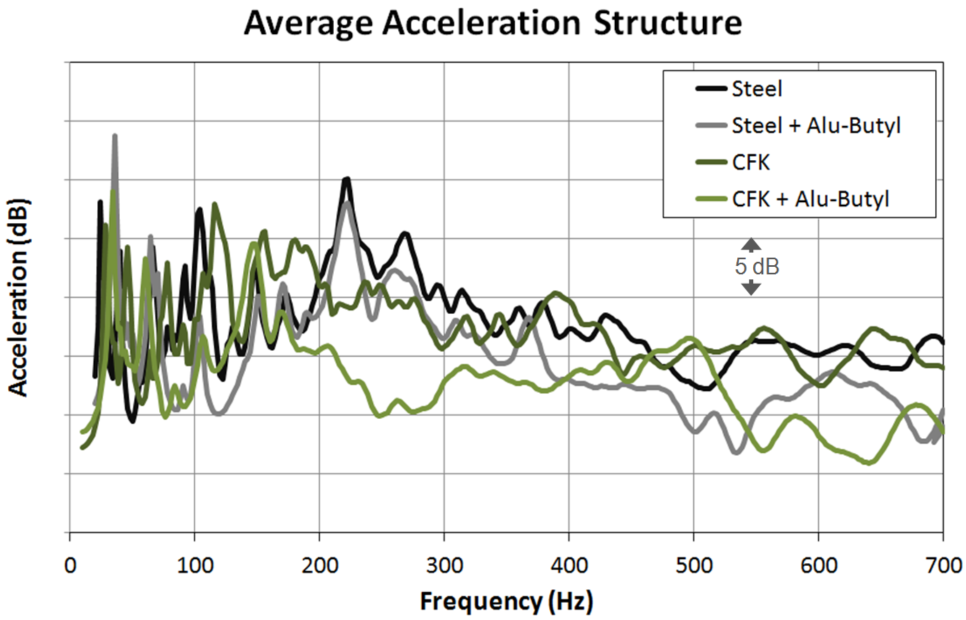

Figure 7: Effect of damping patches on car panels average acceleration

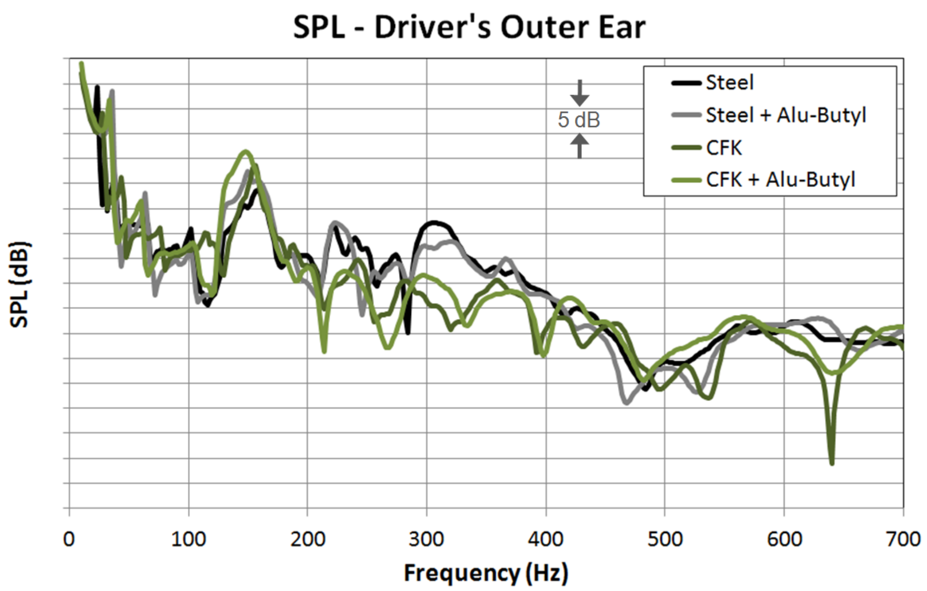

Figure 8: Effect of damping patches on car floor panels' driver's ear SPL

Figure 7 shows the effect of adding damping patches on an unbeaded floor steel or CFRP panels (CFK = CFRP). Adding the damping patches has a clear effect on the first mode of the panel. The panel becomes stiffer and the first mode is shifted by around 10Hz higher in frequency for steel. Therefore, the amplitude of this mode is not lowered. The same kind of behavior can be seen for the next modes. However, for these modes, damping has a really strong effect. At higher frequencies, effect of the damping is more important and leads to a reduction of about 5dB or more for the amplitude of the acceleration.

Figure 8 illustrates a similar comparison for the driver's ear SPL. First peak shift is also observed in the SPL graph. In that case, adding damping patches leads to a diminution of this peak amplitude. For the next few modes, double effect of stiffness and damping can be seen which induces a lower SPL. Then, SPL for the two configurations becomes similar and it becomes difficult to see a benefit in adding damping patches on the car floor panel. The patches influence only locally the SPL.

