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Application of Vehicle Interior Noise Simulation (VINS) for NVH Analysis of a Passenger Car

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ABSTRACT

The overall perception of a vehicle's quality is significantly influenced by its interior noise characteristics. Therefore, it is important to strike a balance between "pleasant" and "dynamic" sound that fits the customer requirements with respect to vehicle brand and class [1]. Typically, a significant share of the interior vehicle noise is transferred through structure-borne paths. Hence, the powertrain mounting system plays an important role in designing the interior noise.

This paper describes an application of the method of vehicle interior noise simulation (VINS) to achieve a characteristic interior sound. This approach is based on separate measurements (or calculations) of excitations and transfer functions and subsequent calculation of the interior noise in the time domain. The VINS procedure allows for explicit separation (and listening) of the sound field into airborne and structure-borne shares and each share can be further decomposed into its constituent paths and directions.

In the vehicle model, which is the basis of the VINS method, the powertrain mounts are represented by frequency dependent transfer functions. In the application discussed in this paper, the influence of mount stiffness on the interior noise is calculated by virtually modifying the characteristics of these transfer functions. Critical noise paths are identified and analytically modified to achieve the desired interior noise characteristics. Finally, engine mounting system hardware changes are conducted and the measured data are compared to the VINS predictions.

INTRODUCTION

The customers decision to purchase a particular vehicle is influenced by objective criteria, such as e.g. fuel consumption, or vehicle cost. On the other hand, the often-unconscious subjective impression, determined by soft factors like vehicle design and sound, play an important role.

The soft factor interior noise influences more than the buying decision; under long term driving condition it also contributes to driver comfort and reduced fatigue, hence increased safety. A more pleasant and, depending on the vehicle class, more dynamic vehicle interior noise is thus an important goal in the vehicle development.

As time goes on, more and more subsystems of the vehicle are defined using CAE tools. Such CAE tools allow target-oriented development and reduce development costs and time to market. Even with the advancements in CAE tools, the complexity of the full vehicle makes it difficult and time consuming to set up a complete virtual vehicle model describing all of its acoustic features. Therefore, the combination of measured data with simulation results is a very powerful tool to achieve a good understanding of the vehicle as a complex acoustic system.

With regard to noise simulation methodology, valuable investigations of the acoustical behavior of the car body were conducted for example by Adam et al [2], by Kropp et al [3] and by Frappier et al [4]. A significant step toward interior noise simulation was made by Williams et al [5] with the exact analysis of the cause-and-effect chain of interior noise generation. Genuit [6] introduces a procedure for interior noise simulation that is more concerned with sound design issues than with engine-acoustic measures. Wiehagen [7] uses the interior noise simulation method for prediction of the impact of engine related modifications on the vehicle interior noise. The details of the simulation method called VINS (Vehicle Interior Noise Simulation) are detailed presented elsewhere [8], [9].

GENERAL PROCEDURE OF VEHICLE INTERIOR NOISE SIMULATION (VINS)

Under various driving conditions, e.g., city stop and go traffic, and during acceleration, the powertrain induced noise share represents a significant portion of the overall interior noise. The engine by itself radiates noise from different components and surfaces. In addition it transfers vibrational energy through the mounting locations. The radiated noise corresponds to the airborne noise share and the noise transferred directly through the structure to the structure-borne noise share. The input of the transmission system under investigation is thus two-fold. The airborne share consists of the noise components from powertrain surfaces and the intake and exhaust orifices. The structure-borne share consists of the powertrain vibrations transferred through the mounts as well as the suspension and drive line.

In order to simulate airborne interior noise, the transfer functions from engine compartment to passenger compartment must be known. The firewall and underbody are critical contributors with regard to airborne noise transmission.

Conventionally, for determination of airborne noise transfer functions, a spherical loudspeaker is placed in the engine compartment with the powertrain removed and an intensive omni directional acoustic field is generated. The interior noise induced is received and recorded by an artificial head at the position of the driver's ears. For the VINS procedure, a reciprocal method is favored (**Figure 1**). A loudspeaker is placed in the passenger compartment, and the supply voltage is controlled (according to the maximum sequence technique) while simultaneously the airborne noise response in the engine compartment is recorded. A PC-based measurement system generates the maximum sequence and calculates the transfer function. Typical airborne noise transfer functions of passenger cars show a steady decrease by approximately 8 dB per octave overlaid by additional resonance effects.

