Evaluation of wheel sound radiation by computer-simulation

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Abstract
For running trains one of the main sources of sound radiation are the wheels. In spite of the improvements reached during the last years there is still a lot of efforts going on to develop new types of “silent wheels” using a new wheel design or different types of wheel absorbers. Important wheel parameters are e. g. the mass, the heat conductivity, the damping and the kind of break usage (disk breaks or block breaks). To predict the sound radiation from the wheels, tools like FEM and BEM are appropriate to use.

Using FEM/BEM some of the new wheel designs currently investigated by the Deutsche Bahn may have lower sound levels than existing ones. Therefore we demonstrate two different techniques to optimize the acoustic behaviour of a wheel on the basis of two new wheel designs.

In both cases the stiffness of the wheel disk was to be enlarged compared to the recent type. One possibility to enlarge the stiffness of the wheel disk is to ondulate it, another one is to add ribs on the conventional flat wheel disk.

Due to the fact that the exact excitation of the wheel is not known, it is difficult to predict the resulting noise level of a certain new designed wheel with an accuracy of a few dB. This paper illustrates for both simulation methods a possibility how to do computer simulation using FEM- and BEM-techniques in order to optimize the acoustical behaviour of a new wheel design without calculating exact noise level values.

1 Introduction
The sound radiation of the wheels (and the track) is caused by their roughness. Because of its roughness, the vibration modes of the wheel will be excited while the wheel is rolling. The vibration amplitudes of these modes are modified by the wheel-track-interaction compared to the amplitudes of the stand alone wheel. Most of the radiated sound is caused by only a few vibration modes like the radial modes $R,0,2$ and $R,0,3$ or the axial modes $A,1,2$ and $A,2,2$ for instance [1]. The frequencies of the modes are in the range from 1500 Hz to 3000 Hz for a conventional wheel design.

One method to lower the sound radiation of a wheel is to reduce the excited vibration modes by raising their frequencies to higher values which will not be excited anymore. So modes with frequencies higher than 4000 Hz will be excited significantly weaker than modes with lower frequencies. Increasing the vibration frequencies of a wheel is generally possible by enlarging the thickness of the wheel disk with the disadvantage of a larger mass. This can be avoided by an enlarged axial stiffness of the wheel disk using an ondulated wheel disk or a ribbed wheel disk.

In the first part of this manuscript, we present a systematic investigation of the effect of ondulating the wheel disk on the frequency of the A,1,2 vibration mode. Ondulating means that the wheel
shape is sinusoidal in the direction parallel to the axle (see figure 5). The task of this study was to design an ondulated wheel which shifts the frequency of the vibration mode $A_{1,2}$ to a preferably high value.

Within the framework of a cooperation of Deutsche Bahn AG, Gutehoffnungshütte Radsatz GmbH (a company of the CARDO RAIL Group) and University of Dresden, we have employed the technique of modal analysis using FEM-tools in combination with considerations of the exciting frequencies generated by the rolling wheel. Since we focused solely on the frequencies we did not make any attempt to predict resulting sound levels.

The second part is devoted to acoustical simulation on wheels with various rib designs fixed on the conventional flat wheel disk which have been performed by Deutsche Bahn in cooperation with IABG. That project reflects the effect of different ribs with respect to the radiated sound level among each other. The treated wheel-track systems have been excited by an uniform acceleration on the wheel-track-contact point without taking wheel rotation into account.

2 Excitation of a wheel

The realistic excitation of a wheel, i. e. its frequency and time dependent acceleration, is difficult to simulate because it depends in a complicated way on the roughness both of the wheel and the track. Therefore some simplifications are necessary.

One simplified excitation model of the wheel is the “uniform excitation”. This model assumes a frequency independent acceleration of $a = 1 \text{ m/s}^2$. Therefore it does not yield realistic radiated sound levels of a single wheel, but it provides at least a comparison of mode shape amplitudes of different similar wheel designs. The same holds for the frequency response functions. This excitation model will be used in the second study reported later on.

For the first investigation more general considerations have been made: the frequency range of excitations can be estimated by considering the frequency of rotation of the wheel, $\omega_0 = v/L$ ($v =$ speed of the wheel in track direction = train speed, $L =$ circumference of the wheel).

