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Modal tyre models for road noise improvement

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A well-tuned tyre–vehicle vibration interaction strongly supports a pleasant road noise. Continental has developed efficient tools and procedures to achieve that. In the frequency range up to 300 Hz, modal tyre models turned out to be very helpful. In this paper we describe the method of simulation and its experimental validation. Using application examples, benefits and limits are shown.

Keywords: Road noise; Tyre; Modal model; Vehicle sensitivity; Tuning

1. Introduction

Today the automotive industry is faced with challenging goals in the field of environmental protection combined with tough cost targets [1]. Both are addressed by the lightweight construction of modern vehicles resulting in less fuel and resource consumption. The light-weight strategy requires appropriate light-weight noise vibration harshness (NVH) solutions. Additional mass for increased stiffness of vehicle structures, for absorbers or for damping layers should be replaced by more smart solutions as far as possible. An NVH method to support this approach is the so-called modal de-alignment. The basic idea is to reduce the structure-borne noise transfer from a sound source to the driver by avoiding the coincidence of source modes with modes of the transferring structure. In this paper the method is applied to tyres, excited by the road contact and acting as the source, while the vehicle structure transfers the road noise to the driver.

In particular, in coarse road noise application the structural displacement amplitudes in the relevant frequency range from 30 to 300 Hz are very small. The system can be assumed to be linear which allows the application of modal models. In that frequency range a tyre shows typically 30 relevant modes. The quality of a full vehicle road noise simulation can be significantly improved by taking into account the modal characteristics of the tyre–wheel assembly.

Building the modal tyre–wheel model based on the detailed tyre and wheel construction specification allows a virtual modal de-alignment. In the simulation the construction parameters will be modified to avoid the coincidence of the frequencies of prominent tyre–wheel modes and specific sensitivities of the vehicle concerning structure-borne noise input at the

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spindle. If the tyre–wheel model is combined with a vehicle model, the effect of design modifications on the interior road noise can be predicted. Only the most promising layouts will be used to build prototype tyres. Typically the method allows a reduction in annoying spectral interior noise peaks by up to 5 dB(A).

2. Simulation model

In order to predict the structure-borne transfer of enforced displacements at the contact patch to the interior of a vehicle the tyre model has to be very detailed. A sufficiently detailed finite-element (FE) tyre model to desribe the dynamics of the tyre up to 350 Hz has over 100 000 degrees of freedom (DOFs). Therefore, it is popular to reduce the number of DOFs with substructuring techniques and to build a modal tyre model.

2.1 *Finite element tyre model*

The finite-element method (FEM) is widely used at Continental to predict the properties of a tyre. With the in-house code GENFEP, various static and steady-state rolling tyre properties such as durability, footprint pressure, rolling resistance, tyre characteristics and wear can be predicted [2, 3]. The software GENFEP is able to handle the geometric nonlinearity due to the large deformation of the tyre and the material nonlinearity. The building process starts with a two-dimensional (2D) FEM model of the tyre where the rim will be attached and the inflation pressure is applied to (figure 1). After having transferred the model into a three-dimensional (3D) representation the camber angle and the static load will be applied, which is a nonlinear displacement of the tyre. This deflection defines the working point of the acoustic relevant tyre vibrations.

2.2 *Modal tyre model*

A modal tyre model describes the dynamic behaviour of a tyre in terms of generalized coordinates. The equations of motion of the tyre in physical coordinates can be written

$$\mathbf{M}\ddot{\boldsymbol{u}} + \mathbf{D}\dot{\boldsymbol{u}} + \mathbf{K}\boldsymbol{u} = \boldsymbol{F}, \tag{1}$$

