Combined Effect of Beads and Carpet on Structureborne Sound Radiated from an Automobile Floor

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Abstract

This paper discusses the effect of adding beads to a floor panel in respect to radiated sound inside a vehicle. It also describes the combined effect of using beads and a carpet to meet a predefined SPL (Sound Pressure Level) target at the driver's ear location. It has been widely demonstrated in the literature that adding beads to a floor panel reduces vibration levels at low frequency by shifting the first few modes to a higher frequency. The gain can be significant especially to reduce the peaks at the first few modes of the unbeaded panel. In many cases, design decisions are taken based on vibration levels in conjunction with the so-called ERP (Equivalent Radiated Power) assumption that assumes a direct relation between radiated power and vibration levels. In fact, using ERP assumes that radiation efficiency is equal to 1 over the whole frequency domain. This is not the case in reality and care must be taken when designing beads to ensure that the radiated sound inside the car is not greater due to increased panel stiffness, increased radiating area and increased radiating edges provided by the beads.

This papers presents a case study where beaded and unbeaded floor panel vibration, radiation efficiency and radiated sound power into an automobile interior cavity are compared. It also describes the impact of such beadings on the carpet design in order to meet SPL targets

Introduction

The work presented here is an extension of a previous study presented in [1] (also published in [2]) 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).

$$\Pi_{rad} = \sigma A \rho_0 C V_{rms}^2 \tag{1}$$

where σ is radiation efficiency, Πrad is the power radiated by the panel, A is the radiating area of the panel, ρC 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 presents the combined effect of panel beading and adding a floor carpet on the sheet metal. A short summary of the previous paper is provided since it is the basis of the present work. In addition, a new light beading pattern is also studied and compared to the previous study. Then follows a short introduction of the modelling method used to represent the carpet. The effect of the carpet on the unbeaded and beaded panel is then illustrated and discussed. Finally, modifications to the carpet and floor structural damping are presented and compared with previous results. A final summary indicates the important aspects of the problems and what should be considered to properly design the floor as a holistic entity that includes the sheet panel, the beading, the damping treatments and carpet.

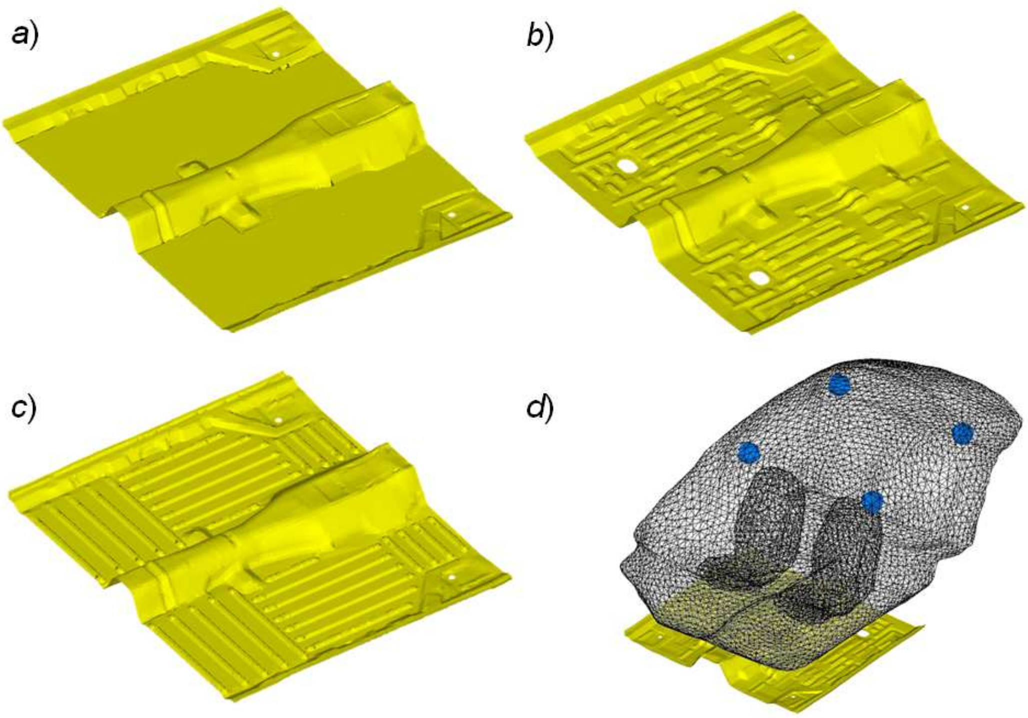

Figure 1: FE Steel floor panel a) unbeaded, b) lightly beaded and c) heavily beaded. d) FE floor panel and FE cavity with position of microphones.

