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Interior Noise Simulation for Improved Vehicle Sound

Norbert W. Alt, Norbert Wiehagen and Michael W. Schlitzer
FEV Motorentechnik

ABSTRACT

In the recent past, interior noise quality has developed into a decisive aspect for the evaluation of overall vehicle quality. At most operating points, the dominating interior noise share is generated by the powertrain.

Interior noise simulation is a new tool for upgrading interior noise. Based on measurements of air- and structure-borne noise excitations caused by the powertrain, the interior noise shares are determined by applying the properties of the transfer paths. By superimposing the individual interior noise shares, the overall interior noise can be predicted.

Well before the engine is operated in the vehicle for the first time, annoying interior noise shares, their causes and their transfer paths can be identified by subjective and objective analysis. This enables the engineer to focus on vital optimization measures as to excitations occurring at the engine as well as to transfer paths in the vehicle.

This paper describes how interior noise simulation can be implemented to improve vehicle acoustics. Aside from the identification of excitations of the pre-optimized engine, procedures for the determination of well-shaped air- and structure-borne noise transfer functions are described.

The interior noise simulation results are compared with the real interior noise and serve as a base for the evaluation of acoustic weak points of the engine and transfer behavior of the vehicle. For noise shares which impair interior noise quality, the dominating transfer path and the extent to which excitation and/or transfer path contribute to the problem are identified. For a component optimization conducted at the powertrain test bench, the influences on interior noise are shown by means of simulation.

Finally, future development trends for simulation and calculation of interior noise are discussed.

INTRODUCTION

For reasons of environmental friendliness, today's motor vehicles are subject to many legislative restrictions concerning pollutant and noise emissions. As for noise emissions, the pass-by level of a passenger vehicle at full load must not exceed 74 dBA or - in case it is driven by a direct-injecting diesel engine - 75 dBA, respectively. Aside from fulfilling these requirements, the acoustic engineer needs to create a distinctive exterior noise characterized by multiple attributes - corporate identity - as well as a pleasant interior noise conveying high riding comfort. In addition to that, the interior noise should coincide with the driver's expectations such as a powerful sound during the acceleration phase.

The interior noise generated by the powertrain predominates at many operating points and influences the aural impression decisively. Being able to simulate this interior noise already at an early stage of the development process of a vehicle (Figure 1) is an indispensable tool for increasing the efficiency of the engineering work on a high acoustic level. The simulation is based on measured values obtained from the new engine and body or data retrieved from data banks of comparable engines and vehicles.

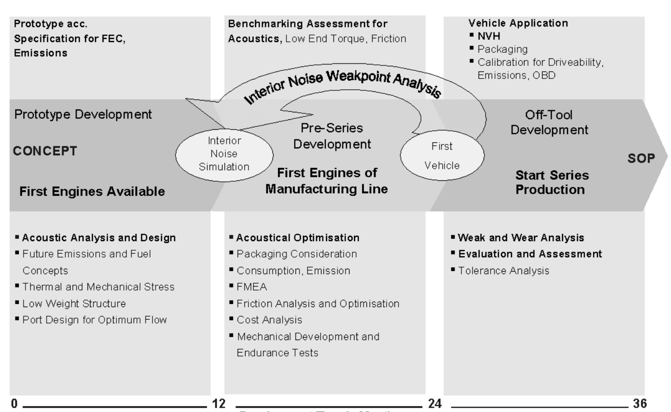

Figure 1: Integration of Interior Noise Simulation into Development Process

A number of valuable investigations have already been published in this context, particularly with regard to the acoustic behavior of the vehicle body. The empirical determination of the main body modes in the lower frequency band by Adam et. al. /1/, the more analytical approach by Kropp et. al. /2/ and the physical description of the interaction of specific body vibrations and standing interior pressure waves by Frappier et. al. /3/ are worthy mentioning. Williams et. al. /4/ made a decisive step toward interior noise simulation by an exact analysis of the interactions leading to an interior noise complaint. Genuit /5/ suggests a procedure for interior noise simulation that facilitates a binaural presentation which, however, is not as much concerned with engine acoustic measures as with sound design issues. The use of interior noise simulation for the evaluation of engine technology measures has been described by Wiehagen /6/.

The interior noise simulation method discussed in this paper originates from the experience of NVH engine development and contains many new features compared with the state-of-the-art.