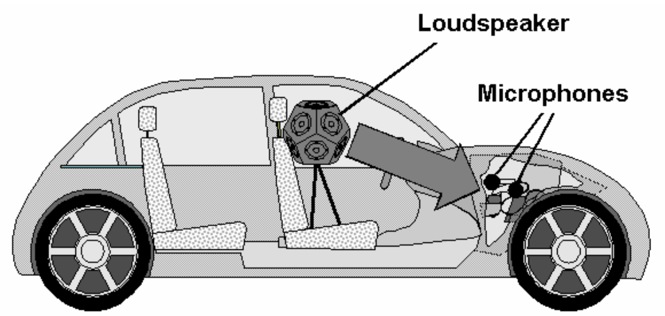

Fig. 1: Reciprocal measurement of airborne noise transfer functions

The main advantages of the reciprocal method are:

- Measurements without removing engine
- Correct measurement of the engine compartment's spatial acoustic behavior
- Individual transfer functions for each engine side

The engine mounts constitute the most important components of the structure-borne noise transfer path. The engine vibration excitation (due to gas and mass forces), the energy transfer through the powertrain mounts and the chassis as well as the sound radiation are contributing to the structure-borne noise share. The overall structure-borne noise transfer function is calculated by consideration of the transfer functions of the mounts, apparent masses and body. (**Figure 2**)

The mount transmissibility is calculated from engine mount acceleration measurements for an engine run up on the vehicle test bench. The value of the mount transmissibility for a certain frequency is calculated based on time sequences with high excitation at that particular frequency. Through this procedure, signal to noise ratio is improved, and the influence of "cross talk" from one mount to another is minimized.

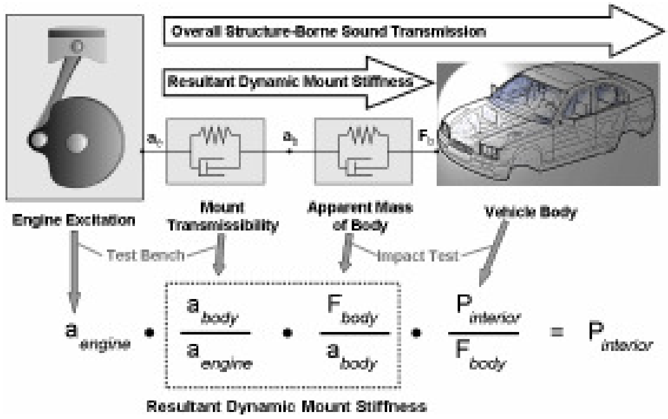

Fig. 2: Measurement and calculation of dynamic mount stiffness and overall structure-borne transfer function

The mount transmissibility depends on the mount properties and on the local stiffness of the chassis. The transfer function, which is a property of the mount itself, is the effective dynamic mount stiffness. This is the product of the mount transmissibility and the apparent mass. The apparent mass is determined via an impact measurement at the vehicle body close to the mount connection.

The vehicle body transfer function is measured simultaneously with the apparent mass by additionally recording the cabin noise sound level at the driver's head position. Finally, the overall structure-borne transfer function is calculated by multiplying the dynamic mount stiffness with the chassis transfer function. The resulting frequency curve typically shows high levels up to approximately 120 Hz and decreases significantly above 500 Hz. This reflects the well-known low pass response of the chassis to structure-borne noise excitation.

During the development phase of the vehicle (even before an engine is installed in the vehicle), potentially annoying interior noise shares, their causes and transfer paths can be identified by using the VINS technique. This analysis is carried out in the time domain so that all phase information from various paths is preserved and the simulated interior sound can be analyzed subjectively (by listening to it) and objectively. The interior sound simulation technique is also used to perform noise path analysis on an existing vehicle. Measured powertrain level data and structure-borne/airborne noise transfer functions are used to synthesize the interior noise. In addition, the interior noise can be decomposed into the dominant airborne and structure-borne paths, which are affected by sources such as engine, intake and exhaust system, powertrain mounts and drive train. The two excitation categories are illustrated in **Figure 3**.