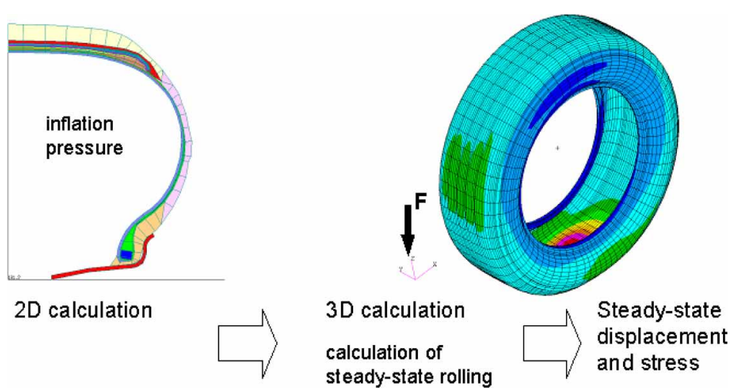

Figure 1. Nonlinear FE calculation of steady state rolling.

where $\mathbf{M}$ is the mass matrix, $\mathbf{D}$ is the damping matrix, $\mathbf{K}$ is the stiffness matrix, $\boldsymbol{u}$ is the vector of physical displacement coordinates and $\boldsymbol{F}$ is the vector of forces acting on the tyre. In order to build a modal tyre model the vibration modes of this system are computed around the stationary working point as a linear perturbation. It is assumed that damping is small and can be neglected. The physical coordinates $\boldsymbol{u}$ are then related to the generalized or modal coordinates ξ_i by the modal transformation

$$\boldsymbol{u} = \boldsymbol{\Phi}\boldsymbol{\xi}, \tag{2}$$

where $\boldsymbol{\Phi}$ is the matrix of mode shapes and $\boldsymbol{\xi}$ the vector of modal coordinates. The tyre is further specified by calculating the generalized modal mass

$$m_i = \boldsymbol{\Phi}_i \mathbf{M} \boldsymbol{\Phi}_i^{\mathrm{T}} \tag{3}$$

and stiffness

$$k_i = \omega_i^2 m_i \tag{4}$$

where $\boldsymbol{\Phi}_i$ is the ith eigenvector and ω_i the corresponding circular eigenfrequency. The modal damping b_i is obtained from experiments.

When the tyre is described by modal parameters, it must be considered how the interface points at which the tyre is connected to the ground and the hub are supported when the vibration modes are computed. At Continental a hybrid method according to MacNeal [4] is applied, where the nodes at the contact patch are fixed and the spindle is free. With the aid of computed reaction forces that are acting on the constrained nodes during a vibration mode it is possible to apply an excitation at the contact patch in the modal tyre model (figure 2).

To make use of the modal tyre model more comfortable, the interface nodes at the contact patch are condensed into the centreline with seven to nine grid points.

If necessary, the wheel elasticity and the air cavity can be included in the modal tyre model. Above 200 Hz the air cavity has to be incorporated in the modal tyre model. This is done by first

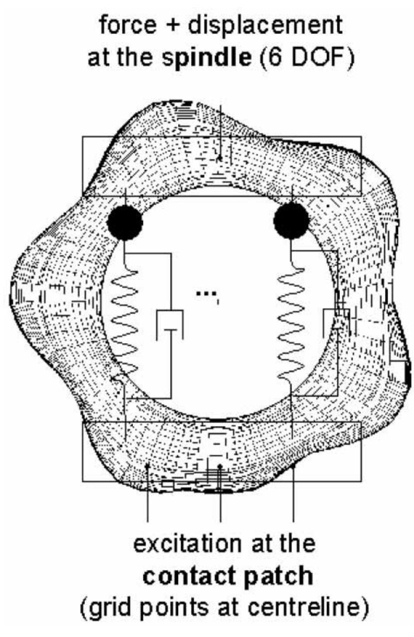

Figure 2. Modal tyre model.

building a modal model of both air cavity and tyre and a following coupling. The incorporation of the wheel as a vibrating substructure can already be necessary below 200 Hz. To include the wheel elasticity the mode shapes and modal parameters of the wheel are computed. From these values, superelements for the contact nodes between tyre and rim are built and included into the eigenfrequency calculation of the tyre [5].

