The influence of modelling parameters on the simulation of car tyre rolling noise and rolling resistance

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Introduction

The simulation of car tyre rolling resistance and exterior rolling noise is a complex task. To achieve valid results with reasonable computational effort, a variety of different physical processes need to be modelled with the right balance between accuracy and simplification. The tyre itself is a highly complex, heterogenous and anisotropic multi-layered structure. Simulation of the tyre dynamics is only possible if the tyre structure is modelled in a simplified way. Next is the tyre/road interaction, which is also impossible to model in every detail. It is by nature non-linear and the interaction processes can cover length scales from several meters down to molecular levels. Finally, radiation calculations need to account properly for the complex geometric amplification effects appearing in the proximity of the contact region. During the development of a suitable simulation tool, design choices have to be made for all these areas. The question is always the same: which level of detail is necessary, which processes need to be included, which parameters are critical, and which are less critical? This study will try to answer some of these question based on experiences which were obtained when a new tyre was implemented into an existing tyre/road noise and rolling resistance simulation tool.

Modelling procedure

The simulation framework is, apart from the modelled tyre, identical to the one described in [1]. The tyre dynamics are modelled using a waveguide finite element approach which combines FE modelling of the cross-section with a wave ansatz for the circumference [2]. Tyre/road interaction is modelled using a non-linear 3D approach which accounts for the alternating relation between contact forces and tyre vibrations [3]. The contact problem formulation reads

$$\mathbf{u}(t_N) = \mathbf{G}_0\mathbf{F}(t_N) + \mathbf{u}_{\text{old}}(t_N) \tag{1a}$$

$$F_e(t_N) = k\, d_e(t_N)\, \mathcal{H}(d_e(t_N)) \tag{1b}$$

$$\mathbf{d}(t_N) = Z_R(t_N) - Z_T(T_N) - \mathbf{u}(t_N)\,. \tag{1c}$$

$\mathbf{u}$ and $\mathbf{F}$ denote the normal tyre displacements and contact forces at time step t_N. $\mathbf{G}_0$ contains the values of the tyre's Green's function for $t_N = 0$. $\mathbf{u}_{\text{old}}$ is the displacement given by the contribution from previous time steps, and $\mathbf{Z}_R$ and $\mathbf{Z}_T$ are the road roughness and tyre profiles. $\mathcal{H}$ is the Heaviside function and the subscript e denotes one individual contact point. To account for small-scale roughness phenomena and the resulting difference between the apparent and real area of contact, contact springs are introduced between the tyre and the road [3]. This is reflected by the spring stiffness k in (1b).

Due to energy conservation, rolling losses can be calculated as input power P_{in} for steady-state rolling conditions [2]. The rolling resistance coefficient C_r is then defined as $C_r = P_{\text{in}}/(F_z V)$, where F_z is the axle load and V the rolling speed. Radiation calculations are based on a half-space BEM approach [4]. Rolling noise is evaluated as mean sound pressure at 321 points on a half-sphere of radius 1 m around the contact point between tyre and road. A-rated sound pressure levels are calculated for the third-octave bands from 100 Hz to 2.5 kHz.

For this study a 175/65 R14 tyre was newly implemented into the simulation tool. Information about the tyre construction and material properties was provided by the manufacturer. The cross-sectional mesh consists of 12 solid elements for the tread and 37 shell elements for the sidewalls and belt. The tyre circumference is discretised into 512 intervals. The rim and the air cavity are accounted for by blocking the tyre motion at the bead and including the pre-tension due to inflation. Losses are implemented as frequency dependent proportional damping. The inflation pressure is 200 kPa, the axle load is 2820 N, and the rolling speed is 80 km/h. The road roughness profile is based on a 3D scan of 15 lateral tracks of a drum-mounted ISO 10844 surface. Rolling is calculated for seven full tyre revolutions of which the last two are evaluated, giving a frequency resolution of 6.1 Hz. More details about the simulation process can be found in [5].

