Tire test stand measurements for blocked forces identification and tire noise auralization

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ABSTRACT

In this paper, we present the process we developed to support automotive OEMs and tire manufacturers with road noise engineering during vehicle design and development. The process uses blocked forces identified from measurements performed on a single tire installed on a test stand on a chassis dynamometer. The forces are then combined with vehicle level transfer functions calculated using a CAE simulation model to predict the sound at the driver ears. The results are imported in a driving simulator for NVH for auralization and sound quality evaluations. The process can be applied from the very early stages of a vehicle development and allows to evaluate the vehicle road noise performance long before the first prototype is available.

1. INTRODUCTION

Automotive OEMs and tire manufacturers allocate significant resources to tire noise engineering with the objective to improve the noise and vibration performance of future cars. This is driven by the customer expectations constantly trending towards quieter vehicle. This trend is even accelerating with the switch towards new energies as electric vehicles (EVs) are quieter at low speed than vehicles with internal combustion engines. Customers are getting used to a quieter vehicle cabin at low speed and expect similar performance when cruising at higher speed when road or wind noise becomes dominant.

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We present in this paper the process we developed to support automotive OEMs and tire manufacturers with road noise engineering during vehicle design and development. We focus on structure-borne road noise, a process for airborne noise has already been proposed in [1]. The process uses blocked forces identified from measurements performed on a single tire installed on a test stand in a chassis dynamometer. The forces are then combined with vehicle level transfer functions calculated using a CAE simulation model to predict the noise at the driver ears. The results are imported in a driving simulator for NVH for auralization and sound quality evaluations. The process can be applied from the very early stages of a vehicle development program, long before the first vehicle prototype is available. With this, engineers can assess the vehicle road noise performance, identify the risks of NVH issue and define design countermeasures if necessary.

We briefly describe in Section 2 the driving simulator for NVH we are using to evaluate the vehicle interior sound. Section 3 is reserved for the measurements performed on the tire stand and the data post-processing to identify the blocked forces at the wheel center. The CAE simulation model is presented in Section 4. The results are illustrated in Section 5 before we conclude and share the future steps in our work.

2. DRIVING SIMULATOR FOR NVH

Driving simulators are now commonly used to support vehicle development programs for attributes including ride and handling, NVH, ADAS, etc. The simulator we used for the work presented in this paper is illustrated in Figure 1.

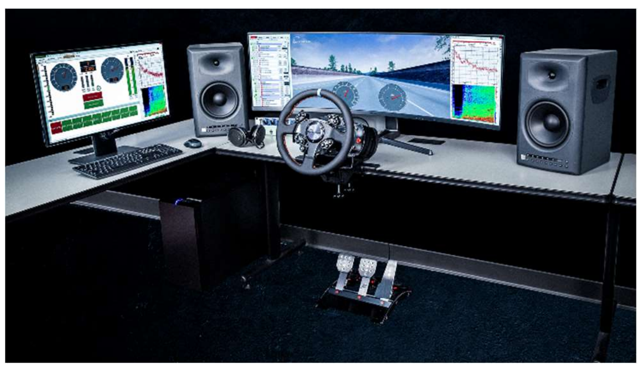

Figure 1: driving simulator for NVH

It is a software solution that allows to interactively reproduce the sound from a vehicle depending on the driving conditions, including the engine or motor RPM, the vehicle speed, the gear, the road surface or any other parameters that can affect the NVH (e.g., sports mode or different regeneration mode for an electric vehicle). The sound replay is interactive and is a function of the driver input on the simulator through the accelerator pedal, gear shifts and steering which may change the road noise if the driving lanes have different profiles. The load on the engine is dependent on the driving conditions and the slope of the road profile. The sound is replayed using headphones or a set of speakers. The replay system is calibrated to verify that the sound produced corresponds to what one would experience if they were driving the same real vehicle.

The interior sound is decomposed into contributions from separate sources including the powertrain, road and wind noise. A similar approach can be used for accessories like turn signals, wipers, HVAC, etc. The powertrain sound is represented as harmonic of the RPM (orders of the engine or motor) and an RPM-dependent background noise. Road and wind noise are represented as velocity-dependent sounds. The software interpolates between the available velocities to calculate the sound at any given vehicle speed.

