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318 Sound Engineering based on Source Contributions and Transfer Path Results*

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ABSTRACT

Apart from other aspects, the interior sound of a passenger car model has to meet customers expectations. For optimizing the interior sound of a passenger car, target sounds have to be met by sound engineering efforts. For such a sound engineering process it is necessary to reduce on the one hand annoying undesired interior NVH aspects and to create on the other hand the relevant and necessary interior sound to meet the target sound as close as possible.

This can only be performed in optimizing the NVH of the different vehicle noise sources with the knowledge and / or modification of the chassis transmission paths. The necessary NVH and sound optimization of the different vehicle noise sources is quite straight forward, however the prerequisite of the exact evaluation of the chassis airborne and structure vibration transfer characteristics is much more complex. However exact chassis transfer path results are necessary to obtain optimum interior sound results.

In this publication, the benefits and drawbacks of current available experimental transfer path analysis (TPA) procedures will be presented and discussed and improved TPA strategies and their potential will be presented, to optimize and increase the efficiency of the passenger car sound engineering process.

1. CURRENT VEHICLE CHASSIS TRANSFER PATH ANALYSIS PROCEDURES AND RESULTS

The transfer characteristics of a vehicle chassis with respect to airborne noise and structure vibration act as a “filter” for the powertrain vibration excitation and airborne noise radiation. Therefore this “filter characteristic” effects to a large extent the vehicle interior noise / noise quality and vibration, which is a basis for vehicle interior noise / noise quality optimization and sound engineering. Since it is reported from acoustic engineers world wide, that the current commercially available acoustic TPA analysis systems to not always give reliable results, a research project was set up by AVL in cooperation with ACC and IEM to understand in detail and describe with sufficient accuracy the transfer characteristics of passenger car chassis.

The first object of this research program is to analyze passenger cars in very much detail with respect to TPA using in parallel some commercially available “standard” TPA systems.

The “standard” TPA procedure uses the inertance matrix obtained at the mounting positions of the powertrain by force excitation. For calculation of the force inputs at these mounting positions during vehicle operation, measured accelerations and the inverted inertances are necessary [1].

If no structural vibration crosstalk between different degrees of freedom is considered, only the main diagonal of the inertance matrix is required. If the crosstalk within each mount is considered the 3x3 block diagonal matrix and for the overall crosstalk within and between all mounts the full inertance matrix is used for the calculation of the forces under vehicle operation (see Figure 1). The drawbacks of the use of more than the main diagonal is, that numerical problems occur due to the ill conditioned matrix if more and more components of the inertance matrix are taken into the force calculation procedure [2].

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\begin{pmatrix} a_1 \\ a_2 \\ a_3 \\ a_4 \\ a_5 \\ a_6 \\ a_7 \\ a_8 \\ a_9 \end{pmatrix} =
\begin{pmatrix}
In_{(1,1)} & In_{(1,2)} & In_{(1,3)} & In_{(1,4)} & In_{(1,5)} & In_{(1,6)} & In_{(1,7)} & In_{(1,8)} & In_{(1,9)} \\
In_{(2,1)} & In_{(2,2)} & In_{(2,3)} & In_{(2,4)} & In_{(2,5)} & In_{(2,6)} & In_{(2,7)} & In_{(2,8)} & In_{(2,9)} \\
In_{(3,1)} & In_{(3,2)} & In_{(3,3)} & In_{(3,4)} & In_{(3,5)} & In_{(3,6)} & In_{(3,7)} & In_{(3,8)} & In_{(3,9)} \\
In_{(4,1)} & In_{(4,2)} & In_{(4,3)} & In_{(4,4)} & In_{(4,5)} & In_{(4,6)} & In_{(4,7)} & In_{(4,8)} & In_{(4,9)} \\
In_{(5,1)} & In_{(5,2)} & In_{(5,3)} & In_{(5,4)} & In_{(5,5)} & In_{(5,6)} & In_{(5,7)} & In_{(5,8)} & In_{(5,9)} \\
In_{(6,1)} & In_{(6,2)} & In_{(6,3)} & In_{(6,4)} & In_{(6,5)} & In_{(6,6)} & In_{(6,7)} & In_{(6,8)} & In_{(6,9)} \\
In_{(7,1)} & In_{(7,2)} & In_{(7,3)} & In_{(7,4)} & In_{(7,5)} & In_{(7,6)} & In_{(7,7)} & In_{(7,8)} & In_{(7,9)} \\
In_{(8,1)} & In_{(8,2)} & In_{(8,3)} & In_{(8,4)} & In_{(8,5)} & In_{(8,6)} & In_{(8,7)} & In_{(8,8)} & In_{(8,9)} \\
In_{(9,1)} & In_{(9,2)} & In_{(9,3)} & In_{(9,4)} & In_{(9,5)} & In_{(9,6)} & In_{(9,7)} & In_{(9,8)} & In_{(9,9)}
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\begin{pmatrix} F_1 \\ F_2 \\ F_3 \\ F_4 \\ F_5 \\ F_6 \\ F_7 \\ F_8 \\ F_9 \end{pmatrix}
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Figure 1: Inertance matrix with indication of main diagonal, three 3x3 block diagonal and all inertance components