PROCEDURE FOR INTERIOR NOISE PREDICTION

Interior noise is generated by a series of complex mechanisms. The individual noise excitations are transmitted from their source to the passenger compartment by a complex, multi-dimensional transfer system. The attempt to describe all components involved in the chain of noise generation is still unproductive. Therefore, the prediction method introduced here is based on an empirical approach.

Vehicle interior noise - depicted in Figure 2 as the center - is determined by a multitude of parameters. Aside from the transfer behavior of the structure adjacent or connected to the passenger compartment, there are a number of excitations that, depending on the operating range, contribute differently.

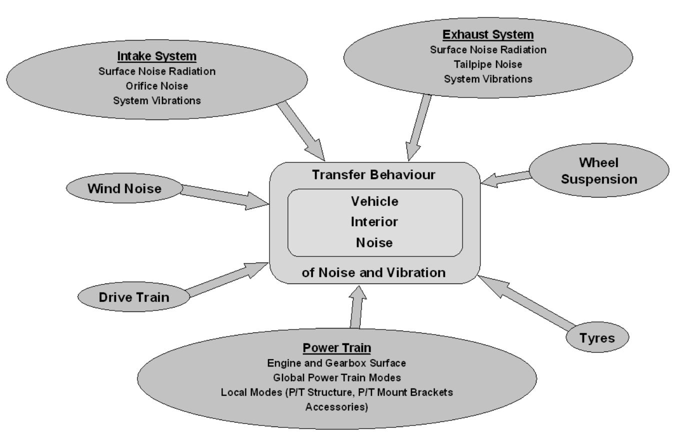

Figure 2: Interior Noise Sources

While wind noise and noises caused by wheel/road contact are important at medium and high speeds, excitations caused by the powertrain, or more precisely engine, transmission and auxiliaries, are predominant at moderate speeds. In addition to that, there are excitations of the intake and exhaust system which, through surface radiation, orifice noise as well as structure vibrations, contribute to the interior noise. Other interior noise contributing sources and transfer paths are the drivetrain and the chassis.

It is inefficient for an empirically based prediction of interior noise, even when using state-of-the-art signal processors, to consider all excitations and transfer paths. By focusing only on the essentials (powertrain including intake and exhaust system and their transfer behavior) interior noise simulation can work fast and efficiently and is therefore a valuable tool for the evaluation of acoustic improvements.

The excitation parameters considered by FEV's interior noise simulation method are based on standard test bench measurements of noise radiated by the engine and the auxiliaries, as well as surface radiation and orifice noise of the intake and exhaust system. Furthermore, the decisive structure-borne noise excitation at the powertrain mounts is recorded. If necessary, the vibration excitation of the intake and exhaust system are also included.

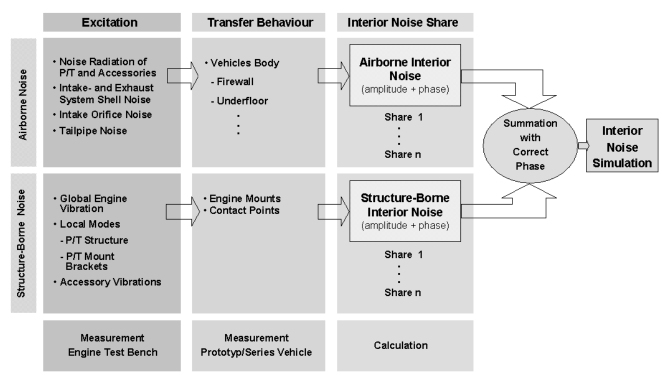

Figure 3: Noise Sources and Corresponding Transfer Paths

The individual interior noise shares are transmitted through numerous paths. Figure 3 shows a schematic classification of the interior noise generation mechanisms. The total airborne noise excitation is made up of a multitude of individual sources Thus, a multitude of individual airborne interior noise shares can be discerned. In particular, the noise radiated by the drivetrain, the auxiliaries, the surfaces of the intake and exhaust system, as well as the orifice noises are major excitations. Predominating transfer paths of the airborne noise are the firewall and the underbody.

The generation of structure-borne noise can be examined analogously. The excitation of the powertrain, that is its global and local vibrations, is amplified mainly by the brackets, the auxiliaries as well as the exhaust system and is impressed at the joints between powertrain and body. Predominating transfer is via the powertrain mounts. For interior noise simulation, the individual powertrain excitations and the transfer paths of the prototype or the production vehicle are measured separately. Subsequently, the interior noise shares are calculated. While considering the phase of noise harmonics (engine orders), all noise shares are summed up. Thus, the interior noise caused by the powertrain can be evaluated. All calculations concerning the transfer functions as well as the noise shares are complex, that is both amplitude and phase must be considered.

