Numerical Vibration Response of Railway Track Retrofitted with Single Degree of Freedom Rail Dampers

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Abstract. A concept and preliminary results of numerical study of changes in the rail track dynamic characteristics are presented in the paper. Change of the characteristics is introduced in order to reduce noise emissions. The theoretical relationship between the studied track dynamic characteristics (TDR - Track Decay Rate) and noise was briefly explained, and then the methodology of FEM modelling for the standard test simulation was described. Three mass modification variants and two rail types were analyzed. Damper connection were modeled as viscoelastic joint. The rail track vibrations were analyzed in both (vertical and lateral) directions separately. Numerical study shows correlation between the railroad track mass increase and its dynamic characteristics and thus the noise emission. Applying dampers with proposed parameters results in more efficient vibration and noise mitigation in lateral direction.

1. Introduction
Rail transport constitutes a source of significant noise and vibrations - mainly due to mechanical vibrations and, to a lesser extent, aerodynamic noise caused by the air flow over the train structure. The reduction of these negative effects without significant financial expenses and excessive complications in the construction and operation of railways should go hand in hand with the development of railway infrastructure. The key to achieving this goal is to understand the sources of noise and vibration and the parameters that may affect them. Subsequently, economically justified technical solutions should be proposed.

In most situations the dominant source of noise from operational railways is that due to the rolling of the wheel along the rail [3, 7, 8, 16]. It is generally agreed that rolling noise is generated by "roughness" on the wheel and rail surface. As a consequence, this results in relative vibrations of the wheel and rail, wherein the amplitude of the vibrations of each element depends on its dynamic properties. The vibrations created in this way are the main source of noise. There are many studies devoted to the theoretical description of the phenomena [12] as well as analytical and empirical models to predict the intensity of the sound emitted for particular types of wheel and track depending on the roughness. The TWINS model (Track Wheel Interaction Noise Software) [9] is the most widely used, thanks to which rolling noise can be calculated as the sum of the sound power emitted from the wheel, rail and underlay.

The dynamic properties of the railway track are crucial in noise prediction, and the appropriate track dynamics model should take into account interactions between different components [15]. The rail has much more damping capabilities than the wheel due to its fastening system. This damping is mainly due
to energy loss during transmission of a mechanical wave from the excitation point along its length. The speed of decay of these waves compared with the distance along the rail determines the so-called effective length of the rail. The damping introduced by fastening to the sleepers is an important parameter for the track, because it affects vibration decay along the track, and thus determines the effective length that is subject to vibrations. The longer the rail section, which vibrates under wheel excitation, the more noise will be emitted. One can distinguish three main sources of this damping: fastening systems (e.g. rail washers), energy transferred to sleepers and embankment, and damping of the rail [10]. Fastening systems block transmission of vibrations along the rail but only at low frequencies. Transfer of the vibrations to the sleepers and embankment (application of rigid supports) has to be limited as sleepers themselves can become the source of noise, and the vibrations have negative effect on objects located nearby the railway. In this context increasing the ability of the rail itself to damp vibrations seems to be the best solution.

The amplitude of the rail vibrations decreases approximately exponentially with the distance along the track (more damping equals faster decay of the vibrations). The parameter used to describe this phenomenon is the decay rate of the vertical and transverse vibrations of the track (Track Decay Rate – TDR [5]) and its value is usually expressed in dB / m. Track decay rate is the most important indicator of the track dynamics in relation to rolling noise reduction. Low values of this parameter lead to a greater effective length of the rail, and thus to high noise emission. High TDR values correspond to less noise and can be obtained e.g. by using rigid washers between the rail and the sleepers. However, soft pads are preferred for non-acoustic reasons, e.g. to minimize concrete damage or vibrations transmitted through the ground. TDR values from measurements are commonly used to calculate rolling noise in models such as TWINS. According to [4], the coefficient can be determined experimentally, using an impact hammer. It is also possible, although technically more difficult, to determine the TDR coefficient by measuring the vibration of the rail during train passage.

The relationship between TDR parameters and the noise emitted can be found e.g. in the annex to the standard [4]. The vibration amplitude of the rail decreases with the distance from the excitation point. This is the point of contact between the wheel and the rail or the point of impact with a hammer during TDR test. Value measured is usually the acceleration or velocity of the rail in the direction perpendicular to the track axis (vertical or horizontal). If the goal is to reduce noise, it is desirable to design a solutions that increase the TDR while keeping sleepers insulation (avoiding introducing stiffer rail pads). One of the more effective solutions are rail mounted dampers [1,14] (Tuned absorber systems). The theory of such vibration absorber was first presented in [13].

The main purpose of the article is to present the methodology for evaluating TDR depending on the dynamic characteristics of the track. Changing the characteristics can be achieved by using rail dampers. Computer simulations with finite element method (FEM) approach were used for initial evaluation of the concept. The analysis was carried out in two stages. In the first stage, the entire track was modeled...
(one rail track, taking into account sleepers and the ballast as elastic foundation). The results from the first stage analysis served as a reference. In the second stage, the track was modeled along with a simplified model of the device (in the form of a discrete mass). The main goal is to develop a rail damper that significantly increases the TDR coefficient and thus is potentially the most effective solution in terms of noise reduction.

