Chassis Noise Optimisation

The vehicle noise and vibration behaviour is one of the essential quality and comfort features for the customer. For a target-oriented vehicle development it is important to understand the excitation mechanisms and transfer paths of interior noises. FEV developed in collaboration with Volkswagen Nutzfahrzeuge a method to simulate the transfer of suspension noises using vehicle level measurements.

1 Introduction

The goal of the test and simulation methods is to provide a means to conduct analysis of excitation and transfer paths for road induced noises. Based upon obtaining a detailed understanding of suspension noises in a particular vehicle, target values for vehicle components can be derived to aid the vehicle refinement process. The presented method for suspension noise simulation is based on a combination of "source" measurements (obtained during vehicle operation) and measurements of appropriate "chassis transfer functions". The interior noise produced by suspension is decomposed into separate components corresponding to certain noise path, such as front axle / rear axle or wishbone / McPherson-strut. Afterwards the separate components of the interior noise are reassembled to provide the total noise field due to all suspension related noise shares. This method allows for detailed analysis of the excitation mechanisms and transfer paths including additional details for each path such as body forces, body transfer functions and interior noise contributions. This method is based on targeted acceleration measurements on the vehicle body and chassis. This method does not rely on the use of separate force transducers, as the necessary sensor installation in the power flow is complex and the investigated system would be explicitly distorted depending on adapter geometry.

2 Principle of Chassis Noise Simulation

The simulation of the powertrain induced vehicle interior noise shares (Vehicle Interior Noise Simulation, VINS) is used as standard method in vehicle development process for several years [1], [2]. The chassis noise simulation (Chassis-VINS) extends this method to road induced noise. Unlike powertrain mounts, the chassis mounts/bushings are rather stiff. Therefore the body reaction to exciting chassis components cannot be disregarded. In addition many of the excitation points are stiffly connected on body side, so that the "crosstalk" within the body has to be considered. The induced forces will be determined based on easily measurable accelerations at the coupling locations of body and chassis. It has to be considered that every induced force generates acceleration at all coupling locations. The main item of this method deals with this crosstalk within the vehicle body, with the goal to eliminate accelerations caused by crosstalk. The chassis interior noise simulation interlinks the measured excitations (under real conditions) with laboratory based transfer functions that describe the vehicle's noise transfer behaviour. The excitations are the accelerations at the chassis coupling points during on-road tests (for example rough asphalt, cobblestone pavement). The impact testing (transfer function measurements) provides a matrix, which connects the force applied at a particular point with the acceleration at all coupling points. By means of a matrix inversion process it is possible to deduce the operational forces from the acceleration during operation. In addition, the interior noise is measured during impact testing, so that appropriate body transfer functions can be calculated. The simulation is done in the time domain and hence, generates an audible interior noise. Therefore the simulated noise can be analysed such as direct measured time data. In order to control the simulation precision the simulated interior noise can be compared to the measured interior noise. By separation of the interior noise in its individual parts, this method allows the acoustics engineer to understand the noise generation process. This allows for targeted optimisation of the vehicle body or chassis in the series development process or to solve specific noise problems.

3 Selection of Relevant Transfer Paths

Before application of this method, the relevant transfer paths for the rolling noise have to be determined. In the context of pre-tests on a chassis-dynamometer and on different road surfaces, the influence of stabilisers, steering gear and driveshaft was classified as being not important. For each vehicle side the following seven transfer paths arise, which consist of three-dimensional components

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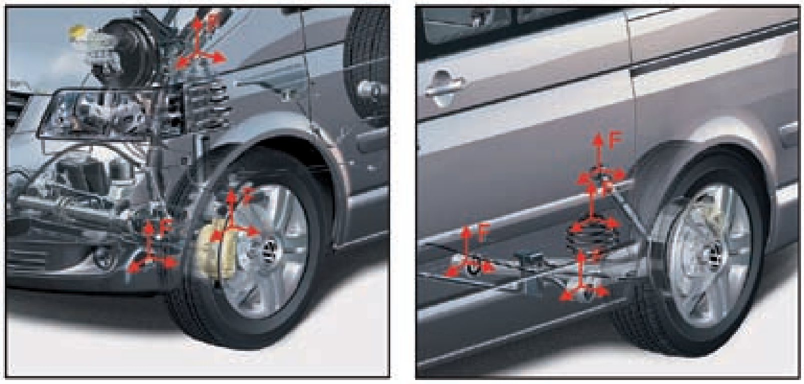