3. Validation

At Continental, different experiments are applied to prove the model quality. In particular, the boundary conditions have to be considered when comparing calculation with experiment.

3.1 *Modal analysis of the tyre*

Figure 3 shows the test rig to measure the mode shapes and modal parameters of a tyre. The tyre is deflected with a hydraulic and fixed at the spindle. With a shaker the tyre is excited and the corresponding response is measured with a laser vibration sensor. The computed mode

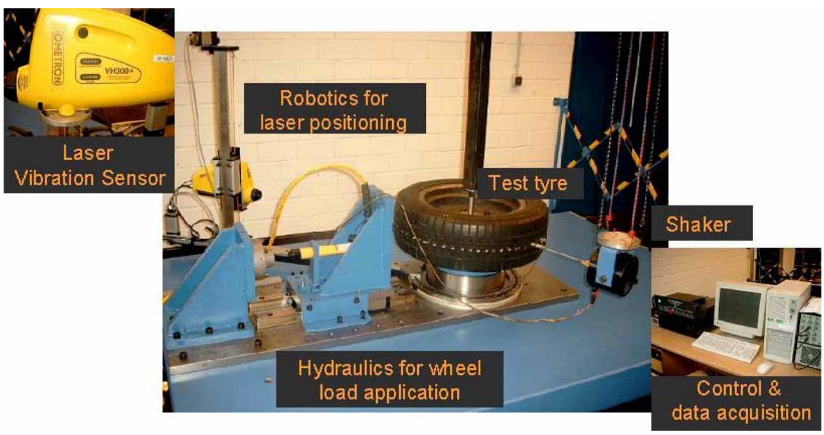

Figure 3. Experimental modal analysis of the tyre.

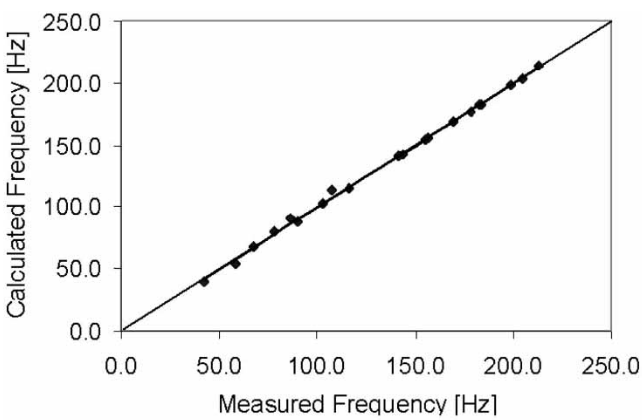

Figure 4. Calculated versus measured natural frequencies of a tyre.