Beading effect on bare floor panel

This section is a summary of a previous work presented in [1,2]. See this reference paper for more detailed information on how to assess the acoustic effect of beading and its relation to bending stiffness. In this example an unbeaded floor panel is compared with a beaded one (see Figure 1: *a* & *c*). A heavily beaded panel is use to better illustrate the effect of beading on noise radiation.

The floor panel is modelled in FE (Finite Elements) and the panel is connected to a FE cavity representing the car geometry (see Figure 1d). The boundary condition used for the floor panel is free-free. The structural FE model was built using 6 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. 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.

The results are shown in Figure 2. 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.

The previous case involved a heavy beaded panel. The same approach was applied to a lightly beaded panel to ensure previous conclusions are still valid (see Figure 1b).

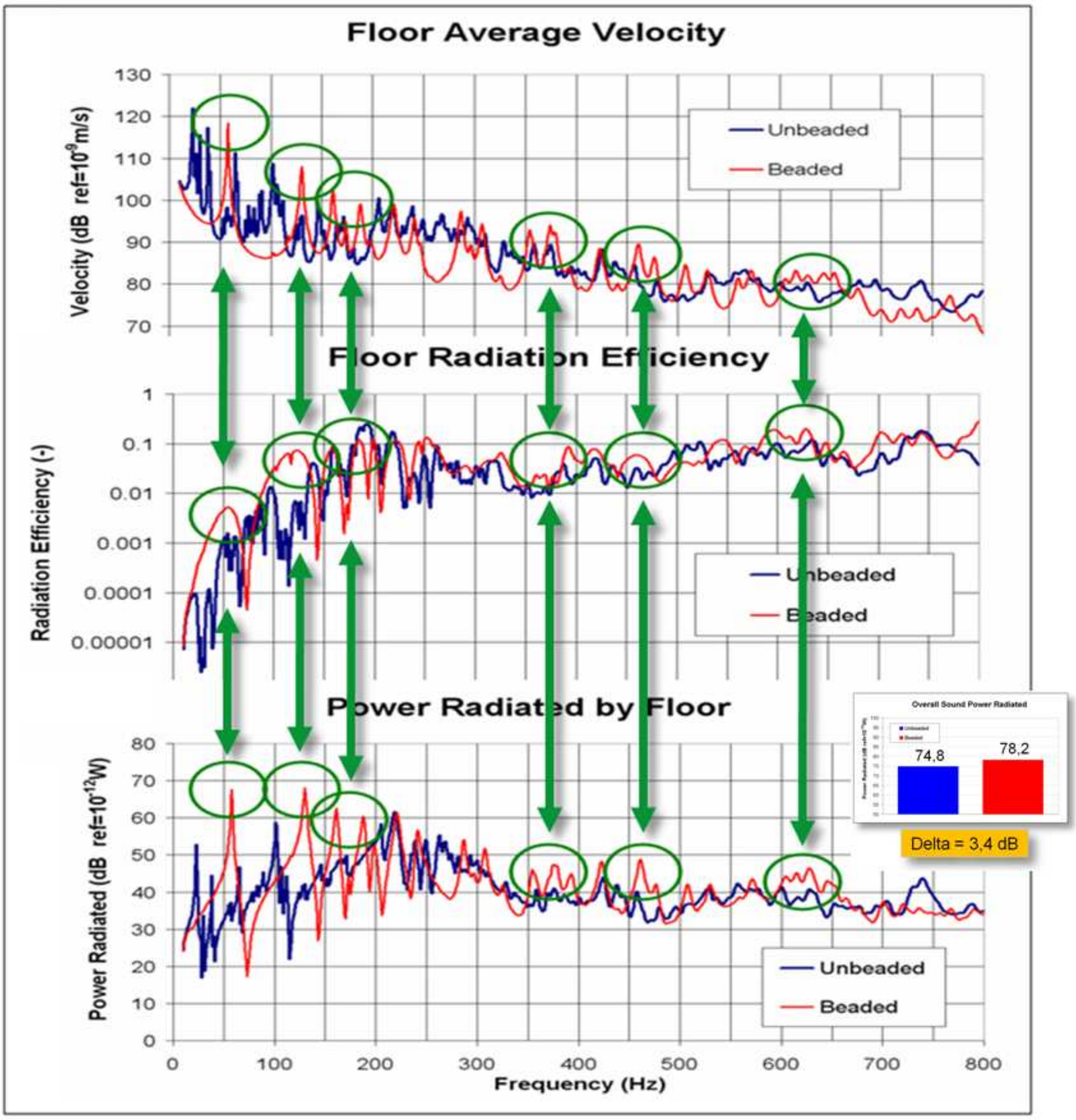

Figure 2: Floor vibro-acoustic results. Circles indicate where both velocity and radiation efficiency peaks coincide to generate higher sound power radiation