Effect of carpet

This section presents the effect of adding a carpet on the different floor panel constructions possibilities studied previously.

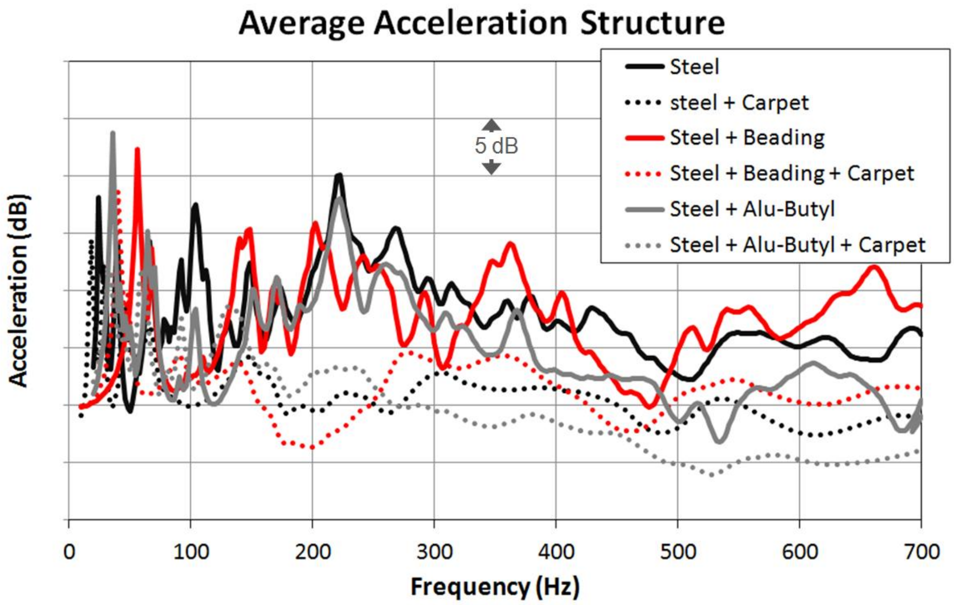

Figure 9: Effect of carpet on various panel construction average acceleration

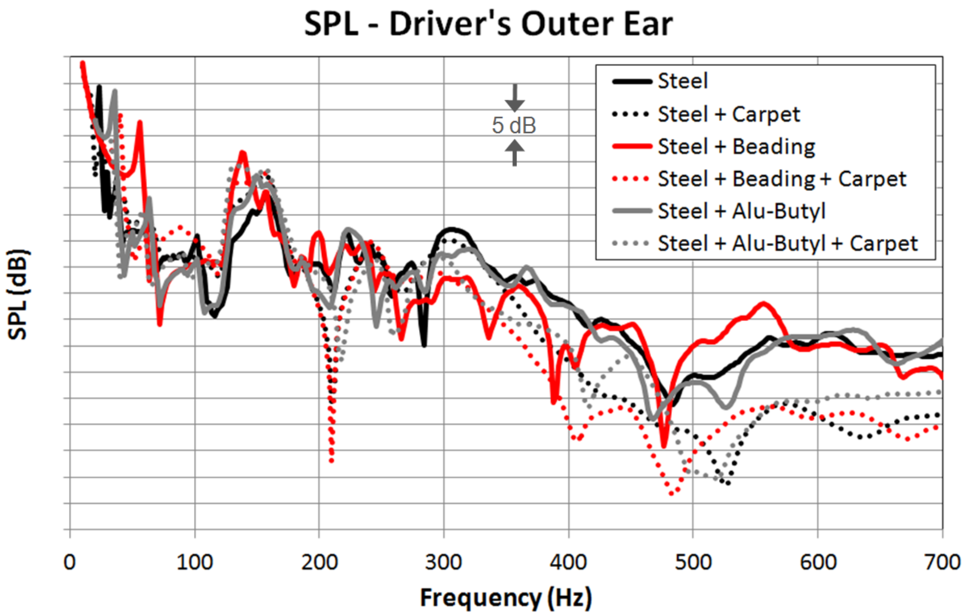

Figure 10: Effect of carpet on various panel construction driver's ear SPL

This carpet is made of 20mm of foam and a 2mm septum (see **Fehler! Verweisquelle konnte nicht gefunden werden.**). Usually, the carpet thickness is varying to adapt itself to the complex geometry of the floor panel and the equipment mounted on it (AC ducts, electric boxes…).

Figure 9 shows the effect of adding a carpet on flat, beaded or flat with damping patches floor which are configurations massively used in the automotive industry. A shift on the first mode can be identified for each configuration. This shift is induced by the added mass created by the carpet. Then the carpet has a clear damping effect which at the lowest frequency cuts down the amplitude of the modes and then generates a diminution of 5 to 10dB.

Figure 10 shows the effect of adding a carpet for the same classical configurations on SPL. Same remarks about the first modes can be made for the SPL results as for the acceleration results. In frequencies higher than 200Hz, the carpet adds an insertion loss on the path and generates a drop of about 10dB for each configuration.