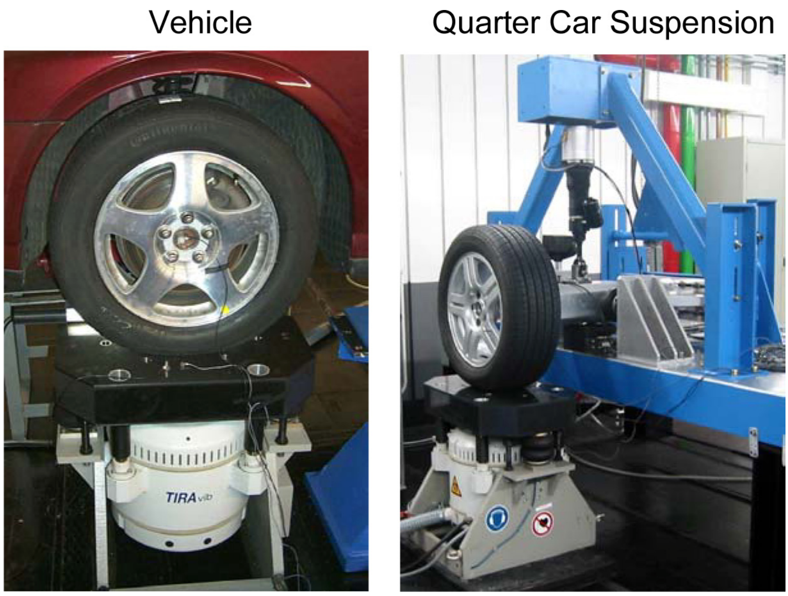

Figure 5. Experimental setup for measurement of the tyre transmissibility function.

shapes and natural frequencies can be compared with this experiment and the model quality can be proved.

It can be seen from figure 4 that the natural frequencies can be calculated very well.

3.2 *Tyre transmissibility function*

If the tyre is plugged to a vehicle, the motion of the spindle point is not restrained. Thus the above-described experiment to determine the mode shapes and modal parameters with fixed wheel is suitable for validating the FE model of the tyre. In order to validate the modal tyre

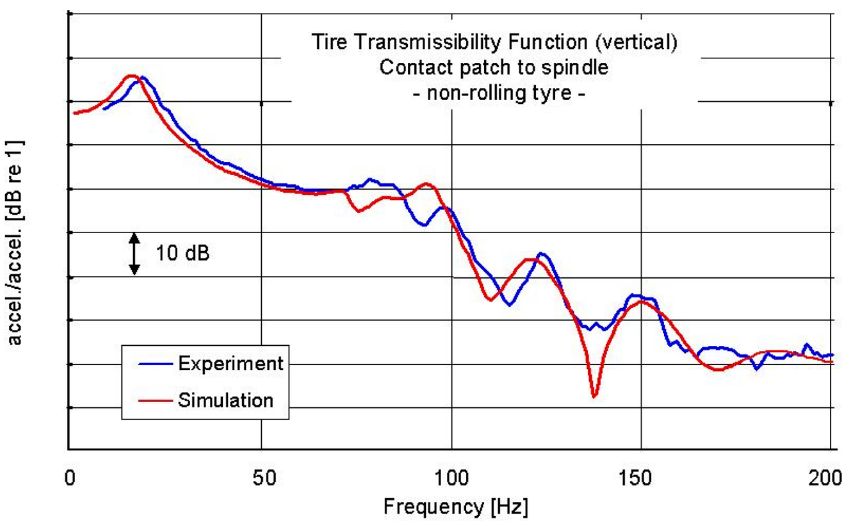

Figure 6. Calculated versus measured tyre transmissibility functions.

model, 'soft' boundary conditions have to be applied. Regarding this concern the so-called tyre transmissibility function is measured, where an excitation is applied at the contact patch and the corresponding response is measured at the spindle point. Figure 5 shows the measurement set-up for when the tyre is attached both to a vehicle and to a quarter-car suspension. From figure 6 it can be seen that the computed tyre transmissibility function matches the measured function well.

4. Application example

Driving on coarse road asphalt a typical rough sound in the frequency range from 100 to 300 Hz is generated. The tyre becomes excited by the pavement texture and reacts with transverse waves of the belt package travelling around the circumference of the tyre [6, 7]. The tyre's sidewall transfers the vibration to the rim, which may have its first natural modes in this frequency range depending on its construction. In the application example described here, the change in the interior noise character of a compact car while driving from smooth to

