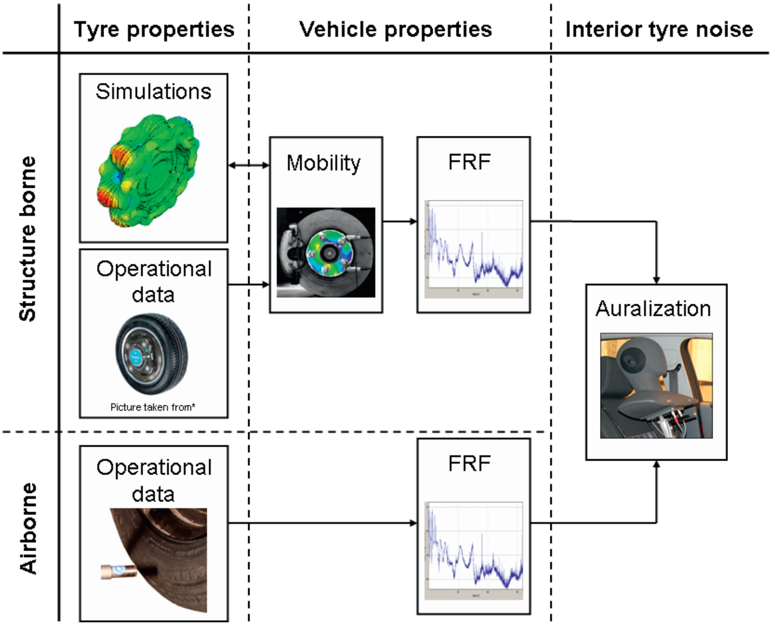
LICENTIATE THESIS

Model Development for Auralization of Interior Tyre Noise



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Model development for auralization of interior tyre noise

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To Max
- you are my force

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Luleå, May

Magnus Löfdahl

Abstract

Increasing competition has set pressure on the product development process to reduce development time and costs. Computer Aided Engineering (CAE) has been used to decrease development times by facilitating early predictions of product performances and qualities. Especially in early phases of product development, models with sufficient accuracy can provide valuable decision supports in order to pass legislations and fulfill customer expectations. Acoustic performance and sound quality are essential parts of the perceived product functionality and quality. A powerful method in product sound development is to combine recordings and simulations into auralizations.

Interior vehicle noise is an important factor in the perceived product quality where tyre noise is a dominant source. The objective of this licentiate thesis was to lay the foundation for an auralization model of interior tyre noise. The aim of the model is to use the results from for example FEM simulations of tyre/road interaction and filter it through experimentally measured transfer functions into the cabin of the car. By varying compounds, components and road profiles in simulations, tyre noise can be auralized in different cars in an early design phase. Tyre noise predictions and auralizations are relevant both in tyre and car development.

The vibrations generated by the tyre/road interaction are transferred through the hubs of the car and into the cabin as structure borne sound. The hub acts as the coupling element and describes the boundary condition for the rim. In paper I, the mechanical mobility of a hub was measured in 6-DOF. Measurement results showed good multiple coherences, reciprocities and low random errors in the frequency range 0-300 Hz. The measured mobilities will be used to transform operational forces and moments into velocities and will be implemented as boundary conditions in FEM simulations.

For auralizations of the air-borne tyre noise contribution, knowledge of the required accuracy in positioning of sources and receivers is essential. In Paper II, variations in perceived sound caused by displacements of source and listening positions were assessed to find the smallest displacement giving a just audible differences. In addition, binaural transmissibility functions were measured from a loudspeaker near a wheel to an artificial head inside the car. Results showed that the accuracy in the positioning of the source and the receiver needed to be smaller than 2 cm to avoid audible differences.

In order to generalize auralizations of interior tyre noise, audible variations in specimens of nominally identical products need to be known. In Paper III, variations in perceived interior sound between tyres of different brands and specimens of nominally identical cars were assessed. The differences between five nominally identical cars were found to be two to three times larger than the difference between two tyre models.

Keywords: Product sound design, sound quality, auralization, interior tyre noise, vehicle acoustics, multidirectional mechanical mobility

Thesis

This licentiate thesis is based on the work presented in the following three papers

Paper I:
M. Löfdahl, R. Johnsson: Mobility measurement and evaluation in 6-DOF applied to the hub of a car. Submitted for publication

Paper II:
M. Löfdahl, A. Nykänen, R. Johnsson: Assessment of changes in automotive sounds caused by displacements of source and listening positions. Resubmitted for publication

Paper III:
A. Nykänen, M. Löfdahl, R. Johnsson: Examination of the variability between artificial head recordings made in different cars of the same brand and model. Proceedings of Inter-noise 2009, Ottawa, Canada

Contents

1 Introduction

Increasing competition on the global market has set pressure on the product development process. New products that fulfill or even surpass the customer expectations have to be developed at lower costs and in shorter times [1]. The efficiency of the product development process plays a key role to get the products out on the market as quickly as possible. Time and costs can be reduced by modeling the final products with functional performances in an early design phase. To meet criterions and demands, more than just the visual design must be handled in the modern industrial design process [2]. Requirements such as legislations must be passed and the customer requirements on perceived product quality should be fulfilled. Functional performance such as comfort, safety and reliability has therefore to be estimated with sufficient accuracy. It is also the functional performance that often set the quality sign of the product and differentiates it from competitors.

Acoustic performance and sound quality are essential parts of the perceived product functionality and quality. Over recent years, product sound design has received increasing attention and has become a natural part of the product development process. Today, companies use sound design to position the brand on the market as well as to differentiate their products from competitors. This has created a growing need for good CAE-tools. Different methods have been developed for example simulation software, measurement techniques and psychoacoustic models. With these methods, sound levels can be predicted, acoustical performance simulated and sound quality evaluated. The needed complexity and accuracy of the prediction depends on the situation. In an early design phase, a rough "sound sketch" can be enough for valuable decision support while more detailed information might be needed in a later phase. More accurate data are therefore usually needed at the end of the development process for satisfying performance predictions and analyses. The specific stage or task in the product sound design process also determines which type of data is the most appropriate, whether it is recordings or simulations.

Recordings are useful in applications such as subjective evaluations and calculations of psychoacoustic metrics such as loudness, sharpness, roughness etc. Attributes of the sound can be described to define a target sound [2]. Recordings can easily be modified and filtered in ways that are hard to achieve using changes to prototypes or simulations. However, accurate recordings often require a well developed prototype and thereby a design process far gone. For each modification, a prototype must be reused or a new created. This is followed by making new recordings and evaluations which is an expensive and time-consuming process. Tracing specific noise components in the recording to their origin in the product can be even harder.

Simulations are usually FEM calculations based on structural dynamic models where modes, deflections shapes and frequency response functions (FRFs) of the product can be calculated. Models can be modified and optimized. The simulation results can for example reveal important information of frequency components that can be perceived as annoying. There is a tendency to simplify models by neglecting information (e.g. degrees of freedoms), nonlinearities, unknown material properties, unknown operating forces and unknown boundary conditions. These simplifications cause simulation errors and can for example limit the frequency range. Some of the errors are due to factors such as complexity, time, costs and quality in the measurement data which the models are based on.

A powerful method in product sound development is to combine recordings and simulations into auralizations. The drawback in one technique can to some extent be compensated with an advantage from another technique. A typical model in such case is divided into substructures where each part is described either by a simulation or experimental data. The substructure models are then coupled into a hybrid model of the product [3, 4]. However, for a correct hybrid model, complete data for both numerical results and measured data are necessary. The accuracy in each part is therefore important. This is especially important in applications where a small error can cause severe errors in a later phase. An example of this is when matrices of measurement data have to be inverted for use as coupling information, such as impedance.

1.1 Tyre noise auralization

The comfort in a vehicle is an important factor for the quality perceived by the customer. The interior noise in a car is highly related to comfort as well as safety issues. The operational noise in the car compartment is mainly generated by tyre, wind, engine and transmission. As the speed increases, tyre noise becomes the dominant source. Tyre noise is generated from the tyre/road interaction and transferred into the car compartment through structure borne and airborne paths. Vibrations caused by the tyre/road interaction are transferred through the hub to the suspension and chassis and radiate in the car compartment as structure borne tyre noise. At lower frequencies, tyre noise consists mainly of structure borne tyre noise. At higher frequencies the contribution of airborne tyre noise dominates. The airborne interior tyre noise is directly radiated from the tyre/road interaction and transferred into the car compartment. Hence the sound pressure level of interior tyre noise depends on properties of the road profile, the tyre and the car. The perceived tyre noise quality is not only influenced by the sound pressure level but also by for example the character of the sound and the listener's expectations. Studies have shown that exposure to low frequency noise affects the subject as annoyance and lack of concentration, sleepiness and tiredness [5, 6]. In electrical and hybrids cars tyre noise is even more prominent and important since the masking effects of the combustion engine noise do not exist.

Airborne tyre noise will also radiate to the surrounding as road traffic noise. Solely in the EU, studies made year 2000 estimated that more than 30% of the population in EU is exposed to sound pressure levels above 55 L_{dn}[1] dB. In addition, 50 million people are exposed to noise levels above 65 L_{dn} dB [7]. Thereby, road traffic noise is a serious environmental problem.

In order to reduce interior tyre noise the whole chain from the road to the ears of the driver and passengers must be taken into account. When reducing tyre noise, it will also affect properties of the tyre such as handling and the dynamics of the car. It is important that the modes of the tyre do not coincide with modes of the suspension. Therefore, not only the properties of the tyre itself are important but also understanding the suspension of the car and the transfer paths through the car. A good knowledge in tyre construction, sound generation mechanisms and the properties of the car is required. By predicting the noise level, components can be improved. A hybrid model able to auralize and predict interior tyre noise is a powerful tool in the development of both tyres and cars. A previous study has shown that the structure borne road noise in the mid frequency range 150-400 Hz can accurately be predicted but require a hybrid formulation based on a full vehicle model [8].

[1] L_{dn} i.e. a day/night level, is a descriptor of noise level based on the energy-equivalent noise level (L_{eq}) over the whole day with a penalty of 10 dB(A) for night time noise (22.00-07.00 hrs).

2 Auralization model of interior tyre noise

The objective of this licentiate thesis was to lay the foundation for an auralization model of interior tyre noise. The idea of the model is to facilitate listening and quality assessments of interior tyre noise at various phases in the development process. Tyre noise predictions and auralizations are relevant both in tyre and car development. It is usually hard to draw conclusion in advance on how the sound will be affected by specific changes in the tyre (or car) construction. In early product development stages it is even harder to say how the interior tyre noise will be perceived in different cars or different sets of tyre on one car. Using a hybrid model based on simulations and experimental data this can be done.

Tyre manufacturers use FEM models of tyres to simulate the tyre/road interaction where compounds, components and road profiles can be changed. Interior tyre noise can be auralized by filtering the outcome of the FEM model through experimentally measured frequency response functions (FRFs) from the tyre to the listener inside a car, Fig. 1. By measuring the structure borne and airborne transfer paths of different cars, tyre noise can be predicted in various situations (roads, tyres and cars).

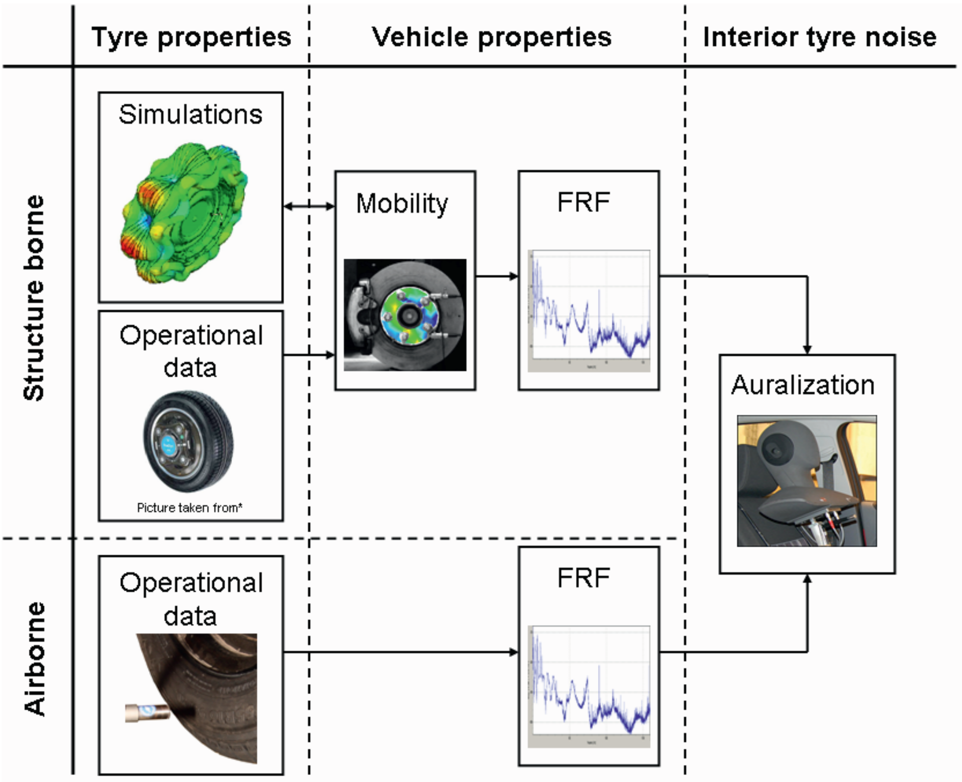

Fig. 1. Auralization model of interior tyre noise.
* Picture is taken from www.kistler.com, RoaDyn® S625 System 2000 – Wheel Force Transducer (WFT) for passenger cars, 2011-05-10

In the suggested hybrid model, the output can be interior tyre noise auralizations where it can be chosen whether the structure born or airborne contribution are modeled, separately or combined. At this instance, the airborne tyre noise can only be based on experimental data such as near field recordings of a tyre. The generation of structure borne tyre noise however, can be based on simulations or operational data. The forces and moments acting on the hub can either be calculated or measured. The auralizations can be used for listening test, assessments or used in psychoacoustic models.

The model can be used in the frequency domain or time domain. The domain in which the simulations are performed determines how the results can be used. If the simulation is performed in the frequency domain, auralizations are not possible since only frequency spectrums will be yielded as results. This can be sufficient for comparing with the modes of the suspension. This can provide valuable information in order to avoid modes of the suspension to interact with tyre modes or resonances. Conclusions on perceived tyre noise quality can hardly be done based only on frequency spectra. When the simulations are performed in the time-domain, auralizations of tyre noise can be made, allowing for aural assessment of sound quality.

2.1 Tyre properties

It is usually the current stage in the development process that determines the available data for the model. In an early stage, simulation or modified earlier products can be used. The simulations are based on structural models (such as FE-models), where in this case both tyre and rim are included. The rim must be integrated in the model since it is the coupling element between the tyre and the structure of the car (the hub). It is important in the simulation that the boundary condition of the rim is included. If the rim is rigidly attached, vibratory velocities cannot be simulated properly since a rigid connection permits no movement and therefore the velocities are incorrectly determined. The velocities are necessary as input for the rest of the model.

For operational measurements of structure borne tyre noise, forces and moments in all degree of freedoms (DOFs) have to be measured. This can be achieved with a transducer such as the Kistlers RoaDyn® hub. Adding microphones for near field recordings of the tyre, signals for structure borne and airborne tyre noise can be measured simultaneously.

2.2 Vehicle properties

2.2.1. Mobility

Vibrations generated by tyre/road interaction are transferred through the hubs and suspension into the car compartment as structure borne sound. The hub acts as the coupling element and describes the boundary condition for the rim. When the full behavior (6-DOF) of the hub is know, this information can be implemented in the FE-model in order to get velocities as output. The information needed to describe the boundary condition in the FE-model is the blocked impedance. However, blocked impedance is hard to measure since it requires that the structure is grounded (blocked) and excited with a velocity in one DOF where forces and moments are measured in all DOFs. Instead it is more convenient to measure the free mobility since it is the inverse of the blocked impedance:

$$Y_{free} = \frac{1}{Z_{blocked}} \tag{1}$$

The free mobility is measured by exciting the free structure with forces and moments and measuring the velocity in all DOFs. The mobility matrix must be measured accurately since it has to be inverted into the blocked impedance. Small errors in the reciprocity (off-diagonal elements) of the mobility matrix will yield large errors in the impedance matrix due to the determinant in the inversion process.

The mobility can also work as a unit converter. For instance in operational measurements when the data is recorded as forces and moments, it has to be converted into velocities for use as input into the frequency response functions (FRF). Using the mobility matrix, these forces and moments can be converted into vibratory velocities according to:

$$\begin{bmatrix} v_x \\ v_y \\ v_z \\ w_\alpha \\ w_\beta \\ w_\gamma \end{bmatrix} = \begin{bmatrix} Y_{xx} & Y_{xy} & Y_{xz} & Y_{x\alpha} & Y_{x\beta} & Y_{x\gamma} \\ Y_{yx} & Y_{yy} & Y_{yz} & Y_{y\alpha} & Y_{y\beta} & Y_{y\gamma} \\ Y_{zx} & Y_{zy} & Y_{zz} & Y_{z\alpha} & Y_{z\beta} & Y_{z\gamma} \\ Y_{\alpha x} & Y_{\alpha y} & Y_{\alpha z} & Y_{\alpha\alpha} & Y_{\alpha\beta} & Y_{\alpha\gamma} \\ Y_{\beta x} & Y_{\beta y} & Y_{\beta z} & Y_{\beta\alpha} & Y_{\beta\beta} & Y_{\beta\gamma} \\ Y_{\gamma x} & Y_{\gamma y} & Y_{\gamma z} & Y_{\gamma\alpha} & Y_{\gamma\beta} & Y_{\gamma\gamma} \end{bmatrix} \begin{bmatrix} F_x \\ F_y \\ F_z \\ M_\alpha \\ M_\beta \\ M_\gamma \end{bmatrix} \tag{2}$$

The required accuracy of the tyre noise auralization determines the required accuracy of the mobility measurements.

