Solving vehicle noise problems by analysis of the transmitted sound energy

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Abstract

The reduction of noise radiation of vehicle body components is one of the main tasks of acoustics engineering work for vehicle design. Different measures can be applied in order to improve noise reduction of structure borne sound driven by powertrain, road and wind excitation.

One of the main applications of energy measurements is the analysis of vehicle body components in view of optimized sound insulation and minimization of radiated structure-borne sound. While there are a lot of possibilities to improve the structural behavior of body panels, e.g. by sheet swaging, application of ribs or use of damping layers, it is not always easy to clearly choose the best method.

The complete energy balance gives sufficient data to choose the best way of noise reduction. Assessment of panel modifications and application of damping layers by analysis of the energy balance is shown by examples. An interesting application is the analysis of acoustic energy transmitted through the tire/rim system.

1. Introduction

If noise radiated by vehicle components shall be reduced, the most important question is about the quantity of sound power coupled into the air-borne sound field, that finally effects passengers ears and often causes annoyance. A basic and useful test of the sound radiating properties of structures is given by measurement of the total radiated sound energy, which easily can be done inside a reverberation chamber. A traditional method is to evaluate the sound-transmission loss as an indicator of sound insulating properties of body panels. This is done inside a facility allowing excitation by an diffuse air-borne sound field. The energy balance of incident and radiated sound power than indicates the performance of material in terms of noise control.

The idea of analysis of the input/output energy balance can be extended regarding excitation by structure-borne sound, which in practice at road vehicles is given by various sources, such as wheel suspensions, engine mounts and many others.

A simplified laboratory simulation of those sources can be gained using an electro-dynamic shaker and measuring the generated mechanical energy.

Secondly, transmission of acoustical energy through mechanical structures can be investigated if not the radiated sound, but mechanical energy coupled to further mechanical structures is recorded [1].

To illustrate the possibilities, some examples are shown which demonstrate development of sound-packages regarding both structure-borne and air-borne sound excitation, assessment of measures applied to reduce panel radiation and reduction of the tire cavity resonance.


The sound-transmission loss (abbreviated here with STL) of a structure is given by a logarithmic expression of the ratio of incident and radiated sound energy (fig. 1). Both values are here
measured inside two reverberation chambers, providing good diffuse field conditions in the frequency range from 100 to 10000 Hz. The FORD facility of the Merkenich Engineering Centre in Cologne consists of rooms with 122 and 275 m³ volume. The main advantage of this method, which as well is applied in buildings acoustics, is that the total power can exactly be measured with few microphones without knowledge of the directivity of radiation. Results, however, can only be gained by averaging specific parts of the total frequency range, leading to third-octave or octave spectra.

The excitation of structure-borne sound can be described by the mechanical input power, calculated from force and velocity measurements at an excitation point (fig. 1). It is important to use the real part of the complex product which stands for the incident energy (effective power). Values of a structure-borne sound transmission loss (abbreviated here with SB-STL) are used to assess the transmission of sound.

3. Application of Sound-Transmission-Loss Measurements upon Vehicle Body Components

Methods of air-borne and structure-borne sound transmission-loss evaluation can be used in parallel to assess sound package performance. An example shall illustrate this procedure: A fully equipped vehicle dashpanel is installed into the transmission-loss facility between both reverberation rooms (fig.2). The dashpanel includes dashboard and heater/ventilation unit. All trim parts, grommets and damping layers which influence the sound radiation are applied. The sound package consists of 2 cm woolen felt, heavy layer and carpet.

The quality of sound-package shall be assessed regarding air-borne engine noise and vibration transmitted via the left hand side engine mount and the front wheel suspension. Effects of diffuse airborne field excitation from the engine side are then compared with those given by shaker excitation at a front chassis connection point (vertical z-direction). Differences in the results reflect the equal distribution of air-borne sound over the total area, while the point excitation with the shaker causes more local effects of sound radiation. E.g. the influence of leakage and grommet sound insulation more effects the air-borne than the structure-borne sound transmission loss. It is obvious that sound insulating properties of sheet metal, sound-package and trim components can be quite different regarding both ways of excitation. Therefore it is recommended to always prove sound insulation under air-borne and structure-borne excitation. Investigation of structure-borne sound paths, however, must include all critical excitation points.