As a basis for a first TPA analysis with current available TPA procedures, an extensive vehicle measurement set up was defined, inheriting all necessary transducers and shaker excitation points at all powertrain mount positions in x, y, z direction and an adequate number of microphones in the engine and passenger compartment. Additionally a large number of transducer and excitation positions were defined for an extensive sensitivity analysis and for an over determination of the inertance matrix with respect to chassis structure vibration and airborne noise [3]. This sensitivity analysis aims at the possibilities to evaluate the accuracy of the results obtained, to evaluate in detail the chassis response (with respect to crosstalk, etc.) and as a basis for the set up of a refined strategy and methodology for an optimized TPA.

As a first step the TPA results obtained with three commercially available acoustic TPA systems on one mid size passenger car powered with a 4 cylinder Diesel engine according to their individual analysis procedures will be presented and discussed. The TPA results obtained with all three systems are based on exactly the same time signals and should therefore yield very similar results for one and the same passenger car. The transfer characteristics from the engine mounts to the vehicle interior have been measured by using minishaker excitations and the airborne noise transfer characteristics by using volume source excitations in the engine compartment. An overview of the TPA results for the narrow band interior 2nd order noise over engine speed at full load in 3rd gear of this passenger car, can be seen in Figure 2.

The results shown in the upper part in Figure 2 are the 2nd order airborne noise and structure vibration contributions from the powertrain and intake / exhaust system to the interior. In the lower part of Figure 2 one can see the comparison between the actual measured 2nd order interior noise level and the overall 2nd order interior noise level calculated from all structure and airborne noise transfer paths by TPA. The results obtained are quite different between the measurement systems and the overall 2nd order interior noise contribution do not really fit the actually measured 2nd order noise.

In Figure 3 a similar 1/6 octave result is shown for the same car for an engine speed of 3850 rpm at full load in 3rd gear. Here the overall interior noise is calculated from the structure vibration and airborne noise TPA and compared with the actually measured overall inte-