For the experimental determination of excitation, noise and vibrations, only standard requirements for acoustic measurements need to be fulfilled. The experimental determination of the transfer behavior, however, is a demanding task and simplifications based on previous experience must be allowed, if the simulation procedure is to stay feasible. In the following, the determination of the transfer properties of the airborne and structure-borne noise paths will be discussed in greater detail.

AIRBORNE INTERIOR NOISE

To simulate the airborne interior noise share, the noise radiation of the powertrain and the transfer function between engine and passenger compartment must be known. In the left part of Figure 4, the conventional approach to the determination of the airborne noise transfer function is outlined: with the powertrain demounted, a spherical loudspeaker radiates a suitable excitation signal into the engine compartment. Simultaneously, the interior noise thus caused is recorded binaurally, for instance at the driver's position, by means of an artificial head. Alternatively, the excitation can be induced with the engine mounted in the vehicle by a miniature loudspeaker which for example is operated all around the powertrain on the six sides. Aside from the disadvantage of limited excitation intensity at lower frequencies, this method also incurs a six fold higher measurement expenditure due to the sequential processing of the individual powertrain sides.

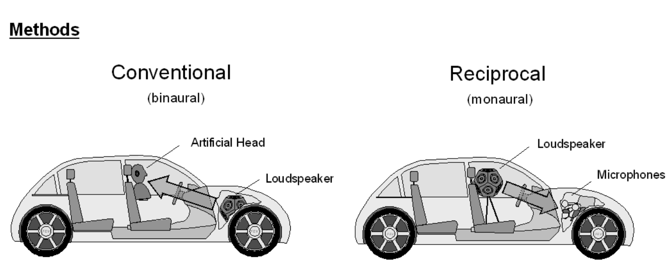

Figure 4: Procedure for Determination of Airborne Noise Transfer Functions

The reciprocal procedure implemented by FEV stands out due to its considerably reduced measurement effort. The procedure is based on Lord Rayleigh's reciprocity law according to which transmitter and receiver can be exchanged in order to determine the transfer behavior of a system, if certain boundary conditions are fulfilled. Consequently, while considering Rayleigh's boundary conditions, the measurement set-up depicted in the right part of Figure 4 can determine the transfer behavior. The excitation, now in the passenger compartment, is caused by means of a loudspeaker, while the response, now in the engine compartment, is measured on the six sides of the powertrain.

Despite efficient determination of the transfer functions, the reciprocal procedure is limited by the fact that it cannot determine binaural transfer functions. Binaural auralization, however, is generally thought of as essential in the field of comparative subjective evaluation of actual interior noises in the audio lab which is also verified by FEV's experience of many years. The purpose of interior noise simulation is, however, to make out acoustic development goals, optimization potentials and weak points of the powertrain and the vehicle at a very early development stage. At that time, uncertainties due to the early prototype stage of the powertrain and the partially imprecise transfer functions are considerably larger than interaural differences.

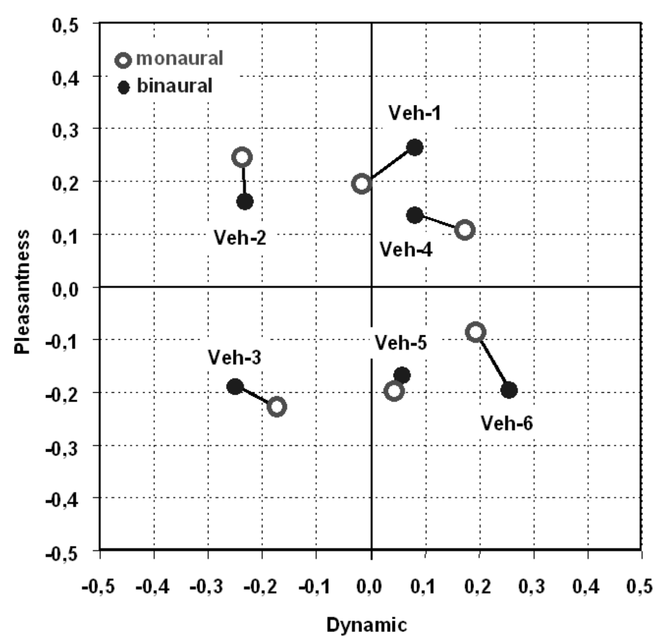

Figure 5: Comparison of Subjective Evaluation Results During Monaural and Binaural Presentation