Our work focused on structure-borne road noise; it originates from vibration generated by the tire/road interaction that propagates through the suspension to the body attachment points as illustrated in Figure 2. The airborne contribution is not included in the work we are presenting. It corresponds to the noise generated at the tire/road interface (i.e., leading and trailing edge) which propagates inside the vehicle cabin through the windows, body panels and leaks. This path is illustrated in Figure 3.

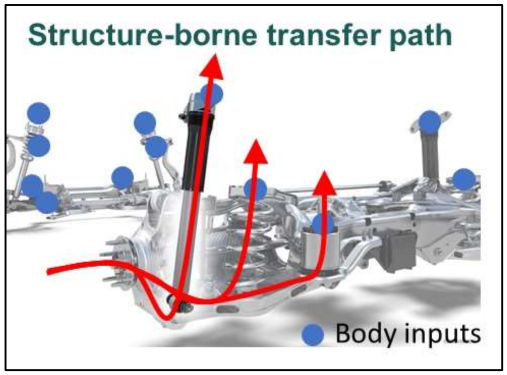

Figure 2: road noise – structure-borne path

Figure 3: road noise – airborne path

Different options are available to decompose the structure-borne road noise into forces and contribution, we have used here measurements performed on single tire installed on a test stand in a chassis dynamometer to identify the blocked forces at the wheel center. These forces will need to be combined with transfer functions to synthesize the noise at the drivers' ears.

3. TIRE TEST STAND MEASUREMENTS FOR BLOCKED FORCES

Blocked forces [2] are used to characterize sources for structure-borne noise. They completely define the source and are independent of the receiving structure. Blocked forces can theoretically be measured on a rigid test rig or be identified in-situ using operational and transfer function measurements. We have followed the latter option to characterize the wheel center forces from tire measurements performed on a test stand.

The tire test stand is represented in Figure 4. It is installed in an NHV chassis dynamometer with the tire in contact with the one of the rollers. The test stand accommodates a large range of tire sizes. During the tests, the roller surface is covered with sandpaper for smooth road profiles or dyno shells (Figure 5) to simulate rough road profiles.

The stand is instrumented with accelerometers, strain gauges and strain sensors. The tires are tested at constant speed for multiple operating speeds ranging from 10 or 20kph up to highway speed or the maximum dynamometer speed. The measurements are repeated for different inflation pressures as well as vertical preloads. The preload is controlled with a force sensor connected to a digital display.

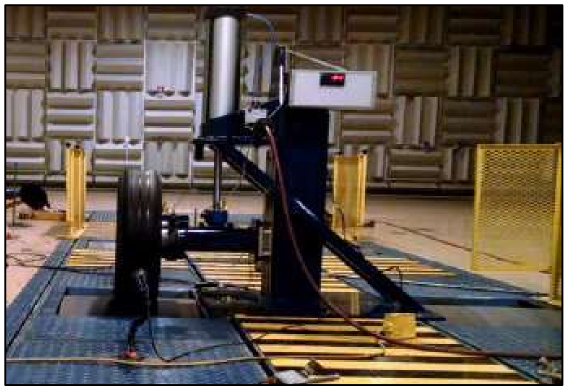

Figure 4: tire test stand

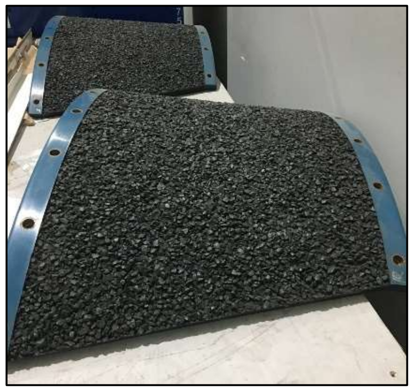

Figure 5: shells to simulate rough roads

The blocked forces at the wheel center are defined in 5 directions: 3 translation and 2 rotations. The moment around the wheel spinning axis is responsible for the vehicle acceleration and is not considered as a source for road noise. The transfer functions between all these inputs and the sensors installed on the test stand are required for the in-situ identification of the blocked forces. While it is straightforward to measure the transfer functions for the translation degrees of freedom (DOFs) using hammer impacts at the wheel center, measuring the transfer functions for the rotation degrees of freedom poses significant challenges. As a solution, we have used the virtual point transformation method described in [3]. We have applied impacts in the 3 translation directions at multiple locations offset from the wheel center. For each location, we have then calculated a coordinate transformation matrix [R_F] that relates the 5 DOFs forces at the wheel center to the impact force applied per:

$$\begin{bmatrix} F_X \\ F_Y \\ F_Z \\ M_X \\ M_z \end{bmatrix} = [R_F]^T \left[F_{Impact}\right] \tag{1}$$