2.2.2. Frequency response functions

To be able to auralize and listen to interior tyre noise, airborne and structure-borne transfer functions have to be measured from the tyre to the eardrums of the listener in the cabin of the car. These are called binaural transfer functions (BTFs). For this application, a convenient way is to use an artificial head, a manikin of the upper body with microphones at the ears. In the automotive industry, this has become a standard procedure for recordings and reproductions for sound quality analysis [9].

For the structure borne path, BTFs from the hub to an artificial head in the car compartment can be measured simultaneously as the mobility measurement of the hubs. Since the hub is excited with a white noise, each BTF can be measured and calculated. BTFs of the airborne transfer paths can for example be measured from a loudspeaker placed near the wheels reproducing a white noise, to an artificial head in the cabin of the car.

Depending on the location of the source and receiver, the character of the sound will change since the (BTFs) will alter. The complex acoustic environment in a car cabin with reflecting and absorbing surfaces close to the passengers affects the effects of displacements of sources and/or listeners. Thereby the accuracy of the positioning of the artificial head is important for the quality of the auralization. This has to be considered when making measurements for building of auralization models of tyre noise.

Another issue that also has to be considered in the auralization model of interior tyre noise is the variation between nominally identical cars. Studies have shown that large variations in acoustic response exist between nominally identical cars [10-12].

2.3 Interior tyre noise auralization

The annoyance that the driver can experience from the tyres is not only a product of the level but also by the character of the sound and the situation in which the sound is heard. This sound quality aspect is also an important aspect of the experienced overall quality of the tyre. With the possibilities to change properties of the tyre, specific components can be judged and evaluated with respect to how it affects the perceived sound quality.

By making auralizations of tyre noise, assessments can be done as well as psychoacoustic models developed to objectively describe the aspects of perceived sound quality.

2.4 Research objectives

The objective of this licentiate thesis was to lay the foundation for an auralization model of interior tyre noise. The following research objectives are covered in the thesis:

- Derive a general approach to obtain the mobility matrix and apply it to a 6-DOF measurement of a complex mechanical structure (the hub of a car).
- Find the just audible differences in headphone reproduced artificial head recordings for displacement of either the source outside the car or listening position inside the car compartment.
- Examine the perceived variability between artificial head recordings made in nominally identical cars.

3 Summary of papers

Each paper can be represented as a part or a complete block in the auralization model of interior tyre noise, Fig. 1. Paper I, represent the mobility block since the mechanical mobility is measured in 6-DOF on the hub of a car. The airborne FRF block is treated in Paper II where knowledge of the required accuracy in positioning of sources and receivers is essential. In Paper III, all blocks are included where variations in perceived interior sound quality between tyres of different brands and specimens of nominally identical cars were assessed.

3.1 Paper I - Mobility measurement and evaluation in 6-DOF applied to the hub of a car

For a correct hybrid model, complete data for both numerical results and measured data are necessary. When coupling substructures, models are often simplified by omitting rotational degrees of freedoms (RDOFs). A simplification such as omitting RDOFs results in incomplete models since only 25% of the structural behavior is described.

The hubs of a car act as the coupling elements for structure borne tyre noise and describe the boundary conditions for the rims. This boundary information can then be implemented into FE-simulations of a tyre and rim. The objectives were to obtain the mechanical mobility matrix by developing a general approach and apply it to measure a hub in 6-DOFs.

By developing a general approach and a measurement set-up applicable to different car models, 6-DOF measurements for a variety of car models can efficiently be done.

3.1.1 Method

To be able to make qualitative 6-DOF mobility measurements, the set-up had to be capable of measuring all 6-DOFs in one set without any changes or alterations. Loading of the structure is then identical for both forces and moments which prevents inconsistencies. To provide a high repeatability and measurement accuracy, the set-up had to be stable and rigid. With no tyre or rim mounted on the hub, the aim was to attach all equipment directly to the braking disc.

Since the original braking disc was not stiff enough to be considered as a rigid body in a sufficiently large frequency range and due to complication with mounting of all measurement equipment to the braking disc, a special braking disc was design and fabricated, Fig. 2.

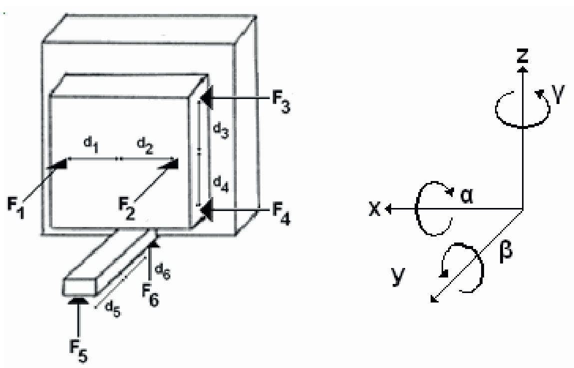

Fig. 2. Square shaped braking disc.

By making the disc into a quadratic shape with a cantilever beam, excitation could be made in all DOFs and the measurement points could be evenly spaced. The disc was milled from a steel plate with the same mass centre and weight as the original front braking disc to prevent additional mass loading. By making the disc solid and thicker the structure behaved as a rigid body in a larger frequency range.

3.1.2 Results

LDV measurements showed that the braking disc was rigid up to 858 Hz. To assure no contribution from the resonance frequency, only results up to 300 Hz will be shown. The measured point mobility in all 6-DOFs can be seen in Fig. 3.

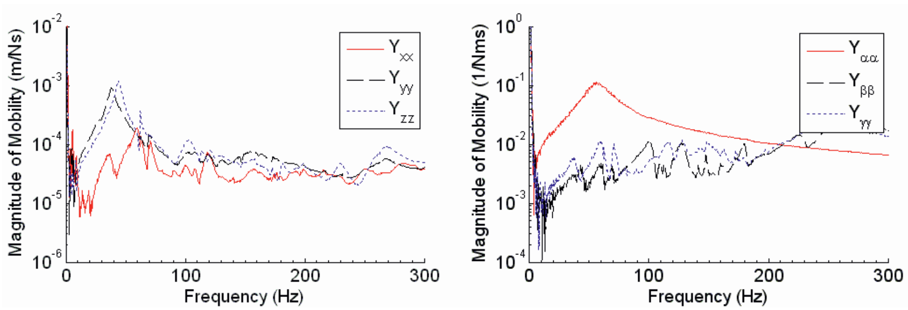

Fig. 3. Point mobilities for translations Y_{xx}, Y_{yy}, Y_{zz}. and rotations $Y_{\alpha\alpha}$, $Y_{\beta\beta}$, $Y_{\gamma\gamma}$.

The linearity of the system and the quality in the measured mobility functions can be evaluated by examining the symmetry in the off-diagonal (reciprocal) elements in the mobility matrix. In Fig. 4, the reciprocity is viewed between translational and rotational mobility terms.

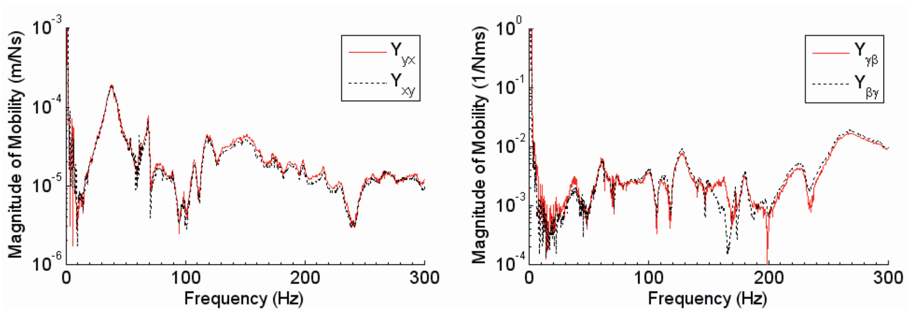

Fig. 4. Reciprocity between translational mobilities Y_{xy} and Y_{yx} and rotational mobilities $Y_{\beta\gamma}$ and $Y_{\gamma\beta}$.

The mobility showed similar results in the reciprocal directions which indicate a good measurement quality.

An ordinary input to output coherence function can not be properly used for quality evaluation when the response at the output is the result of contribution from multiple mutually correlated inputs. By conditioning out the linear dependence between all inputs using optimum linear least squares relationships, the inputs become mutually uncorrelated and the multiple coherence function can be calculated where the quality and noise in the measurement can be estimated, Fig. 5.

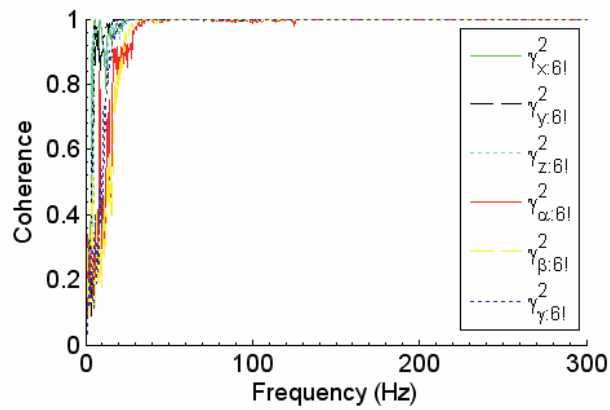

Fig. 5. Multiple coherence for all DOFs.

In Fig. 5, the multiple coherence is close to one over 20 Hz for all DOFs except in the α- and β-directions which are close to one over 40 Hz.

3.1.3 Discussion

The general method derived to determine and measure the mechanical mobility in 6-DOF has been shown to work well. With a measurement set-up applicable to different car models, hubs of different cars can be measured in a straight forward way and be used in the auralization model of interior tyre noise.

The magnitude of the mobility is approximately the same in all DOFs. This implies that the direction which has the highest force or moment input excitation will be the degree of freedom which will yield the highest vibratory response which in turn will be transferred to the rest of the structure. Therefore, RDOFs cannot be omitted unless it is known in advance that the input forces and moments in RDOFs are small in comparison to those in translational DOFs. Results in the frequency range below 300 Hz showed a good quality in reciprocity and multiple coherence.

Improvements can be made by splitting the measurement in two frequency intervals, 0-20 Hz and 20-300 Hz. More force can be delivered to the hub at frequencies below 20 Hz which would improve the quality in the measurement. To extend the frequency range above 300 Hz, the construction of the braking disc could be improved by strengthening of the cantilever beam. If the strengthening leads to that the bending mode at 2003 Hz become the lowest, then the possible measurement range could be expanded to 650 Hz. This would cover the frequency range where the structure borne noise from the tyre dominates over the airborne noise in the car compartment.

3.2 Paper II - Assessment of changes in automotive sounds caused by displacements of source and listening positions

For auralizations of the airborne tyre noise contribution, knowledge of the required accuracy in positioning of sources and receivers is essential, for example when BTF measurements shall be performed at different cars. It is important that the auralizations are reproduced with sufficient accuracy when sound quality studies are made. In Paper II, variations in perceived sound quality caused by displacements of source and listening positions were assessed to find the smallest displacement giving a just audible difference.

3.2.1 Method

Changes in perceived sound quality caused by displacements of either the source or the receiver position were studied through a listening test to find the just noticeable difference. Two typical automotive sounds were considered, tyre and engine noise, since they both are prominent sounds under driving conditions but have different characters. The engine sound was recorded from the same car as for which the displacement sensitivities was measured. The tyre sound was recorded at 75 km/h with a specially designed trailer for near field recording of tyres. The recording was made with a studded tyre on a dry asphalt road.

The measurement setup was designed to reproduce automotive sounds through a loudspeaker (source) placed outside the car, and recorded with an artificial head as receiver, placed in the front passenger seat, see Fig. 6.

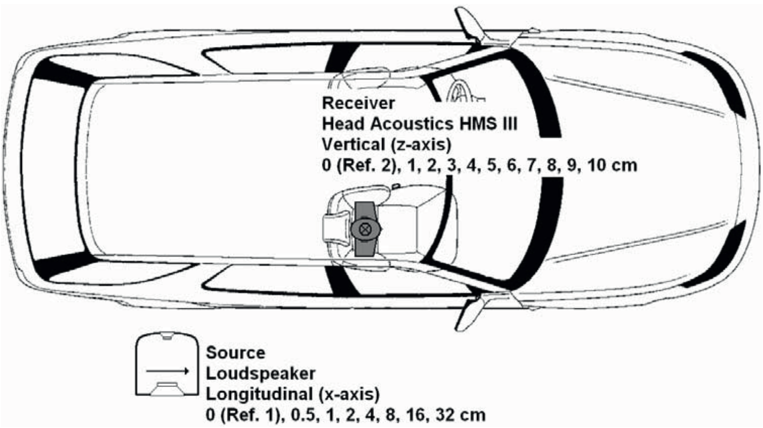

Fig. 6. Reference positions and displacement of source and receiver

Displacements of the loudspeaker in the longitudinal-direction (heading of the car) and the artificial head in the vertical-direction (height of the listener) where measured. For every position of the loudspeaker and artificial head, both the engine and tyre sounds were reproduced through the loudspeaker and recorded with the artificial head.

The listening test compared pairs of reference stimuli (recorded at the reference positions) and object stimuli (recorded at the displaced positions). In addition the subjects were asked to judge the difference between the stimuli as differences induced by displacements may not necessarily be perceived as impairments. The difference was judged on a scale ranging from 0 to 100 where 0 = very pronounced and 100 = not audible.

3.2.2 Results

In Fig. 7 and 8, the numbers of correct identifications for each sound pair are reported. The results showed that displacements up to 1 cm were not audible. Differences caused by displacements larger than 2 cm were heard by almost all participants. This shows that the accuracy in positioning needed to avoid audible differences in artificial head recordings in a car compartment is 1 cm.

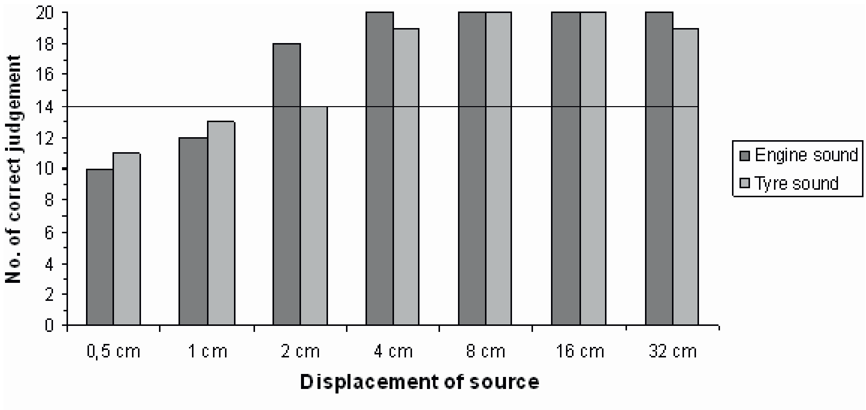

Fig. 7. Correctly identified sound pairs for displacement of source

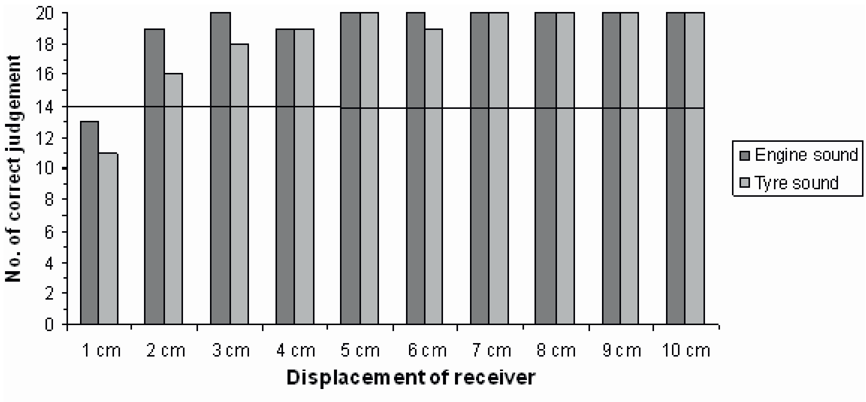

Fig. 8. Correctly identified sound pairs for displacement of receiver

In Fig. 9 to 10 means and 95 % confidence intervals for the normalised difference ratings between the reference position and the displaced position are reported as functions of the displacement. In Fig. 9 comparisons are made between the stimuli when the source is displaced. No significant differences can be seen between tyre and engine sound for displacements of the source. In Fig. 10 tyre and engine sound are compared for displacements of the receiver. For this case the differences between the reference and the object stimuli were judged higher for engine sounds for displacements up to 6 cm. Hence, engine sounds seem to be more affected perceptually by displacements of the receiver than tyre sounds. This may be explained by more high frequency content in the engine sounds. Comparing Fig. 9 and Fig. 10 shows that the perceptual change in sound quality was approximately the same when the receiver was displaced as when the source was displaced the same distance.