To quantify the influence, investigations have been conducted at FEV with monaural and binaural presentations, respectively. Figure 5 shows the results with regard to the main criteria "pleasant" and "dynamic" of the interior noise caused by the powertrain. The subjective evaluation of the noise recordings during FEV's standardized test cycles differs only by approximately 0.1 points. However, in individual cases, larger differences also occurred. These were caused by distinct interaural differences due to standing waves in the passenger compartment. Such influences can be taken care of later during detail optimizations of the complete vehicle. To consider these influences during the development phase of the powertrain is, however, inefficient.

To determine the airborne noise transfer function according to the reciprocal method, FEV uses the maximum length sequence (MLS) measuring technique. In a PC-based measuring system with specialized software for generating the maximum length sequence and calculating the transfer function, the disturbing noise shares are eliminated by correlating a transmitted signal which is known with the received signal. Using the maximum length sequence method, the signal to noise ratio can be increased by 40 to 60 dB.

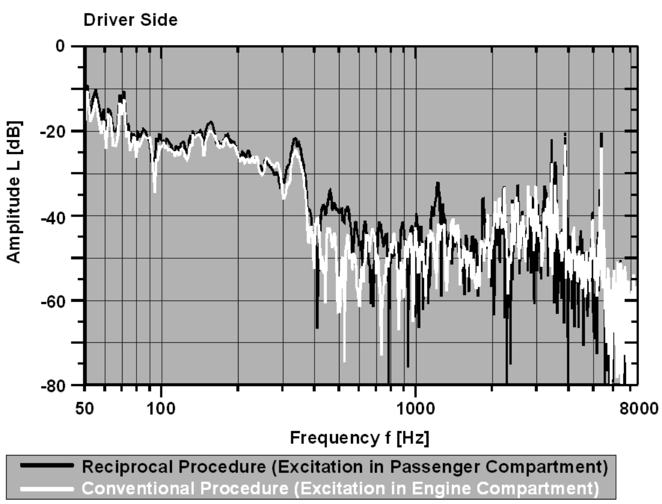

Figure 6: Comparison of Conventionally and Reciprocally Determined Airborne Noise Transfer Functions

Figure 6 compares the results of the reciprocal measurement with those of the conventional measurement by means of frequency curves of the airborne noise transfer function. While considering Rayleigh's boundary conditions, both procedures were used with the powertrain demounted to determine the transfer function from one point in the engine compartment to the driver's head position. The frequency curves correlate extremely well. Obvious differences in the frequency range around 500 Hz and 1300 Hz are less significant than differences that may occur when two bodies of identical design are compared.

STRUCTURE-BORNE INTERIOR NOISE

The powertrain mounts are among the main transfer paths of structure-borne interior noise shares. Establishing the transfer behavior of the components involved (powertrain mount and body) poses the most challenging task in interior noise simulation.

The generation process of structure-borne interior noise is represented in the upper half of Figure 7 as a transfer system model. It shows the excitations represented by the cranktrain with its gas and mass forces, the interface mount/body and finally the radiation of the structure-borne interior noise by the body. According to system theory, the interface between mount element and body is divided into mount transmissibility and apparent body mass.

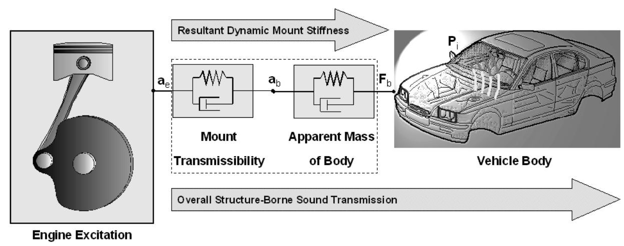

Figure 7: Model for Generation of Structure-Borne Interior Noise Share

The mount transmissibility describes the ratio of the accelerations in front of and behind the mount. The acceleration behind the mount is not only dependent on the transfer properties of the mount but also on the stiffness and the apparent mass of the attached system (the body), respectively. For a full description of the mount properties, the apparent mass has to be determined preferably at the location of the powertrain mount /7/. With the procedure implemented here, the force at the body can be calculated from the engine acceleration at the powertrain mounts. This brings the advantage of avoiding highly expensive direct measurement of the forces at the mounts, as will be explained later on.