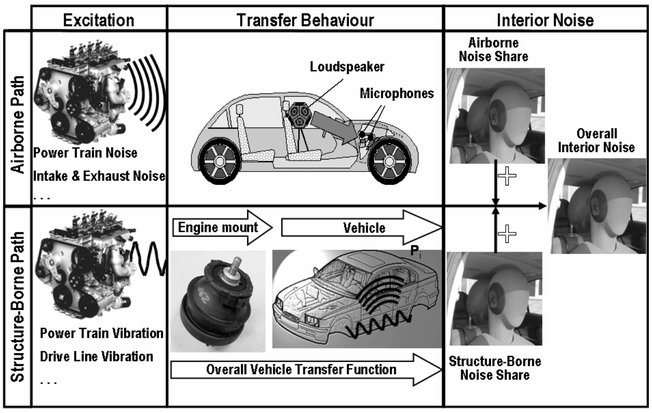

Fig. 3: Procedure of Vehicle Interior Noise Simulation (VINS)

The excitation signals used in the interior noise simulation must have the correct phase relation, but occasionally the excitations cannot be measured simultaneously (test performed at different times and/or locations). Therefore, a tool to synchronize different measurements was developed. It operates in the time domain and is based on simultaneously measured camshaft trigger signals.

After synchronisation the measured excitations are filtered by the vehicle transfer functions. Results of the procedure are audible noise shares of the two categories airborne and structure-borne noise, along with their individual components. In addition, the shares can be combined to an audible overall noise or even to audible noise elements, containing only a selection of the measured transfer paths. Having the individual noise shares is a very important feature for trouble shooting work, allowing subjective and objective diagnosis of the disturbing noise events.

INTERIOR NOISE SIMULATION APPLIED TO MINIVAN

In a cooperative project between VW and FEV, the procedure of vehicle interior noise simulation (VINS) was applied to a minivan with a four cylinder diesel engine and manual transmission. The goal of this work was to achieve a detailed understanding of the passenger cabin sound generation mechanisms focusing on the powertrain mounting system.

The noise simulation is based on measurements taken from:

- Powertrain test bench (powertrain sound radiation)
- Chassis dynamometer (powertrain and driveline structure-borne excitation, intake and exhaust orifice noise, mount transmissibility)
- Impact and sound transmission measurements (airborne and structure-borne transfer functions)

Figure 4 shows the overall level of simulated cabin noise compared to the direct measurement for a full load run up in the 3rd gear. The direct measured noise is not an input parameter for the simulation, but if it is available, it is used to validate the simulation. The plot shows in general a good correlation of the sound pressure levels for the measured and simulated noise. The resonance at 3800 rpm is slightly overestimated by the simulation.

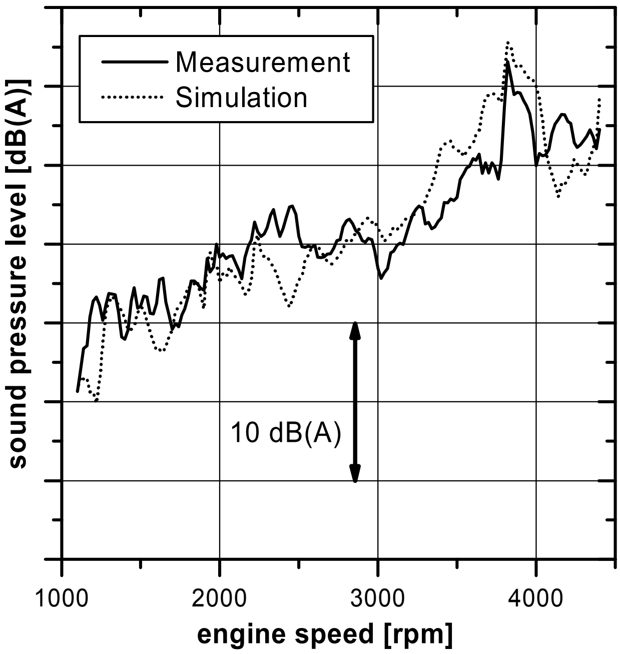

Fig. 4: Interior noise level of direct measured v/s simulated vehicle interior noise (full load)

A more detailed impression of the spectral content is given in Figure 5 in the form of Campbell diagrams (corresponding to the overall levels shown in Figure 4).