While in reality both wheel and rail have their own roughness, we have modelled the wheel/track interaction by assuming a perfectly smooth track surface and absorbing the combined roughness completely into the wheel roughness. The dominant wavelengths of the roughness are known to be in the range from $l = 0.03 \text{ m}$ to $0.06 \text{ m}$.

Assuming a periodic roughness with wavelength $l$ and a circumference $L = 3 \text{ m}$ of the wheel leads to $z = 50$ up to $z = 100$ equidistant perturbations on the wheel. Therefore the wheel will be excited by the frequency $\Omega_1 = z\cdot\omega_0$.

Subject to these simplifications the range of excitation frequencies of the wheel will be up to $f = 2200 \text{ Hz}$ for a maximum train speed $v = 240 \text{ km/h}$. Assuming a periodic excitation also whole-numbered multiples excitations are possible, of course. That means that also the higher order excitations of modal frequencies $\omega_i = k\cdot\omega_1$ ($k = 1, 2, ...$) are possible. By a short calculation considering the energy behaviour it can be shown that for a rolling wheel only those vibration modes with mode number $m$ equal to order $k$ can be excited in contrast to the standing wheel.

This is the reason for trying to lift the dominant resonance frequencies of the wheel, i. e. the frequency of the mode $A_{1,2}$, to values beyond $4400 \text{ Hz}$.

3 Optimization of a wheel with ondulated wheel disk

This optimization process was performed in three steps. The intention of each step was to increase the frequency of the vibration mode $A_{1,2}$. The only boundary condition in each calculation was the mass of the wheel limited to 400 kg.
In the first step the geometry of the cross section was modified by optimizing the twelve different coordinates and thickness values shown in figure 1, starting from the cross section of a conventional wheel.

![Fig. 1 Cross section of the wheel: parameters to be optimized](image)

Each variation of the cross section was followed by a modal analysis with respect to the A,1,2-mode (figure 2).

![Fig. 2 Cross section of the wheel during optimization process (A,1,2 – mode)](image)

A 2D-model of the cross section of the wheel was applied using harmonic finite elements in order to get highest accuracy within short calculation times. Finally, the A,1,2-frequency could be lifted up to 2640 Hz.

In a second step the best ondulation design was calculated. Again, the simulation was done using a reduced CAD model of the wheel composed of one ondulation segment (with appropriate boundary conditions). In order to find the optimum ondulation number, the best amplitude, and the optimum thickness of the wheel disk a large meshed 3D-model was sufficient (figure 3).
Fig. 3 3D-FEM-model of one sector of the ondulated wheel

Again, each change of the wheel geometry was followed by a modal analysis with respect to the A,1,2-mode (figure 4).

Fig. 4 Modal frequency of the ondulated wheel shown in figure 3 at $f = 3340$ Hz
For the ondulation number equal to 10, we have found the frequency of the A,1,2-mode at 3450 Hz. Higher ondulation numbers of 12 or 14 result in slightly higher frequencies, but the more the wheel disk is ondulated, the more difficult and more expensive is its manufacturing. Therefore, the best compromise for the ondulation number is 10. Variations of the amplitude of the sinusoidal wave do not lead to significant changes of the investigated frequency as do not do variations of the thickness of the ondulated wheel disk.

Finally, a high quality fine meshed 3D-FE-model of that wheel was built up (37360 Elements) using the optimized geometrical values (figure 5). A modal analysis of this wheel gives all resonance frequencies in the range up to 5000 Hz (see figure 6 for instance).

![Fig. 5 Fine meshed 3D-FE-model of the ondulated wheel under investigation](image)

![Fig. 6 Modal frequency of the ondulated wheel shown in figure 5 at f = 709 Hz](image)
4 Investigation of the influence of various rib designs on the sound radiation

The aim of the second project was to investigate the effect of certain ribs added to the wheel disk of a conventional wheel. Therefore a combined wheel-track-FE-model was set up taking symmetry conditions into account (figure 7). The wheel and the track (length 6 m) were given a common node at the contact point. In order to excite the system at this node, its mass was defined to be m = 10^6 t, which is much more than the wheel-track-system itself (“large mass system”). To simulate a uniform excitation, a force F = m•a was applied to the system (m = 10^6 t, a = 1 m/s^2) [2].