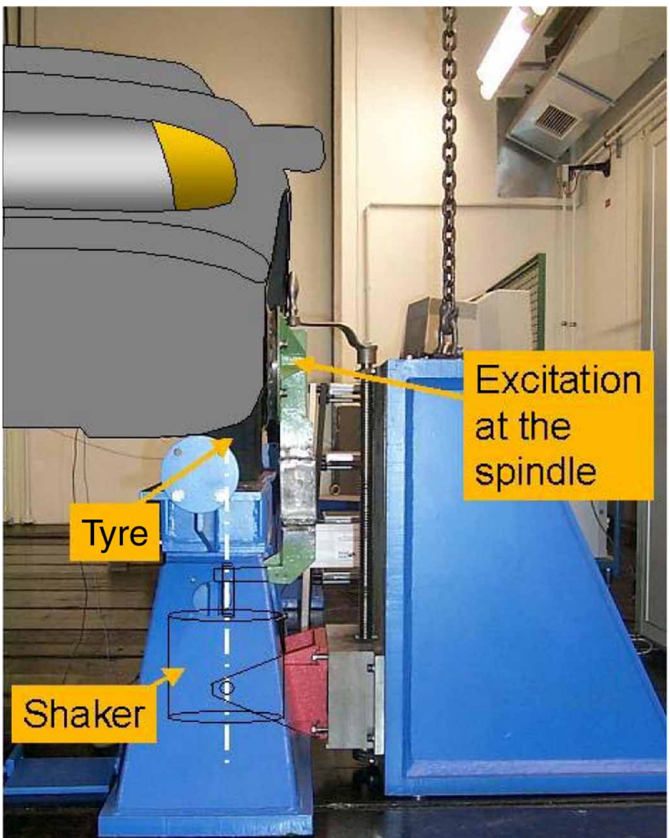

Figure 7. Test stand to evaluate the sensitivity of vehicles to vibration input. Electromagnetic shakers are used for excitation at any point of interest, for example at the spindle from 2 to typically 300 Hz. Transfer and transmissibility functions from the input force and acceleration to noise and acceleration output at locations in the vehicle relevant for comfort will be calculated. The test stand is suitable for vibration input as from tyre or brake non-uniformity as well as for structure-borne noise transfer from road, engine and other sources.

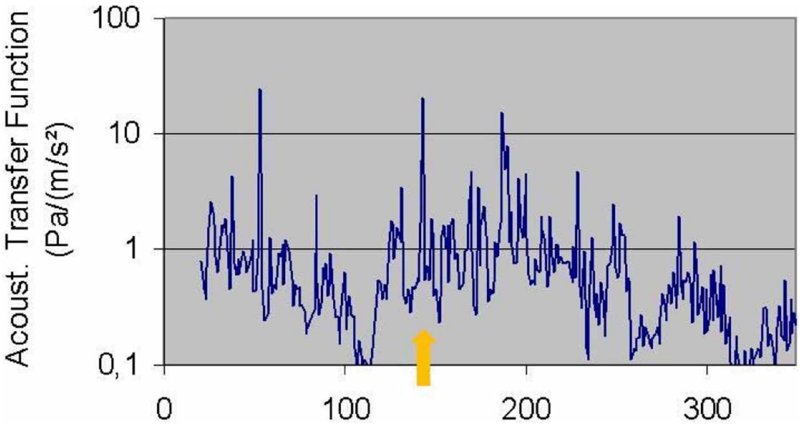

Figure 8. Acoustic transfer function from acceleration at the spindle to the interior noise, measured at the vehicle vibration sensitivity test stand (see figure 7). In this example the vehicle has a high structure-borne noise transfer peak close to 150 Hz.

coarse road asphalt was too intense. An objective sensitivity analysis using shaker excitation successively at all spindles in all DOFs together with the detection of the generated noise inside the vehicle (figure 7) was applied. The vehicle turned out to have a high sensitivity to front axle input at approximately 150 Hz (figure 8).

Based on in-house tyre design guidelines, modifications of the stiffness and tension of different parts of the tyre were introduced into the FEM model of the tyre. From the modal model the corresponding transmissibility from the contact patch to the spindle were calculated (figure 9). Typically a maximum shift of the natural frequencies of $\pm 10\%$ can be realized, which may result in significant changes in the amplitude of the transmissibility at certain frequencies. In parallel the effect of the design changes on other tyre performance criteria has to be checked carefully to avoid target conflicts. In the present example the best performance balance allowed a reduction of the 150 Hz noise peak from approximately 60 dB(A) to 55 dB(A), resulting in a much more pleasant sound (figure 10).