The goal of the noise simulation procedure is to find the main characteristics of the measured noise in the simulated results. Another feature of the VINS technique is its ability to obtain results in a relatively quick manner. This allows the application of the simulation procedure during the typical vehicle NVH development process. After the first application of the interior noise simulation, the detailed analysis of the interior noise generation mechanisms is quite simple compared to other approaches. The individual interior noise shares already exist as audible time signals, and the relevant transfer functions are also available. Typically, the first step is the decomposition of the cabin noise into its airborne and structure-borne shares. The corresponding sound levels (and Campbell diagrams) of the investigated minivan are shown in **Figures 6 and 7**.

The sound level curves indicate that the interior noise is clearly dominated by the structure-borne share. Over the entire engine speed range, the level of the structure-borne share is 5-10 dB(A) above the level of the airborne share. The Campbell diagrams of the corresponding noise shares (Figure 7) show that the structure-borne noise is dominant in the low frequency range up to approximately 1 kHz. Only the second engine order of the airborne noise contributes significantly in this low frequency range.

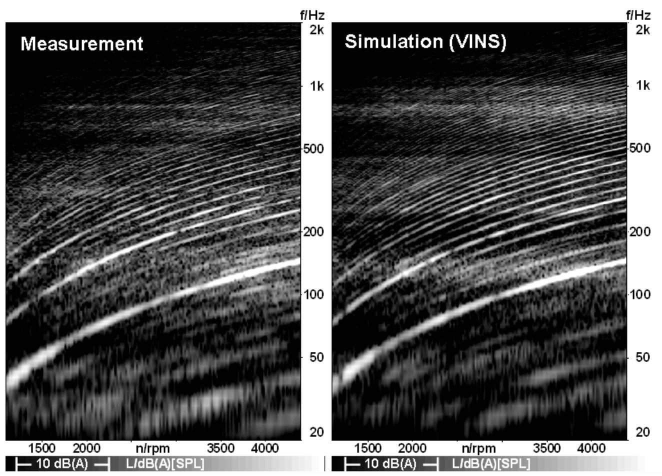

Fig. 5: Campbell diagram of direct measured v/s simulated vehicle interior noise (full load)

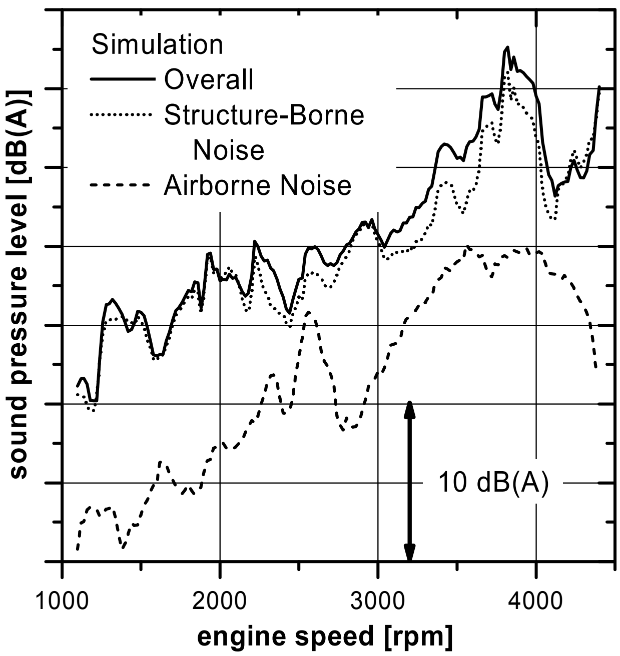

Fig. 6: Decomposition of interior noise level into airborne and structure-borne share (full load)

As an example of further decomposition and detailed analysis, the noise share of the rear roll restrictor path is analyzed. **Figure 8** shows the level of the rear roll restrictor noise share in comparison to the overall structure-borne share. In the engine speed range between 1200 rpm and 1800 rpm the rear roll restrictor noise share contributes significantly to the overall noise. After clarification of the dominant noise path for a certain interior noise feature, the standard VINS evaluation procedure includes further decomposition of the critical noise paths into excitation and transfer function. In the case of the rear roll restrictor the excitation (i.e. the roll restrictor vibration at powertrain side), shows no distinctive characteristics. Therefore the transfer functions will be considered in more detail.