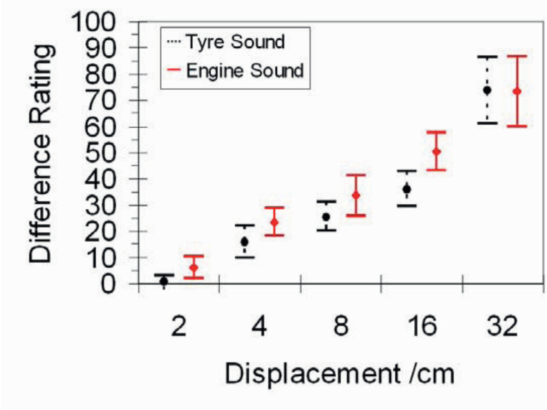

Fig. 9. Means and 95 % confidence intervals (internal s) for displacements of the source.

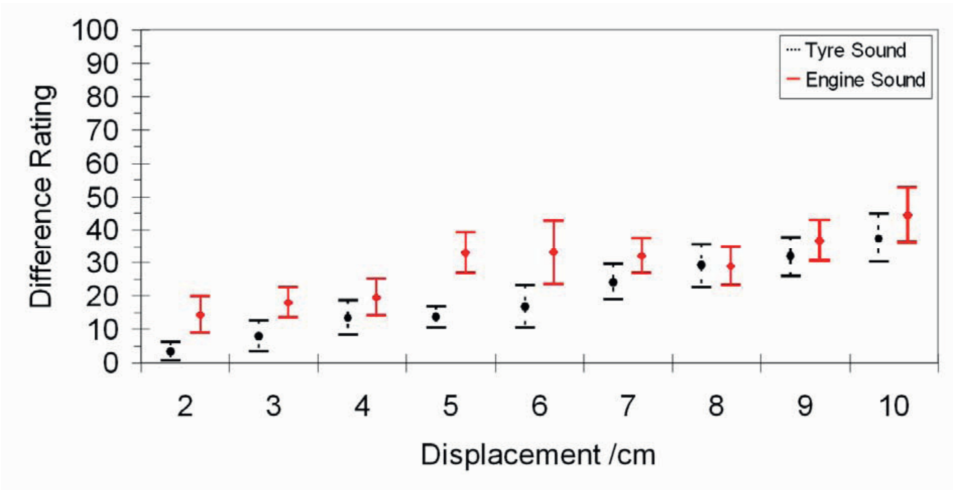

Fig. 10. Means and 95 % confidence intervals (internal s) for displacements of the receiver.

3.2.3 Discussion

The results showed that artificial head recordings are affected by small displacements in location of both the source and the receiver. In this study, the just audible difference was less than 2 cm for displacements of a source outside a car or a receiver inside the car. Therefore, the accuracy in the positioning of the source and the receiver need to be smaller than 2 cm if audible differences shall be avoided. This has to be considered in the auralization model when comparisons of auralizations are made in order to represent equivalent situations and conditions.

The difference ratings showed, as expected, that larger displacements lead to larger perceived differences in sound quality. They also showed that engine sounds were more affected perceptually by displacements of the receiver than tyre sounds. This may be explained by the high frequency content in the engine sounds. This kind of sounds therefore require higher precision in the placements of sources and receivers compared to less impulsive sounds with less high frequency content, like the tyre sound used in this study.

3.3 Paper III - Examination of the variability between artificial head recordings made in different cars of the same brand and model

When auralizations of interior tyre noise shall be compared, it is important to be aware of the variability between measurements made under similar conditions. In Paper III, the interior sounds of five nominally identical passenger cars were recorded binaurally. The recordings were made using two different sets of tyres. Based on the recordings, psychoacoustic metrics were judged in a listening test and the variability in perceived sound quality between nominally identical cars was assessed.

3.3.1 Method

Sound stimuli for the listening test were created by binaural recording of interior sound in five nominally equal cars (mid-sized estate cars of the same brand, model and production year, and equipped with the same interior trims). The same sets of rims and tyres were used on all cars. From now on the cars are labeled Car 1 to 5. All cars were less than one year old with mileage between 9500 and 20000 km. This was considered to be a typical first year mileage for a buyer of a new car, and hence the results should be representative for the spread in sound quality as experienced by owners of new cars. In order to minimize measurement uncertainty, all recordings were made at constant speed, 60 km/h, on a 70 m long straight section of a test track. The surface of the track was dry asphalt. During the tests the temperature varied between -15 and -5 ºC. The test sounds were recorded as a part of a study on winter tyre sounds. Therefore, studded winter tyres were used. In order to create differences in sound character two different sets of tyres were used (from now on labeled Tyre 0 and Tyre 1). Both were of the same dimension, 205/55-16, but from different manufacturers. Measurements were made using a Head Acoustics HMS III artificial head placed in the front passenger seat. All measurements were repeated twice.

In a listening test, the subjects were asked to judge how well the sound stimuli were described by 4 verbal attributes, loud, sharp, rough and annoying, using a verbal attribute magnitude estimation method.

3.3.2 Results

The perceived differences between the sounds of the 2 tyres and the 5 cars were analyzed using ANOVA. Significant effects ($p \leq 0.05$) were found for judgments of Loud and Annoying, Fig. 11 and Fig. 12. Tyre 0 was judged as being louder and more annoying. An interesting observation is that there are significant differences in judgments of Annoying between the different specimens of the nominally equal cars. The difference between the least and the most annoying car was three times as large as the difference between the two tyres.

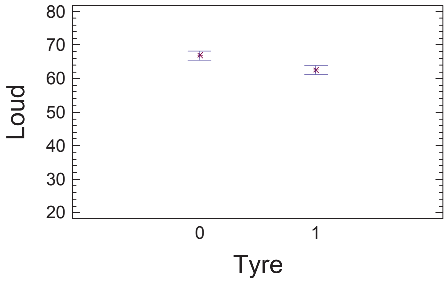

Fig.11. Means and 95 % Tukey HSD intervals for judgments of Loud based on recordings in the 5 cars using both sets of tyres.

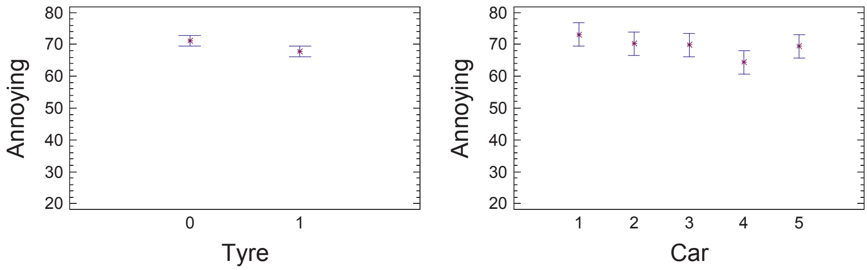

Fig. 12. Means and 95 % Tukey HSD intervals for judgments of Annoying based on recordings in the 5 cars using both sets of tyres.

3.3.3 Discussion

It was found that variations in interior sound between different specimens of nominally equal cars may be larger than variations caused by the use of tyres of different design. This was apparent for judgments of loudness and annoyance, where the differences between cars were between two and three times larger than the difference between the two tyres. It is of great importance to be aware of such variations, especially if the number of stimuli is limited.

4. Conclusions

In this thesis, the foundation has been laid for an auralization model of interior tyre noise. Several issues have to be considered when developing an auralization model in order to reproduce realistic auralizations. In this thesis it has been shown that variations in interior sound between different specimens of nominally equal cars may be larger than variations caused by the use of tyres of different design. To highlight differences in the auralized interior tyre noise rather than variation between nominally identical cars it is important that all measurements are performed on the same specimen of a car. Between different measurements of car properties, variations occur due to displacements in positioning of measurement equipment. These variations have to be considered in the auralization model when comparisons of auralizations are made in order to represent equivalent situations and conditions. It has been shown that larger displacements lead to larger perceived differences in sound quality. The just noticeable displacement was less than 2 cm for a source outside a car or a receiver inside the car. Therefore, the accuracy in the positioning of the source and the receiver need to be smaller than 2 cm if audible differences shall be avoided.

Results have also shown that in order to auralize interior structure borne tyre noise, all 6-DOF are needed. The hub of a car was measured in 6-DOF where the magnitude of the mobility at the hub is approximately the same in all 6-DOFs. This implies that the direction which has the highest force or moment input excitation will be the degree of freedom which will yield the highest vibratory response which in turn will be transferred to the rest of the structure. Therefore, RDOFs cannot be omitted in the auralization model unless it is known in advance that the input forces and moments in RDOFs are small in comparison to those in translational DOFs.

5. Future research

In order to reduce interior noise a good knowledge in tyre construction, sound generation mechanisms and the properties of the car is required. With further development this can be provided in the suggested auralization model of interior tyre noise. As a first step, complete auralizations of operational measurements have to be made where both structure borne and airborne transfer path are included. Results can be validated by comparison with operational recordings made on the same cars as for which the hub mobility and BTFs are measured. With a complete operational model, the quality in the auralizations can be evaluated. A suggested next step is to compare the simulations against operational measurements. With a validated and evaluated model, auralizations of interior tyre noise can be incorporated in the development process in order to increase the efficiency.

Further research can also be made of each individual block of the model. One such example is to investigate the perceived effects of RDOFs in the structure borne transfer paths. Research could also be made to see if it is possible to add simulations of airborne tyre noise with the simulations of structure borne noise.

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Paper I

Mobility measurement and evaluation in 6-DOF applied to the hub of a car

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Mobility measurement and evaluation in 6-DOF applied to the hub of a car

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ABSTRACT

To develop competitive products, sufficient estimations of the final product and its performance are of great importance for decision supports in early design phases. To predict functional performances such as noise and vibration levels, complex models are usually needed. A typical model is divided into substructures where each part is described either as a simulation or an experimental model. For full structural behavior, rotational degrees of freedoms (RDOFs) are needed. Moment excitation and measurement is required which can be difficult to achieve. By applying a multiple input multiple output (MIMO) technique both translational and RDOFs can be measured simultaneously without alterations in the measurement set-up. The objectives of this study were to develop a general approach based on the MIMO technique to determine the mobility matrix and apply it to measure the mechanical mobility in 6-DOF for a complex mechanical structure. The hub of a car was chosen as measurement object where a specially designed braking disc was fabricated for direct attachment of shakers and transducers. The quality of the full mechanical mobility matrix was evaluated using multiple coherences, reciprocities and random errors. The results showed a good quality in the frequency range 20-300 Hz.

Keywords: Mobility measurements, Multidirectional mechanical mobility, 6-DOF, MIMO technique, hub, vehicle suspension

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1. INTRODUCTION

Global competition has increased the demand to develop products at lower costs and in shorter timelines that fulfill or even surpass the customer expectations. The product design process has drastically changed with CAE. To develop competitive products, good predictions of the final products performances are required already in early design phases. Functional performances such as noise levels, vibration levels, comfort, reliability, and safety have to be estimated with sufficient accuracy in order to ensure fulfillment of legislation and customer requirements [1]. It is often the functional performances that set the quality sign of the product and differentiates it from competitors.

To attain satisfying performance prediction, complex models of the final product are often needed. Today the majority of product developers in industries such as automotive and aerospace uses and relies on structural dynamic models. A typical model is divided into substructures where each part is described either as a simulation or an experimental model. The substructures can be coupled into hybrid models and used to simulate combined parts or the complete structure [2-5]. However, for a correct hybrid model, complete data for both numerical results and measured data are necessary. When coupling substructures, models are often simplified by omitting information. It is often not known which degrees of freedom those (DOF) are important. Due to factors such as complexity, time, costs and quality in the measurement data, rotational degrees of freedom (RDOFs) are often omitted [6]. RDOFs are often ignored in techniques such as transfer path analysis (TPA), experimental modal analysis (EMA), statistical energy analysis (SEA), substructuring and in the coupling techniques component mode synthesis (CMS) and frequency response function based substructuring (FBS). This simplification results in incomplete models with varying outcome of incorrect results.

Several studies have shown the importance of RDOFs along with its residual terms in coupling [7-10]. A problem when measuring RDOFs is to obtain a pure moment excitation, which means that no net force is applied to the structure [11, 12]. The source signal filtering technique [13] overcomes this problem by compensating the input signals with the caused additional force. However, a drawback of the technique occurs with difficulties in data processing when it is not automated. A method where both forces and moments can be measured simultaneously in a single measurement and set-up is the multiple input multiple output (MIMO) technique [11, 12, 14-16]. With one set-up, the loading of the structure is the same in all measurement directions which improves the quality of the data. Excellent reciprocity is required if the mobility should be inverted into an impedance matrix for future use in couplings. The MIMO technique has earlier been tested and validated on a simple beam and plate structure in 2- and 3-DOF [14, 17]. However, as the number of DOFs increases to determine the mobility of a structure, the complexity and number of equations increases as well. To ensure that errors are avoided, this process can become time-consuming. By developing a general approach, the derivation procedure can be simplified and fewer calculations are required.

The objectives of this study were to obtain the mechanical mobility matrix by developing a general approach based on the MIMO technique, and apply it to a 6-DOF measurement of a complex mechanical structure. The objective was further to apply methods to evaluate the quality in the measurement results without requirements of simulation comparisons. An accurate 6-DOF description of a mechanical system is suitable for later use in a hybrid model. Depending on how the hybrid model will be used, the frequency range has to be considered and set requirements for the measurement set-up.

This mobility measurement is intended to be a part of a hybrid model where interior tyre noise can be auralized and predicted. Today, a common and challenging problem is to reduce tyre induced interior noise. By predicting the noise level, components can be improved. A previous study has shown that the structure borne interior tyre noise in the mid frequency range 150-400 Hz can accurately be predicted but require a hybrid formulation based on a full vehicle model [18]. The idea behind this study is to build a new hybrid model that is less complex by considering the structure of the measured mobility as the coupling element between the tyre/road interaction and tyre noise in the cabin of the car. For structure borne interior tyre noise, the hub of the car is considered as the coupling element. A 6-DOF measurement of the hub describes the boundary condition for the connection of the rim. Implementing this boundary information into FE-simulations of a tyre and rim, the vibrations at the connection can be simulated. By filtering this information through binaural transfer functions measured from the hub to an artificial head in the cabin of the car, interior tyre noise can be auralized. Operational measurements of forces and moments [19] can as well be used as input to the model for interior tyre noise auralizations.

With a general approach and a quality evaluation procedure, hubs at different cars can be measured in a straight forward way. One aim of the measurement set-up was to make it general and applicable to different car models. This makes it possible to auralize interior tyre noise in a variety of car models. By varying compounds, components and road profiles in the FE-model, interior tyre noise can be estimated and auralized in an early design phase without building prototypes. Even a sound sketch [20] can be sufficient as an indication in an early design stage. Car manufacturer can also use 6-DOF mobility measurements to update existing FE-models.

2. THEORY

2.1 Mobility

A point on a structure has in general 6 degrees of freedom (6-DOF), 3 translations (x-, y-, and z-direction) and 3 rotations (α-, β-, and γ-direction). The motion of a point in 1-DOF can be described by the definition of mechanical mobility:

$$Y_{translation} = \frac{v}{F} \quad \text{or} \quad Y_{rotation} = \frac{w}{M} \tag{1}$$

The variable v is the complex velocity-response in any direction, and F is the complex excitation force in any direction. The response can either be a translational (v) or a rotational (w) velocity and the excitation force can either be a rectilinear force (F) or a moment (M) [21]. The 6-DOF input vector $\mathbf{f_M}$ and velocity-response vector $\mathbf{v}$ can be written as:

$$\mathbf{v} = \begin{bmatrix} \mathrm{v}_{\mathrm{x}} \\ \mathrm{v}_{\mathrm{y}} \\ \mathrm{v}_{\mathrm{z}} \\ \mathrm{w}_{\alpha} \\ \mathrm{w}_{\beta} \\ \mathrm{w}_{\gamma} \end{bmatrix} \qquad \mathbf{f_M} = \begin{bmatrix} F_x \\ F_y \\ F_z \\ M_\alpha \\ M_\beta \\ M_\gamma \end{bmatrix} \tag{2}$$

The point can as well be an area, as long as it is rigid and the size is small in comparison to the wavelength in the structure. This condition state the upper frequency limit where the theory of mobility is valid for an area [22]. To fully describe the structural behavior of an area, the interactions between all DOFs have to be taken into account. This results in a 6-by-6 mobility matrix **Y**:

$$\mathbf{Y} = \begin{bmatrix} Y_{xx} & Y_{xy} & Y_{xz} & Y_{x\alpha} & Y_{x\beta} & Y_{x\gamma} \\ Y_{yx} & Y_{yy} & Y_{yz} & Y_{y\alpha} & Y_{y\beta} & Y_{y\gamma} \\ Y_{zx} & Y_{zy} & Y_{zz} & Y_{z\alpha} & Y_{z\beta} & Y_{z\gamma} \\ Y_{\alpha x} & Y_{\alpha y} & Y_{\alpha z} & Y_{\alpha\alpha} & Y_{\alpha\beta} & Y_{\alpha\gamma} \\ Y_{\beta x} & Y_{\beta y} & Y_{\beta z} & Y_{\beta\alpha} & Y_{\beta\beta} & Y_{\beta\gamma} \\ Y_{\gamma x} & Y_{\gamma y} & Y_{\gamma z} & Y_{\gamma\alpha} & Y_{\gamma\beta} & Y_{\gamma\gamma} \end{bmatrix} \tag{3}$$

Only 25% of the mobility matrix is described if the RDOFs are neglected and thereby is the structural behavior not fully described. From Eq. (2) and Eq. (3), the mobility in 6-DOF in matrix vector format is:

$$\mathbf{v} = \mathbf{Y}\mathbf{f_M} \tag{4}$$

From Eq. (4), a 6-DOF system is described by 6 force inputs, 6 velocity-responses and 36 mobility terms. Since the system is underdetermined it can not be independently solved in this instance. Additional procedures are therefore necessary.