Figure 3 compares average acceleration for unbeaded, light beading and heavy beading. Both beaded panels display the same behaviour at low frequency shifting the peaks of the first few modes towards higher frequency. The overall tendency at higher frequency shows that the heavy beaded panel yields lower vibration levels. If one would consider only lower vibration levels or a shift to higher frequency of the first few modes as a design criteria, the heavy beading would be chosen. On the other hand, if one also considers the acoustic radiation of such panels, one would conclude differently.

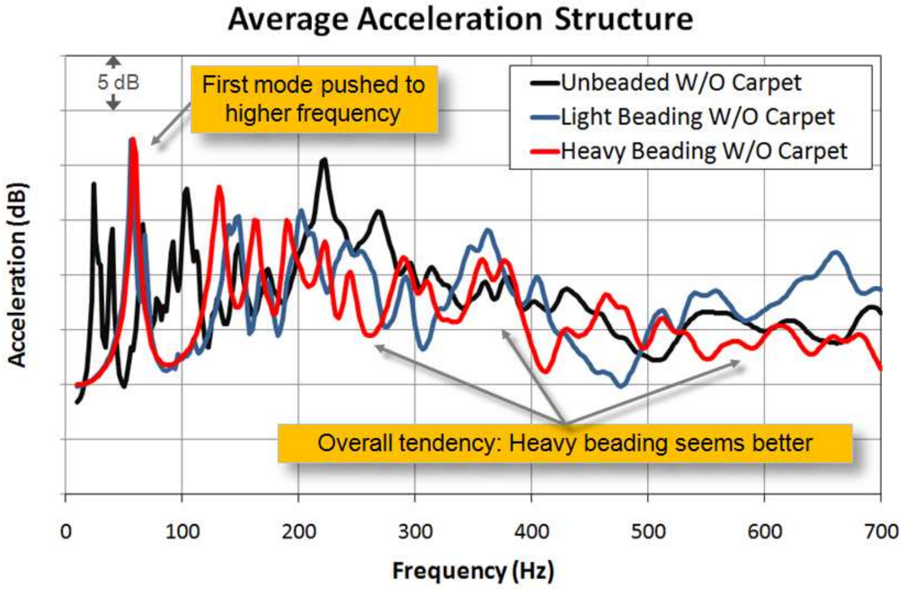

Figure 3: Comparison of average acceleration for unbeaded, lightly beaded and heavily beaded panel