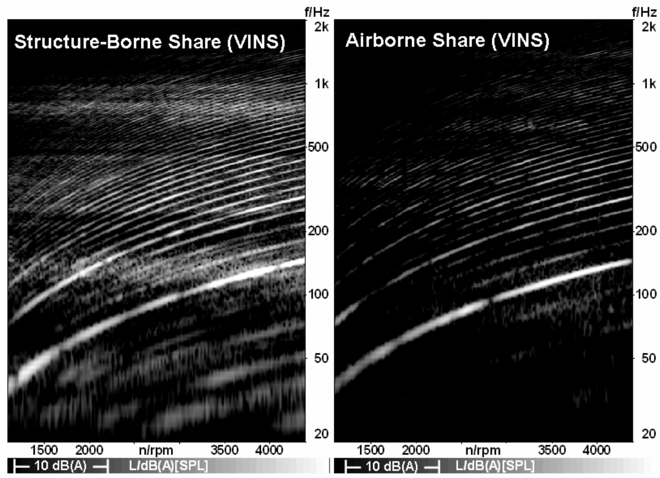

Fig. 7: Spectral content of airborne and structure-borne share of interior noise (full load)

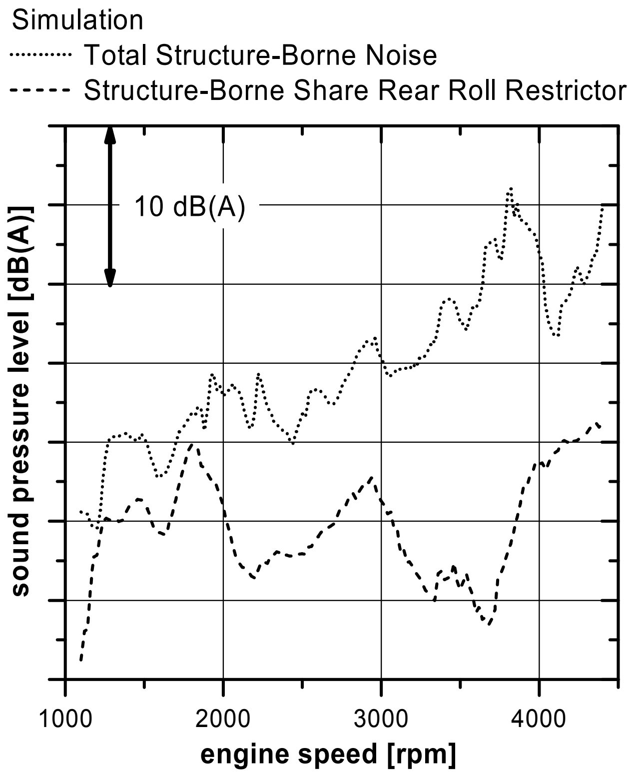

Fig. 8: Comparison of simulated structure-borne noise and the noise share from the rear roll restrictor (full load)

In **Figure 9** the different relevant transfer functions of the rear roll restrictor, vertical direction, are plotted in a single plot as follows:

- Effective dynamic mount stiffness - connecting mount acceleration at the engine bracket with the corresponding force at the chassis
- Chassis transfer function from force at the mount location to sound pressure in the vehicle cabin
- Resulting overall structure-borne transfer function

Figure 9 illustrates how mount and body transfer function contribute to the resonance in the overall transfer function in the vicinity of 40-50 Hz. Within this frequency range the chassis transfer function has a resonance which leads to a resonance in the structure-borne transfer function. In a similar way the other shares of the roll restrictor, excited by acceleration in longitudinal and lateral direction, can be analyzed. This analysis is usually performed for all important noise paths (with corresponding excitations and transfer functions) focusing on the selected characteristic of the interior noise. The results of such an analysis is a table indicating the dominant noise path(s) and corresponding influence of excitation and transfer functions for each noise feature considered (**Figure 10**).