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STL = 10 \log \left( \frac{P_S}{P_E} \right)
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SB-STL = 10 \log \left( \frac{P_{\text{mech}}}{P_E} \right)
\]

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P_{\text{mech}} = \text{Re} \left( F_v^* \right)
\]
4. Minimization of Panel Radiation

4.1 Application of Damping Layers

A rectangular sheet metal (80 x 120 cm, 0.8 mm vehicle steel) is tightly fixed into a heavy frame. Structure-borne sound is generated by an electrodynamic shaker and directly induced upon the sheet metal. The measured sound transmission-loss with structure-borne sound excitation shows values in the range 12 - 23 dB (fig. 3). The frequency response of the structure depends on the geometry and the boundary conditions given by the heavy frame. If the total area is covered by damping layer (5mm bituminous layer, applied by heat), sound insulation can be increased up to 19 - 32 dB. In general, the optimum way of damping layer application is to cover all radiating panels. This of course will cause weight and cost increase which appear to be not acceptable in case of small and medium size cars.

Thus, in terms of weight and cost reduction the size of layers must be minimized at vehicle body components. This is normally done by application of a layer with a specified area upon the centre part of the radiating sheet. In the experiment 35% of the panels area have been covered, which often seems to be a good compromise of effort and effect of noise control. The results demonstrate the remaining reduction of radiated sound power.

In the case of direct excitation of a vehicle panel, e.g. by crashing stones or water splash, the application of damping layers is most effective if the excited area is covered directly. In the given example, a bituminous layer of 9% of panel area applied around the excitation point provides similar noise reduction as the 35% sheet situated in the centre area. Thus, with respect to the position where the excitation happens, in special cases a clear reduction of radiated sound can be caused by very few material.

Measurement of the total balance of sound energy therefore proves to deliver sufficient data to optimize layer placement and to minimize material effort at vehicle body components, before layers are applied to a total vehicle and a fine tuning can be done.
4.2 Modification of Panel Structure

Some differences in performance of damping layers can be seen if a panel is not excited directly, but via a rigid frame. This is a typical case of structure-borne sound excitation via chassis frames and side members, transporting engine and road noise to the passengers compartment. The example described next shows the effects of structural modifications and application of damping layers upon a framed structure.

Rectangular sheet metal test objects are fixed with rivets inside a metal frame (tubular steel 30x50mm). This frame is elastically mounted into a heavy wooden frame as part of the test window of the transmission-loss facility. Mechanical energy is excited with an electro-dynamic shaker fixed upon the metal frame. The input energy is measured by an impedance pickup.

The sound radiation of a flat sheet metal (120x80cm, thickness 0.8 mm) is measured as baseline. The influence of several modifications is then tested as given in fig.4. Four pieces of bituminous damping layer are used.

Data of the panels surface velocity are gained by Laser scanning techniques (fig. 4). This gives an overview of the vibrational behaviour of flat (unmodified) and modified sheet metal. Data are here prepared as “response value” = surface velocity / excitation force, averaged in the frequency range from 10 to 1000 Hz.

While vibrational energy is equally distributed over the flat sheet metals total area, the spatial distribution is influenced by the structure of frame, ribs and swages. It is remarkable that various ways of sheet metal modification naturally induce different spatial distributions, but the total mechanical energy is rather insignificantly reduced by those measures. Application of a bituminous damping layer, however, causes strong reduction of the vibrational magnitudes irrespective of additional ribbing.

All measured results are related to the structure borne sound-transmission loss of the flat unmodified sheet metal. The improvement of structure-borne sound transmission loss induced by the modifications described above is shown within fig.5 for third-octave frequencies from 100 to 1000 Hz. Note that positive values indicate a reduction of radiated sound energy.
The single values shown in fig.5 summarize the total reduction of sound energy within the frequency range 100 - 1000 Hz. Swaging combined with ribbing here leads to increase of sound radiation (1). Results can be concluded as follows:

- swaging a vehicle sheet can cause small reduction of radiated energy (e.g. total result << 1dB) but can in contrary lead to improvement of radiation efficiency: e.g. 5dB as 1/3-octave result at 315 Hz
- additional ribs as found at a space-frame structure show small reduction of radiated energy, e.g. < 1dB and can also increase sound radiation, particularly at high frequencies
- combination of the investigated swaged panel and the space-frame ribs causes strong increase of sound radiation (total result > 2dB), in particular above 300 Hz.

