Designing the damping treatment of a vehicle body based on scanning particle velocity measurements

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Abstract

Car manufacturers are constantly seeking methodologies to enhance acoustic performance whilst meeting demanding weight and cost targets. Most optimisation strategies are designed via numerical simulations and later adjusted through modal and noise testing. Although traditional approaches are fairly effective, they require a very laborious and time-consuming process. An alternative procedure is hereby proposed to identify key radiation areas and control the structural-acoustic system by adding damping pads. Multiple scans are performed very close to the surfaces of interest acquiring the normal acoustic particle velocity distribution. The direct visualisation of this information can be used to find leakage as well as problematic modes across the structure. All data can be acquired with the vehicle in static conditions, using a monopole source or shaker as the source of excitation. The suggested measurement approach simplifies dramatically the refinement process of the damping package of an entire vehicle, reducing the traditional testing process to just a few days. Details about the measurement methodology as well as its implementation are explained in this paper along with an example in a car section which is validated with on-road tests. As shown, this approach provides an extensive amount of vibro-acoustic information for every vehicle section, ultimately helping to design an effective damping treatment.

Introduction

The vibro-acoustic properties of a car cabin play a key role in the perception of a vehicle quality. All vehicles include an acoustic package comprising various components such as absorbers, barriers, dampers and isolators which aim to improve the noise performance [1]. Increasing pressure from weight and emissions targets means that automotive OEMs are being challenged to meet customer expectations for NVH performance in ever more efficient ways, and of course, without inflating the cost of production.

Surface damping materials are very effective at reducing structure borne noise [2]. Passive damping materials have been used since the early 1960s in the aerospace industry. Over the years, advances in material manufacturing and the development of more efficient analytical and experimental tools to characterise complex dynamic behaviours enabled to expand the usage of these materials to the automotive industry [3]. Nowadays, multiple viscoelastic damping pads (as shown in Figure 1) are usually attached to the body in order to attenuate higher order structural panel modes that significantly contribute to the overall noise level inside the cabin.
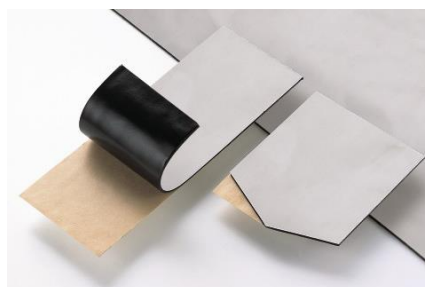

Figure 1: Example of a viscoelastic damping pad used in the automotive industry.

The rapid development in computational processing power has enabled the use of numerical simulations to determine the dynamic response of complex structures with a reasonable accuracy [4]-[8]. These techniques are effective especially in the early stages of the design cycle. However, substantial discrepancies are often found between simulations and experiments which led to development of hybrid techniques [9] or purely experimental methods [10].

Traditionally, experimental techniques are used to optimise the size and location of damping treatments. In particular, laser vibrometer type tests are often conducted on body in white structures enabling the fast acquisition of a large number of measurement points with a good spatial resolution. However, testing a complete vehicle is mostly unfeasible, requiring to evaluate every subsystem individually, hence limiting the usability of this technology in a fast and efficient way.

Alternatively, acoustic particle velocity sensors have been proven suitable for performing non-contact vibration measurements. Structural vibrations can also be acoustically measured using particle velocity sensors located near a vibrating structure. Several studies have revealed the potential of particle velocity sensors for characterising structural vibrations [11]-[16], which remarkably accelerates the entire testing process when combined with scanning techniques such as Scan & Paint [17].

The main purpose of this study is to introduce a measurement methodology for designing an effective damping treatment of a vehicle by means of scanning particle velocity measurements. The fundamental relationship between particle velocity and structural vibrations are hereby introduced. Furthermore, in the following sections several experimental examples are evaluated with the proposed methodology, including a validation on a complete vehicle and comparison with a damping package obtained by traditional techniques.