Figure 1: Force induction points at front and rear axle

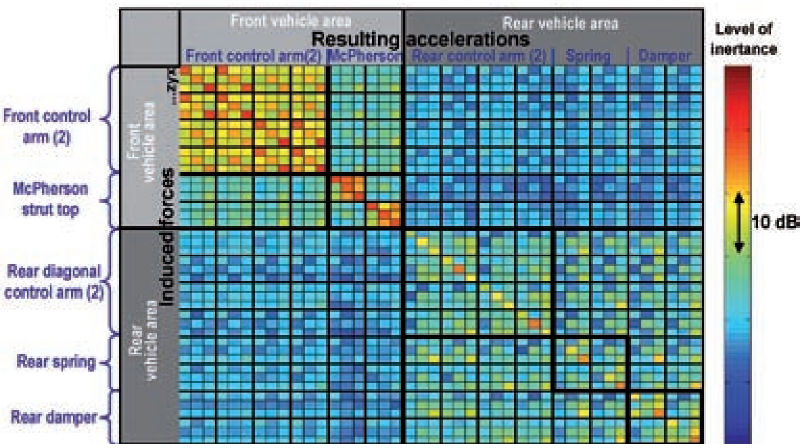

Figure 2: The inertance matrix describes the reaction (acceleration) of the vehicle body at the contact points on force induction at certain contact points. Colour shows average level of inertance function from 11 Hz to 355 Hz

and therefore result in a total of 42 transfer functions for the vehicle, **Figure 1**:
- McPherson-strut bushing
- front transverse link with 2 bushing
- rear diagonal link with 2 bushings
- rear damper
- rear spring.

4 Measurements of Raw Data

The measurement of operational acceleration at the coupling points takes place on a cobblestone pavement. As indicated previously, no additional force transducers are used during the vehicle tests. In a separate step the body transfer behaviour is determined by impact testing: The chassis is disconnected from the test vehicle and the vehicle body decoupled mounted on air cushion. In order to determine the body transfer functions and the crosstalk within the body, the decoupled body was excited by an impact hammer at all coupling locations. In response to force excitation, the acceleration at all coupling points were measured to determine the crosstalk. At the same time the interior sound pressure level at different ear positions in the cabin were measured to determine the corresponding vibro-acoustic transfer functions. The measurement of all relevant accelerations results in a complete inertance matrix, whose elements describe the frequency response functions of each combination of „transmitting" and „receiving points" of chassis to body coupling points. The frequency-selective crosstalk can be graphed, **Figure 2**, by the inertance matrix averaged over certain third octave bands or averaged over the overall frequency range. In Figure 2, the areas of front wishbone suspension show high crosstalk due to the common connecting subframe. In addition, a strong crosstalk was shown within the individual dimensions of suspension strut. Between rear and front axle only a comparatively low crosstalk exists.

5 Simulation of Chassis Noise

In order to determine the body forces out of the accelerations measured on the test course, it is necessary to calculate the dynamic mass matrix (force per acceleration) by inversion of the inertance matrix (acceleration per force). The real forces on the chassis coupling points arise from multiplication of measured acceleration with the dynamic mass matrix. Via subsequent multiplication of the forces with the measured vibro-acoustic transfer