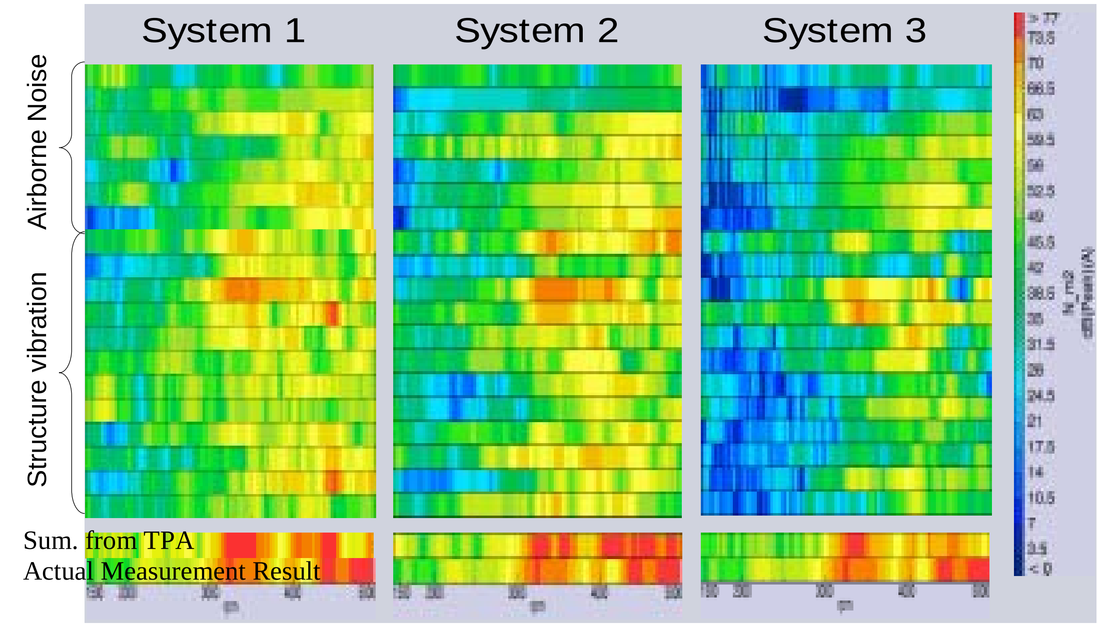

Figure 2: 2nd order TPA results of one mid size passenger car using three different analysis systems (upper part) and comparison of overall interior 2nd order TPA noise results actually measured and obtained via TPA (lower part)

rior noise level. Again the results obtained are quite different between the measurement systems and do not really fit the actually measured overall interior noise level. Deviations of up to more than 10 dBA can be recognized at certain frequency bands.
Such differences in path contribution results obtained with different TPA systems would actually impede any noise / noise quality or sound engineering process on a vehicle.

Using the so called “reciprocal” excitation by a volume source in the vehicle interior and analyzing the resulting accelerations at the powertrain mount positons one can obtain a transfer function with the dimension ($1/m^2$) from the volume acceleration (m^3/s^2) to the vibration acceleration (m/sec^2) which has the same dimension as the force (N) to interior noise (N/m^2) transfer function. Apart of the high excitation energy needed for producing adequate vibration accelerations (especially for premium passenger cars) at the powertrain mount positions there is no force information at the powertain mounts available with this approach.

For vehicle noise / noise quality improvement and sound engineering, however the effect of the magnitude distribution of the powertrain excitation forces is of considerable importance since their modification is one option for interior noise quality modification and optimization.

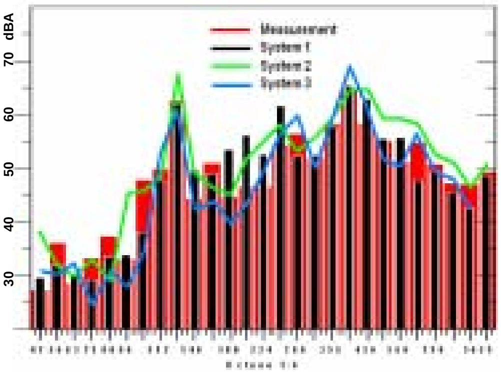

Figure 3: 1/6 octave interior overall noise TPA result at 3850 rpm, full load, 3^{rd} gear for one mid size passenger car using three different measurement systems compared with the actual measured overall interior noise level

2. RESULTS OF THE SENSITIVITY ANALYSIS

To understand in more detail the transfer characteristics of a vehicle chassis the before mentioned sensitivity analysis was performed. One aim was to analyze possible errors in the measurement set up, the extent of the crosstalk within each engine mount and between all engine mounts as well as the overall response of the vehicle chassis for further TPA development.