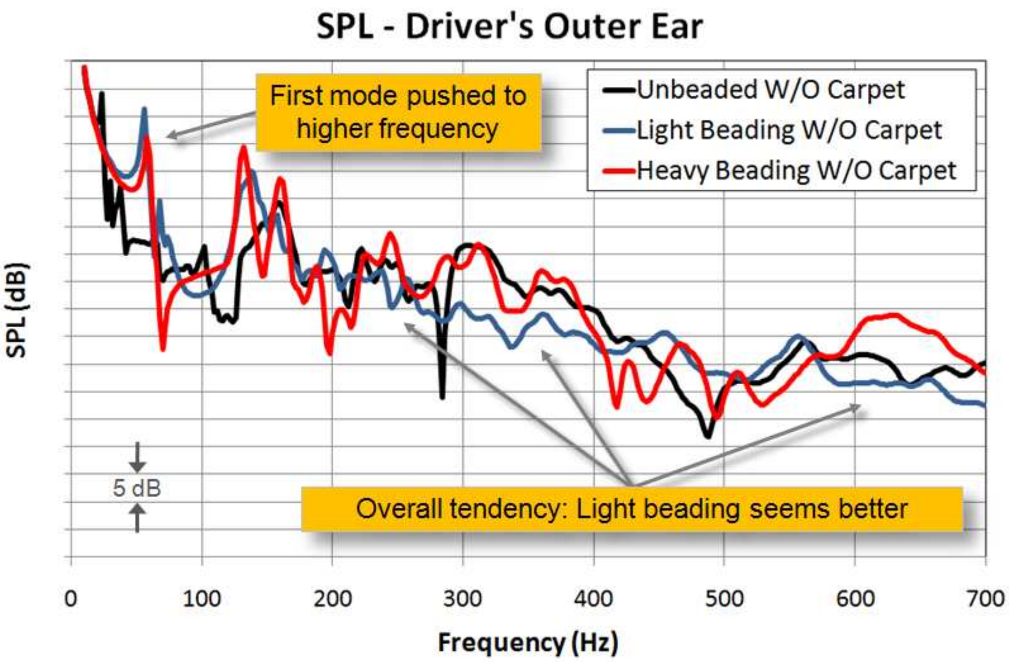

Figure 4: Comparison of SPL at driver's ear for unbeaded, light beading and heavy beading of a floor panel

Figure 4 shows a similar comparison for the SPL at the driver's ear. The first peak shift is also observed in the SPL graph and at that frequency the light beading panel generates 5dB more that the heavy beading and around 20 dB more than the unbeaded panel. The overall tendency at higher frequency suggests that the light beading would yield a lower SPL at the driver's ear. One can conclude that beading a panel pushes the first few modes of the structure to higher frequency and that these new peaks can be higher in amplitude than the unbeaded case. The beaded panels have different effect depending if acceleration or SPL is used as a design criteria.

Adding carpet to the floor panel

Biot Theory Approach

In the past ten years, the use of Biot parameters to represent acoustic trims made with porous material has been steadily increasing in all major industry sectors such as automotive, aerospace, train and marine industry [4,5,6,7,8,9,10]. This progress has been greatly due to the fact that affordable techniques to characterize porous material have been developed. Direct measurement of all five Biot parameters is possible and indirect methods have even simplified testing to a simple impedance tube (Kunt's tube) measurement [11,16]. The five Biot parameters are flow resistivity, porosity, tortuosity, viscous and thermal characteristic length. For a foam material, the structure of the foam is also accounted for using the following properties: Density, Young's modulus, Poisson's ratio and structural damping. An elastic porous material is made up of a solid skeleton portion, or frame, and a fluid portion. The energy transfer of three different wave types within the material controls the vibro-acoustic performance of elastic porous materials. These three types of waves are composite waves in that each wave type is present in both the frame and the fluid. Of these composite wave types, one compression wave and one shear wave have properties predominately influenced by the frame properties. The properties of the other compression wave are strongly influenced by the fluid properties. The behaviour of these waves is defined by the physical properties of the fluid and bulk properties of the elastic porous material. The elastic porous model is used for foam materials where the stiffness of the frame is important in vibro-acoustic response of the noise control material. The energy exchange between structural energy and acoustical energy within a foam material typically provides much of the desired energy absorption. The full elastic porous model requires all the fluid properties and the elastic bulk properties. For full details of the theoretical development refer to the documentation provided in [12]. The set of Biot parameters represent the intrinsic properties of the porous material and can therefore be used in any predictive models using either a transfer matrix method (TMM) or a finite element representation of the poro-elastic material (PEM).

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 [13,14,15] for a complete theoretical derivations of these equations. As described in [5], 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 Z_s is the mechanical impedance of the master structure (car body in white), A_c is the acoustic admittance of the internal cavity. C_{sc} 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).

Trimmed floor study

Designing floor beading in isolation, without accounting for the carpet that will be placed onto the floor panel might be detrimental to the design of the carpet itself and can actually require a higher level of vibro-acoustic performance from the carpet. This section presents results for unbeaded and beaded steel panels with typical floor carpet often used in the automotive industry. For this study, the carpet chosen is a spring/mass system made of one layer of foam and a mass layer. The foam is on average 20mm thick and the mass layer (septum) 2mm thick. (See Table 1 for detailed properties). The foam thickness layer varies to adapt to the structure underneath and a detailed FE mesh is created to represent this complex geometry (see Figure 5). The carpet coupling to the structure is sliding over the floor surface area and only attached at edge nodes to represent real life installation. Modelling guidelines and examples on how to create FE meshes for acoustic trim such as carpets, headliners, seats, dash insulators, package tray, trunk floor and sides can be found in [17,18].