Particle velocity, surface velocity and acceleration

It is worth clarifying the relationship between particle velocity, surface velocity and surface acceleration in order to understand how particle velocity measurements can be helpful to characterise the vibro-acoustic behaviour of a complex structure. For an excited body that vibrates in a stationary harmonic regime with normal velocity $v_n(\mathbf{x}_0)$, it can be established that

$$a_n(\mathbf{x}_0) = \frac{\partial v_n(\mathbf{x}_0)}{\partial t} = \frac{\partial\left(v_0 e^{j(\omega t - \phi)}\right)}{\partial t} = j\omega v_0 e^{j(\omega t - \phi)} = j\omega v_n \qquad [\text{m/s}^2] \qquad (1)$$

where $a_n(\mathbf{x}_0)$ is the normal surface acceleration at the point $\mathbf{x}_0$; v_0 is the vibration speed and ϕ is an arbitrary phase value. Hence, there is a linear relationship between surface velocity and acceleration that allows for the direct computation of both quantities by simply measuring one of them. In contrast, the particle velocity in front of a vibrating body depends upon a term describing how efficiently vibrational energy is converted into acoustic excitation in the form of normal velocity as well as the acoustic field generated by any other surrounding sources captured at the sensor position $\mathbf{x}$, hence

$$u_n(\mathbf{x}) = v_n(\mathbf{x}_0)Z_r(\mathbf{x}_0, \mathbf{x}) + \sigma_n(\mathbf{x}) \qquad [\text{m/s}] \qquad (2)$$

where $Z_r(\mathbf{x}_0, \mathbf{x})$ is a term which relates the surface displacement and the particle velocity; $\sigma_n(\mathbf{x})$ is the variance of the additional acoustic signal perceived at $\mathbf{x}$. For a simple case, such as a baffled circular piston in the absence of noise, the particle velocity u_n measured on the axis of a rigid circular piston of radius b is related to its surface velocity v_n as such [18]

$$u_n(\mathbf{x}) = v_n\left(1 - \beta e^{-j2\gamma}\right)e^{j(\omega t - k\delta)} \qquad [\text{m/s}] \qquad (3)$$

where

$$\beta = \delta/\sqrt{b^2 + \delta} \text{ and } \gamma = k\left(\sqrt{b^2 + \delta^2} - \delta\right)/2 \qquad [1] \qquad (4)$$

where δ denotes the distance between the measurement point and the surface, i.e. $\|\mathbf{x} - \mathbf{x}_0\|$. The terms $Z_r(\mathbf{x}_0, \mathbf{x})$ and $\sigma_n(\mathbf{x})$ are usually unknown for most practical applications since both depend upon the characteristics of the radiating source and the measurement environment. However, as it has been proven in [19], the impact of those terms is minimised near the vibrating surface. Therefore, non-contact vibration measurements using particle velocity sensors should be performed as close as possible to the vibrating body in order to capture an accurate representation of the vibro-acoustic response. The uncertainty introduced by these unknown terms may significantly bias an absolute quantification of the vibro-acoustic excitation, but it certainly provides a reliable tool to characterise the energy distribution that is perceived at the acoustic boundary near the structure.

Direct sound mapping using Scan & Paint

The measurement system used in this paper is the scanning technique "Scan & Paint" [17][20]. The acoustic signals of the sound field are acquired by manually moving a P-U probe across a measurement plane whilst filming the event with a camera. In a post-processing stage, the sensor position is extracted by applying automatic colour detection to each frame of the video. The recorded signals are then split into multiple segments using a spatial discretisation algorithm, assigning a spatial position depending on the tracking information. Therefore, each fragment of the signal will be linked to a discrete location of the measurement plane. Next, spectral variations across the space are computed by analysing the signal segments. The results are finally combined with a background picture of the measured environment to obtain a visual representation which allows us to "see" the sound pressure, particle velocity or sound intensity spatial distribution. Figure 2 presents the equipment involved along with the basic measuring steps, in this example to detect acoustic leakage of a vehicle using a sound source inside the cabin.

Figure 2: Illustration of the measurement equipment (left) and the basic steps of the Scan & Paint method: manual scan while recording a video (middle) and assessment of the sound maps (right).

The use of scanning methods incorporating pressure and particle velocity sensors together within one single probe allows for a detailed study of a sound field. Additionally, a fixed reference pressure microphone can be used to preserve the relative phase information at the different grid positions. In this paper, measurement results are focused on normal particle velocity maps, since the objective is to identify structural areas with a high vibro-acoustic excitation.