2.2 Moment excitation

To create a moment without a net force, two excitation points are normally required as a minimum. A typical method is to attach a lever to a rigid part of a structure, Fig. 1. The weight of the lever should be light in comparison to the structure so the mass loading of the structure and change of mass centre become minimal.

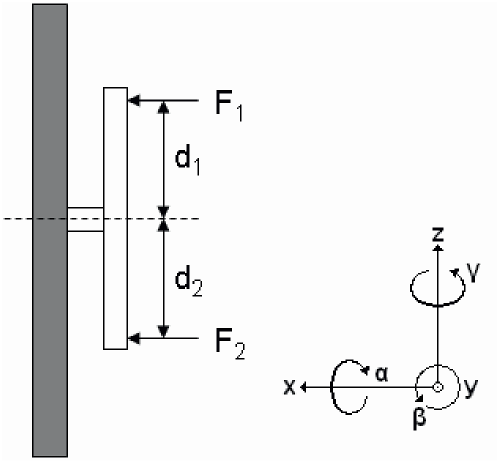

Fig. 1. Excitation of translation and rotation using a T-lever.

When force F_1 is equal to F_2 with a 180 degrees phase shift, pure moment excitation (F=0) is obtained. Due to mass loading by the measurement equipment, this condition can not be fulfilled. To overcome this, a method called source signal filtering technique can be applied [13]. Adjustments are made at each excitation frequency to compensate for unwanted forces which results in erroneous translation- and rotational velocities. This method is time consuming unless the compensation process can be automated [13].

2.3 MIMO system

A MIMO technique takes away the need of a pure moment excitation by simultaneously using multiple uncorrelated input signals where each input is represented at an excitation point. If two signals F_1 and F_2 are applied to the structure in Fig. 1, both a moment and a force are applied at the attachment point of the lever. The input signals are uncorrelated but since they act on the same structure the forces become partly correlated. The force and moment applied to the structure in Fig. 1 are:

$$F_x = F_1 + F_2 \tag{5}$$
$$M_\beta = F_1 d_1 - F_2 d_2 \tag{6}$$

Eq. (5) and Eq. (6) can be written in matrix form according to:

$$\begin{bmatrix} F_x \\ M_\beta \end{bmatrix} = \begin{bmatrix} 1 & 1 \\ d_1 & -d_2 \end{bmatrix} \begin{bmatrix} F_1 \\ F_2 \end{bmatrix} \tag{7}$$

The input vector $\mathbf{f_M}$ is divided into a distance matrix **D** and a force vector **f**. The distance matrix describes the geometry of how the structure is excited in the different DOFs. Eq. (7) can be expressed in matrix vector format:

$$\mathbf{f_M} = \mathbf{Df} \tag{8}$$

When the system is expanded for full 6-DOF excitation, either a more complex single lever is needed or a combination of multiple simpler levers. One example of a 6-DOF lever can be found in [23].

Inserting Eq. (8) into Eq. (4) the MIMO system can be described by the location of the excitation points:

$$\mathbf{v} = \mathbf{YDf} \tag{9}$$

Transposing Eq. (9) and multiply it with the complex conjugate of the input vector is equivalent to applying a least square principle for noise reduction at the output [24]. This procedure extends the system from an underdetermined system to an independently solvable mobility terms:

$$\mathbf{f}^*\mathbf{v}^\mathbf{T} = \mathbf{f}^*\mathbf{f}^\mathbf{T}\mathbf{D}^\mathbf{T}\mathbf{Y}^\mathbf{T} \tag{10}$$

Multiplying the complex conjugate of the input vector with the transpose of the output vector yields the input/output cross-spectral density matrix [15]:

$$\mathbf{S}_{Fv} = \frac{2}{T}E\left(\mathbf{f}^*\mathbf{v}^\mathbf{T}\right) \tag{11}$$

The input spectral density matrix is obtained when the complex conjugate of the input vector is multiplied with the transpose of itself:

$$\mathbf{S}_{FF} = \frac{2}{T}E\left(\mathbf{f}^*\mathbf{f}^\mathbf{T}\right) \tag{12}$$

Multiplying both sides in Eq. (10) with (2/T) and using the expected values, the formula can be expressed with spectral density matrices:

$$\mathbf{S_{Fv}} = \mathbf{S_{FF}}\mathbf{D}^\mathbf{T}\mathbf{Y}^\mathbf{T} \tag{13}$$

which can be rearranged into:

$$\mathbf{S_{FF}}^{-1}\mathbf{S_{Fv}}\left(\mathbf{D}^\mathbf{T}\right)^{-1} = \mathbf{Y}^\mathbf{T} \tag{14}$$

By transposing Eq. (14), the mobility matrix is solvable in a general approach for any number of DOFs:

$$\mathbf{Y} = \mathbf{S_{Fv}}^\mathbf{T}\left(\mathbf{S_{FF}}^\mathbf{T}\right)^{-1}\mathbf{D}^{-1} \tag{15}$$

Since all spectral density functions can be calculated and exported from the data acquisition system, the mobility can be solved in a convenient way. It can be noted that the product of the inverse spectral density matrix and the input/output cross-spectral density matrix describes the input/output frequency response function (FRF) matrix. Eq. (15) can thereby be written as:

$$\mathbf{Y} = \mathbf{H}_{\mathbf{Fv}}^{\mathbf{T}} \mathbf{D}^{-1} \tag{16}$$

2.4 Quality evaluation of mobility data

Validation of mobility data by comparing measurements against simulation results can be difficult to achieve for larger and more complex structures. Simulation models often lack the full body behavior since RDOFs are neglected. Without comparisons, some interpretations of the measurement quality can however be done by using methods such as reciprocity, coherence functions and random error.

2.4.1 Reciprocity

The mobility matrix describes a linear mechanical system. The diagonal in the matrix correspond to the point mobility in each DOF. For a linear system the off-diagonal mobility functions should be symmetrical. For example, if the mobility terms Y_{xy} is equal to Y_{yx}, a good reciprocity is obtained. By examining these reciprocal elements, the quality of the measurement can be evaluated.

2.4.2 Multiple and partial coherence functions

An ordinary input to output coherence function can not be properly used for quality evaluation of a MIMO system since the response at the output is contributed from multiple mutually correlated inputs. By conditioning out the linear dependence between all inputs using optimum linear least squares relationships, the inputs become mutually uncorrelated and the partial coherence function can be estimated. To obtain as good linear dependence and thereby as high coherence as possible, the inputs has to be ordered. Various procedures exist, such as ordering after ordinary coherence functions [16]. The partial coherence functions can be summed into the multiple coherence function which reveals the quality and noise in the measurement. The multiple coherence function satisfies the condition $0 \le \gamma_{out:in}^{2}(f) \le 1$ and can be calculated according to [15]:

$$\gamma_{j:q!}^{2} = 1 - \prod_{i=1}^{q} (1 - \gamma_{ij\cdot(i-1)!}^{2}) \qquad \text{where} \qquad \begin{matrix} \mathrm{i} = 1, 2, ..., \mathrm{q} \\ j = x, y, z, \alpha, \beta, \gamma \end{matrix} \tag{17}$$

The index $\mathrm{i} = 1,2,...,\mathrm{q}$ represents the number of inputs and index j the requested DOF in the MIMO system. Each partial coherence function is calculated as:

$$\gamma_{ij\cdot(i-1)}^{2} = \frac{\left|S_{ij\cdot(i-1)!}\right|^{2}}{S_{ii\cdot(i-1)!} S_{jj\cdot(i-1)!}} \tag{18}$$

The partial coherence functions are determined from conditioned auto and cross spectral density functions. Each conditioned auto and cross spectral density function can be calculated from the previously computed spectra of the original spectral functions used in Eq. (15). The partial coherence functions satisfy the condition $0 \le \gamma^2_{ij\cdot(i-1)}(f) \le 1$. The complete procedure of deriving each partial coherence functions can be found in [15, 16].

2.4.4 Random error

The quality of each input/output frequency response function in Eq. (16) can be estimated by using the partial coherence function. Since each function is considered as normally distributed, the normalized random error can be calculated according to [16]:

$$\varepsilon\left[\left|\hat{H}_{ij}\right|\right] = \frac{\sqrt{(1-\gamma^2_{ij\cdot(i-1)!})}}{\left|\gamma_{ij\cdot(i-1)!}\right|\sqrt{2(n_d+1-i)}} \tag{19}$$

The variable n_d represents the number of averages used in the measurement. When the normalized random error is sufficiently small, the uncertainty of the phase factor estimates can be approximated as:

$$\Delta\hat{\phi}_{iy} \approx \varepsilon\left[\left|\hat{H}_{iy}\right|\right] \tag{20}$$

3. MEASUREMENT SET-UP

In this study, a medium size station wagon was used as test object and the left front hub was the measurement object. The set-up had to be capable of measuring 6-DOF in one set without any changes or alterations. Loading of the structure is therefore identical for both forces and moments which prevents inconsistencies. To provide a high repeatability and measurement accuracy, the set-up had to be stable and rigid. With no tyre or rim mounted on the hub, the aim was to attach all equipment directly to the braking disc. This implies that transformation or additional post processing of the mobility matrix is not necessary.

Previous studies have shown that a moment can be created by a pair of shakers [12] or impact hammers [25]. Due to long measurement sets and good repeatability, electrodynamic shakers were considered as beneficial for this application compared to impact hammer excitation. The direction and positioning of the shakers are also more accurate compared to the impact hammer testing. Shakers fulfill the condition of a general set-up that only requires minor adjustments between car models.

3.1 Braking disc

Attachment points as far apart as possible are desirable to create large moments. Due to the circular shape, convenient positions are hard to find. In addition a front disc usually has air-channels for cooling which make the structure more flexible. This implies that the frequency range of a rigid body is decreased since the frequency of the first bending mode is lowered compared to a solid disc. Therefore, the excitation points have to be attached to the stiffer center part of the disc which can result in too close positions to each other. Adjacent excitation points leads to an equation system that tends to be singular, followed by an increased measurement error. A smaller distance between the input forces also decreases the moment excitation. This has to be compensated with larger input forces and thereby larger shakers.

For these reasons, a special breaking disc was fabricated where the excitation points could be placed further apart, Fig. 2. By making it into a quadratic shape, no levers are needed and the excitation and measurements points can be evenly spaced. For excitation in the α-direction, a cantilever beam had to be mounted at the bottom of the square disc. The disc was milled from a steel plate with the same mass centre and weight of the ordinary front braking disc to prevent additional mass loading. The distance from the centre to a corner is similar to the radius of the circular disc. This makes the area of the square disc smaller compared to an ordinary circular disc. To keep the mass constant, the loss of area is compensated in thickness for the quadratic shape disc. For the transducers, this extra thickness would imply that the mobility is defined in a different plane compared to a regular braking disc. To avoid transformation calculations, small segments were lowered in the disc for the transducers. In this way the mobility is measured in the same plane as an ordinary disc. By making the disc solid and thicker the structure behaves as a rigid body in a larger frequency range. The modal properties of the brake disc were measured with a scanning Laser Doppler Vibrometer (PSV 300, Polytec GmbH). The first bending mode of the disc was detected at 858 Hz at the cantilever beam used for exciting a moment in the α-direction. For the plate structure of the disc, the first bending mode was detected at 2003 Hz.

Since no wheel was mounted, the suspension had to be loaded. A combination of a mechanical structure (steel block) together with rubber sheets was placed in between the floor and the lower suspension arm to simulate the spring and damping properties of the tyre and rim. The loading of the hub was checked by measuring the height from the ground to the center of the hub. When the height was the same as with a wheel mounted, the loading was considered as equivalent.

3.2 Excitation and shaker arrangement

The aim was to attach all measurement equipment directly to the braking disc. All shakers had to fit inside the wheelhouse and underneath the car. Because of the limited space, small shakers were necessary. With a maximum deliverable force of 17.8 N, LDS V201 electrodynamic shakers powered by LDS PA 150 amplifiers was chosen.

To excite the structure in 6-DOF, 6 shakers were used, arrangement pair wise, orthogonal and uncoupled to each other. To fit one pair underneath the car, it had to be parked on 50 mm concrete slabs. The other two pairs were mounted on individual mechanical benches loaded with blocking masses. Each shaker pair was mounted so that the distance and alignment could be adjusted accurately. The excitation points and applied forces can be seen in Fig. 2.

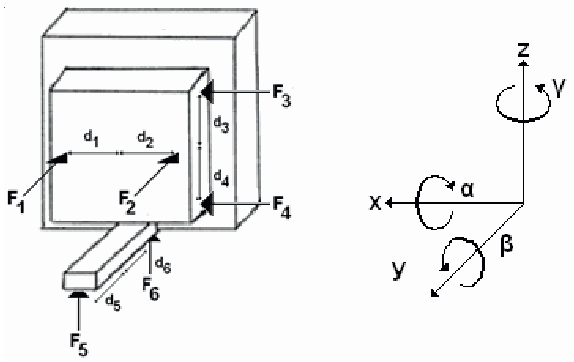

Fig. 2. Square shaped braking disc.

From Fig. 2, the square shaped braking disc is excited in 6-DOF by the forces and caused moments:

$$F_x = F_3 + F_4 \tag{21}$$

$$F_y = -F_1 - F_2 \tag{22}$$

$$F_x = F_5 + F_6 \tag{23}$$

$$M_\alpha = F_5 d_5 - F_6 d_6 \tag{24}$$

$$M_\beta = F_3 d_3 - F_4 d_4 \tag{25}$$

$$M_\gamma = -F_1 d_1 + F_2 d_2 \tag{26}$$

The distances d_1- d_4 were 70 mm while d_5 was 66 mm and d_6 34 mm. Eq. (21) to Eq. (26) can be expressed in matrix format by the distance matrix and force vector according to Eq. (8):

$$\begin{bmatrix} F_x \\ F_y \\ F_z \\ M_\alpha \\ M_\beta \\ M_\gamma \end{bmatrix} = \begin{bmatrix} 0 & 0 & 1 & 1 & 0 & 0 \\ -1 & -1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 1 \\ 0 & 0 & 0 & 0 & d_5 & -d_6 \\ 0 & 0 & d_3 & -d_4 & 0 & 0 \\ -d_1 & d_2 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} F_1 \\ F_2 \\ F_3 \\ F_4 \\ F_5 \\ F_6 \end{bmatrix} \tag{27}$$

3.3 Transducers

The forces applied to the braking disc were measured with force transducers mounted on the disc. Since the structure is excited in 6-DOF, the force transducers will move in same directions. This movement will load the transducer with unwanted tangential forces which causes a measurement error. The size of the error will increase with the height of the transducer as it works as a lever. In order to minimize the tangential force and the measurement error, small force transducers should be used instead of impedance heads

[26]. In this experiment B&K 8230 force transducers were used. To minimize the error further, stingers used for energy distribution from the shakers to the force transducers should have a low bending stiffness but be stiff in compression. To fulfill this, stingers were turned from Polyoxymethylene-plastic (POM).

To determine the full structural behavior at the center of the hub (the spindle), multiple accelerometers was used. To theoretically obtain as high frequency range as possible, the accelerometers should be placed as far out as possible on the structure. However, it has to be considered that the area has to be much smaller than wavelength governing the propagation [22].

Small triaxial-accelerometers were placed close to each corner where all distances a_1- a_6 were 60 mm, Fig. 3. With light accelerometers (B&K 4524B) the mass loading of the structure was neglected.

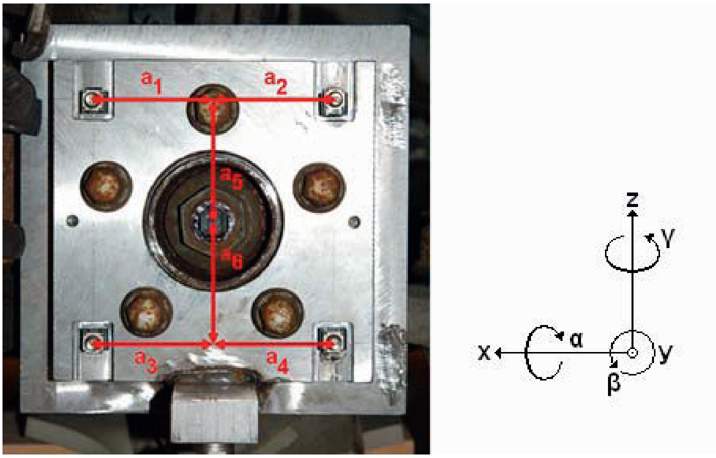

Fig. 3. Positioning of accelerometers.