Table 1: Material and physical properties of base panel and floor carpet

	Thick (mm)	E (Pa)	ρ (kg/m^3)	ν (-)	η (%)	Φ (-)	σ (N.s/m^4)	α_∞ (-)	Λ (10^{-6}m)	Λ’ (10^{-6}m)
Steel Floor	0.8	2E11	7820	0.29	4.0	--	--	--	--	--
Foam	20	4.2E4	63.8	0.3	17.0	0.93	1.9E4	1.7	40	120
Septum	2	--	2500	--	--	--	--	--	--	--

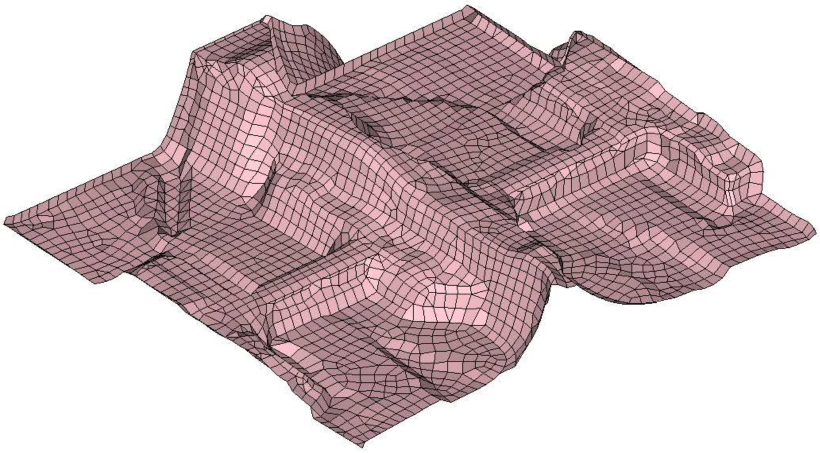

Figure 5: Floor carpet seen from below. Notice thickness variation and complex geometry.

Adding carpet to floor panel

First results presented in Figure 6 and Figure 7 show that adding carpet to the unbeaded floor panel reduces acceleration the first peak by 4 dB and first SPL peak by as much as 12 dB. The overall tendency of the rest of the frequency range do not show obvious best performer for the acceleration case and shows that the unbeaded and light beading with carpet provide largest SPL reduction. The general effect of the carpet on average panel vibration is to completely attenuate the peaks after the first few peaks. The carpet effect on SPL is less pronounced and affects the first few peaks as well as the overall tendency by reducing the SPL at higher frequency: for this carpet starting at 300 Hz and up.