The transfer functions at the wheel center are then calculated using an inverse methodology assembled over all the transformation matrices:

$$[FRFs_{WC}] = [FRFs_{Meas}]([R_F]^T[R_F])^{-1} \tag{1}$$

This approach assumes that the test stand is perfectly rigid. The validity of this assumptions can be verified by animating the transfer functions or by comparing the transfer functions measured against the ones calculated from the estimated transfer functions at the wheel center and the inverse of the coordinate transformation matrix. The comparison can be performed using an indicator like the FRAC or Frequency Response Assurance Criterion [4]. Our results showed that the test stand can reasonably assumed rigid up to approximately 500Hz.

The blocked forces are estimated from the transfer functions and the operational measurement data using:

$$[F_{BL}] = iFFT\left(\left[FRFsH_{WC}^{Indicator}\right]^{-1}\right) * [X_{indicator}] \tag{1}$$

In this equation, the vector [X] regroups the frequency spectrum of all the indicator signals. A combination of accelerations and strains is required to cover a broad range of frequencies from approximately 20Hz up to several 500 Hz. Prior to the inversion, the indicator signals are normalized using their average amplitude across the frequency range as scaling factor. Without this, the strain signals would have a limited influence on the results as their amplitude is order of magnitude lower than the accelerations.

The force identification results are validated by comparing the indicator signals calculated from the forces and the transfer functions against the measured ones. The comparison is performed on all the indicators, but since most were used for the force identification, this serves more as a consistency check than a real validation. In addition, a few indicators signals were defined as reference target signals, they were measured at the same time as all the other indicators, but we did not use them in the identification process. These indicators will then provide an independent validation of the results.

Figure 6 compares the calculated sum of contributions of all the wheel center forces against the measured signal for one of the reference target accelerations. The results show a good correlation over the entire frequency range analyzed.

The same steps were repeated for the same tire tested at multiple constant speeds for several inflation pressures and vertical preloads and 2 road profiles (smooth and coarse shell). The data is used to create a library of excitation forces at the wheel center to predict the structure-borne road noise performance of automotive vehicles under development. An example of wheel center forces is represented in Figure 7.