Measurement methodology

The addition of damping material to a vibrating structure helps reducing the radiated sound as well as the vibration levels transmitted to the components that are mounted or attached to it. Surface damping simultaneously reduces both stress and deflection by converting mechanical energy into heat. Since the main effect achieved with damping treatments on panels is the reduction of vibration amplitude at resonance, it is key to identify the areas with a high surface displacement and hence large bending. Given the relationship between acoustic particle velocity and surface displacement presented above, localising the normal particle velocity maxima will directly indicate suitable locations for placing damping pads.

Most experimental vibro-acoustic methods developed for optimising surface damping use a shaker to introduce a known input force and evaluate the structural response of each panel via laser vibrometer measurements. This process is repeated multiple times for several excitation points, such as suspension or body mount attachments. Sequentially attaching the shaker to various points ensures that all noise transfer paths are excited and therefore all flexible regions on structural panels are identified. Although traditional experimental approaches can be effective, they are also cumbersome, very time consuming and expensive.

The first step assessed in this study in order to achieve a significant reduction of the total testing time was an evaluation of the spatial variability introduced by changing the source type and location. The vehicle under investigation had a complete acoustic package installed and only the powertrain was dissembled. Floor panels were measured several times with a single shaker attached to the front and rear suspension mountings as well as with a monopole sound source under the floor and rear wheel arches. Despite the local level variations between all the measurements, it was observed that the monopole excites most of the structural modes independently of its position, which was not necessarily the case when the shaker was used. A possible explanation for this finding is that using an acoustic source will excite the structure by the pressure field that spreads through a fairly large area. Since the main goal is to identify panel resonances across the vehicle body in a time efficient manner, using an acoustic source can very beneficial, despite the fact that the operational forces acting over the structure become unknown unless the pressure field in the incident side is also measured. Consequently, for the sake of simplicity, scanning measurements were only performed with a single monopole source located either in the wheel arches, underneath or inside the vehicle cabin, depending on the panel evaluated.

The second crucial element that reduces the overall testing time is the usage of a scanning method that enables to produce visual representation of the acoustic field in a matter of minutes. In contrast with laser vibrometers, where most of the testing time is due to positioning the device appropriately, most of the measuring time is spent during the manual sweeps of the probe above the panel of interest. Investigating a different area just implies changing the camera location and scan the new surface. As a result, this technique becomes highly suitable for most panels inside or outside the vehicle without the necessity of dissembling the different subsystems from the main vehicle body, or where positioning a laser vibrometer appropriately would be difficult.

Improving a structure without damping

An example of the particle velocity maps obtained for a floor section are presented in Figure 3. As shown, the direct mapping of the normal particle velocity yields large spatial variations across the evaluated area (>25 dB) even for low frequencies, revealing the critical panels with high excitation.
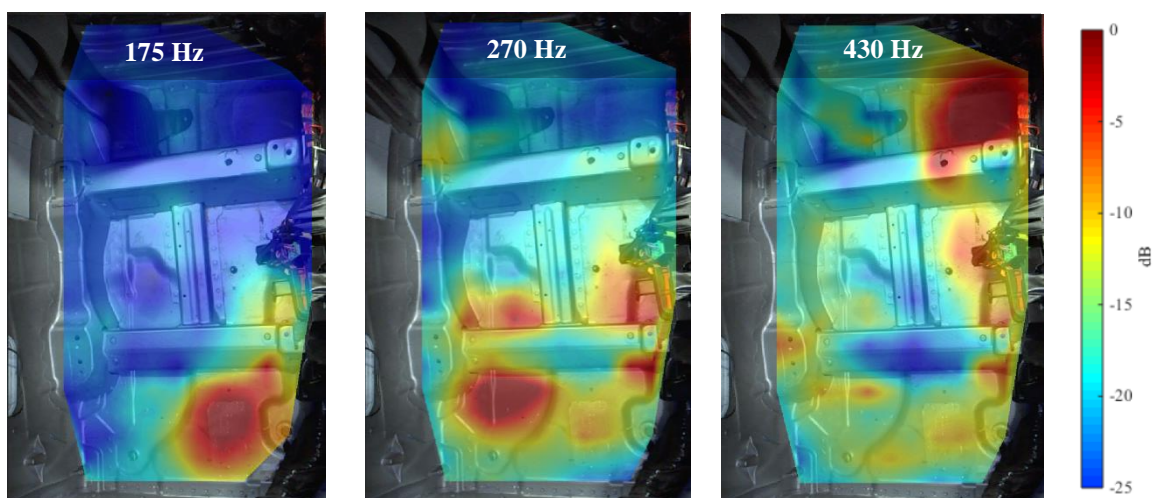