Initial simulation configuration

In this section simulation results are presented which are for an initial simulation configuration in which certain modelling parameters have been specified in form of educated guesses based on previous modelling experiences. In subsequent sections the most important parameters for tyre, contact, and radiation modelling are then varied systematically to assess their influence on the simulation results.

The total A-rated sound pressure level $L_{p,A,\text{tot}}$ is 91.1 dB. A simulated third-octave band rolling noise spectrum is shown in Figure 1. Validation of the results is difficult as no sound pressure measurements exist for the simulated tyre. For comparison reasons sound pressure levels for similar conditions from a measurement of a 205/55 R16 tyre are also shown in Figure 1. The simulation underestimates the measured spectra by up to 10 dB for lower frequencies and around 2 dB in the upper frequency region. At the peak value of the simulated spectra, i.e. at 1250 Hz, the difference to the measured value is 4 dB. It

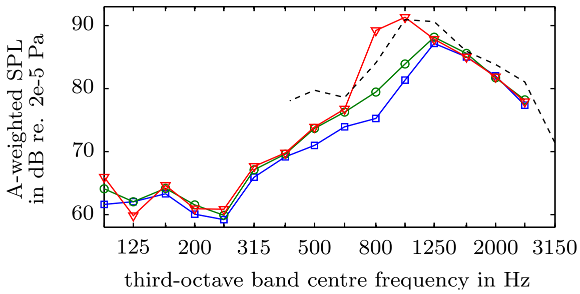

Figure 1: Sound radiation for optimised (□) and original (○) material values, missing P_{i3}/Q_{i3} coupling terms (▽), and measurement results (– –) for a similar, but not identical tyre.

is difficult to assess if these differences are relevant or not because of the different tyres used in simulation and measurement. Differences in measured sound pressure levels of several decibels for different tyres under otherwise identical operation condition are not uncommon [6].

For the initial configuration the rolling resistance coefficient C_r is 0.55 %. Again, no measurements exist for the simulated tyre. A comparison with results reported in literature is insofar difficult as the measurement technique has a large impact on measured rolling resistance values [2]. Nevertheless, the simulations seem to underestimate typical values provided in literature by around 50 %, possibly due to the damping implementation which underestimates the losses in the tread area [5].

Tyre properties

The material data provided by the manufacturer does not account for the changes which occur during the moulding process; neither does it account for discretisation effects in the simulations. Due to this, adjustments of the material parameters are necessary. An optimisation process was performed based on comparisons of measured and simulated mobilities for 25 different positions on the tyre. As an example, in Figure 2 input mobilities for simulations based on original tyre design material data and optimised data are compared with measurements. Although there is already good agreement between measured mobilities and simulated mobilities based on tyre design data, clear differences between the mobilities before and after the optimisation are also visible.

Differences between optimised and non-optimised material data are also visible in the rolling noise spectrum in

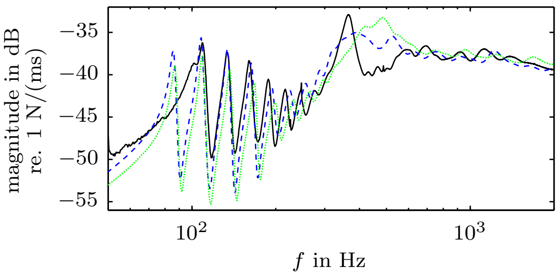

Figure 2: Input mobilities on tread centre line from measurement (—) and simulations using original (· · ·) and optimised material parameters (– –).

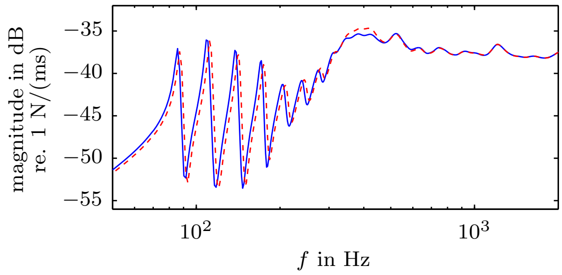

Figure 3: Input mobility on tread centre line for tyre models with all coupling terms (—) and no P_{i3}/Q_{i3} terms (– –).