When determining the transfer characteristics, the vehicle chassis vibration excitation can be performed by impact hammer or miniature shaker according to the required force input and space available [1,2]. In todays engine compartments space is normally very limited so small errors in x, y, z excitation direction are sometimes inevitable. To estimate the effect of such an excitation direction error, according results can be seen in figure 4 with respect to the frequency response function (FRF) from one engine mount to interior noise under the two different angles of 0° and 15° of shaker excitation. As can be seen, such a deviation of 15° in excitation direction which can easily happen also unintentionally under severe space restrictions in the engine compartment, yields errors of nearly up to 10 dB in the FRF at a number of frequency bands.

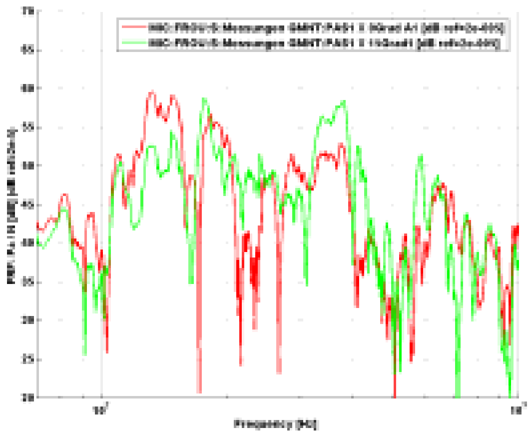

Figure 4: FRF from one engine mount to interior noise under two different angles of shaker excitation (0°, 15°)

Secondly to understand and obtain the magnitude of the crosstalk within one powertrain mounting position and between all powertrain mounting positions in x, y, z direction a number of tests were performed and some results are shown in figure 5 and 6. In figure 5, the crosstalk energy at one powertrain mount which is transferred in the two other directions compared to the energy transfer in excitation direction is shown. Zero dB indicates that the same energy as the input energy is transferred in the other two directions. A positive dB number indicates more crosstalk energy and a negative dB number indicates less crosstalk energy is transferred into the other two directions. As can be seen in figure 5 the crosstalk within each powertrain mounting position has a magnitude of up to +10 dB (see left side of figure 5) which induces an error in the TPA calculation if this crosstalk is not considered also of up to 10 dB for some frequency bands. (see right side of figure 5)

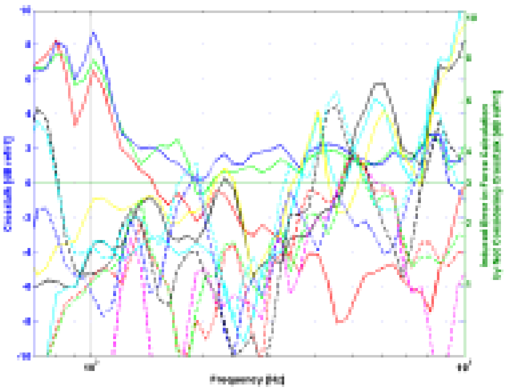

Figure 5: Crosstalk at chassis side engine mount position within each of all powertrain mounting positions in x, y, z direction

In figure 6, zero dB indicates the same crosstalk energy is transferred in all the other remaining powertrain mounts in x, y, z direction as in the excitation direction at one powertrain mount.

As can be seen in figure 6 the crosstalk between the excitation and the remaining powertrain mounting positions in x, y, z direction has a magnitude of up to +4 dB (see left side of figure 6) which induces an error in the TPA calculation if this crosstalk is not considered of up to 5 dB. (see right side of figure 6)

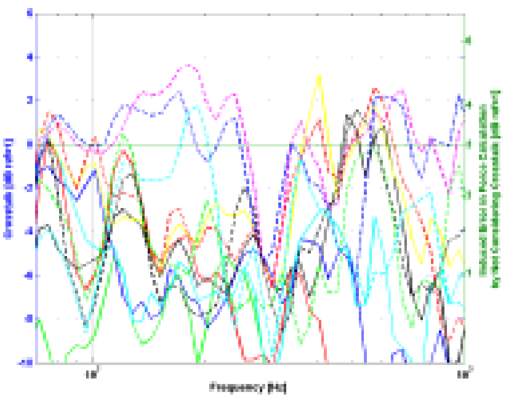

Figure 6: Chassis crosstalk between 4 powertrain mount positions in x, y, z direction