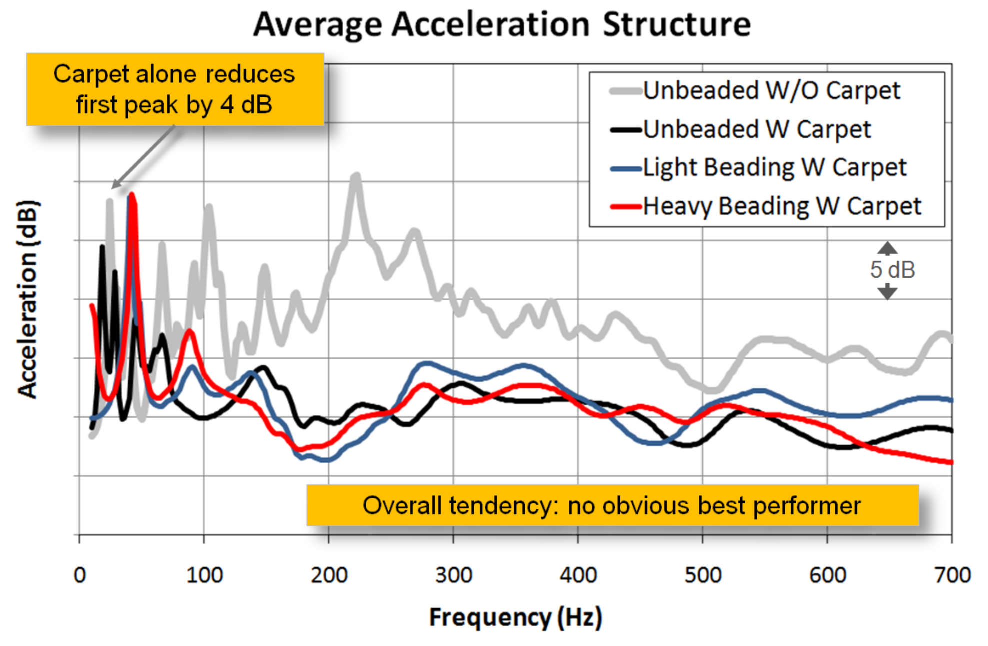

Figure 6: Effect of carpet on unbeaded and beaded panels average acceleration

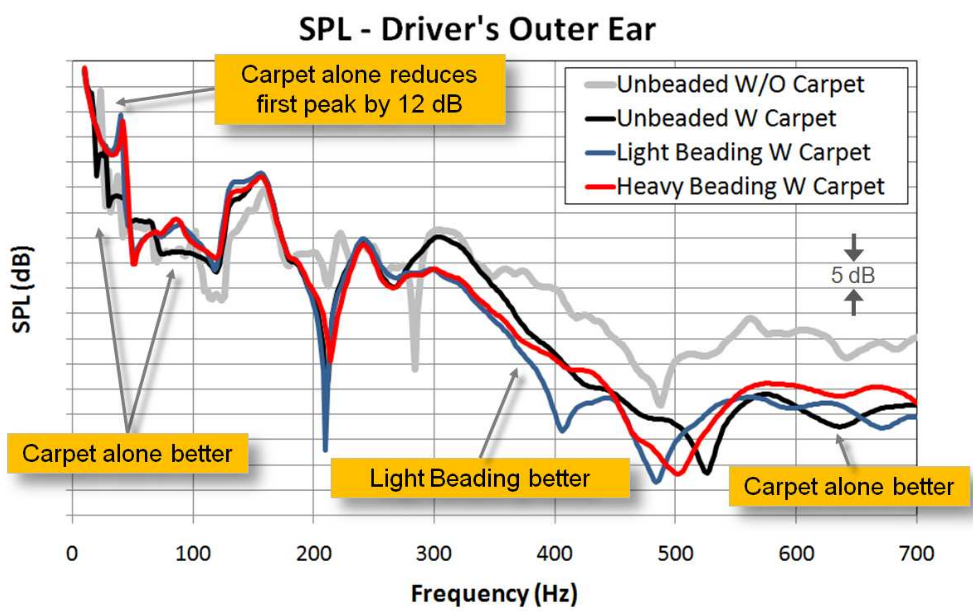

Figure 7: Effect of carpet on unbeaded and beaded panels driver's ear SPL

Effect of adding carpet on unbeaded panel vs beading only

It is interesting to note that the effect of adding a carpet on the unbeaded panel has an important positive effect on both acceleration and SPL. Figure 8 shows the effect of adding carpet onto the unbeaded panel (black) as opposed to only beading the panel (red, blue).

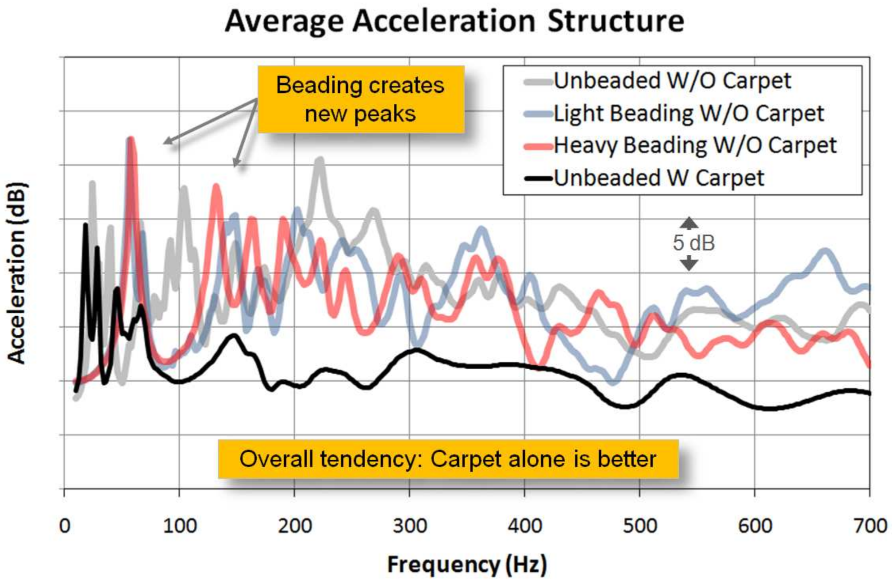

Figure 8: Effect of adding carpet on on unbeaded panel vs beading on acceleration