The velocities at the center point of the hub in all DOFs are described by summarizing the vibratory responses according to Eq. (28) to Eq. (33): This summation can either be done in time domain or frequency domain.

$$\dot{x} = \frac{A_1(\dot{x}) + A_2(\dot{x}) + A_3(\dot{x}) + A_4(\dot{x})}{4} \tag{28}$$

$$\dot{y} = \frac{A_1(\dot{y}) + A_2(\dot{y}) + A_3(\dot{y}) + A_4(\dot{y})}{4} \tag{29}$$

$$\dot{z} = \frac{A_1(\dot{z}) + A_2(\dot{z}) + A_3(\dot{z}) + A_4(\dot{z})}{4} \tag{30}$$

$$\dot{\alpha} = \frac{\left(\dfrac{A_1(\dot{y}) + A_3(\dot{y})}{2}\right) - \left(\dfrac{A_2(\dot{y}) + A_4(\dot{y})}{2}\right)}{a_5 + a_6} \tag{31}$$

$$\dot{\beta} = \frac{\left(\dfrac{A_1(\dot{x}) + A_2(\dot{x})}{2}\right) - \left(\dfrac{A_3(\dot{x}) + A_4(\dot{x})}{2}\right)}{a_1 + a_2} \tag{32}$$

$$\dot{\gamma} = \frac{\left(\dfrac{A_1(\dot{x}) + A_3(\dot{x})}{2}\right) - \left(\dfrac{A_2(\dot{x}) + A_4(\dot{x})}{2}\right)}{a_1 + a_2} \tag{33}$$

3.4 Data acquisition and auxiliary equipment

For data acquisition, B&K PULSE software was used along with B&K LAN XI interfaces, 3160-B-4/2 and 3050-B-6/0. All transducer settings, measurements and calculations of input and input/output spectrums were done in PULSE. Random signals were generated and fed to the power amplifiers for the shakers. Both auto-spectrums of the input signals and cross-spectrums between input and output signals were calculated in the frequency range 0-800 Hz with 0.25 Hz resolution. This resolution was considered as enough to detect narrow and close peaks. A Hanning window with 50% overlap was used and each measurement consisted of 30 averages. A high-pass filter with a cut-off frequency of 0.7 Hz was used for all transducers. The complete 6-DOF measurement set-up of all transducers and shaker can be seen in Fig. 4.

Fig. 4. Complete 6-DOF measurement set-up.

4. RESULTS

The LDV measurements showed that the structure is rigid up to 858 Hz. As a rule of thumb for transducers, the upper frequency limit of a transducer is set to one-third of its resonance frequency [24]. Applying the same statement for this measurement, no influences from bending modes can be expected below 300 Hz. Therefore, only results up to 300 Hz will therefore only be shown.

4.1 Mobility

The measured point mobility in all 6-DOFs can be seen in Fig. 5 and 6.

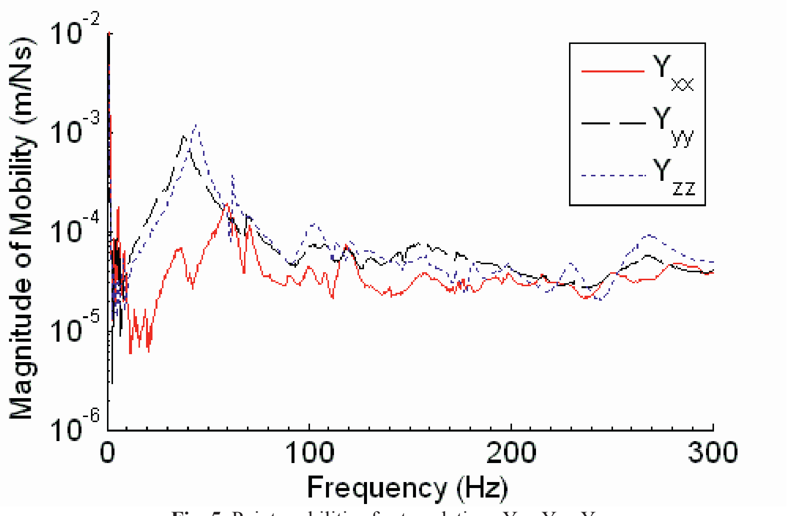

Fig. 5. Point mobilities for translations Y_{xx}, Y_{yy}, Y_{zz}.

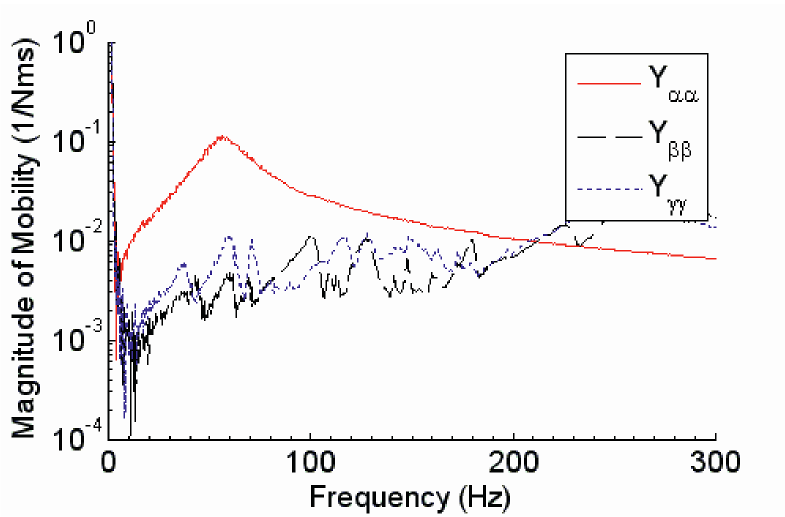

Fig. 6. Point mobilities for rotations $Y_{\alpha\alpha}$, $Y_{\beta\beta}$, $Y_{\gamma\gamma}$.

4.2 Reciprocity

The linearity of the system and the quality in the measured mobility functions can be evaluated by examining the symmetry in the off-diagonal (reciprocal) elements in the mobility matrix. In Fig. 7 to 9 the reciprocity is viewed between translational and rotational mobility terms.

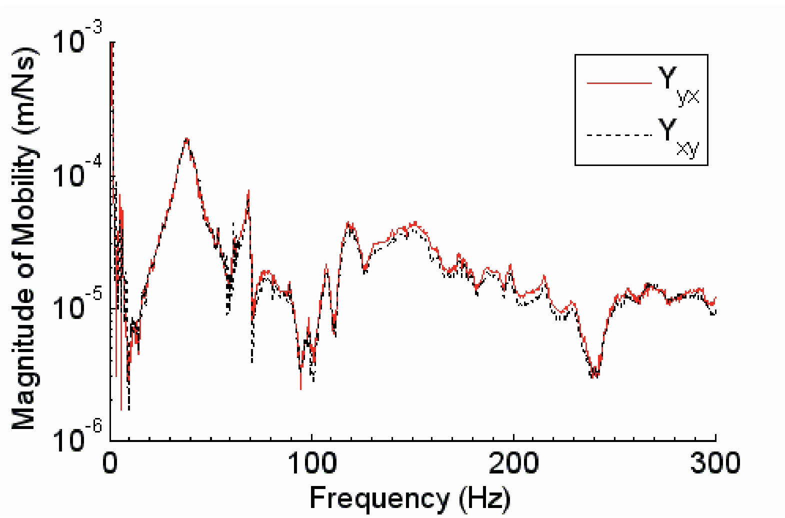

Fig. 7. Reciprocity between the translational mobilities Y_{xy} and Y_{yx}.

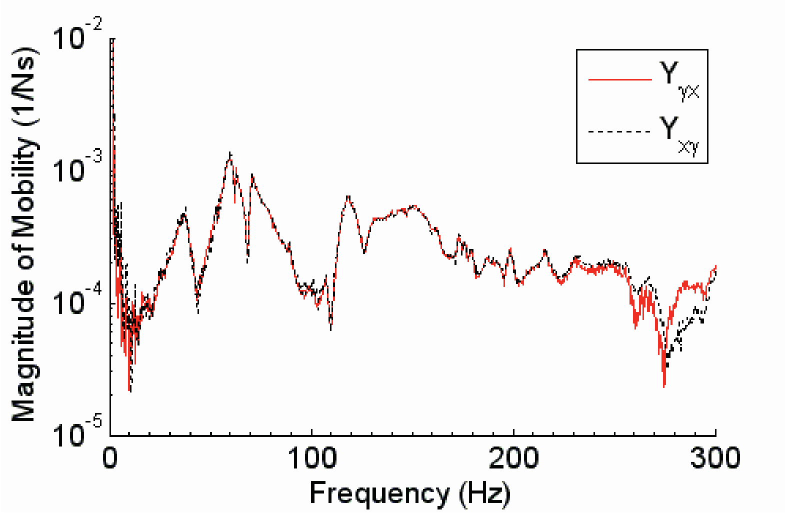

Fig. 8. Reciprocity between the translational mobility $Y_{x\gamma}$ and the rotational mobility $Y_{\gamma x}$.

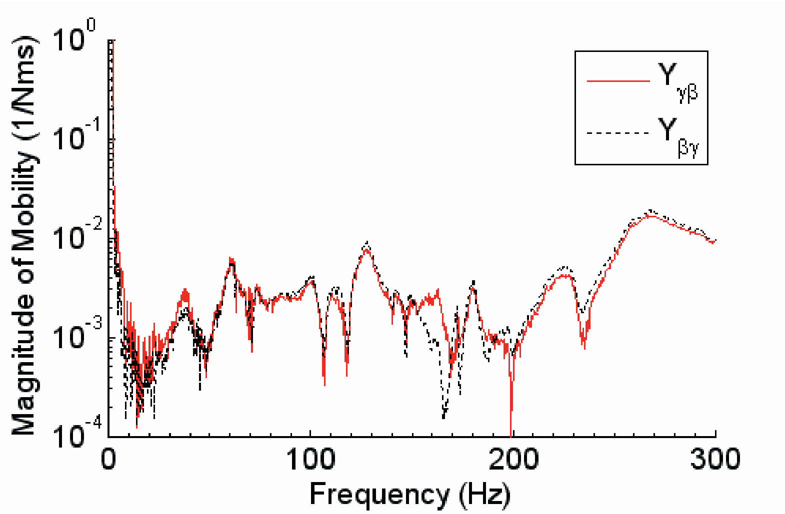

Fig. 9. Reciprocity between the rotational mobilities $Y_{\beta\gamma}$ and $Y_{\gamma\beta}$.

The reciprocity in other direction shows similar results which indicate a good measurement quality.

4.3 Multiple and partial coherence functions

To achieve as high multiple coherence as possible, good partial coherence functions are essential. The quality in each partial coherence function is determined from their linear estimate. The estimates are in turn dependent on the ordering of the inputs. A common way is to arrange the inputs after each ordinary coherence function [16], Fig. 10.

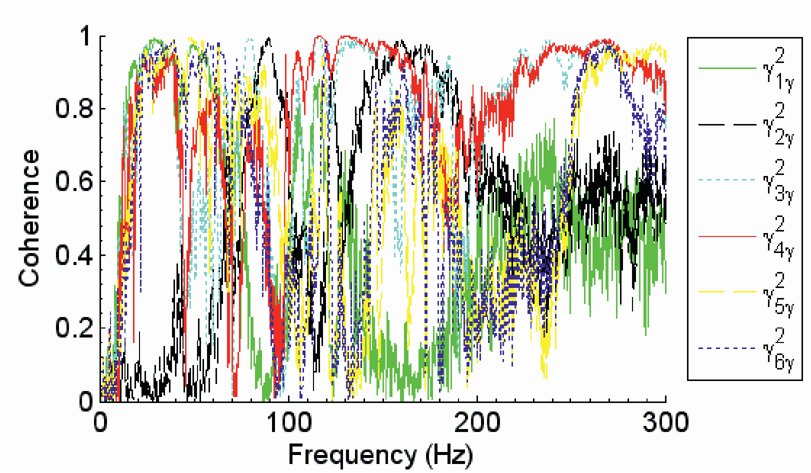

Fig. 10. Ordinary coherence functions from all 6 inputs to output in γ-direction**.**

For a larger MIMO system, as in this case with 6 inputs, the ordering process is harder to achieve. In Fig. 10, the inputs with the highest ordinary coherence function change over the frequency interval. For optimum linear estimates, a new ordering has to be made for each shift between the coherence functions. If the procedure is not automated, it can be time consuming. In this case, the ordering was made from the calculated mean value of the ordinary coherence functions in the frequency interval 0-300 Hz. Each partial coherence function was calculated according to Eq. (18). The complete procedure of partial coherence calculation can be found in [15, 16].

The multiple coherence was calculated from Eq. (17), see Fig. 11.

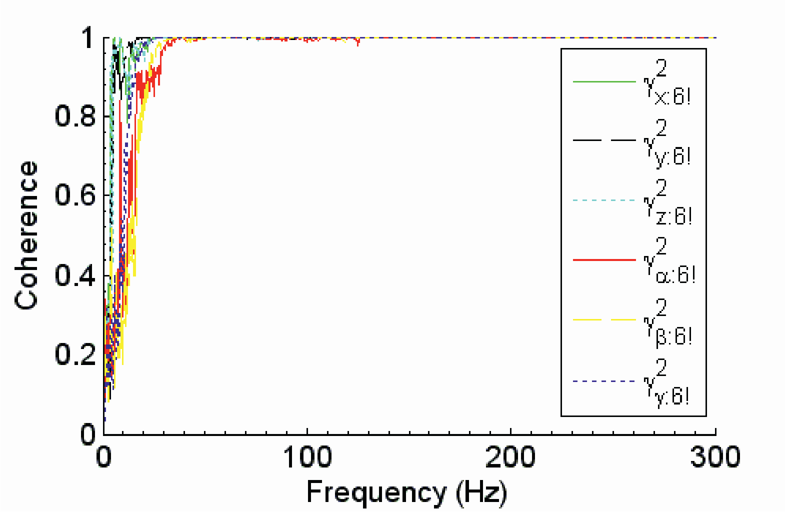

Fig. 11. Multiple coherence for all DOFs.

In Fig. 11, the multiple coherence is close to one over 20 Hz for all DOFs except in α- and β-direction which are close to one over 40 Hz.

4.4 Random error

The random error is calculated from Eq. (19). Random error in the frequency response functions from each input to the output in γ-direction is shown in Fig. 12.

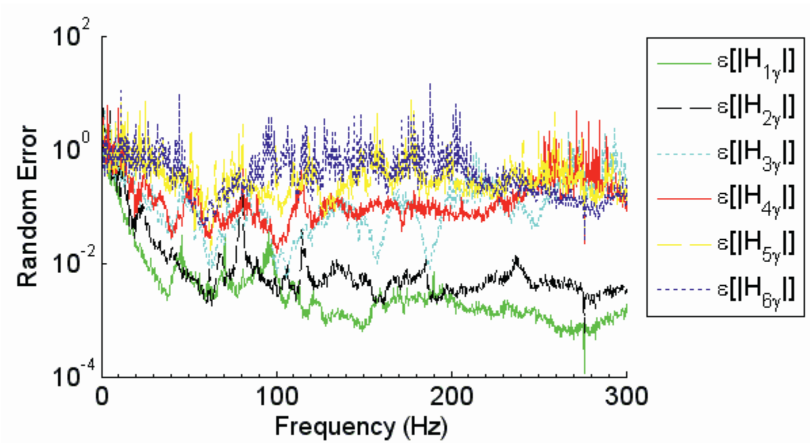

Fig. 12. Random error of frequency response functions in γ-direction.

In Fig. 12, the shakers exciting in the γ-direction (1 and 2) provide a much smaller random error compared to the other inputs. All random error functions are lower than 10 which can be considered as sufficiently low [16], and therefore is Eq. (20) fulfilled. The uncertainty of the phase factor estimate can therefore be approximated to the random error in gain factor of the frequency response functions. The random error in other DOFs shows a random error in the same magnitude as in the γ–direction.

5. CONCLUSION

The general method derived to determine and measure the mechanical mobility in 6-DOF has been shown to work well even for a complex structure. This method can be implemented in a variety of applications were the structural behavior needs to be known. The magnitude of the mobility is approximately in the same in all DOFs. This implies that the direction which has the highest force or moment input excitation will be the degree of freedom which will yield the highest vibratory response which in turn will be transferred to the rest of the structure. Therefore, RDOFs cannot be omitted if the input force or moment is known in advance as little in comparison with translational DOFs. Results in the frequency range below 300 Hz showed a good quality in reciprocity and multiple coherence. Normally, a good reciprocity can be hard to achieve for RDOFs. Despite the non-optimum input ordering, the multiple coherence is close to one except at in the frequency interval 0-20 Hz and 0-40 Hz. This indicates a model with low noise and good estimates of the linear relationships in the partial coherence functions. The low multiple coherences below 20 Hz are mostly due to low forces from the shakers. This could be improved by splitting the measurement in two frequency intervals, 0-20 Hz and 20-300 Hz.

To improve the frequency range above 300 Hz, the construction of the braking disc can be made by strengthen the cantilever beam with a wedge according to Fig. 13. The measurement range will be increased since the first bending mode is shifted upwards. Modifications of the disc have then to be made to not affect the mass center of the disc. If the strengthening leads to that the bending mode at 2003 Hz become the lowest, then the possible measurement range could be expanded to 650 Hz. This would cover the frequency range where the structure borne noise from the tyre dominates over the air-borne noise in the car compartment.

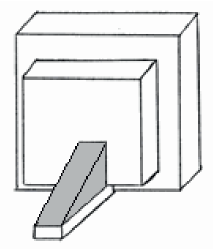

Fig. 13. Improvements of the braking disc.

A drawback with this method is the installation and preparation time of the measurement set-up. To achieve high quality measurements, transducers and stingers has to be aligned accurately which is time-consuming. Due to the developed general method applied to the measurement set-up, changes in the measurement set-up could be evaluated in fast way. Adjustments were made and evaluated in a rather fast and easy way.