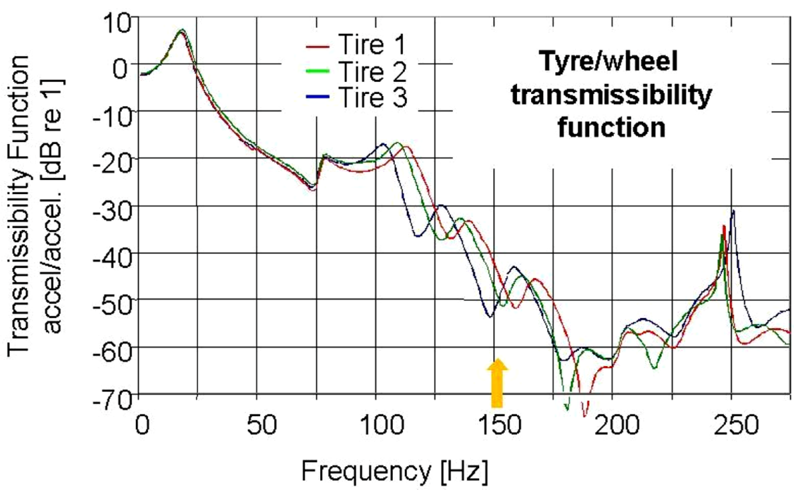

Figure 9. Transmissibility from footprint acceleration to acceleration at the wheel centre calculated from a modal tyre–wheel model. The amplitudes at approximately 150 Hz were significantly changed by tyre design modifications.

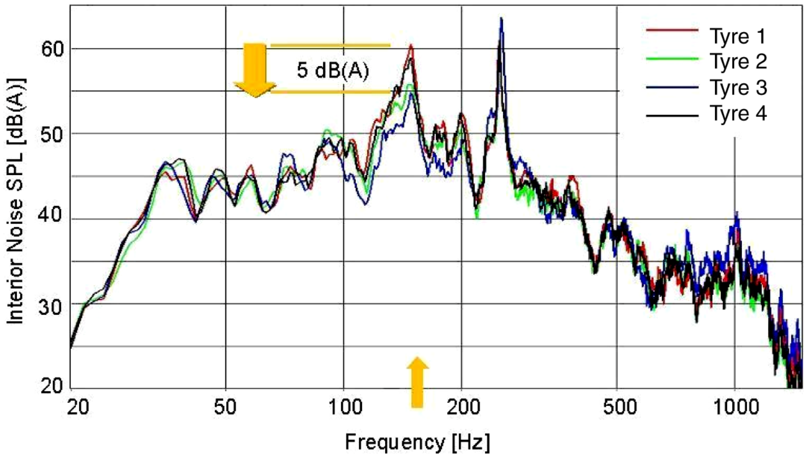

Figure 10. Interior noise spectrum, compact class car on coarse road asphalt at 80 km h^{-1}. Tyres having a lower patch to spindle transfer function at 150 Hz (see figure 9) fit better to this specific vehicle sensitivity, resulting in a reduced 150 Hz interior noise level.

5. Conclusions

The approach at Continental to predict the dynamic tyre characteristic up to 300 Hz was shown. With the aid of a substructuring technique an easy-to-handle modal tyre model is built. Various experiments are performed to validate the modal tyre model. The use of the modal tyre model as a straightforward design tool to reduce coarse road noise was demonstrated.

However, a modal representation of the tyre is limited to linear phenomena below 300 Hz. Further the 'rolling effect' is not completely incorporated. As soon as the tyre is rolling, a description of belt vibrations with mode shapes can only be an approximation. A description of belt vibration as travelling waves is more accurate [8].

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