Figure 3: Particle velocity colormaps for the main critical frequencies of right floor section without damping material.

After identifying the main panel resonances and critical areas, several configurations of damping were assessed (material type, position and thickness). In practice, this implies scanning the area of interest for each configuration, obtaining direct feedback on the impact that placing several damping pads have on the structure. Figure 4 shows the spatially averaged particle velocity spectra before and after a damping treatment, along with velocity colormaps. As it can be seen, the damping treatment applied reduces significantly the acoustic output of this panels, mainly reducing the excitation of the resonance frequencies displayed above. A fairly homogenous distribution of energy across the area evaluated is obtained after applying the damping pads.
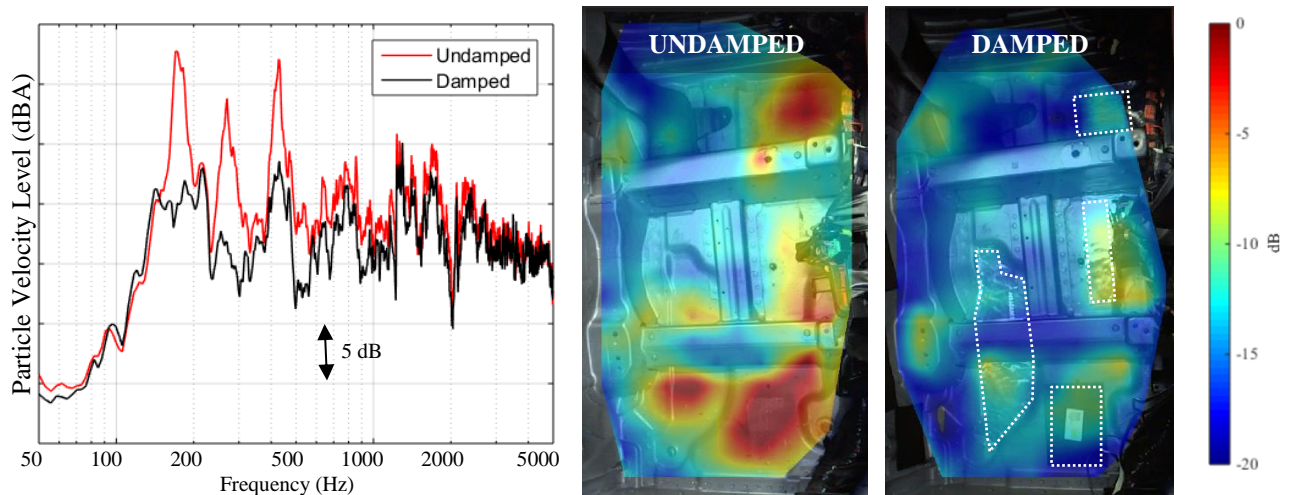

Figure 4: Spatially averaged particle velocity spectra (left) and broadband colormaps of the floor without (middle) and with (right) a damping treatment.

Enhancing an existing damping pack

The proposed measurement methodology was applied to an entire vehicle supplied with a complete acoustic package, including a baseline damping treatment designed with traditional methods. In this case, the investigation had to consider additional damping introduced by trim and carpets that are attached to the vehicle panels and could not be modified. The main purpose of this test example was to optimise the existing panel damping pack of a vehicle addressing the following goals when possible:

- Move or remove existing pads that are ineffective and/or unnecessary
- Ensure existing pads have a suitable size and thickness
- Identify undamped problem panels/areas and to demonstrate the effectiveness of placing or moving an additional pad to such areas
- Relocate parts from the bottom of surfaces to the top where possible due to a durability issue of hanging parts coming loose
- Overall removal of weight and part count with no significant detriment to NVH