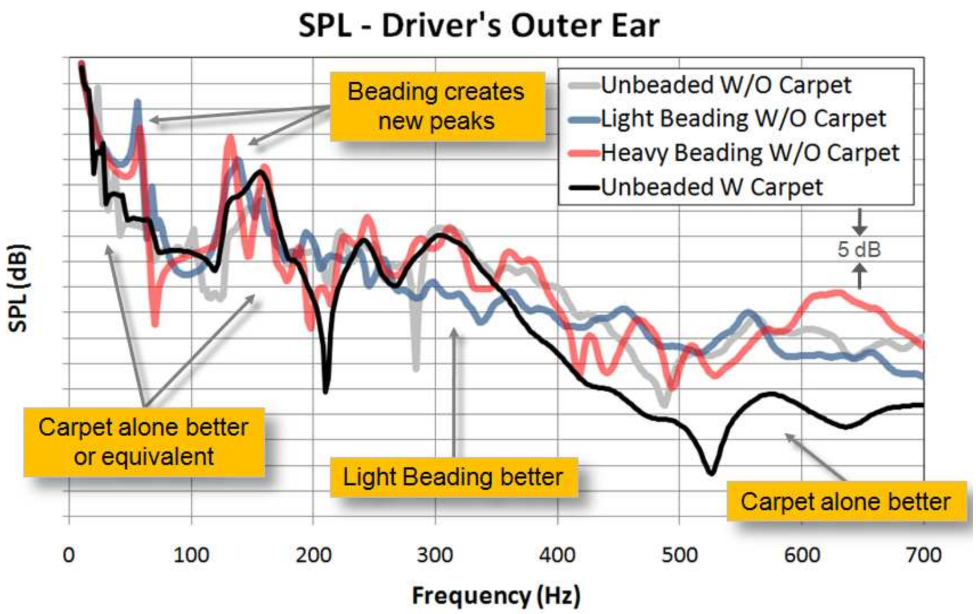

Figure 9: Effect of adding carpet on unbeaded panel vs beading on SPL

The carpet partly attenuates the first few peaks and the rest of the peaks on the frequency spectrum are completely attenuated while the beading produces new peaks and in some cases higher acceleration levels (up to 15 dB) than without beading at all. The effect of the carpet on the SPL reduces the first few peaks amplitude by 15 dB while the rest of the spectrum is attenuated starting at around 200 Hz (**Fehler! Verweisquelle konnte nicht gefunden werden.**).

Beading provides a better performance than the carpet alone on the acceleration at the first few peaks (15 dB reduction) which unfortunately do not translate into a significant reduction in SPL. Beading is also better on SPL at around 300 Hz. In fact, for the studied case, one would assume that using the carpet as the main method of controlling vibration and SPL would be adequate.

Effect of increasing structural damping

Another way to decrease vibration and SPL levels is by increasing structural damping of the floor panel. The default value used so far is 4%. Computations with a structural damping of 10% were conducted to assess if a further reduction in vibration and SPL can be obtained. As can be observed in Figure 10, a slight gain can be obtained from increasing structural damping. Real life damping treatment would also add mass and stiffness to the floor panel. An increase in performance should be expected. This has not been accounted for in this study.