The mount transmissibility is determined on a vehicle dynamometer by measuring the accelerations on both sides of the mount, that is on the engine and body side (a_{engine}, a_{body}). From the difference spectrum, the mount transfer function is calculated. To determine the apparent mass of the body at the mount location, the body is excited by a force (F_{body}) and the resulting acceleration (a_{body}) at the body-side mounting point is measured. By multiplying the mount transfer function with the apparent mass, the effective mount stiffness F_{body}/a_{engine} is determined. The vibro-acoustic transfer function, necessary for determining the overall structure-borne noise transfer function, is calculated as the ratio of the noise caused in the passenger compartment to the exciting force.

The signal processing of the mount accelerations measured on a vehicle dynamometer has to fulfill special requirements. While at the mounts the engine acceleration is almost exclusively caused by the powertrain, the body accelerations include a substantial number of interfering shares. Aside from accelerations transmitted by the inspected mount, other accelerations also occur - for instance accelerations transmitted through other mounts and/or excited by the chassis.

The accelerations transmitted by the mount can be roughly divided into shares coherent with the excitation and non-coherent interfering shares.

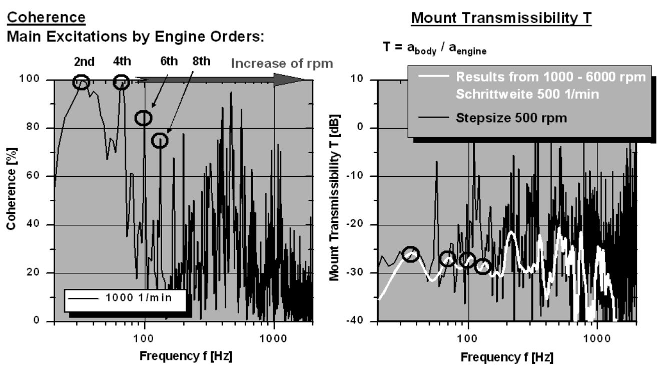

Figure 8: Coherence Based Procedure for Determination of Mount Transmissibilty

Figure 8 shows FEV's coherence based procedure for the determination of mount transmissibility. The left part of the figure depicts the coherence of body acceleration with engine excitation for 1000 min^{-1} which clearly shows that only at frequencies corresponding to the 2nd, 4th, 6th and 8th engine order, that is in case of high vibration excitation, high coherences occur. Only for these frequencies, the mount transmissibility can be determined with the required precision (see mark in Figure 8).

To obtain a sufficient number of points with high coherence, the acceleration measurements are conducted at several stationary speeds, for instance every 500 min^{-1}. As a result, up to 1 kHz, that is even beyond the frequency band of dominating structure-borne interior noises, high-coherent points with a good frequency resolution of 16 Hz are obtained. From the acceleration ratios at these frequencies, the mount transfer behavior is determined as illustrated in the right diagram of Figure 8. Compared to the transmissibility measurement only at 1000 min^{-1}, the shortcomings of measurement without coherence examination become apparent: Considerable interferences lead to significantly banked mount transmissibility.

The procedure for the measurement of the apparent body mass at the powertrain mount is outlined in Figure 9. Simultaneously, the excitation force applied to the body with an impulse hammer and the resulting acceleration are measured at the same location. Also in this case, an examination of the coherence of the excitation and the response is compulsory for the signal analysis.

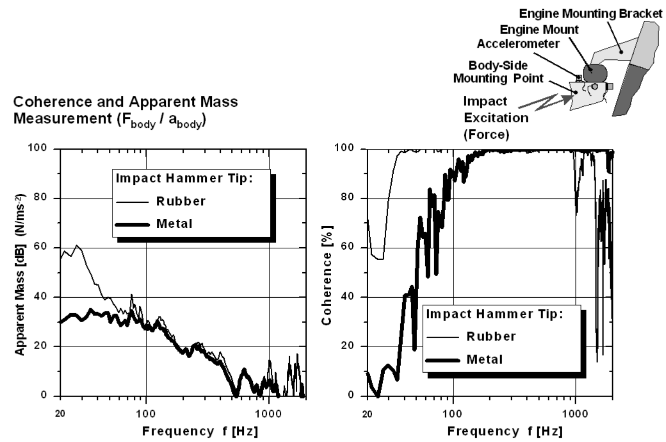

Figure 9: Influence of Hammer Excitation on Determination of Apparent Body Mass at Powertrain Mount

The comparison of coherences of excitations with a steel hammer tip and a rubber hammer tip shows that, for the steel tip, sufficient coherences exist only above 100 Hz, whereas for the rubber tip, the limit is pushed down to 30 Hz. However, in the higher frequency range, the steel tip shows significantly higher coherences.