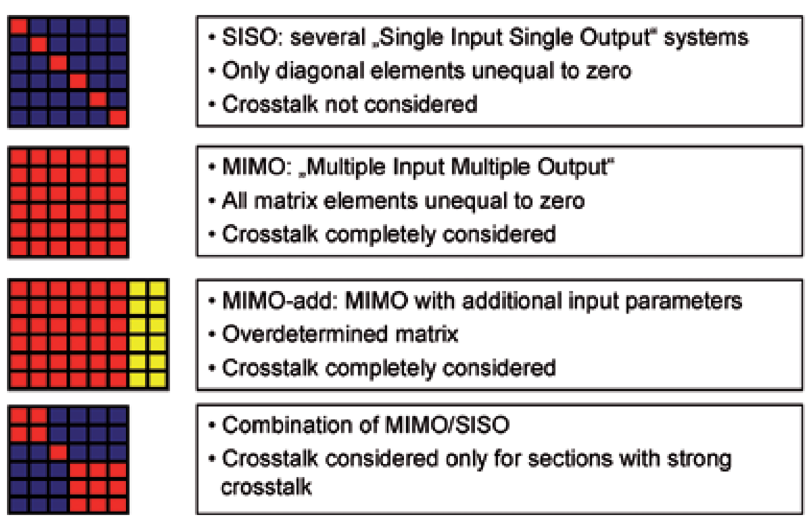

Figure 3: Different options to set up the inertance matrix before inversion

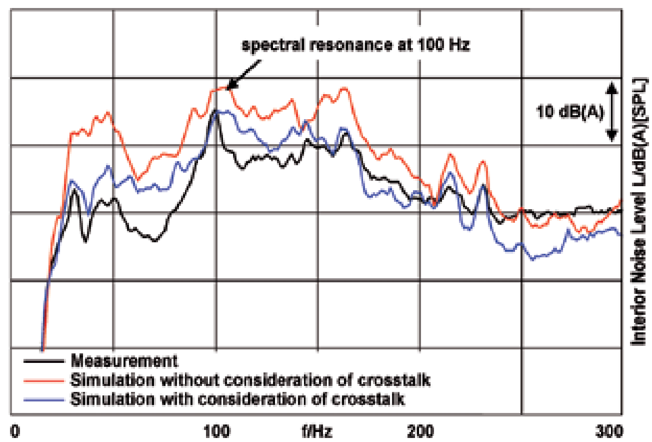

Figure 4: Comparison of measured and simulated interior noise with and without consideration of crosstalk

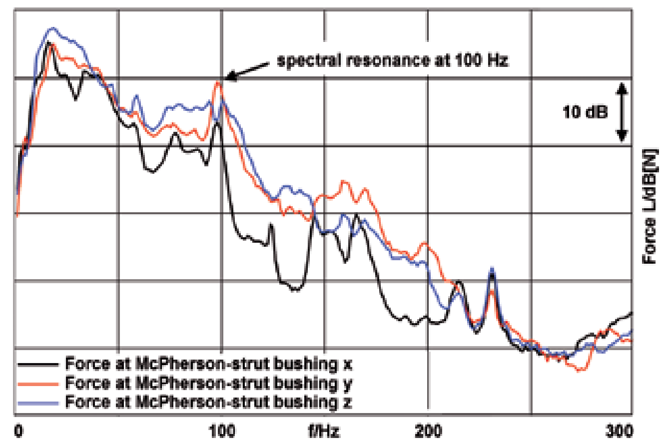

Figure 5: Calculated forces of the McPherson-strut showing a peak at 100 Hz

summed-up with correct phase, provide the total vehicle interior noise from the considered paths. For the inversion of the matrix different approaches can be chosen, **Figure 3**. In the easiest case, without consideration of crosstalk, all matrix elements beside the diagonal ones are set to zero, and the system splits up into subsystems, each having one input and one output parameter (Single Input Single Output, SISO). If all matrix elements are used the crosstalk is completely included (Multiple Input Multiple Output, MIMO). In this example the matrix consists of 42*42 = 1764 frequency dependent inertance functions. The measurement of vehicle body accelerations at additional positions leads to an overdetermined matrix. As an alternative also a combination of the SISO and MIMO variant can be used, which considers crosstalk only in regions where it is strong. Before use of these simulation results there has to be a reconciliation of measuring and simulation. The simulated interior noise was compared with the interior noise of the test track driving, which was measured for checking purposes. The comparison of the simulation, without consideration of body crosstalk, with the measurement shows a significant variance as expected and cannot be used for further analysis, **Figure 4**. The method with crosstalk compensation however shows a significantly improved correlation with the measurements. The characteristic ranges with high levels around 100 Hz and within the range of the tyre cavity modes above 200 Hz are reproduced well with crosstalk compensation. The missing component of the direct airborne tire noise was noted to be negligible in the range up to 300 Hz. By the breakdown of the interior noise in separate transfer paths the dominant paths can be identified and analysed quickly. Paths without level-dominant components can be directly excluded from further investigations with hardware modification. The paths with level-dominant spectral components at the interior noise will be further decomposed into excitation (forces) and transfer (body transfer functions). **Figure 5** shows an example of a spectral peak in the force signal of the body excitation by the McPherson-strut in the range of 100 Hz. The peaks occurring in this frequency range result in a high contribution of these noise paths in the interior noise at 100 Hz. Besides the excitation also the body transfer behaviour can influence the interior noise. However, in this case the vibro-acoustic transfer functions do not contribute to the peak at 100 Hz in the noise paths of the McPherson-strut.