The results can further be improved by deriving a general method for ordering and calculating the conditioned partial coherence functions.

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Paper II

Assessment of changes in automotive sounds caused by displacements of source and listening positions

M. Löfdahl, A. Nykänen, R. Johnsson

Assessment of changes in automotive sounds caused by displacements of source and listening positions[1]

Magnus Löfdahl[2], Arne Nykänen[3], Roger Johnsson[4]

ABSTRACT

Artificial head recordings are commonly used to measure and evaluate sound quality. In sound quality assessments, spatial qualities are crucial both for correct localization and separation of sources. A change in the locations of the source and the receiver will alter the character of the sound since the binaural transfer path will alter. Examples of automotive sounds were reproduced through a loudspeaker in the source position outside a car and recorded with an artificial head inside the car. Changes in perceived sound quality caused by displacements of either the source or the receiver position were studied through a listening test to find the just noticeable difference. In addition, binaural transmissibility functions of airborne sounds were measured directly to compare the perceived changes to the visual changes in the binaural transmissibility functions. The results showed that artificial head recordings and measurements of binaural transmissibility functions are affected by small displacements in location for either the source or the receiver. The accuracy in the positioning of the source and the receiver need to be smaller than 2 cm if audible differences shall be avoided. Variability due to displacements causes changes in sound quality that must be considered in order to draw correct conclusions from sound quality analyses based on artificial head recordings and binaural transmissibility function measurements.

Primary subject classification: 13.2.1; Secondary subject classification: 79.2

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1. INTRODUCTION

Binaural recordings and reproductions have become a standard procedure in sound quality evaluations. A common and convenient method is to use artificial heads to binaurally record the sound pressure in the ears. Most of the spatial information is preserved and the sounds can later be reproduced at their original level in the ears of the listener[2]. In sound quality assessments, the spatial qualities are crucial both for correct localization of sources, and for separation of sources from reverberation and background noise[3-4]. The perceived localization is determined by a number of acoustical cues such as interaural time differences (ITD), interaural level differences (ILD) and spectral cues provided by pinna filtering. Results from experiments reveal that four factors influence the relative salience of these cues, *a priori* knowledge of stimulus characteristics, source frequency content, the reliability and plausibility of the cues, and the consistency of the cues across the frequency spectrum[5]. Depending on stimulus used and frequency range of the stimulus, the least audible differences for ITD vary from 9 to 62 μs[6] and the least audible differences for ILD vary between 0.6 to 2 dB. In a real situation where the binaural technique is used, imperfections exist in these cues which cause a localization blur. The localization blur describes the smallest possible change of position of the sound source that causes a just noticeable change of perceived position. In a real life situation, the localization blur has been shown to vary from 1-4° [7].

In the automotive industry, artificial head recordings are commonly used to measure and evaluate the interior sound quality of cars[8]. The environment in the cabin of a car can be considered as complex with reflecting and absorbing surfaces close to the driver and passengers. Effects of displacements can therefore be assumed to be increased. However, it can also be argued that the environment in the cabin creates a diffuse sound field where displacements are less crucial. The noise in the cabin of the car will depend on the transfer paths. Not only are the transfer paths dependent on the frequency range of the generated noise, but also spread through different airborne and structure borne transfer paths. Therefore it can be argued that the effects of changes in transfer paths are more important than the distance and angle between the source and the receiver.

Some studies have been done to assess the variability in the acoustical response of interior sound between nominally identical cars[9-11]. None of these studies have assessed the required precision in the positioning of sources and receivers in binaural measurements used in a complex environment such as a car compartment. However, some studies have been done to determine human sensitivities to irregularities and defects in electro-acoustical transfer functions used for sound reproduction[12-15]. Electro-acoustical transfer functions are commonly smooth compared to transfer functions typically found in mechanical structures of, for example vehicles. The precision of transfer functions measurements has been discussed in several papers[16-20] where the results are commonly presented as graphs. However, it is usually difficult to draw adequate conclusions on the audibility of differences in transfer functions based on such graphs. Since, the results of transfer function measurements usually have to be interpreted as effects on sound, the understanding of the audibility of alteration of transfer functions are essential. The audible effect of changes in frequency resolution of binaural transfer functions used for auralizations of vehicle interior sounds was studied by Nykänen et al.[21]. The auralizations were made from the transfer path of an engine sound into the cabin of a truck. Results showed that auralizations made through BTFs with a resolution of 4 Hz or higher or smoothed with maximum 1/96 octave moving average filters were perceived as similar to artificial head recordings.

The objective of this study was to find the just noticeable differences for displacements of either the source outside the car or listening position in a car compartment. To address the frequency dependence of the transfer paths, two typical automotive sounds were considered, tyre and engine noise. They were chosen since both are prominent sounds under driving conditions but have different characters. Typically, engine sounds have more high frequency content than tyre noise. The aim was not to make an authentic reproduction but rather evaluate the sounds in a typical automotive environment. The result of this study can therefore not be generalized for all kinds of sounds, source positions and receiver positions but may serve as a guideline for binaural measurements done in a car cabin. The just noticeable difference for displacements was determined from a listening test based on comparing recordings made at displaced positions against a reference position. The perceived effect of displacements was also compared against measured differences in the response functions from source to receiver with and without displacements.

2. Method

2.1 Evaluation Methodology

An effective method to detect small differences in reproduced sounds is through the use of a test method called “double-blind triple-stimulus with hidden reference”[22]. It has been developed by the International Telecommunication Union and has been found to be sensitive, stable and accurate in detection and magnitude estimation of small impairments. This method is similar to an ABX-test[23] with additional subject estimation of perceived differences. One subject at a time is presented with three stimuli (“A”, “B” and “REF”). A known reference (from now on called reference stimulus) is always stimulus “REF”. A hidden reference identical to the reference stimulus and an object stimulus are simultaneously available but are randomly assigned to “A” and “B”. The subject is asked to assess the impairments on “A” compared to “REF”, and “B” compared to “REF”. The subject may switch between stimuli at will. The subject is forced to judge one of the sounds (“A” or “B”) as equal and the other different from the reference (“REF”). The method was slightly modified, instead of judging impairment, subjects were asked to judge the difference between the stimuli as differences induced by displacements may not necessarily be perceived as impairments. The difference was judged on a scale ranging from 0 to 100 where 0 = very pronounced and 100 = not audible. The listening test was controlled by the subject through a computer interface as shown in Fig. 1. The subject assessed each displacement once. During the listening test, the subject was free to switch between sounds in any order and as many times as desired. On switching, the first sound was ramped down over a 20 ms period followed by a 30±10 ms pause before the new sound was ramped up over 20 ms. Sound pairs with no audible differences were identified by analysis of the proportion of correct identifications while difference ratings were analyzed for correctly identified sound pairs.

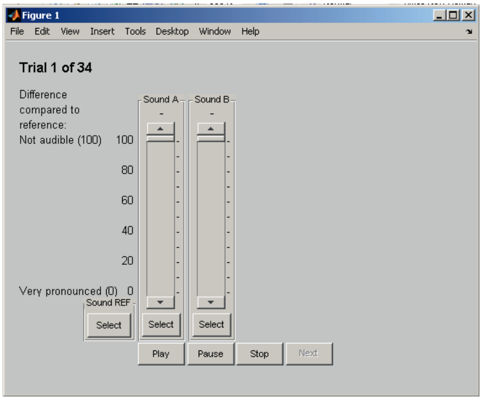

Fig. 1: User interface used in the listening test.

2.2 Measurement setup

The measurement setup was designed to reproduce automotive sounds with easy adjustments of source- and receiver position, hence not to make authentic reproductions of the automotive sounds. All measurements were made at Luleå University of Technology in a semi-anechoic room. The two automotive sounds (tyre and engine noise) were reproduced through a loudspeaker (source) placed outside a car, and recorded with an artificial head (Head Acoustics HMS III) as receiver, placed in the front passenger seat, see Fig. 2. A microphone (Brüel & Kjær 4190) was placed 5 cm in front of the loudspeaker to measure the radiated sound. The position of the artificial head was comparable to the position of a person with the approximate length of 180 cm.

As source reference position, the loudspeaker was placed on the floor in the semi-anechoic room, near the right rear wheel with a distance between the centres of the tyre and loudspeaker of 33 cm, Fig. 2.

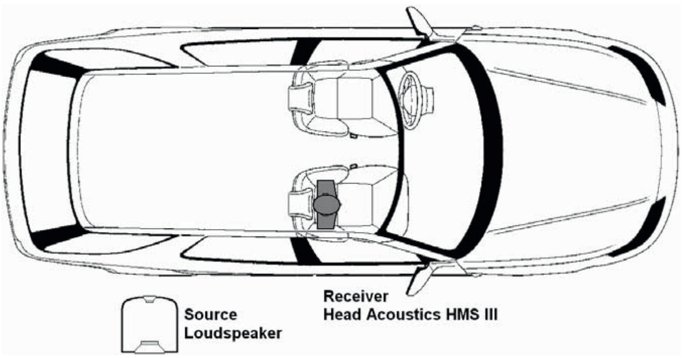

Fig. 2: Measurement set up of source, receiver and car

Data acquisition was made with a Brüel & Kjær PULSE system. All measurements were done on a medium size station wagon. All sounds were reproduced from a CD played through a TEAC CD-5. The signals were fed to a crossover filter (Behringer Ultradrive

Pro) and splitted 3-ways to a power amplifier, Rotel 933. The amplifier was then connected to the loudspeaker. To be able to reproduce the automotive sounds at realistic sound pressure levels, a loudspeaker capable of playing high levels in a wide frequency range was required. A loudspeaker with a compact design and nearly half-spherical directivity was used, see Fig. 3. It was based on a coaxial mid-range and treble element (SEAS T18RE) on the front and a woofer unit (Peerless 269SWR) on the back. Due to the physical distance between the loudspeaker elements, compensations were made in time when reproducing the sounds. The frequency response function for the loudspeaker is shown in Fig. 4.

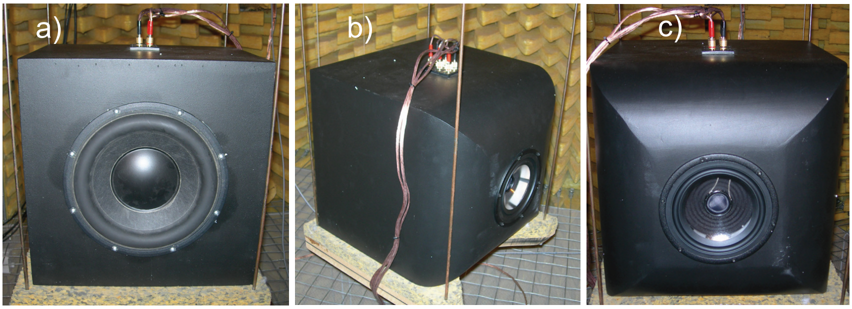

Fig. 3: Loudspeaker used for direct measurements of binaural transmissibility functions and for reproduction of the engine and tyre sounds. The size of the loudspeaker cabinet was 400x400x420 mm. a) woofer unit on back side. b) perspective view. c) coaxial unit on front side.

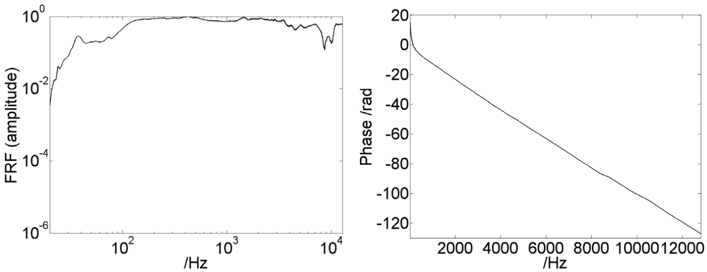

Fig. 4: Amplitude and phase of the frequency response function of the loudspeaker used for direct measurements of binaural transfer functions and for reproduction of engine and tyre sounds.

2.3 Measurement of Binaural Transmissibility Functions

Transfer functions are commonly used for analysis and prediction of the contribution of different sources to the sound in the compartment of vehicles[16-18, 20]. They are often measured reciprocally, since it is usually simpler, faster and cheaper than direct measurements[18]. Another important advantage with reciprocal measurements is that the sensor (microphone) can be placed closer to sound radiating surfaces and in smaller spaces than a sound source (loudspeaker)[19].

Despite the close distance between the microphone and the centre of the loudspeaker cone, the microphone is influenced by the surroundings and the actual source strength is not measured. According to the definition of standards, this implies that it is not a transfer function that is measured but rather a transmissibility function. A transmissibility function is defined between two response functions while a transfer function is calculated by Laplace transformation of the unit impulse response function[24]. In this study direct measurements of binaural transmissibility functions were chosen, since this allowed the use of the same measurement setup both for making binaural recordings of reproduced source sounds and measurements of the binaural transmissibility functions. Therefore, transmissibility functions from the microphone position to each ear of the artificial head were measured using white noise, with 1 Hz frequency resolution. The sampling frequency was 65.5 kHz. A Hanning window was applied to signal sections with a 50 % overlap. The transmissibility function H was estimated according to Eqn. 1,

$$H_1(f) = P_{xy}(f) / P_{xx}(f) \qquad (1)$$

where P_{xy} is the cross power spectral density of the input x and the output y. P_{xx} is the power spectral density of the input x.

2.4 Creation of Object Stimuli

The engine sound was recorded from the same car as for which the binaural transmissibility functions were measured. The microphone (Brüel & Kjær 4190) was placed 22 cm above the centre of the engine block. The engine sound was recorded at 2000 rpm, no gear and with open hood. The tyre sound was recorded at 75 km/h with a specially designed trailer for near field recording of tyres[25]. The recording was made with a studded tyre on a dry asphalt road.
To achieve controlled listening conditions stationary sounds were used, i.e. tyre sound at constant speed and engine sound at constant engine speed. The frequency spectrum of both sounds can be seen in Fig. 5. The volume was adjusted so that the levels of the reproduced sounds were equivalent to their originally recorded levels.

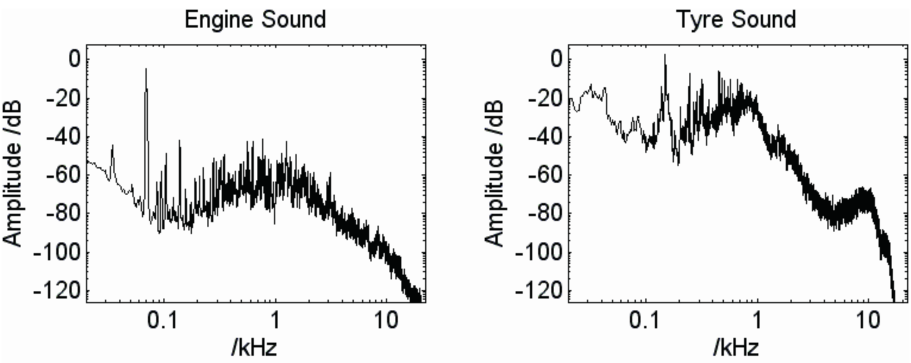

Fig. 5: Spectrum of engine and tyre sound

Two reference positions were used depending on whether the source or the receiver was displaced. When the source was displaced, the receiver (artificial head) was positioned corresponding to a 180 cm tall person. As source reference position (Reference 1), the loudspeaker was placed near the right rear wheel. The loudspeaker was then displaced 0.5, 1, 2, 4, 8, 16 and 32 cm from the Reference 1 position in the longitudinal direction (heading of the car), Fig. 6.

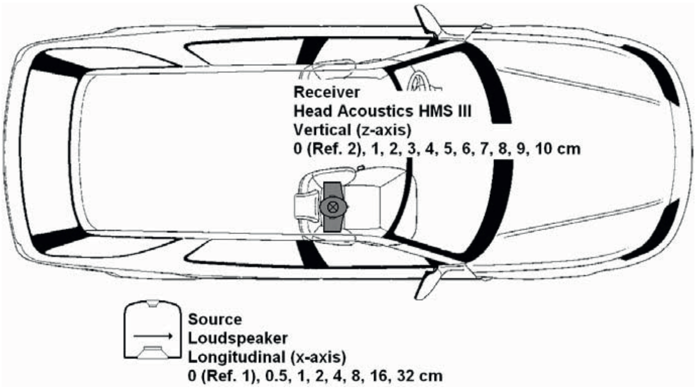

Fig. 6: Reference positions and displacement of source and receiver

When the receiver was displaced, the source was positioned at Reference 1 (no displacement). The receiver reference position (Reference 2) corresponded to a 185 cm tall person. The artificial head was then lowered 1, 2, 3, 4, 5, 6, 7, 8, 9 and 10 cm in the vertical direction, Fig. 6. For every position of the loudspeaker and artificial head, both the engine and tyre sounds were reproduced through the loudspeaker and recorded with the artificial head.

2.5 Listening test

The listening test compared pairs of reference stimuli (recorded at the reference positions) and object stimuli (recorded at the displaced positions). Displacements of the loudspeaker in the longitudinal-direction (heading of the car) and the artificial head in the vertical-direction (height of the listener) were studied. The order of tests was randomised for each subject and was preceded by a training session which consisted of 6 pairs of sounds. The stimuli used in the training session were randomly chosen from the listening test. The sounds were reproduced through equalised headphones, Head Acoustics HPS IV. The sound pressure level of the stimuli was 80 dB(A).