The frequency curve of the apparent body mass at the mount and the influence of the hammer material is depicted in the right diagram of Figure 9. In the shared frequency range of high coherences, the frequency curves match well. Considering the results of the coherence examinations, the apparent mass is determined by using a rubber tip in the lower frequency range and a steel tip in the higher frequency range.

Mount transmissibility and apparent body mass together lead to the resultant dynamic mount stiffness. Direct measurement on a vehicle dynamometer by means of force sensors is extremely costly with respect to preparation and implementation and therefore not recommended. However, in order to verify the method used by FEV for the determination of the resultant dynamic mount stiffness, measurements of the acceleration excitation at the powertrain and of the force transmitted into the body were conducted. The resultant mount transfer functions determined by either method are shown in Figure 10. If the previously described special requirements on signal processing are considered, the results of this practicable method for the determination of mount transmissibility and apparent body mass are almost as accurate as those of the highly expensive and in many cases impracticable force measurement. The simple measurement yields good results, perhaps a smoothing of the frequency curve would be advisable.

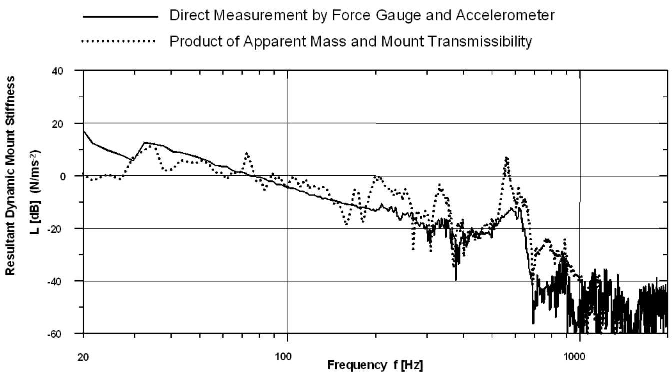

Figure 10: Comparison of Calculated and Measured Resultant Dynamic Mount Transfer Functions

From the acceleration at the powertrain mounts, the force onto the body is determined by means of the resultant mount transfer function. For the resulting interior noise, the vibro-acoustic transfer function of the body needs to be ascertained.

Using a hammer impact to determine the apparent masses, the hammer force and the very acceleration at the impact location are measured. By an additional measurement of the impact-generated interior noise, the vibro-acoustic transfer function can be determined.

For the entire structure-borne noise path, the noise transfer behavior is calculated from mount transmissibility, apparent mass and body connected in series. The resulting frequency curve is shown in Figure 11. High peaks occur up to approximately 120 Hz. Above approximately 500 Hz, total transfer is considerably diminished; the typical low pass character of the structure-borne noise transfer becomes obvious.

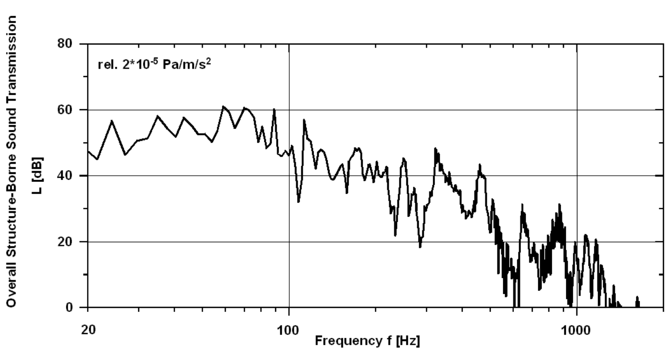

Figure 11: Complete Transfer Function of Structure-Borne Interior Noise

APPLICATION OF INTERIOR NOISE SIMULATION

After the transfer functions as well as the airborne and structure-borne noise excitations have been measured, the interior noise can be simulated by means of specific filtering software. For this purpose, FEV has developed a software based on the standard tool Matlab.