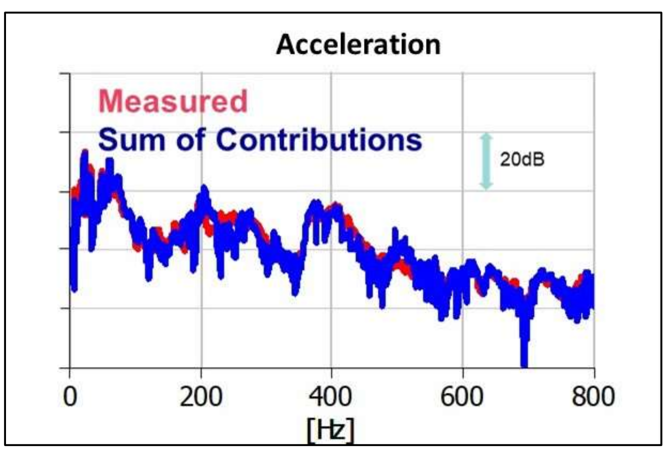

Figure 6: results validation

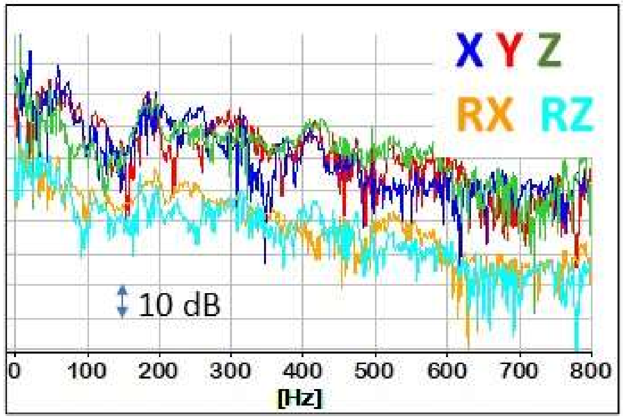

Figure 7: example of wheel center forces

4. CAE SIMULATION FOR FULL VEHICLE TRANSFER FUNCTIONS

4.1. Finite Element Model

The simulation model is used to predict the structure borne transfer functions from the tire attachments to the driver and passenger head positions (Figure 8). To connect simulated and measurement results with the blocked force method, the forces and moments for 5 DOFs at the tire attachment are provided for all 4 tires. Those are the transfer functions of all force directions (x-, y- and z-direction) and the transfer functions for the moment in x- and z-direction.

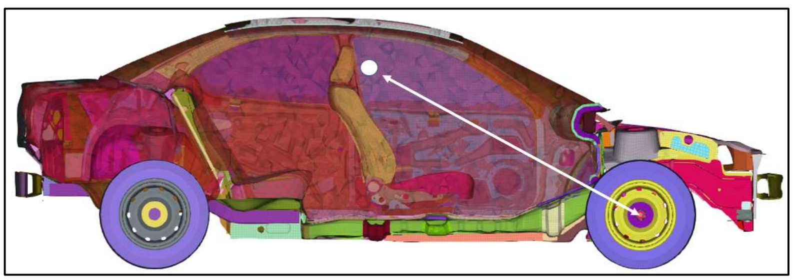
Figure 8: example of transfer function front right tire to driver's head position

The simulation model is a FE (Finite Element) Nastran PEM (Poroelastic Material) model [8] [9]. This is a state-of-the-art modeling technique for this kind of applications [5] [6] [7]. The model consists of

- the body and suspension model (Figure 9),
- the cavity (Figure 10), and,
- the acoustic trim (Figure 11).

Figure 12 shows the excited suspension attachment nodes. The tires are kept in the simulation, as this is required when using blocked forces. The head positions are shown in Figure 10.
The structure and fluid model respect the maximal frequency of the modal extraction. The structure model has 2762 modes up to 600 Hz. The fluid cavity has 441 modes below 1000 Hz.
The structural model has about 500,000 elements. The cavity model consists of about 300,000 elements and the acoustic trim model has 120,000 elements. The element sizes of the trim models respect the maximal computation frequency of 400 Hz.

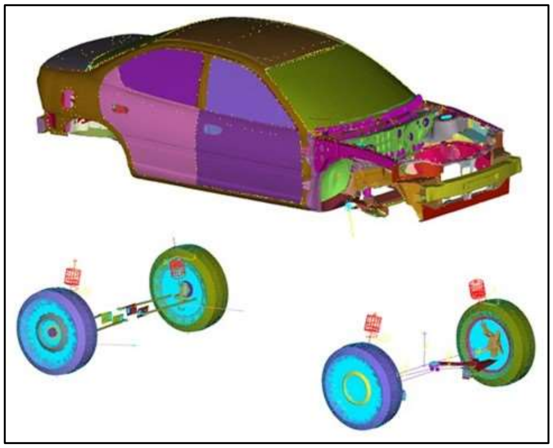

Figure 9: the body model and the suspension model being parts of the structure model

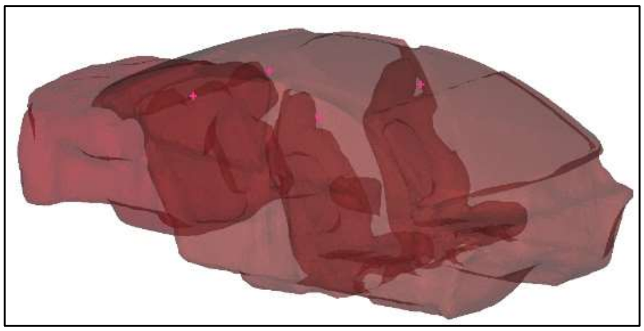

Figure 10: the cavity model, including the microphone positions of driver and passengers

The body and suspension models are connected, see Figure 12, allowing to transmit the forces at the wheel centers to the inside of the cavity. The trim model allows to correctly take damping, absorption and insulation effects into account.