6 Vehicle Modification

In order to validate the simulation, different hardware modifications to the test vehicle were made. In the following a modification of the McPherson-strut path will be examined. The analysis of the simulated noise shares shows that the paths of the McPherson-strut contribute to the interior noise at front seats. As shown in part a of **Figure 6**, the contribution of the strut paths causes a significant effect at front seating position up to 4 dB(A) in a broad frequency range. For the overall noise at the rear seating position these paths do not contribute according to Chassis-VINS. That means that strut-path modifications are expected to have an effect on the noise at the front but not at the rear seating positions. In order to validate these results the strut isolation to body was improved by using additional decoupling elements, in an attempt to reduce the force application at the suspension strut dome. As shown in part b of Figure 6, the measured interior noise confirms the prediction of the Chassis-VINS process. The outcome of this is a reduction of the interior noise level at front seating position and only a minor effect at rear seating position. The frequency range of the reduction is shifted to higher frequencies in comparison to the simulation. This can be traced back to the fact that the decoupling elements used in experimental testing do not possess the linear behaviour as it is assumed in the simulation.

7 Summary

The further development of the vehicle interior noise simulation (VINS) in order to generate chassis interior noise simulation, Chassis-VINS enables the acoustics engineer to better understand the creation and transfer behaviour of road in-

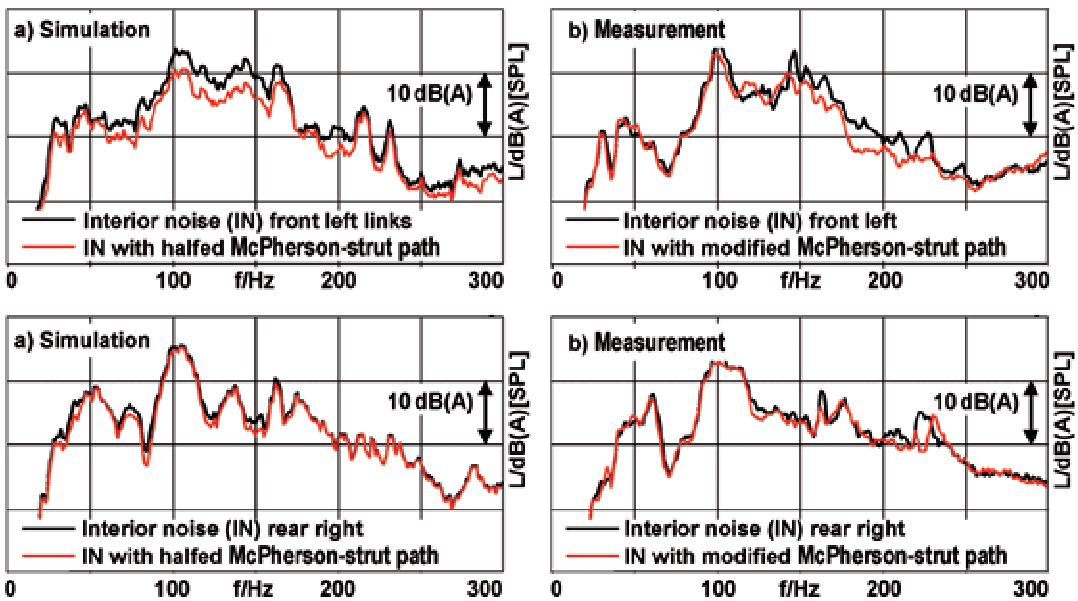

Figure 6: Effect of a modified noise path via the McPherson-strut on the measured and simulated interior noise

duced noises and vibrations. In addition, this method provides the possibility to define and review component target values by measuring forces and transfer functions. As a further advantage the method works without the use of external force transducers that could alter the system as well as prevent certain driving manoeuvres. As well in the early development and testing phase as for the solution of production start-up problems this method provides important information in order to optimise vehicle Noise, Vibration, Harshness (NVH). Therefore, target conflicts between vehicle dynamics and NVH can be handled more effectively with this method. By separate examination of transfer function weaknesses and given force peaks this method assists in achieving an inter-coordinated, consistent excitation and transfer behaviour of each transfer path to the interior noise. The hardware modifications, performed in order to validate the method, confirm the road capability of this method for the development process.

References

[1] Eisele, G.; Wolff, K.; Alt, N.; Hüser, M.: Application of Vehicle Interior Noise Simulation (VINS) for NVH Analysis of a Passenger Car". SAE 2005-01-2514

[2] Alt, N.; Wiehagen, N.; Schlitzer, M. W.: Interior Noise Simulation for Improved Vehicle Sound. SAE-Paper 2001-01-1539