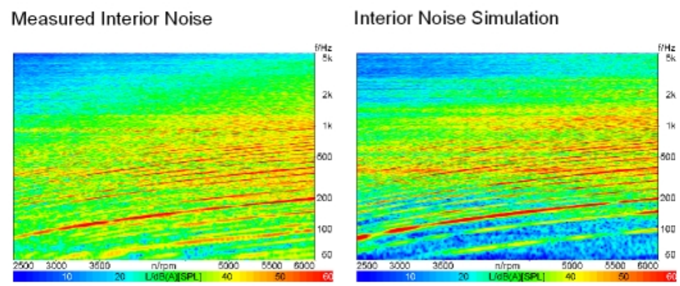

Figure 12: Real vs. Simulated Interior Noise

Figure 12 shows the comparison of the simulated interior noise with the real interior noise by Campbell diagrams. In both diagrams, the signal components of the 2nd engine order that predominate the sound hearing impression and the relatively high levels between 200 and 450 Hz, 550 and 700 Hz and at 1100 Hz are clearly visible. Significant differences between real and simulated noise occur in the frequency range below approximately 100 Hz, mainly due to excitations and transfer paths that have not been considered, such as the drivetrain and the chassis.

Concerning the powertrain induced noise share, aural comparison yields very good correspondence between real and simulated noise, even if - of course - equality cannot be expected. However, experienced NVH specialists can still draw accurate conclusions from this and at an early stage make suggestions as to necessary noise reduction measures regarding powertrain excitation and transfer paths, respectively.

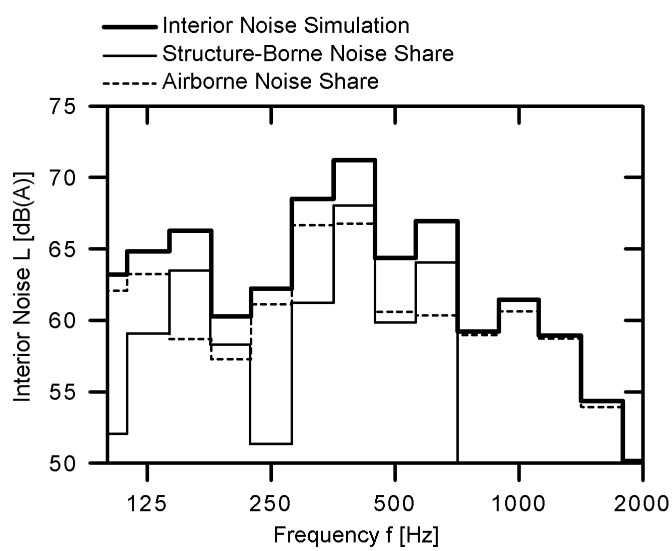

Figure 13: Airborne and Structure-borne Noise Shares in Total Interior Noise

By means of simulation, interior noise shares can even be quantified. Figure 13 compares the airborne and structure-borne noise shares with the total interior noise as third-octave spectra normalized and averaged over a speed run-up.

In the frequency range up to the 630 Hz third-octave, airborne and structure-borne noise shares determine the interior noise to a similar degree. Unusually high levels occur in the 400 Hz third-octave due to an increase in airborne noise, but mainly due to the structure-borne noise share which will be illustrated hereafter. At higher frequencies, the airborne noise share predominates.

One important feature of the interior noise simulation procedure is that the interior noise shares which are transmitted for instance over six airborne and 12 structure-borne noise paths (four induced mounts, three spatial directions) are individually available. Thus, in-depth analyses show for the noise share of mount no. 1, x-direction, a distinct peak in the 400 Hz third octave (Figure 14, upper part). By comparing the engine vibration excitation and the transfer function with the structure-borne interior noise, the excitation peak displayed in the lower part of Figure 14 can be identified as the source. It is caused by a resonance of the P/T mount bracket.

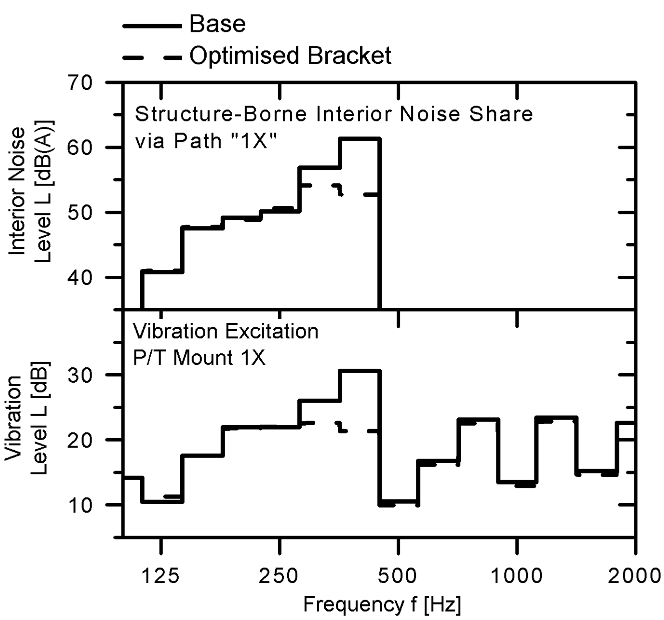

Figure 14: Influence of Induced Mount Bracket Optimization on Vibration Excitation at Engine Mount and Thus Caused Interior Noise Share