The powertrain of the vehicle was dissembled and the cabin divided into multiple sections that could be measured with a single camera view. A monopole source was used to excite the structure at multiple key locations: it was positioned under the boot during the measurements performed in the passenger floor and tunnel; inside the cabin for characterising the boot and rear wheel arches; and in the wheel arches while measuring the rear quarters, rear seats and heel boards. It should be noted that it is not absolutely necessary to remove the power train of the car as almost all body in white panels can be assessed externally or by removal of just seats, carpet and centre console. The use of an acoustic excitation does not require direct access to the opposite side of the panel as sound will readily diffract around most obstacles.

The first detailed scan of the baseline vehicle allowed to identify the potentially most problematic areas, as well as any acoustic leakage of airborne sound through sealing, trim and carpets. The floor sections had significant damping material attached underneath the carpets, which had to be improved by taking into account the complete system.

As an example, the different stages covered during the investigation of a floor section are presented in Figure 5. The measurement performed for the baseline configuration showed that there was an area with a significantly high excitation being transmitted through the carpet. After removing this element, surprisingly the highest velocity region appeared at a different location as when the carpet was attached. In addition, most areas with existing pads had a low surface velocity.
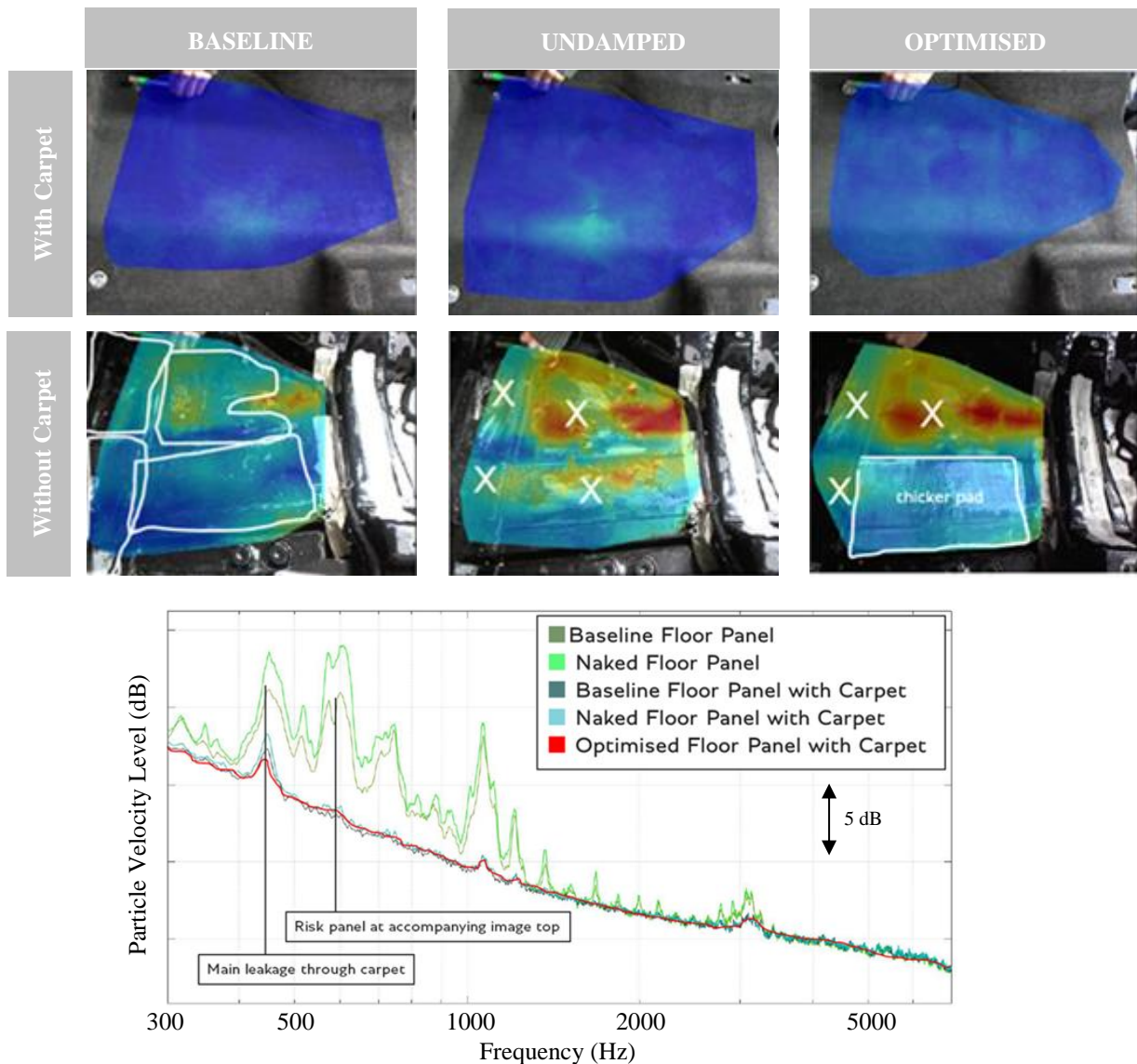