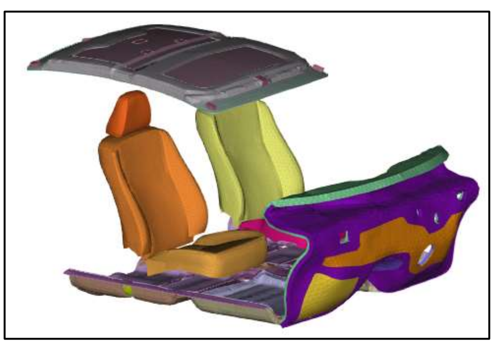

Figure 11:the trim model consists of the firewall insulation panel, the headliner, the front seats and the carpet

This model uses the same modeling techniques as in (5), with the published high-quality correlations with measurements. The model includes poroelastic materials, modeled with the Biot theory and the reduced impedance matrix method (5). This method couples efficiently with vehicle models in modal coordinates.

Figure 12: structural model with excitations

4.2. CAE Simulation Results

The results are computed up to 400 Hz. The transfer functions from the left front tire to the driver's ears are shown in Figure 13. The transfer functions from the front right tire to the driver's ears in Figure 14 and the transfer functions form the rear right tire to the driver's ear in Figure 15. All these figures show the excitation force in x-, y- and z-direction and the moments in x- and z-direction.

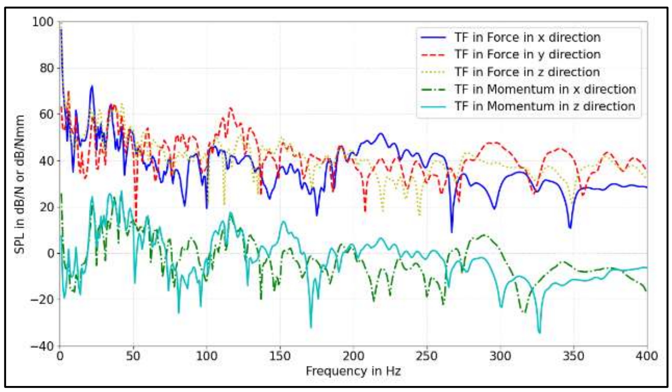

Figure 13: transfer functions front left tire to the driver's ear in dB/Pa for forces and in dB/Nmm for moments

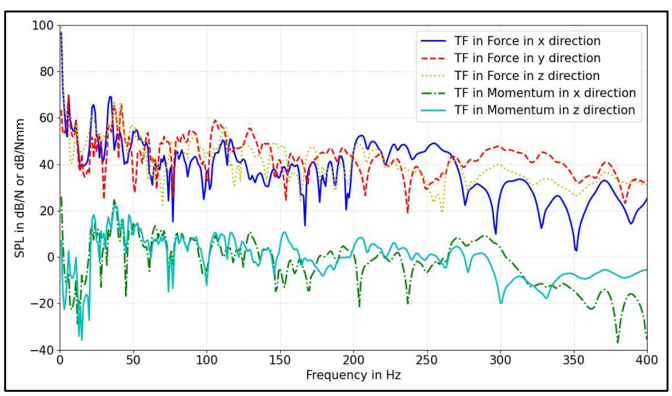

Figure 14: transfer functions front right tire to the driver's ear in dB/Pa for forces and in dB/Nmm for moments

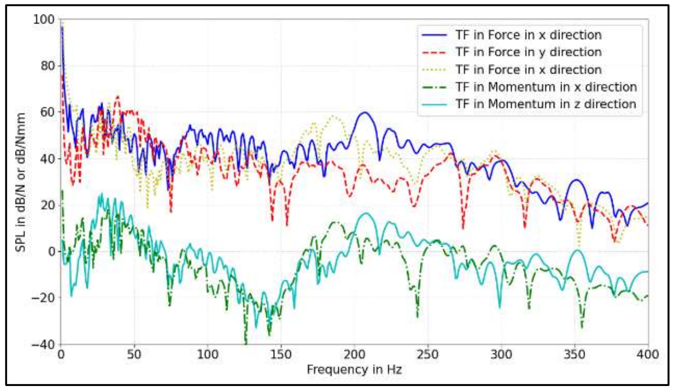

Figure 15: transfer functions rear right tire to the driver's ear in dB/Pa for forces and in dB/Nmm for moments