In the course of the optimization work, the P/T mount bracket was considerably stiffened, reducing excitation by approximately 9 dB (Figure 14, lower part). The reduction in the interior noise transmitted through this path is depicted in the upper part of Figure 14. Due to the almost linear transmission behavior, the individual noise through path 1x is also reduced by approximately 9 dB in the frequency range of the mount bracket resonance.

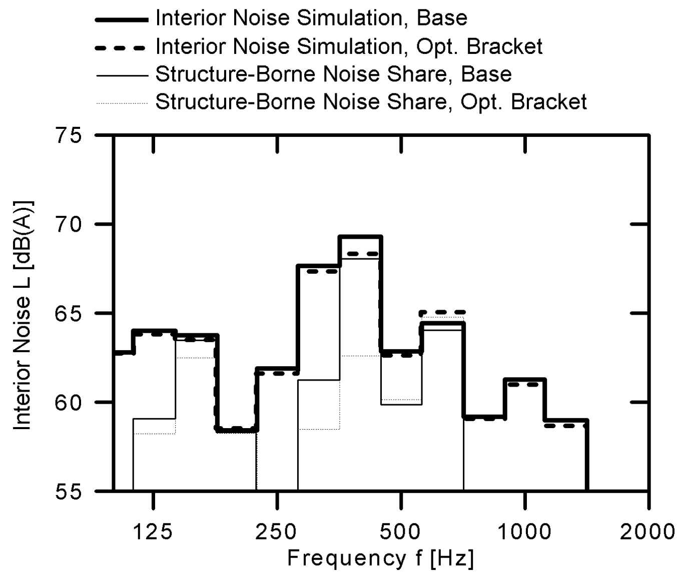

Figure 15: Influence of Induced Mount Bracket Optimization on Interior Noise

While individual interior noise optimization can be easily estimated by the transfer function, the interior noise simulation method also facilitates the subjective and objective evaluation of the influence on the total interior noise. Figure 15 shows the influence of the P/T mount bracket optimization on the total structure-borne interior noise as well as the total interior noise consisting of both noise shares. The total structure-borne interior noise share is reduced by about 6 dB in the frequency range of the mount bracket resonance. Allowing for the airborne noise share, this still reduces the total interior noise by about 1 dB.

By means of interior noise simulation, conspicuous interior noise phenomena can be assigned to individual airborne or structure-borne noise shares. Moreover, for these noise shares, it can be distinguished between excitation and transfer behavior as the weak point. This knowledge facilitates a purposeful optimization of the relevant vehicle components.

The modifications resulting from this can be evaluated by means of interior noise simulation with regard to their interior noise relevance. In addition to the aforementioned objective third-octave data, subjective evaluations of the audible interior noise simulation have proven to be efficient.

By modifying the transfer functions, changes in the vehicle are first calculated concerning their effect on interior noise. Thus, the extent of modifications can be defined beforehand and the improvement of interior noise quality can be quantified objectively as well as subjectively.

OUTLOOK

The determination of excitations and the weighting with corresponding transfer functions will be extended in the future to other interior noise shares, such as due to drivetrain, chassis and wheel suspensions.

Based on interior noise simulation by means of empirically determined excitations, future developments will increasingly deal with the modeling of excitations. The numerical calculation of the induced structure-borne vibrations, with their low frequent excitations, as well as the P/T mount bracket resonances, with their higher frequent excitations, is already a state-of-the-art procedure. Calculations of drivetrain vibrations are currently conducted by means of multi-body systems and finite element analysis models. Therefore, it makes sense to base interior noise simulation additionally on numerically determined excitations.

Furthermore, apart from the calculation of interior noise, the calculation of passenger compartment vibrations is also an important development tool which is implemented for instance for different engine mount concepts while considering body transfer behavior.

Turning today's interior noise simulation procedure into a tool for creating the "virtual vehicle" is the great future challenge. Thus, weak points of the induced could be identified and optimized at an even earlier stage of the development process.

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