The results presented in figure 5 and figure 6 strongly enforce the consideration of crosstalk for the “standard” TPA analysis. This consideration however leads in vehicle application to numerical problems due to ill conditioned inertance matrices. This is mainly due to the number of antiresonances in measurement results at frequencies where the background noise level is reached and the coherence becomes low. Over all measurement results these errors are more or less even distributed over the whole frequency range. Here the errors from measurement data may be even amplified and lead generally to errors in the contribution determination. Therefore a method considering crosstalk without generating these numerical errors is one of the goals of our research project.

Due to differences in the path contributions determined by the different “standard” TPA systems in a first step a “standard” TPA verification approach is currently performed where operational excitation forces are simulated and therefore exactly known since they will be in parallel applied by a number of shakers instead of the powertrain excitation under operation. This allows a further detailed investigation of the results obtained by different TPA systems and will give valuable input for developing an optimized TPA procedure.

3. APPROACHES TO OBTAIN OPTIMUM VEHICLE CHASSIS TRANSFER BEHAVIOR RESULTS

Apart from the “standard” TPA procedure presented in detail with its benefits and drawbacks a number of other procedures are feasible to obtain the transfer behavior of a vehicle chassis.

In figure 7 an extensive summary of possible analysis procedures to obtain the vehicle chassis transfer characteristics and its verification methods are listed. Apart from the verification methods it can be seen that the analysis methods can be divided into two groups. The first group uses force calculations to determine the contributions of the transfer paths (also including the “standard” procedure). The second group calculates the path contributions by only concentrating on the measured accelerations at chassis and powertrain mounting positions. For some of these methods even only measurements under vehicle operation are sufficient [4]. An extensive comparison between the methods listed in figure 6 concerning the detailed results obtained for path contributions and the possible difference between simulated and measured results has not been published yet.