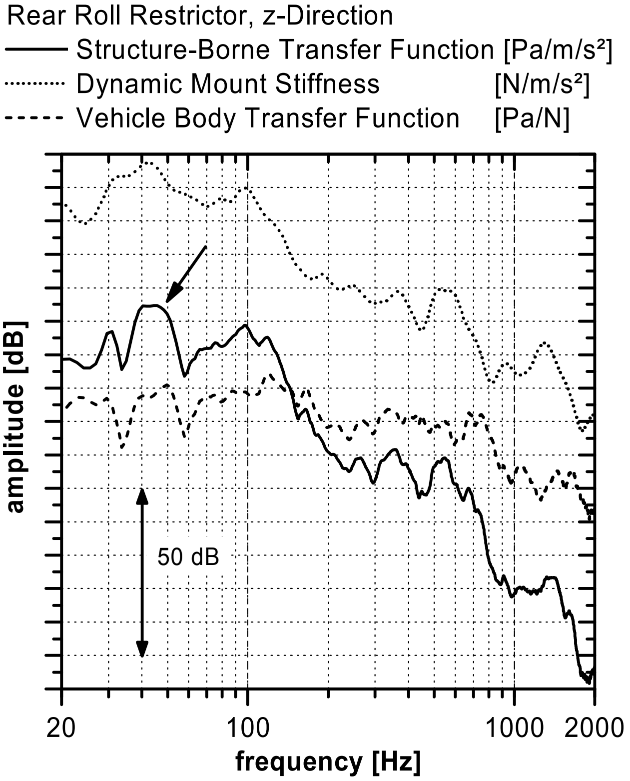

Fig. 9: Transfer functions of the rear roll restrictor, vertical direction

Interior noise feature: Frequency band between 1200 rpm and 1800 rpm									
noise path	engine mount	gear-box mount	front roll restr.	rear roll restr.	drive-line left	drive-line right	P/T radia-tion	Intake orifice	exhaust orifice
noise path contribution				●	●	●			
excitation									
transfer function				●	●	●			
mount stiffness				●					
body transmission				●	●	●			

Fig. 10: Example of noise path overview of a selected interior noise feature

MODIFICATIONS

After the vehicle model of the interior noise simulation is complete and all input data are available, various modifications can be easily analyzed. In the following section, the exchange of a roll restrictor and the variation of control arm bushing stiffness are given as examples of such modifications.

MODIFICATION OF THE ROLL RESTRICTOR

In the process of vehicle development, the idea of using common parts for different vehicles plays an important role. In this context, a stiffer rear roll restrictor was considered and the vehicle interior noise simulation (VINS) was used to calculate the effect of the stiffer roll restrictor.

In the VINS vehicle model the powertrain mounts are represented by frequency dependent transfer functions. The effect of a stiffer mount is therefore considered by an increase of the corresponding transfer function, which represents the dynamic mount stiffness. The stiffnesses of the two considered roll restrictor mounts are known from a component test bench. The stiffness of the rear roll restrictor in the vehicle model was increased according to these results. Initially, the increased stiffness influences only the noise share of the rear roll restrictor, however after the summation of all noise shares the effect on the interior noise becomes audible.

Figure 11 shows the effect of the stiffer rear roll restrictor on the level of the simulated interior noise.

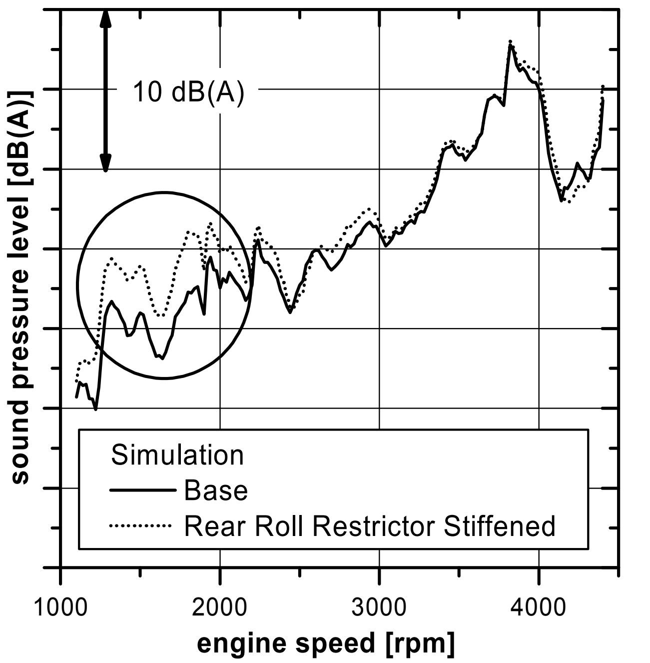

Fig. 11: Simulated overall noise level of base variant v/s modified mounting system with stiffer rear roll restrictor

The increased stiffness mainly influences the overall level in the lower engine speed range below 2200 rpm.

This corresponds with the structure-borne noise share of the rear role restrictor noise level shown in **Figure 8**, which is more significant for lower engine speeds.