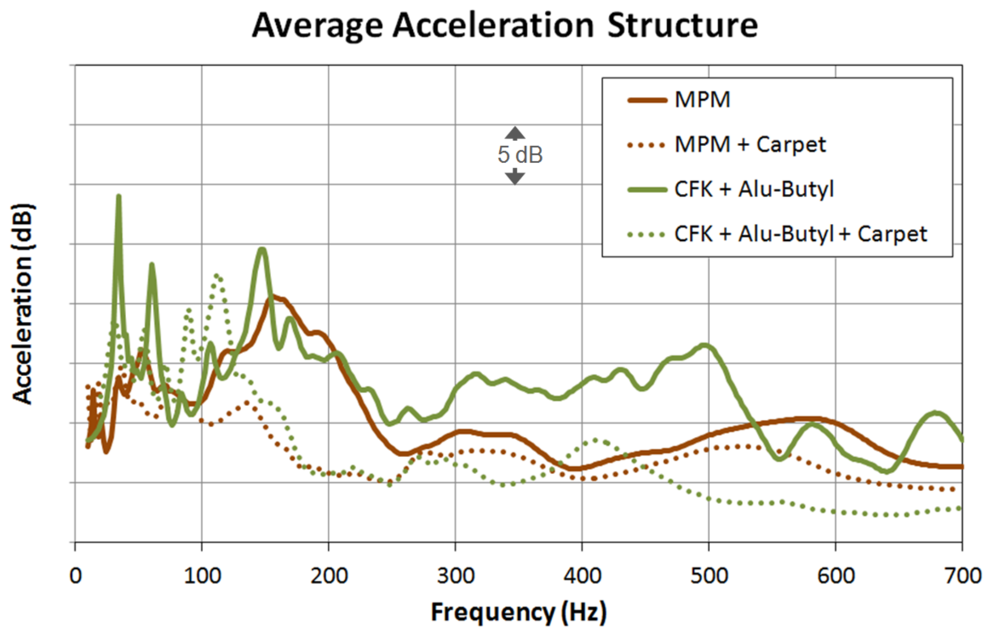

Figure 11: Effect on average floor acceleration of carpet on MPM and CFRP

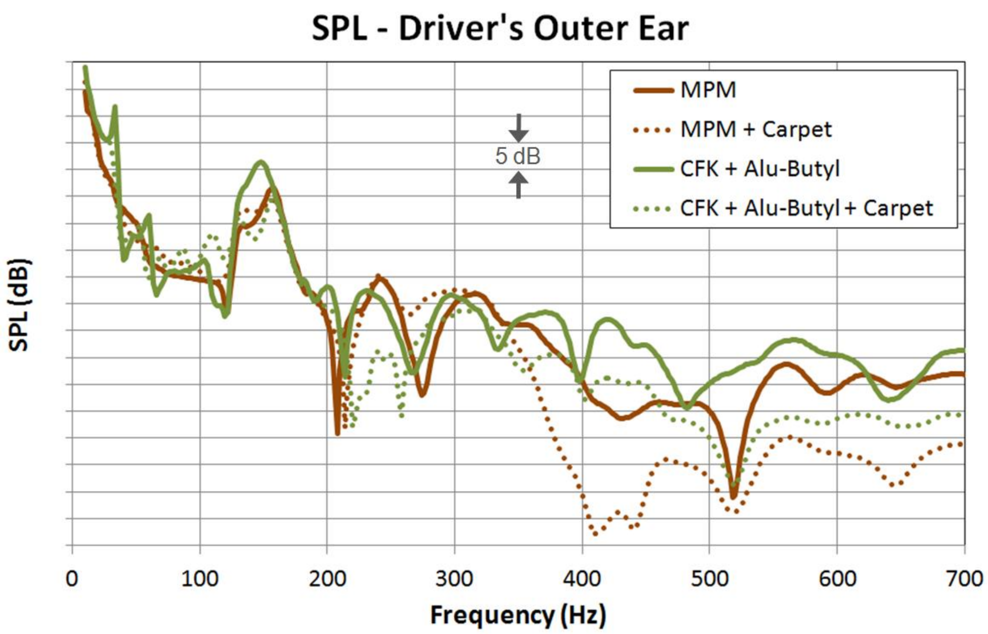

Figure 12: Effect on SPL at driver's ear of carpet on MPM and CFRP

Figures 11 and 12 show the effect of adding a carpet on an innovative floor construction. For these configurations, adding a carpet generates the same effect as on the classical configurations (damping added and insertion loss on SPL)

Conclusion

The previous results show that one can model with todays techniques the effect of adding beads, damping layer, MPM and CFRP along with the addition of carpet using FEM. The effect of combining the different solution often yields results that are difficult to guess beforehand. It is important to note that focusing on lowering down vibration levels is not necessarily going to provide proportional gains on SPL. An approach that focuses on perceived SPL by the automobile occupants and an holistic design process that minimizes conflicting solutions would yield floor component design that are more effective, cheaper and lighter.

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