- the only measure that guarantees strong reduction of sound radiation is the application of damping layers, here causing a total reduction of > 3dB. In this case there is no 1/3-octave band showing increase of radiated energy.

5. Sound Transmission through the Tire/Rim System

Compared with the overall damping properties of the tire/rim system, transmission of vibrational road input through the system is increased in a narrow frequency band around 230 Hz. This causes by improved transfer of energy into chassis components and body, where it is radiated as airborne sound, leading to road noise annoyance due to a pure tone that is clearly audible inside the passengers compartment. This effect is described within literature [2] and has been detected during road tests using artificial head recordings and
vibration measurements at the wheel suspension. The frequency of the tone is nearly independent from engine and wheel speed and is not directly connected to the modal properties of the rim. It has been assumed that this effect is caused by a resonance of the air-borne sound field inside the tire.

A test setup has been designed to detect this resonance effect (fig.6): An electro-dynamic shaker induces broad-band vibration inside the bottom of the tire (vertical z-direction). The vibrational input energy is calculated from input force and velocity given by an impedance pickup. The output energy of the system transmitted from the hub upon a simple test structure (aluminium beam) is measured with a second pickup. The properties of the wheel system in transmitting road induced energy can then be described by the ratio of mechanical input to output energy, which are equivalent to sound-transmission-loss values. It was not intended to find an exact description of the total system in terms of broad-band road noise excitation, but to gain clear understanding of the phenomena regarding the tire cavity resonance.

Additionally, the radiation of air-borne sound by rim and tyre can be evaluated using diffuse field measurements inside the reverberation chamber.

With this method it has been shown that the decrease of sound transmission loss of the wheel system (see fig. at 230 Hz) in fact is caused by the resonance inside the tire cavity but is not essentially effected by the modal properties of the rim [3].

The resonance frequency depends on an effective length of the air tube inside the wheel. The resonance appears when the wavelength is equal to this cavity length. An upward directed force at the tire bottom (road surface) then causes a sound pressure maximum in the lower sphere of the cavity, while sound pressure is forced to minimum at the top of the cavity. Therefore an upward directed force acts at the hub, strongly supported by the cavity resonance. If during the dynamic input change at the tire bottom a negative force
(downward) is applied, the air-borne resonance moves the hub downwards. In practice, the movement of the tire/rim system is more complicated, including lateral components and showing two resonance peaks with a distance in frequency that depends on vehicle speed.

The resonance effect can completely be eliminated by insertion of absorbent material into the cavity (fig. 7). This is clearly demonstrated within the diagram: At 230 Hz, insertion of mineral fibre material leads to reduction of transmitted energy by more than 20 dB (!). Results of further measurements showed that different materials with absorbent properties can be used successfully. Modifications of the mechanical properties of the rim, however, do not reduce the transmitted noise energy at 230 Hz, but in contrary can intensify the resonance.

The results show that application of material which is feasible for long term road conditions will solve the 230 Hz road noise problem. Application of cavity absorption causes nearly no weight increase, but removes energy from the air-borne field of the tire cavity, that otherwise would be coupled into the mechanical suspension system, from where it cannot be removed without application of mass.
6. Conclusion

Measurement and analysis of the energy balance within acoustical systems has proven to be an important tool in design of noise control measures. The main advantage is that all physical units which enable a comprehensive description of acoustical phenomena are taken into account. The usual methods based on measurement of frequency response functions contain uncertainties regarding the best descriptors while only one unit (regarding force/pressure or movement) is included to describe input and output of a system. Magnitudes of those single descriptors only in special cases show proportionality to energy values. Thus, the interpretation of sound transmission phenomena by analysis of frequency response functions is not easy and can contain errors if the system is not well known before.

Comparison of input and output energies in terms of an energy balance, however, help to understand the mechanism of noise controlling measures regarding the effective energy reduction. All important units are included. The results are significant for a specific configuration of driving impedance (e.g. of the shaker) and output impedances used in the experiment. This is not a problem if radiation of sound into the air-borne sound-field is investigated because the field impedance of air is nearly constant (except small changes by temperature and humidity). Procedures for analysis of structure-borne sound transmission, however, need specific standardisation of test structures.

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References