5. ROAD NOISE EVALUATIONS

The road noise evaluations are performed in the driving simulator for NVH. The blocked forces at the wheel center are combined with the transfer functions to generate the noise contribution. The process requires an inverse FFT as both the blocked forces and transfer functions are defined in the frequency domain. The amplitude of the forces is kept constant over the 4 wheels and several options can be considered for the phase difference. We can use a constant phase difference (e.g., all in phase, out-of-phase) or a random one to simulate incoherent inputs at each wheel. Alternatively, the driver and passenger side can be considered incoherent with a phase delay between the front and rear wheels on each side calculated using the vehicle speed and wheelbase.

The interior sound is calculated for all the tire speeds tested on the stand (20, 40, 60 and 80kph). When driving on the simulator, the results are interpolated to reproduce the road noise at any given vehicle speed in-between.

An example of road noise results is represented in Figure 16. It represents the FFT spectrum of the vehicle interior sound when driving at 20kph assuming incoherent inputs from each wheel. Only the road noise contribution is represented on this plot. In practice, for meaningful subjective evaluations, it is important to complement the simulated structure-borne road noise to include the powertrain, airborne road noise and wind noise. These sounds can be defined from measurements data acquired on the road, chassis dynamometer and wind tunnel if available. Using a prototype, predecessor or even a benchmark vehicle is often sufficient to give the proper context. Using CAE simulation data may provide a more efficient alternative if the vehicle design is optimized for all the NVH attributes at the same time.

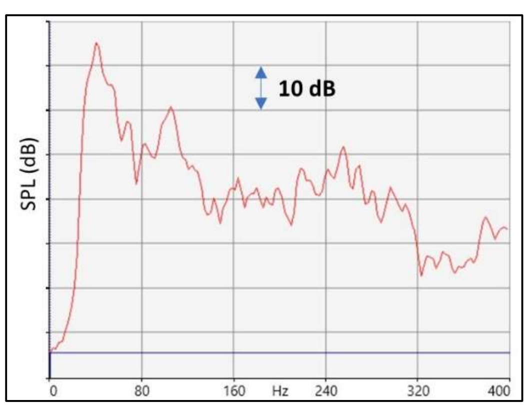

Figure 16: example of road noise results

Being able to perform subjective evaluation and experience the sound in the driving simulator is invaluable. It is of course always possible to compare the interior noise level against targets but being able to subjectively evaluate the interior sound really helps make better design decisions for NVH. It is easier to accept a target overshoot or get approval for an additional noise reduction when the targets are met. If improvements are required, the results will help guide the definition of design countermeasures. It is straightforward to identify which wheel or direction is responsible for elevated sound pressure level. In addition, the model can be used to define if changes in the source terms or transfer functions should be prioritized to improve the performance. Once design changes have been implemented in the model, the transfer functions can be updated and the sounds recalculated to evaluate the effect on the road noise performance.

But the key benefit of this process is to be able to perform all this long before any prototype is available. It really allows to identify risks of NVH issues very early on leaving plenty of development time to define design countermeasures if necessary and perform a multi-attribute optimization.

6. CONCLUSIONS

We have presented the process we developed to evaluate a vehicle structure-borne road noise performance using blocked forces identified from tire test stand measurements and transfer functions from CAE simulation. A driving simulator for NVH replaces the use of physical prototypes to evaluate and experience the vehicle interior sound. This process is available from the very early stages of a vehicle development program and is effective at minimizing the risks of late NVH issues. In our current work, we are developing a library of tires, each tire is tested on our test stand to identify the blocked forces at the wheel center for different inflation pressures, preloads, road profiles and vehicle speeds. This library will be used to support future vehicle programs. We are also working on extending the process to include vibration feedback at tactile points using a driving simulator for sound and vibration.

7. REFERENCES

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