The stiffer roll restrictor was available as an identical part from another vehicle. Therefore, the prediction of the interior noise simulation could be checked without great effort by an interior noise measurement of the modified vehicle. The result of this measurement confirms the predicted level increase in the lower engine speed range (**Figure 12**).

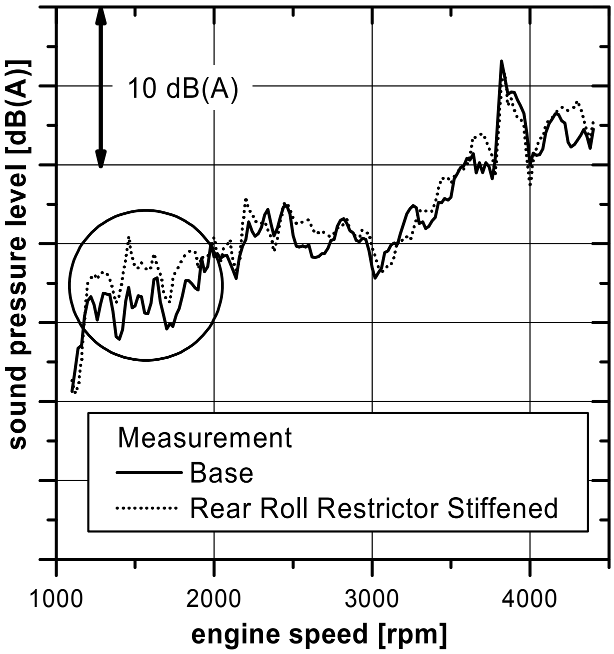

Fig. 12: Measured overall noise level of base variant v/s modified mounting system with stiffer rear roll restrictor

VARIATION OF CONTROL ARM BUSHING STIFFNESS

In the lower frequency range, the driveline contributes significantly to the interior noise. Therefore, variation of control arm bushing stiffness is considered as a further example of the application of the VINS technique. **Figure 13** shows the decomposition of structure-borne interior noise share of the vehicle into powertrain induced and driveline induced noise.

The resonances of the driveline noise share (at 2000 rpm and at 3800 rpm) contribute significantly to the structure-borne noise share and hence to the overall noise (**Figure 14**).

One possibility to reduce the driveline induced interior noise share is to decrease the stiffness of the connection from the transverse control arm to the chassis. The structure-borne noise is transferred through the driveline to the wheel hubs and furthermore through the control arms back to the chassis.

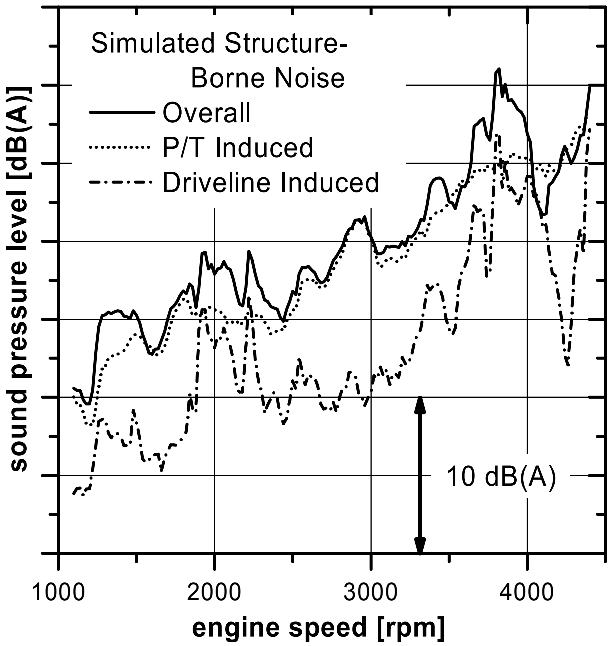

Fig. 13 Decomposition of simulated structure-borne noise level into powertrain induced and driveline induced noise