Figure 5: Particle velocity colormaps of the cabin floor for multiple conditions (upper and middle) and spatially averaged spectra (bottom). The outlined areas indicate the position of damping panels and the crosses denote that a damping panel was removed.

Next, the damping package of the supplied vehicle was removed. After assessing the panel excitation for the undamped condition, a thicker damping pad was attached over just the area that had a perceivable excitation when the carpet was mounted. The experimentally optimised surface damping distribution was then evaluated by scanning again the section with and without the covering carpet.

As shown in Figure 5, the modified damping treatment with a thicker pad introduces a risk at 600 Hz, since the dominant mode at that frequency was no longer attenuated by damping pads. However, the combined damping achieved when the carpet is mounted shows that the spatially averaged particle velocity spectrum has better performance overall and a similar performance at the risk frequency, as compared to the original configuration. Furthermore, the colormap of the modified configuration including the carpet does not have any apparent maxima, the energy is low and homogeneously distributed across the scanned surface, therefore achieving a significant improvement over the baseline configuration. Subjective evaluation or measurements taken after an optimisation study will confirm the performance, however should a new problem occur, the existing notes can be brought back and the problem panels quickly traced.

In summary, the direct visualisation of the particle velocity distribution of a vehicle section enables localisation of the areas where the structure has high excitation. Either if there is acoustic leakage or a modal amplification, the colormaps can be used to gain understanding about the critical locations where noise is introduced into the cabin. Furthermore, due to the complexity and numerous layers of different materials that a car interior requires, it is useful to assess every area with multiple configuration to reach the best compromise between a good acoustic performance, weight and cost of the damping treatment applied.

On-road validation

The vehicle studied in this paper was tested before and after performing modifications on the damping treatment. The complete investigation lasted for 5 days, including data acquisition, analysis and evaluation of the proposed configuration. Sound pressure measurements were carried out at the front and rear of the cabin for multiple asphalt surfaces. Later on, the vehicle powertrain was again assembled and tested on the road. Results obtained for two very different asphalt surfaces are shown in Figure 6.

As shown, the acoustic performance of the modified vehicle is equal or better in all the conditions evaluated. It can be noted that despite weight increasing slightly in this instance, the number of panel treatments was reduced by approximately 15 %, yielding a similar cost reduction for the total pack plus additional saving through warehousing costs and production time.

A second vehicle model to which the proposed methodology was applied achieved a 30 % reduction in the number of panel treatments and even 10 % reduction in the damping package weight. Unfortunately, further details cannot be shared at this time due to their confidential nature.