Figure 1. The curve for the non-optimised material values is on average 1 dB higher for all third-octave bands apart from 125 Hz and 2 kHz. Total rolling noise increases from 91.1 dB for optimised values to 92.2 dB for non-optimised values. The C_r value for non-optimised material data is 0.51 %. This is 9 % lower than for the optimised material values.

Due to the angle at which the steel cords in the tyre belt are oriented with respect to the circumferential direction, some of the shell elements in the WFE tyre model have stiffness terms P_{i3} (Q_{i3}); $i = 1, 2$; which couple in-plane strain and shear (bending and twist of curvature). The calculation of these stiffness terms can be problematic as the belt layers combine the highest extensional stiffnesses with a very small thickness. Minor errors in the determination of the thickness can lead to considerable differences in the calculated stiffness of the shell element. Additionally, a previous WFE tyre model which has been successfully used for rolling resistance and rolling noise simulations [3], did not include the mentioned coupling terms as the necessary material input data was not available. Because of these two reasons, an investigation of the influence of these coupling terms is worthwhile.

Input mobilities obtained without coupling terms are shown in Figure 3. The omission of the P_{i3}/Q_{i3} coupling terms has nearly no influence on the mobility at all. The biggest difference compared to the configuration with all coupling terms is an increase of roughly 0.7 dB for the peak of the first belt bending order around 390 Hz. Similar results are obtained for transfer mobilities over the whole tyre, where additionally some smaller differences are observed above 1.5 kHz.

This limited effect can be explained by the number of shell elements which are affected by omitting the P_{i3}/Q_{i3} terms. Major parts of the tyre contain two belt layers in which the steel cords are oriented at antisymmetric angles. For these areas the coupling effects cancel each other out and the P_{i3}/Q_{i3} terms vanish. Merely a 7 mm long section in each tyre shoulder area contains only one belt layer and accordingly has non-zero P_{i3}/Q_{i3} terms.

The influence of the coupling terms on rolling resistance is only marginal, too. The C_r reduces to 0.54 % for no P_{i3}/Q_{i3} terms. A surprising result is obtained for the third-octave band rolling noise spectra, see Figure 1. Rather drastic changes can be observed when the P_{i3}/Q_{i3} terms are ignored. The A-weighted sound pressure levels

k in kN/m	10	30	50	60
$L_{p,A,\mathrm{tot}}$ in dB	82.4	88.2	91.1	92.7
C_r in %	0.53	0.54	0.55	0.56

Table 1: Dependency of rolling resistance and rolling noise on the stiffness of the contact springs.

increase by more than 11 dB in the 800 Hz and 1 kHz third-octave bands and by about 2 dB to 3 dB in all other bands. The spectrum also matches reasonably well with the measurements for the 205/55 R16 tyre. It is remarkable that by removing two stiffness terms affecting two small areas such an extreme change is obtained. An investigation of calculated transfer mobilities to 24 further points on the tyre and contact force spectra (for both of which results are not shown here), does not provide an explanation. This might instead by given in [4], where it has been shown that wave components of low order in cross-sectional and/or circumferential direction are the dominating vibrational sources for sound radiation over the complete frequency region. The slightly increased response around 390 Hz in the input mobility for the tyre without P_{i3}/Q_{i3} terms, see Figure 3, might indicate an increased response from one or more of these dominating modes. However, further investigations are necessary to proof this, until then this is pure speculation.

Contact modelling parameters

A critical parameter tyre/road contact modelling is the contact spring stiffness k in equation (1b). Results for values of k ranging from 10 kN/m to 60 kN/m are shown in Tab. 1. The influence on the rolling resistance coefficient C_r is limited. However, sound radiation is strongly effected by the contact spring stiffness. Figure 4 shows that the position and amplitude of the maximum in the third-octave band spectrum depends highly on k. From the lowest to the highest stiffness the peak amplitude increases by more than 15 dB. For the highest frequency bands the amplitude differences are around 6 dB. Below 800 kHz the changes are around 3 dB.