Performing a modal analysis leads to the vibration modes. With a frequency response analysis afterwards the vibration amplitudes at each node of the FE mesh (wheel and track) can be determined. These values have been projected onto the surface nodes of a corresponding BEM-model of the system (figure 8).
From the acceleration on the surface of wheel and track, the sound radiation can be calculated by certain software tools like SYSNOISE [3]. Due to the above described limitations of the model of uniform excitation, the calculations will not lead to realistic quantitative sound level values, but they provide an acoustic evaluation of different wheel-track-systems with varying types of ribs with respect to each. In order to compare different rib types, calculations were carried out with three different wheels shown in figure 9.

![FEM-models of the differently ribbed wheels](image)

Fig. 9 FEM-models of the differently ribbed wheels (lineal rib, C-rib, S-rib) under investigation

While the modal analysis allows the visualisation of the vibration modes (see figure 10), the frequency response analysis enables an evaluation of the effect of each single resonance frequency (see figure 11).
Fig. 10 Vibration mode A,1,2 of the C-ribbed wheel at $f = 2434$ Hz

Fig. 11 Comparison of the Input Power of the lineal ribbed wheel and the C-ribbed wheel
The acoustical simulation using FEM- and BEM-tools allows a qualitative visualisation of the radiated sound levels with respect to sound values and direction (figures 12, 13). So it can be shown for instance that the sound radiation caused by the track, dominating for low frequencies, effects only the near distance region while the radiation due to the wheel which is overbalancing for higher frequencies is the main radiation source for large distance terms.

![Spatial sound pressure distribution of the C-ribbed wheel at f = 1245 Hz](image)

Fig. 12 Spatial sound pressure distribution of the C-ribbed wheel at f = 1245 Hz
5 Results

The intention of this report was to briefly describe two different simulation methods in order to acoustically optimize a railway wheel. The most important results are summarized in the following. In general, a stiffer wheel disk leads to an improvement because the frequencies of some of the modes of vibration relevant for sound radiation are shifted to such large values that they will not be excited anymore. Also, the higher stiffness of the wheel disk leads to lower mode shape amplitudes of some of the excited vibration modes.

The following table compares the optimized ondulated wheel as described in section 3 with the corresponding wheel with plane wheel disk. The shift of the modal frequencies up to higher values is remarkable.

Fig. 13 Spatial sound pressure distribution of the C-ribbed wheel at $f = 1935$ Hz
The main focus of the second project reported was to compare the effect of various ribs attached to a conventional wheel. In general all different ribs lead to a stiffer wheel disk which results in a lower input power in the wheel-track-system when excited. The lowest input power will be obtained with the C-type rib. Figure 14 compares the input power of the standard wheel with the same wheel when C-ribs are attached.

![Fig. 14 Comparison of the Input Power of the standard wheel (without ribs) and the C-ribbed wheel](image)

6 Discussion
Both calculation methods are based on a modal analysis and show possibilities to investigate and optimize a wheel with respect to its acoustic behaviour.
Best quantitative values of the vibration frequencies and reliable statements about the wheel under investigation require sufficiently large model sizes, which requires powerful computers and long calculation times. But also reduced FE-models will lead to useful results for a certain optimization task.
While simulation of the ondulated wheel took only a few hours of computing time on a standard workstation, the computation of the different ribbed wheels including the complete frequency response analysis and the prognosis calculation of the sound levels took several weeks.
Shortening of computing times is mostly possible by reducing the model size or by splitting the complete calculation into several parts for a separate investigation of the low and the high frequency region. Usually the FE-element has to be such big that six elements fits the treated wavelength.
In addition, 2D harmonic elements may be used in certain cases instead of a complete 3D-model calculation. This reduces the calculation significantly.
Reducing the computing times when optimizing the acoustic properties of a wheel is particularly important, because the “acoustic optimization” naturally affects the mechanical properties of the wheel. This might conflict with other requirements concerning e. g. tightness, strength of the thermal behaviour. Consequently, acoustic requirements can be met in the design process the easier, the more readily corresponding simulations are available.
7 Outlook

Further important statements on the acoustic behaviour of the investigated wheel design can be obtained by combining both reported methods. So the simulation of the ondulated wheel similar to the computation of the ribbed wheel will show up possible disadvantages caused by the shifted frequencies (or perhaps significant lower input power compared to the ribbed wheels). On the other hand, the C-ribbed wheel can now be optimized regarding the base geometry of the wheel and the geometry of the C-formed rib using a combination of both methods, starting with the first one.

Bibliography


Keywords
Wheel, sound radiation, computer simulation, FEM, BEM