Conclusions

A measurement methodology to design the damping treatment of vehicle body is hereby proposed and demonstrated for a full vehicle on the road. Manual scanning measurements via Scan & Paint are used to characterise the vibro-acoustic distribution near the vibrating structure. The short time required to undertake the tests allows for capturing a detailed acoustic dataset in a fast and efficient way. Furthermore, the extensive data gathered can be effectively used to design an optimal damping treatment, and directly evaluate the direct impact of the changes implemented. The use of acoustic particle velocity becomes key to identify high excitation areas due to the direct link between this quantity and the surface displacement when near-field conditions are achieved. Experimental results show that a significant reduction in the number of damping pads can be obtained with the proposed methodology, yielding a significant cost and/or weight reduction whilst preserving or improving the cabin acoustic performance.
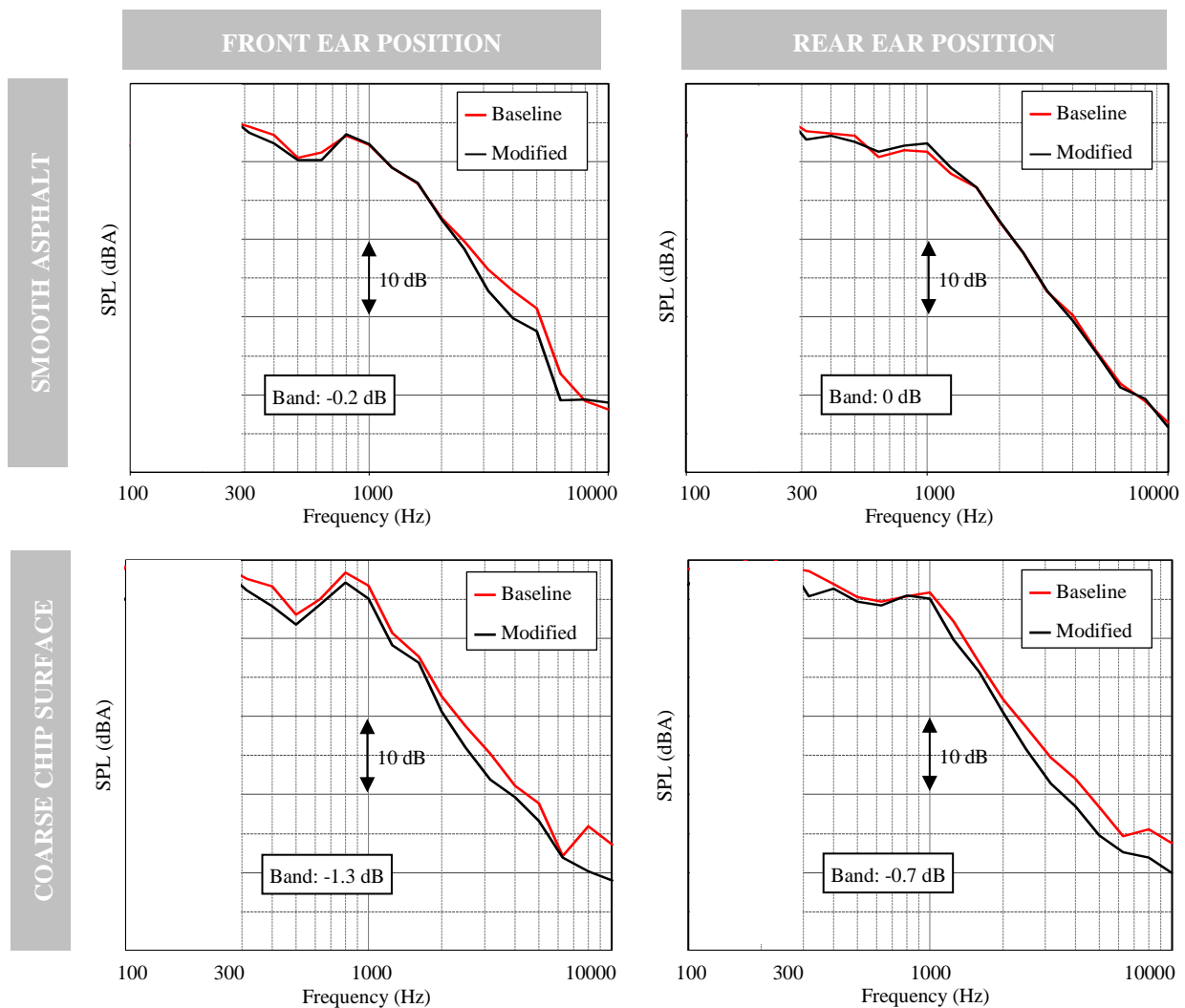
Acknowledgements

Figure 6: Sound pressure spectra measured at the front (left) and rear (right) of the vehicle cabin for two different asphalt surfaces: smooth (top) and coarse chip (bottom), including the sound pressure level differences in the frequency bands displayed.

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