It is difficult to asses what the cause for these changes is. The stiffness of the contact springs affects the tyre deformation during contact. Tyre and contact springs act as a set of two springs in series. The overall stiffness at a contact point e is given as $k_{\mathrm{tot}} = (k_{\mathrm{tyre}}^{-1} + k^{-1})^{-1}$, where k_{tyre} is the apparent contact stiffness of the tyre.

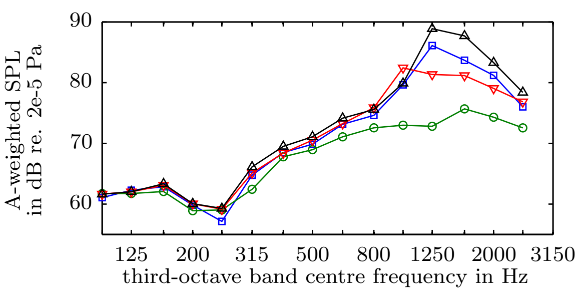

Figure 4: Sound radiation for values of k of 10 kN/m (○), 30 kN/m (▽), 50 kN/m (initial config.) (□), and 60 kN/m (△).

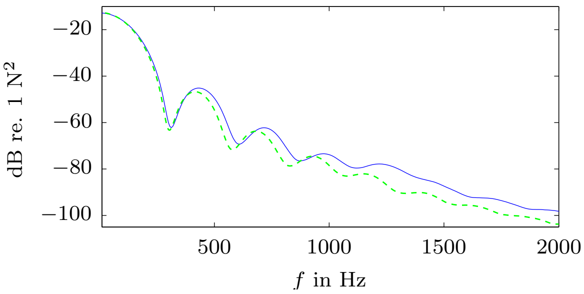

Figure 5: Contact force power spectra at tread centre for $k = 30$ kN/m (−−) and $k = 50$ kN/m (initial config.) (—).

Depending on the actual values of k_{tyre} and k, both the tyre and the spring will show a certain deformation due to the contact force. For a softer spring, for example, a larger part of the total deformation is allocated to the spring instead of the tyre. This difference affects the excitation of the tyre structure, even though it is not the contact displacement but the contact force which is the input variable into the WFE tyre model. The local behaviour of the contact force is dependent on the contact stiffness. The distribution of contact forces over the contact region or the force spectrum are influenced by k. An example is given in Figure 5, where the average contact force power spectra over two revolutions for a position on the tread centre line is shown for contact stiffnesses of 30 kN/m and 50 kN/m. For all frequencies the higher contact stiffness also results in a larger contact force. The differences between the contact force spectra are 0 dB at 100 Hz, 1.6 dB at the peak in the 420 Hz region and 1.1 dB for the peak in the 960 Hz region. For the peak around 1170 Hz the difference increases to 4.3 dB. In Tab. 1 the difference in $L_{p,A,\mathrm{tot}}$ between stiffnesses of 30 kN/m and 50 kN/m is 2.9 dB. The third-octave band maxima are 82.7 dB at 1 kHz for the lower stiffness and 86.1 dB at 1.25 kHz for the higher stiffness. For the latter the peak is also more pronounced. These tendencies (lower frequencies and amplitudes for the lower stiffness, higher amplitudes and frequencies for the higher stiffness) correlate quite well with the observed differences in force spectra. Also the shift of the third-octave band maxima from 1 kHz (soft spring) to 1.25 kHz (stiff spring) could be explained by the sudden increase in difference between the contact forces from 1 kHz to 1.25 kHz.