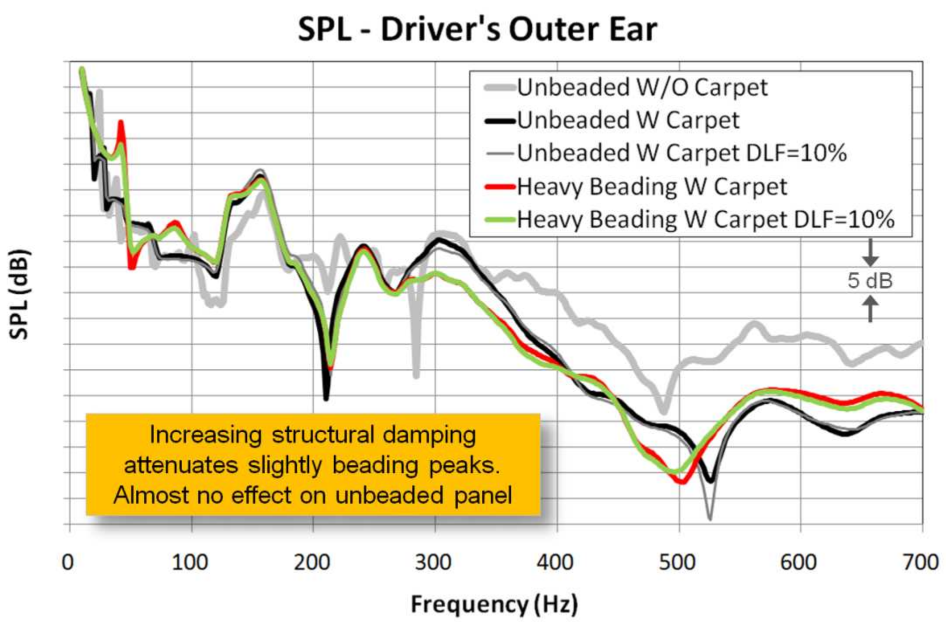

Figure 10: Effect of increasing steel panel structural damping to 10% on SPL

Effect of doubling carpet thickness

Finally, the effect of doubling the carpet thickness is presented in Figure 11. One would hope that increasing the carpet thickness might eliminate the new peaks created by the presence of the beading at low frequency. At the first few peaks, it reduces slightly the amplitude of the peaks and at higher frequency (above 250 Hz) the response is significantly lower, probably too low compared to vehicle requirement. Doubling the thickness of the carpet used in this study therefore cannot eliminate the negative impact of the beading at lower frequency.

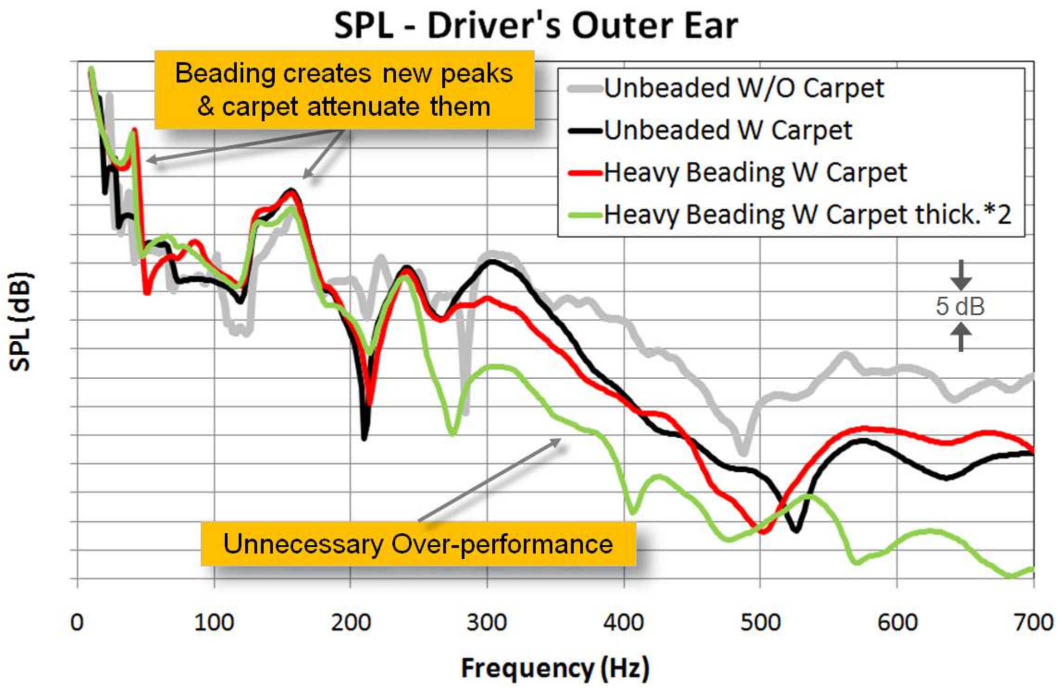

Figure 11: Effect of doubling carpet thickness on beaded panels

Conclusion

This paper has demonstrated the importance of taking into consideration not only the vibration response but also the acoustic aspect of a problem when designing beads in general. The assumption that the sound radiated power scales directly with vibration levels therefore considering the value of radiation efficiency to be equal to one is only valid above the coincidence frequency (around 12 500 Hz for non-beaded and 5 000Hz for beaded steel 1mm). Below coincidence, radiation efficiency can vary over several orders of magnitude. The sound power radiated is proportional to the product of radiation efficiency and velocity and therefore any match in peaks of these two quantities will yield a higher noise radiation. The effect of adding a carpet and increasing structural damping has also been presented. The effects of adding the carpet on the un-beaded panel has shown that this method alone can yield an important reductions in both vibration and SPL. Finally, optimization of beads should always be done using an acoustic criteria such as sound power radiation or SPL at a location in the vehicle since effect of beading does not have the same impact on vibration than it has on acoustics. Furthermore, optimization of beading should be done by considering the floor carpet and damping treatment that are intended to be used therefore the optimization objective function should be a combination of beading, carpet and damping treatment effect on acoustic response.

Additional Note

All computations were performed using VTM (Vehicle Trim Modeler) from ESI.

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