A total of 20 subjects participated, 2 women and 18 men. All were volunteers. The mean age was 28 years (standard deviation 6 years). All subjects had self-reported normal hearing.

2.6 Analysis of the Listening Test

The analysis of the results was divided into two parts. First, sound pairs where the object stimulus (displaced position) could not be separated from the reference stimulus (reference position) were identified. This was made by comparing the number of subjects who correctly identified the object stimulus with the total number of subjects. The hypothesis was that the difference between the object stimulus and the reference stimulus could be heard. The null hypothesis was that the object stimulus sounded the same as the reference stimulus. For a binomial distribution, 14 out of 20 subjects needed to be correct to reject the null hypothesis at 95 % confidence level. This way, sound pairs too similar for correct discrimination between the object stimulus and the reference stimulus could be identified. For the pairs where the object stimulus was correctly identified by the group, the difference between ratings of the reference stimulus and the object stimulus was analyzed. Before analysis, the score difference rating were normalised with respect to the mean for the individual subject and for the group, according to Eqn. 2 – 4,

$$Z_{ij} = x_{ij} \frac{\bar{x}_{group}}{\bar{x}_i} \qquad (2)$$

$$\bar{x}_{group} = \frac{\sum_{i=1}^{m} \sum_{j=1}^{n} |x_{ij}|}{mn} \qquad (3)$$

$$\bar{x}_i = \frac{\sum_{j=1}^{n} |x_{ij}|}{m} \qquad (4)$$

where x_{ij} = score difference for subject i and sound pair j, Z_{ij} = normalised difference for subject i and sound pair j, $\bar{x}_i$ = mean of absolute score difference for subject i (sound pairs where the object was not successfully identified by the group were excluded), $\bar{x}_{group}$ = mean of absolute score difference for all subjects (sound pairs where the object was not successfully identified by the group were excluded), m = number of subjects and n = number of sound pairs (excluding pairs where the object was not successfully identified by the group).

3. Results

In Fig. 7 and 8, the numbers of correct identifications for each sound pair are reported. The results showed that displacements smaller than 1 cm were not audible. Differences caused by displacements larger than 2 cm were clearly heard by almost all participants. This shows that the accuracy in positioning needed to avoid audible differences in artificial head recordings in a car compartment is 1 cm.

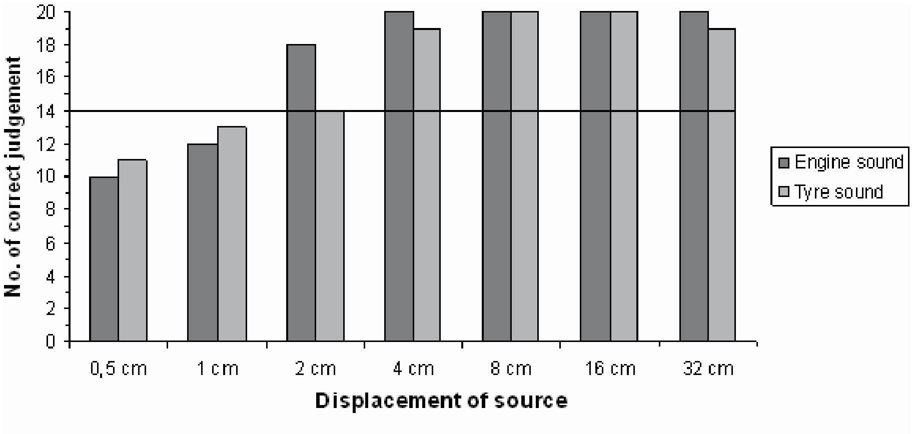

Fig. 7: Correctly identified sound pairs for displacement of source

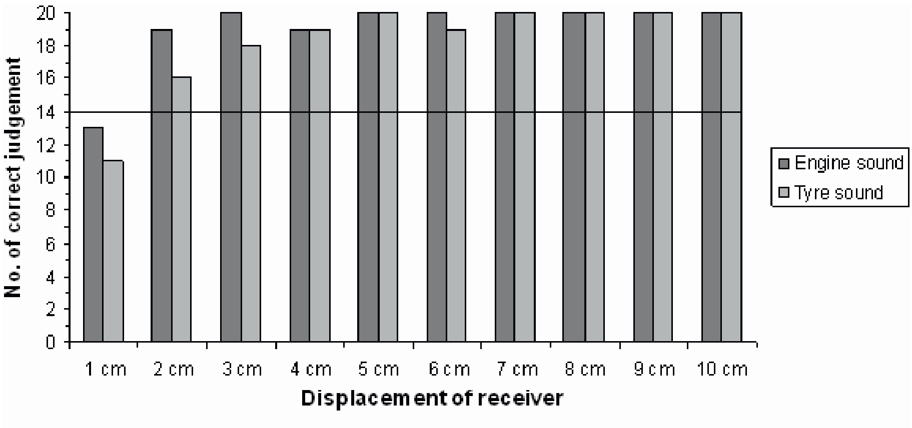

Fig. 8: Correctly identified sound pairs for displacement of receiver

In Fig. 9 and 10 means and 95 % confidence intervals for the normalised difference ratings between the reference position and the displaced position are reported as functions of the displacement. In Fig. 9 comparisons are made between the stimuli when the source is displaced. No significant differences can be seen between tyre and engine sound for displacements of the source. In Fig. 10 tyre and engine sound are compared for displacements of the receiver. For this case the differences between the reference and the object stimuli were judged higher for displacements up to 6 cm. Hence, engine sounds seem to be more affected perceptually by displacements of the receiver than tyre sounds. This may be explained by more high frequency content in the engine sounds. Comparing Fig. 9 and Fig. 10 shows that the perceptual change in sound quality was approximately the same when the receiver was displaced as when the source was displaced the same distance.

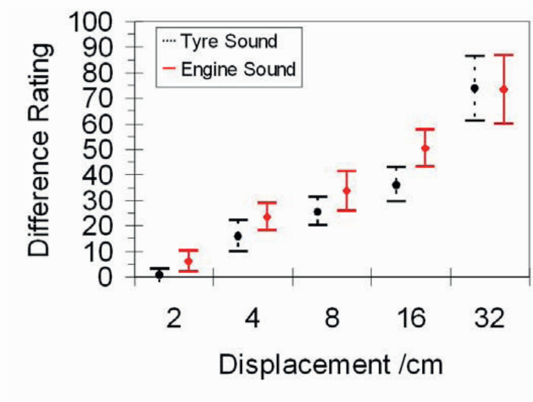

Fig. 9: Means and 95 % confidence intervals (internal s) for displacements of the source.

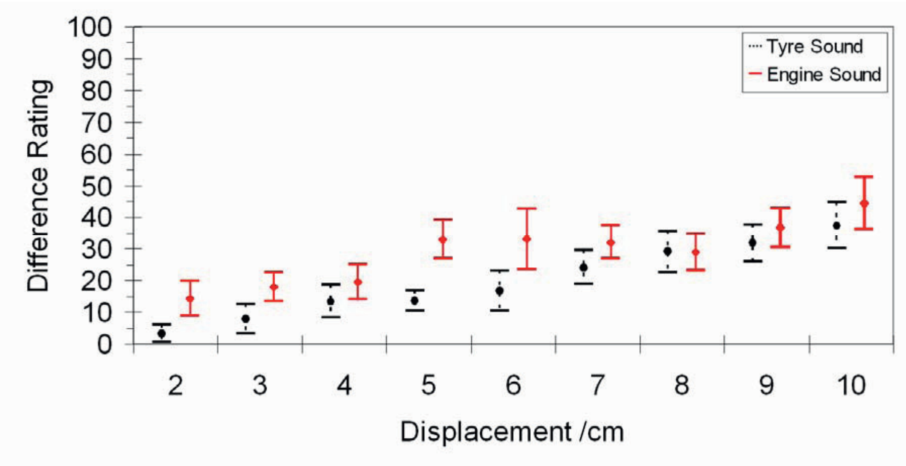

Fig. 10: Means and 95 % confidence intervals (internal s) for displacements of the receiver.

The results from the measurement of binaural transmissibility functions were limited to 8192 Hz. At higher frequencies, the signal levels of the artificial head were below the crosstalk between the channels in the measurement equipment. All functions were calculated from the microphone in front of the loudspeaker to the right ear of the artificial head. This was because the right ear was close to a reflecting surface which might have increased the sensitivity to displacements due to early reflections. The coherence function for reference position 1 (no displacements) is viewed in Fig. 11.

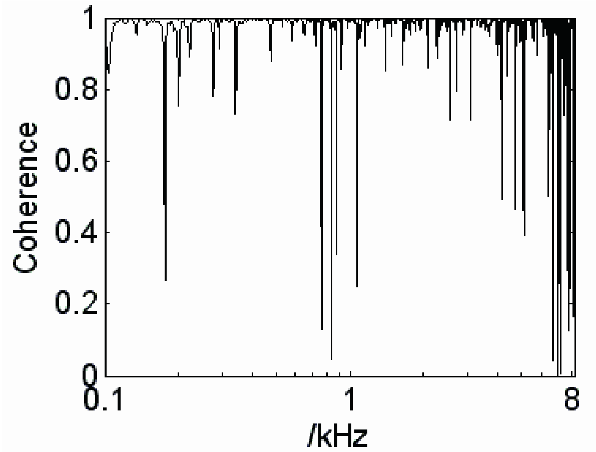

Fig. 11: Coherence function from source to right ear with no displacement

In Fig. 12 to 14, magnitude and difference of binaural transmissibility functions are shown. The magnitude plots compare the reference transmissibility function with the displaced source transmissibility functions of 1, 2 and 32 cm. To highlight the magnitude difference between the transmissibility functions, a difference plot is shown. In Fig. 12, the binaural transmissibility functions with a displacement of 1 cm are almost identical to the transmissibility function measured for the reference positions. The listening tests showed that this displacement was inaudible. In Fig.13 the displacement was 2 cm and was audible. With larger displacements the differences increase and become more prominent, both visually and audibly, Fig. 14. However it is very hard to judge perceived differences from only visual representations. Comparing the difference of a 32 cm displacement with the 1 and 2 cm displacements, the differences in frequency range is lowered as the displacement is increased.

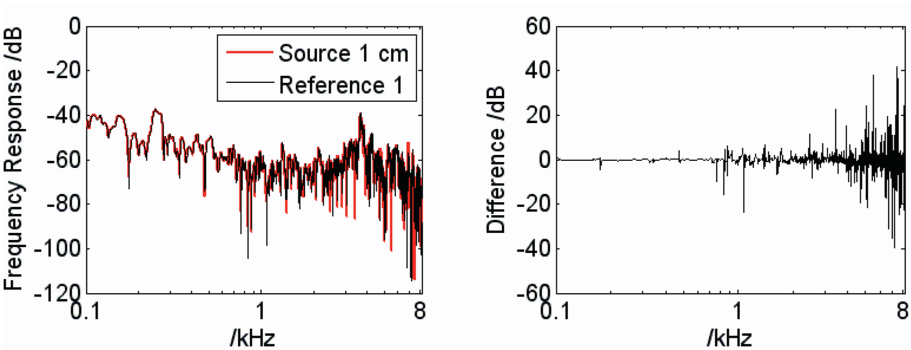

Fig. 12: Magnitude and difference of binaural transmissibility functions to the right ear caused by 1 cm displacements of the source.

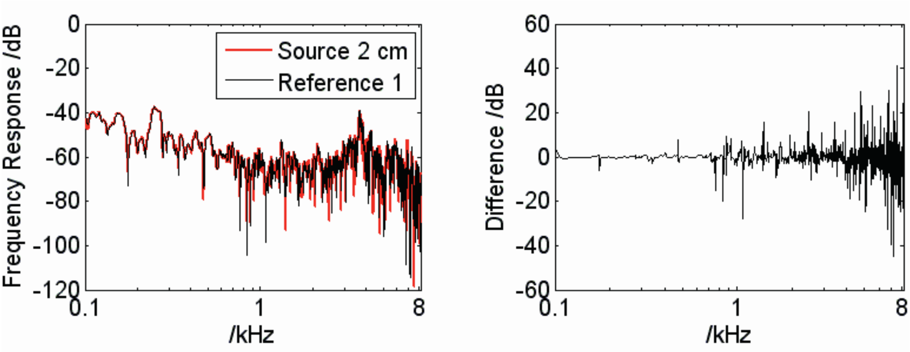

Fig. 13: Magnitude and difference of binaural transmissibility functions to the right ear caused by 2 cm displacements of the source.

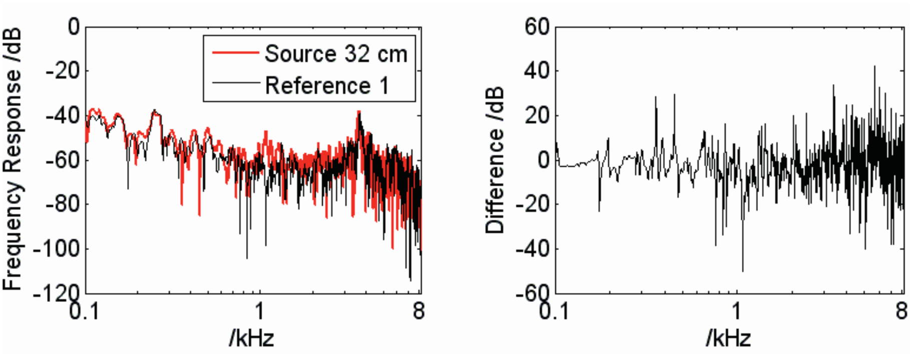

Fig. 14: Magnitude and difference of binaural transmissibility functions to the right ear caused by 32 cm displacements of the source.

Phase differences due to the displacements are shown in Fig. 15. No large differences in the shape of the phase curve were seen. The slope was slightly changed as the time delay between the reference microphone and the receiver changed due to the displacement.

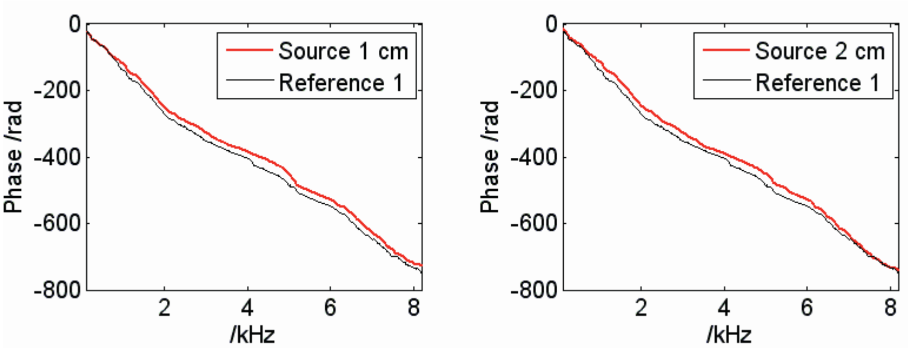

Fig. 15: Phase of binaural transmissibility functions to the right ear caused by 1 and 2 cm displacements of the source.

In Fig. 16 and 17, the magnitude and difference of binaural transmissibility functions are shown when the receiver is displaced. The visual difference in the binaural transmissibility functions for a displaced receiver is approximately the same as when the source is displacements in Fig. 12 and Fig. 13.

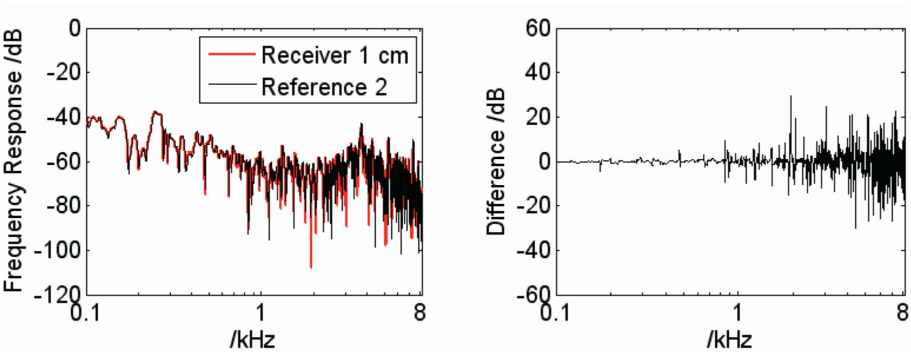

Fig. 16: Magnitude and difference of binaural transmissibility functions to the right ear caused by 1 cm displacements of the receiver.

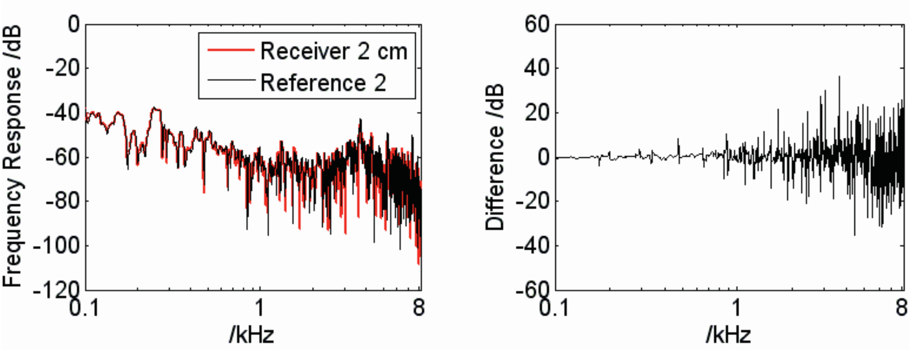

Fig. 17: Magnitude and difference of binaural transmissibility functions to the right ear caused by 2 cm displacements of the receiver.