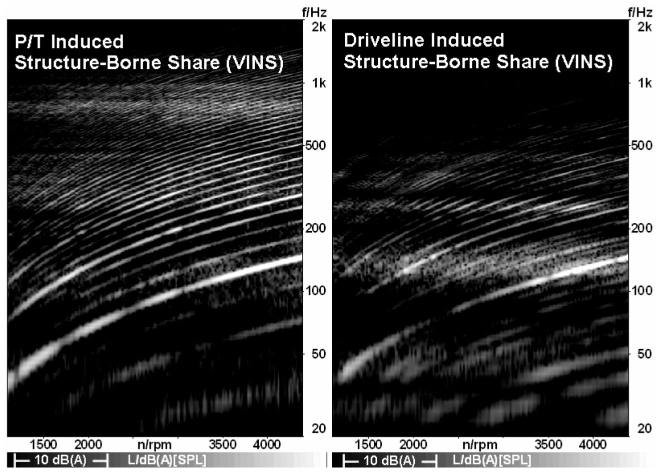

Fig. 14 Frequency content of simulated powertrain induced and driveline induced structure-borne noise shares

Such a change does not only affect the interior noise but also other vehicle attribites, such as driveability. Therefore, such countermeasures should be coordinated with other relevant aspects of vehicle development.

As an example the effect of reduced control arm bushing stiffness is shown in **Figure 15**. The control arm bushing stiffness is reduced by 50% in all directions. As expected from the driveline share (Figure 13), the main influence can be seen at the two resonances at 2000 rpm and at 3800 rpm. Here the overall interior noise level is reduced by up to 4 dB.

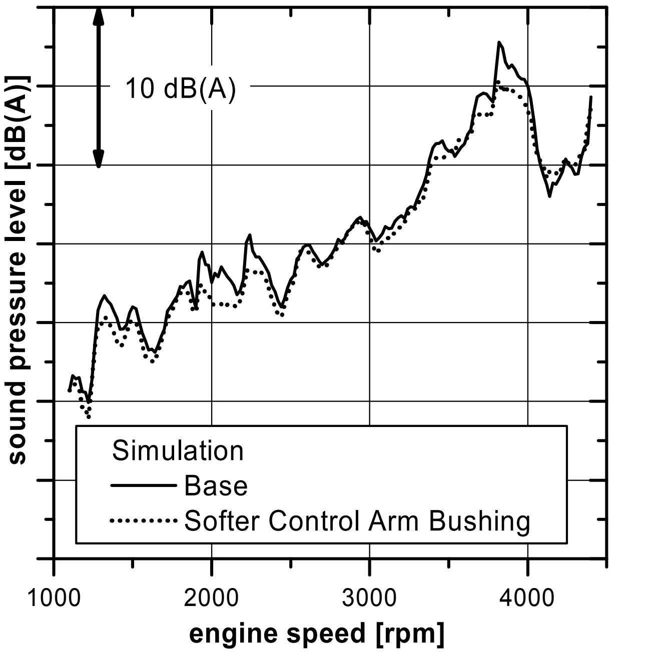

Fig. 15: Simulated effect of reduced control arm bushing stiffness on interior noise

SUMMARY

The application of vehicle interior noise simulation (VINS) in the vehicle NVH development process was described in this paper. A case study on a minivan was used to demonstrate the utility of the VINS technique. With the main structure-borne and airborne noise paths included, a good correlation between simulated and measured noise was achieved. The VINS technique allows the decomposition of the interior noise into the individual noise shares for root cause determination and subsequent problem resolution. The case study highlighted the effect of a stiffened roll restrictor and the simulation result was confirmed through a vehicle interior noise measurement. The effect of control arm bushing modifications on the interior noise was also estimated.

OUTLOOK

Future efforts will be aimed at modeling powertrain excitations analytically, and substituting the modules which were initially empirically determined. Thus, the synchronization tool for non-simultaneous excitation measurements (as described above) can also be used for analytically derived signals. This is currently done for results of CFD (computational fluid dynamics) calculations for intake or exhaust orifice noise [10].

The numerical calculation of the power train related excitation of low-frequency solid body vibrations and higher-frequency excitations of engine mount brackets modes is state-of-the-art. Calculation of the power train vibrations by means of MBA (multi-body analysis) and FEA (finite element analysis) models is performed currently as well.

Future work is aimed at advancing the, interior noise and vibration simulation towards the creation of a Virtual NVH Engine and/or Vehicle. With such a tool, weak points of the power train, the mounts and the vehicle body could be detected and optimized at an even earlier stage of the development process. The experience gained with the development and usage of the VINS methodology could also be applied to simulate the vehicle's exterior noise. Such a tool, in conjunction with the VINS methodology, could efficiently guide the vehicle development process to meet both interior and exterior noise and sound quality requirements. .

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