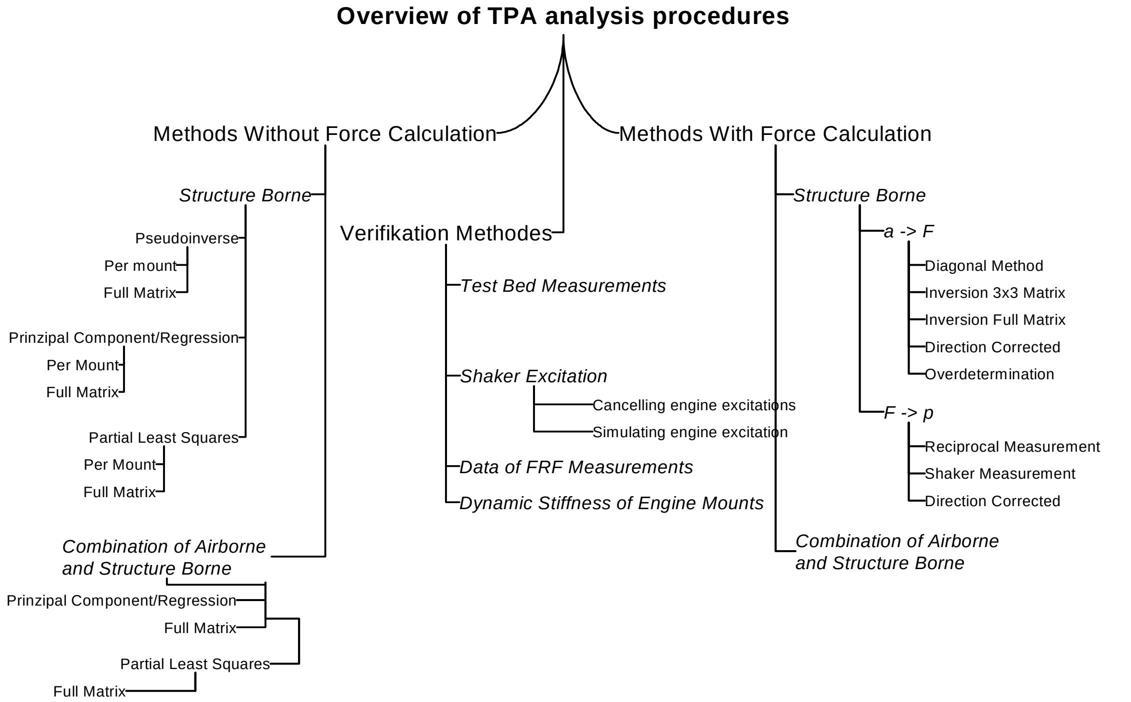

Figure 7: Overview of a number of procedures to obtain chassis transfer characteristics

Having shown the strong influence of a 15° deviation in excitation direction on the obtained FRF (see figure 4), the development of new FRF determination procedures that are based on a number of arbitrary excitation directions is under investigation [5]. In this new approach more equations will be available than unknown variables to determine the exact magnitude and direction of the force input at

each powertrain mount. Such a method would lead to more precise results with respect to excitation errors in everyday work even without changing the “standard” TPA procedure.

As an alternative approach, a purely vibration acceleration based procedure can be developed. The acceleration data can be obtained either in FRF measurements or under real or simulated vehicle operation conditions. One preliminary example is given in figure 8 comparing the actual measured 2nd order interior noise level and the prediction based on a pure real operational acceleration based method [4]. In figure 8, it can be seen that the results are quite promising. Similar to the drawbacks mentioned in the reciprocal approach in chapter 1, no information about the operational forces at all powertrain mount degrees of freedom can be obtained by this result. As mentioned before, here we loose a valuable input for vehicle sound engineering. To obtain these force data necessary for vehicle sound engineering, we have to come back to the inertance relation between force and acceleration which – in this case - only serves for an a-posteriori force determination and not for the calculation of the actual interior noise contribution. This approach is also currently under investigation in our research project.

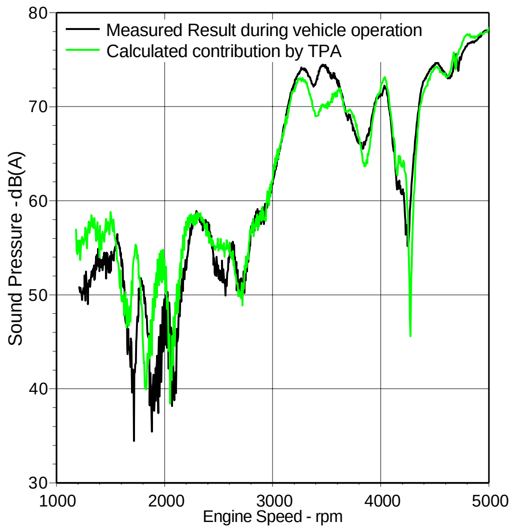

Figure 8: Actual 2nd order interior noise level compared to 2nd order interior noise level obtained by an acceleration only based TPA

Furthermore, a robust and preferably application-specific method for the inversion of the inertance matrix can be further utilized. For this, the above mentioned method for exact excitation force determination which can be regarded as a special constrained inversion procedure is currently also investigated in combination with high-sophisticated mathematical tools like partial least squares for improving as fare as possible the ill condition status of the inertance matrix.

4. SUMMARY / CONCLUSIONS

Since reliable results for transfer paths and excitations forces are necessary for effective vehicle noise / noise quality improvement and sound engineering a research program has been set up to develop methodologies which will meet these targets. In this publication, first of all, the pro and cons of current TPA procedures are discussed and presented by actual results obtained on passenger cars. Further on, first possibilities to refine the current TPA approaches and / or to define and develop new transfer characterization methodologies which are currently under investigation and development have been presented. It is intended that the final outcome of this research project will be presented in the near future.

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