4. Conclusions

The results showed that artificial head recordings and measurements of binaural transmissibility functions are affected by small displacements in location of both the source and the receiver. In this study, the just noticeable difference was less than 2 cm for displacements of a source outside a car or a receiver inside the car. Therefore, the accuracy in the positioning of the source and the receiver need to be smaller than 2 cm if audible differences shall be avoided. The results of this study can not be generalized for all kinds of sounds, source positions and receiver positions but since two typical automotive sounds were used (tyre sound and engine sound) and the measurements were made in a car cabin they may serve as a guideline for binaural measurement made on cars.

The difference ratings showed, as expected, that larger displacements lead to larger perceived differences in sound quality. They also showed that engine sounds were more affected perceptually by displacements of the receiver than tyre sounds. This may be explained by the high frequency content in the engine sounds. This kind of sounds therefore require higher precision in the placements of sources and receivers compared to less impulsive sounds with less high frequency content, like the tyre sound used in this study.

The study also showed that it is very hard to judge perceived differences from only visual representations of binaural transmissibility functions. Variability due to displacements causes changes in sound quality that must be considered in order to draw correct conclusions from sound quality analyses based on artificial head recordings and binaural transmissibility function measurements.

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Paper III

Examination of the variability between artificial head recordings made in different cars of the same brand and model

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Examination of the variability between artificial head recordings made in different cars of the same brand and model

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ABSTRACT
Artificial heads are commonly used for the recording of samples intended for sound quality evaluations. When results from listening tests and psychoacoustic analyses shall be compared it is important to be aware of the variability between measurements made under similar conditions. To study this kind of variability, the interior sounds of five passenger cars of the same brand, model and production year were recorded binaurally. The recordings were made using two different artificial heads (Head Acoustics HMS I and HMS III) and a binaural microphone headset (Head Acoustics BHM). The recordings were made at constant speed on a test track with controlled surface roughness. Based on the recordings loudness, sharpness, roughness and annoyance were judged in a listening test. The results show how the sound quality can be expected to vary between specimen of cars of the same brand and model. This observed variability was compared to the accuracy of the three different binaural microphones used.

1. INTRODUCTION
The use of artificial head recordings has become a standard method for sound quality evaluations within the automotive industry[1]. While played through headphones true to life reproductions are achieved. Sound quality is commonly assessed through listening tests based on the artificial head recordings. Such tests become very extensive if large amounts of samples are to be examined. Therefore, the number of samples is often limited. To avoid drawing too extensive conclusions from a limited set of samples it is important to be aware of the expected variability between measurements made under similar conditions. Some studies on acoustic response variability in cars have shown large variations among nominally equal cars[2-4]. However, it is difficult to draw conclusions on the variability in perceived sound quality based on such measurements. The main objective of this study was to study the variability between artificial head recordings made in five nominally equal passenger cars. Tyre sound was used as a case to assess whether the variations between the cars should be considered as substantial when compared to typical variations caused by a single component of the car. In addition, the listening tests were repeated

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using recordings made with three different binaural microphones. The objective of this addition was to examine whether differences between nominally equal cars are better detected with some of the binaural microphones.

2. METHOD

A. Recording of sound stimuli

Sound stimuli for listening tests were created by binaural recording of interior sound in five nominally equal cars (mid-sized estate cars of the same brand, model and production year, and equipped with the same interior trims). The same sets of rims and tyres were used on all cars. From now on the cars are labeled Car 1 to 5. All cars were less than one year old with mileage between 9500 and 20000 km. This was considered to be a typical first year mileage for a buyer of a new car, and hence the results should be representative of the spread in sound quality as experienced by owners of new cars. In order to minimize measurement uncertainty, all recordings were made at constant speed, 60 km/h, on a 70 m long straight section of a test track. The surface of the track was dry asphalt. During the tests the temperature varied between -15 and -5 °C. The test sounds were recorded as a part of a study on winter tyre sounds. Therefore, studded winter tyres were used. In order to create differences in sound character two different sets of tyres were used (from now on labeled Tyre 0 and Tyre 1). Both were of the same dimension, 205/55-16, but from different manufacturers. All measurements were made using a Head Acoustics HMS III artificial head. In addition, recordings with Tyre 0 were repeated using a Head Acoustics HMS I artificial head and a Head Acoustics BHM binaural microphone headset. The artificial heads were placed in the front passenger seat. The binaural microphone headset was worn by the driver. Data acquisition was made using a Brül & Kjær frontend (Type 3032A) and Brül & Kjær Pulse software. All measurements were repeated twice. A summary of the set of recorded stimuli is found in Table 1.

Table 1: The recorded stimuli.

Binaural microphone	HMS III artificial head	HMS I artificial head	BHM binaural microphone headset
Number of nominally equal cars	5 (Car 1 to 5)	5 (Car 1 to 5)	5 (Car 1 to 5)
Sets of tyres	2 (Tyre 0 and 1)	1 (Tyre 0)	1 (Tyre 0)
Number of repetitions	2	2	2
Total number of stimuli	20	10	10

B. Reproduction of sound stimuli

Sound stimuli were reproduced using equalized headphones (Head Acoustics HPS IV). For recordings made with the HMS III head, headphone equalization filters supplied by Head Acoustics were used. For the BHM headset and the HMS I artificial head, free field equalization filters designed at Luleå University of Technology were applied to the recordings. Then, the recordings were reproduced using the free field equalization filters supplied with the headphones.

C. Listening test

In a listening test 15 subjects were asked to judge how well the sound stimuli were described by 4 verbal attributes, loud, sharp, rough and annoying, using a verbal attribute magnitude estimation method[5]. All subjects were male and had self-reported normal hearing. The mean age was 26 years (SD 5 years). The judgments were made using a computer interface with scales ranging from "not at all" to "extremely", see Figure 1. In the analysis, the scales were divided into 101 equally long sections, numbered from 0 (not at all) to 100 (extremely). The subject results were normalized with respect to the group mean and standard deviation. The stimuli were looped and played continuously at the same level as they were recorded at (approximately 78 dBA).

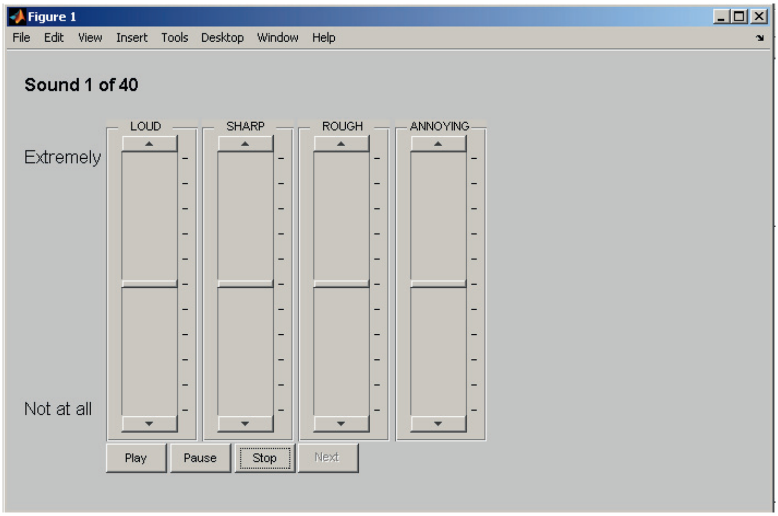

Figure 1: Computer interface used for judgements in the listening test.

3. RESULTS AND DISCUSSION

A. Perceived differences between the two tyres and the five cars

The perceived differences between the sounds of the 2 tyres and the 5 cars were analyzed using ANOVA. In these comparisons only recordings made with the Head Acoustics HMS III artificial head was used. Significant effects ($p \leq 0.05$) were found for judgments of Loud and Annoying (see Table 2 and Figures 2 and 3). Tyre 0 was judged as being louder and more annoying. An interesting observation is that there are significant differences in judgments of Annoying between the different specimens of the nominally equal cars. The difference between the least and the most annoying car is three times as large as the difference between the two tyres. No second order interaction effects were observed.

Table 2: ANOVA for the dependent variables Loud and Annoying with the factors Car (5 levels) and Tyre (2 levels) based on recordings with the HMS III artificial head.

Analysis of Variance for Loud – Type III Sums of Squares

Source	SS	Df	MS	F-Ratio	P-Value
Main Effects					
A: Car	882.4	4	220.6	1.58	0.1803
B: Tyre	1420.7	1	1420.7	10.16	0.0016
Interactions					
AB	323.0	4	80.7	0.58	0.6792
Residual	40550.9	290	139.8		
Total (corrected)	43176.9	299			

Analysis of Variance for Annoying – Type III Sums of Squares

Source	SS	Df	MS	F-Ratio	P-Value
Main Effects					
A: Car	2358.1	4	589.5	2.73	0.0294
B: Tyre	843.9	1	843.9	3.91	0.0490
Interactions					
AB	1505.6	4	376.4	1.74	0.1404
Residual	62599.0	290	215.9		
Total (corrected)	67306.5	299			

All F-ratios are based on the residual mean square error.

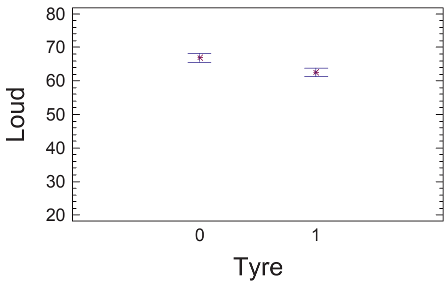

Figure 2: Means and 95 % Tukey HSD intervals for judgments of Loud based on recordings in the 5 cars using both sets of tyres. Only recordings made with the HMS III artificial head are included.

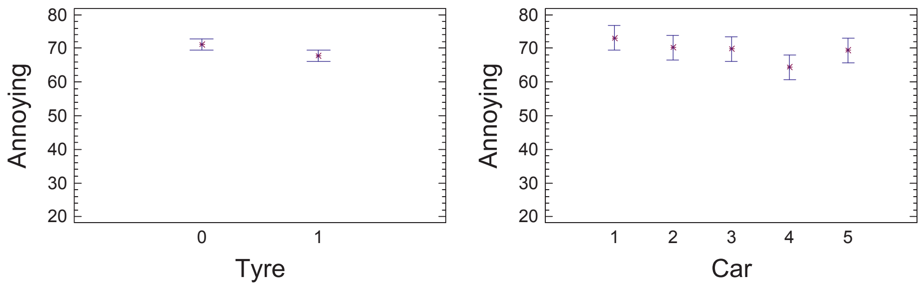

Figure 3: Means and 95 % Tukey HSD intervals for judgments of Annoying based on recordings in the 5 cars both sets of tyres. Only recordings made with the HMS III artificial head are included.

B. Perceived differences between recordings made with the different binaural microphones

ANOVA was used to analyze the perceived differences between recordings made using the three different binaural microphones (Head Acoustics BHM, HMS I and HMS III). Recordings from all five cars fitted with Tyre 0 were used. The effects of two factors, Car (5 levels, Car 1 to 5) and Mic (3 levels: BHM, HMS I and HMS III) were analyzed. Second order interactions were included. It was hypothesized that if interaction effects occur, the choice of microphone could influence the results. No such interactions were found (see Table 3). From Figure 4 it is obvious that the results are depending on which microphone that is used. Different microphones give different loudness ratings. Since sharpness, roughness and annoyance are influenced by loudness[6,7,8], the differences in ratings of these attributes may be explained as an effect of different loudness levels. This could be corrected by appropriate modifications of the microphone equalizations. Since no interactions were found between Mic and Car, more reliable conclusions on the variability between the five nominally equal cars may be drawn by merging the data collected with all three microphone setups. This results in 6 repetitions for each car (2 repetitions for each microphone). Significant differences ($p \leq 0.05$) between the cars are found for the variables Loud and Rough. The difference in judged loudness between the loudest and quietest car is twice as large as the differences in judged loudness between the two tyres (compare Figures 2 and 5). The outdoor temperature may affect the interior sound of the car. However, the temperature was quite stable during the tests (from -15 to -5 °C). The recordings of the six repetitions with one car were made during approximately 3 h. Since the lowest temperatures were found in mornings and evenings, there was a spread in temperatures between repetitions of each car. Further studies are required to guarantee that outdoor temperature does not influence the results.

Table 3: ANOVA for the dependent variables Loud, Sharp, Rough and Annoying with factors Car (5 levels) and Mic (3 levels: BHM, HMS I and HMS III).

Analysis of Variance for Loud – Type III Sums of Squares

Source	SS	Df	MS	F-Ratio	P-Value
Main Effects					
A: Mic	75599.9	2	37800.0	241.1	0.0000
B: Car	3939.6	4	984.9	6.28	0.0001
Interactions					
AB	1669.8	8	208.7	1.33	0.2258
Residual	68202.5	435	156.8		
Total (corrected)	149412.0	449			

Analysis of Variance for Sharp – Type III Sums of Squares

Source	SS	Df	MS	F-Ratio	P-Value
Main Effects					
A: Mic	28613.3	2	14306.7	47.70	0.0000
B: Car	1589.7	4	397.4	1.33	0.2597
Interactions					
AB	3690.1	8	461.3	1.54	0.1418
Residual	130474.0	435	299.9		
Total (corrected)	164368.0	449			

Analysis of Variance for Rough – Type III Sums of Squares

Source	SS	Df	MS	F-Ratio	P-Value
Main Effects					
A: Mic	23439.8	2	11719.9	40.91	0.0000
B: Car	3133.8	4	783.4	2.73	0.0286
Interactions					
AB	2487.0	8	310.9	1.09	0.3723
Residual	124622.0	435	286.5		
Total (corrected)	153682.0	449			

Analysis of Variance for Annoying – Type III Sums of Squares

Source	SS	Df	MS	F-Ratio	P-Value
Main Effects					
A: Mic	78483.1	2	39241.5	154.85	0.0000
B: Car	1592.0	4	398.0	1.57	0.1811
Interactions					
AB	2958.9	8	369.9	1.46	0.1699
Residual	110239.0	435	253.423		
Total (corrected)	193273.0	449			

All F-ratios are based on the residual mean square error.

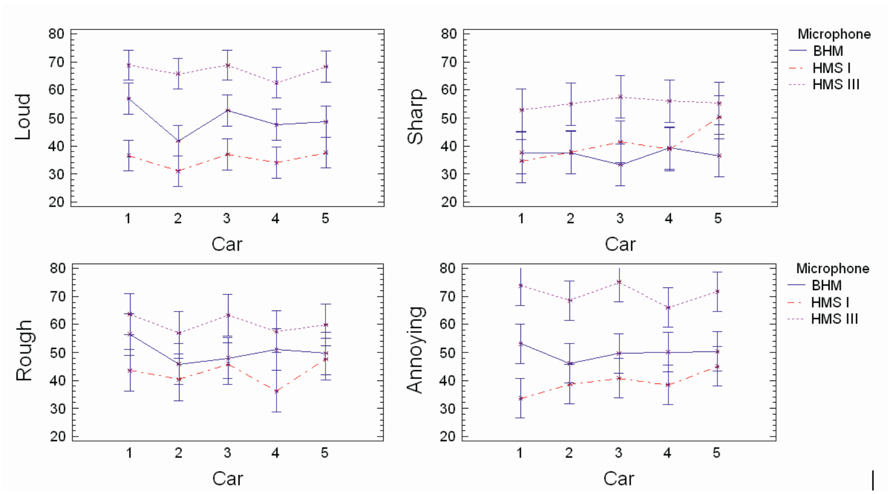

Figure 4: Means and 95 % Tukey HSD intervals for judgments of Loud based on recordings in the 5 cars using Tyre 0 for comparison of results based on the three different binaural microphones used.

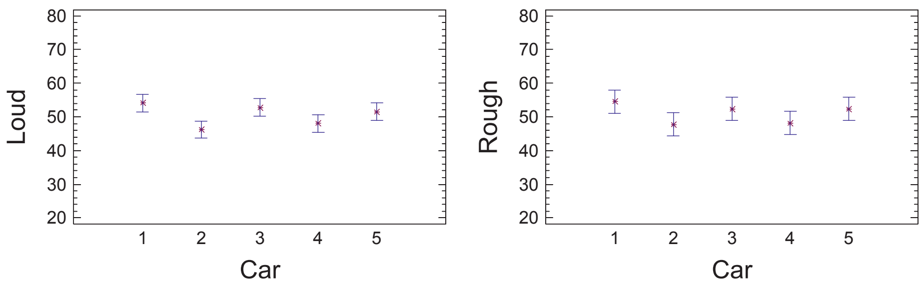

Figure 5: Means and 95 % Tukey HSD intervals for judgments of Loud and Rough based on recordings in the 5 cars using Tyre 0. Two repetitions using each of the 3 binaural microphones were made, in total 6 repetitions.

4. CONCLUSIONS

It was found that variations in interior sound between different specimens of nominally equal cars may be larger than variations caused by the use of tyres of different design. This was apparent for judgments of loudness and annoyance, where the differences between cars were between two and three times larger than the difference between the two tyres. It is of great importance to be aware of such variations, especially if the number of stimuli is limited. Recording of all stimuli using a single car should be avoided.

There were significant differences in judgments depending on which binaural microphone that was used. Changes in equalization would probably correct for this. Since no interaction effects were found between used microphone and studied car, the conclusion was that all of the three binaural microphones should perform well as long as they are properly equalized.

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