Radiation modelling parameters

In the contact region high surface velocities on the tyre coincide with very narrow gaps between tyre and road. These near-singular conditions can lead to numerical problems. As a countermeasure, the tyre is slightly lifted above the road surface for the BEM calculations. As the horn effect is very sensitive to geometric modifications, this lift can have a substantial influence on sound radiation. In [7] it is shown that for a 205/55 R16 tyre a lift of 1 mm allows accurate modelling of the horn effect below 3 kHz. The same value is used for the initial configuration here. However, in contrast to the tyre investigated in this study which has distinct circumferential voids, a slick tyre was considered in [7]. It is possible that this

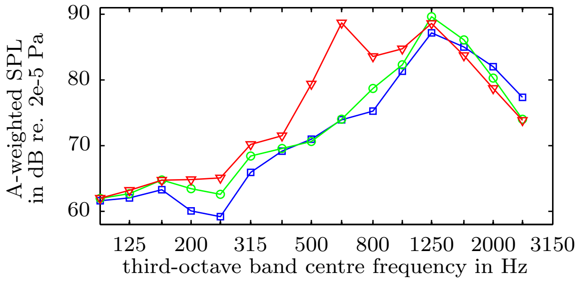

Figure 6: Sound radiation for lifts of 0.5 mm (○), 1 mm (initial config.) (□), and 1.5 mm (▽).

difference results in a different sensitivity with regard to the height of the tyre above the rigid plane. In order to investigate this, sound radiation is calculated for lifts from 0.5 mm to 1.5 mm, see Figure 6. For 0.5 mm the sound pressure levels are higher than those for the standard of 1 mm for frequencies up to 400 Hz and between 800 Hz and 1.6 kHz. In the important 1.25 kHz third-octave band the sound radiation for 0.5 mm is 3.5 dB higher than for 1 mm lift. For 2 kHz and 2.5 kHz 1 mm gives roughly 2 dB higher levels. Drastic differences are obtained for a raise of 1.5 mm. A distinct peak develops at 630 Hz which is 15 dB higher than levels obtained with the lower lifts. The levels in the surrounding third-octave bands are also significantly higher than before. The cause for this peak cannot be clearly identified. It is possible that the region between the road and the lifted tyre forms a duct-like structure with a resonance frequency around 630 Hz. Regardless of the possible cause, it is certain that a lift of 1.5 mm does not accurately model the sound radiation from a tyre in contact with the ground anymore.

These results are in good agreement with the observations made by Brick [7] who calculated the horn effect for lifts of 0 mm, 1 mm, and 10 mm. She reported nearly exactly the same differences between lifts of 0 mm and 1 mm as are obtained for 0.5 mm and 1 mm here. Slight differences between the two studies are only observable at frequencies above 2 kHz. Due to the differences in lift, her results for 10 mm cannot be reasonably compared to the ones for 1.5 mm obtained here.

Concluding remarks

Variations between non-optimised and optimised material values can be observed in the rolling loss and rolling noise simulations. With less than 10 % difference for the rolling resistance and a 1.1 dB difference in the overall sound pressure level it is questionable if a laborious material parameter optimisation is necessary for all applications. For relative comparisons between different tyres it might be sufficient to directly use the material values obtained from the design process.

It is remarkable which pronounced changes in the rolling noise spectra are obtained by a slight modification of two small regions in the tyre structure. It was proposed that this might be due to the differences in excitation of individual low order modes which are significant radiators. The available data needs to be further analysed to see if any proof for this theory can be found.

The contact springs are a critical parameter for rolling noise simulations. The chosen k has a large influence on the contact force spectra in the frequency regions relevant for rolling noise. It is not clear how the correct spring stiffness can be determined. In [8] a determination procedure is described for non-linear contact springs but this requires very detailed road surface data.

The tyre geometry in the contact zone greatly influences the sound radiation. The lift of the tyre above the road surface is more difficult to assess. It could be reasonable to assume that the lowest value which does not give numerical problems should be chosen. The risk is that it might be difficult to judge when exactly numerical errors occur. Obvious numerical instabilities or an increase in calculation time were not be observed with the reduced heights in this study. Furthermore, for horn effect simulations it has been shown that tradeoff between amplitude and frequency accuracy exists which is determined by the amount of lift of the tyre above the road surface [7].

The most important outcome of this study is given by Figures 1 to 3 which show that it is not generally possible to relate variations in mobilities to changes of rolling resistance or rolling noise and vice versa.

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