2. Methodology
In order to predict a rail track dynamic characteristics FEM simulation was conducted. The methodology of baseline track modelling is presented in previous paper [2]. However, in this article rail track vibrations were analyzed in both directions – vertical and lateral (previously only vertical). FEM analysis was carried out in the Abaqus® environment. The entire system was modeled in 2D space (vertical and lateral direction separately).

The rail was modeled with a continuous beam. Discrete viscoelastic joints (rails with sleepers and sleepers with the ground) were created using special elements designed for this purpose (so-called engineering features). A linear load was assumed in the form of a rectangular pulse, distributed evenly over a distance of 6 cm in the middle of the span. The duration of the pulse is 0.0023 s and its amplitude is $2.775 \cdot 10^6$ N/m, which is supposed to simulate a hammer impact during the standard TDR test. The beam has been divided into 15 cm finite elements with quadratic shape functions. The model reproduces a fragment of a track with a length of 120 m. However, because the symmetry conditions have been used, the length of the model is 60 m.

The dynamic analysis was carried out in two steps (standard solver – implicit method). The first step is the load phase lasting for 0.0023 s. The second step represents free vibrations in the time of 0.20 s. A constant increase of time in the following analysis moments was assumed, which was $8.0 \cdot 10^{-5}$ s, allowing for identifying of vibrations with a frequency of 5000 Hz. After the calculation, an acceleration signal, recorded at the nodes, was acquired at the same distance as during the standard test procedure [4].

In previous paper also track [2] with dampers mounted was modeled. Single damper was represented by a discrete mass element rigidly connected to the rail. In this stage of research the rail dampers are single degree of freedom oscillators. 1DOF mass is connected to the rail with viscoelastic discrete connection. The scheme of the model is presented in figure 2. The rail track parameters were adopted mainly on the basis of previous studies available in the literature [6, 11] and are summarized in table 1.

| Table 1. Parameters for baseline track |
|---------------------------------------|
| unit       | vertical | lateral |
| Rail pad stiffness | N/m      | 2.00E+8  | 6.90E+7  |
| Rail pad viscosity | N∙s/m    | 1.00E+6  | 3.45E+5  |
| Sleeper mass (equivalent) | kg       | 150      | 300      |
| Ballast pad stiffness | N/m      | 5.00E+7  | 4.00E+7  |
| Ballast pad viscosity | N∙s/m    | 1.00E+5  | 8.00E+4  |
The parameters for rail dampers were assumed arbitrarily for purposes of preliminary numerical study, and summarized in Table 2. The value of rail damper stiffness was selected so that the vibration eigenfrequency is about 1000 Hz with a weight of 2 kg.

Table 2. Parameters for rail dampers

| Unit                     | Value   |
|--------------------------|---------|
| Rail damper stiffness    | 2.51E+7 |
| Rail damper viscosity    | 1.00E+3 |
| Rail damper mass (equivalent) | 2   |

Acquired acceleration signal was filtered and post processed according to the standard procedure [4] and then TDR was calculated.

3. Results and discussions
The dynamic characteristics of the track used to assess the effectiveness of dampers is highly processed and aggregated information. Before calculating the TDR, the output and intermediate calculation results were analyzed. Selected results will be presented and discussed below. In the case of acceleration figures the load step is highlighted with a green color and the free vibrations are plotted on a white background.

Figure 3 presents acceleration signals acquired in the point of rail excitation. All the signals are aligned, so it seems as one. It is because the dampers do not influence significantly first, intensive stage of the rail vibration near the impact point.
Figure 3. Acceleration signals acquired in the point of rail excitation (see figure 2) – track with rail profile 60 E1

Less intensive vibrations, acquired 3.0 m from excitation point are influenced slightly by adding the rail dampers (see figure 4). The maximum amplitude is lower, when rail damper is mounted. Also the amplitude decreases with time faster, especially for 2 kg damper.

Figure 4. Acceleration signals acquired 3.0 m from point of rail excitation – track with rail profile 60 E1

Figures 5 and 6 present averaged amplitude of acceleration signals (calculated as moving average of the signals absolute value). The reason of this filtering is to clearly present signals intensity. By analyzing these results, one can see that 2 kg damper performs the best in vibration mitigation. The efficiency of other dampers is slightly worse. However, all the dampers decrease the rail vibrations when compared with baseline (regular) track. One could probably draw a conclusion that the best performance of 2 kg damper is related with better frequency tuning. Nevertheless it should be further investigated.
Finally the results in both directions (vertical and lateral) and both rail profiles (49E1 and 60E1) are presented in figures 7-10. In most of the cases, application of rail damper (especially with mass of 2 kg) increased TDR in some frequency ranges without decreasing in others (what could be observed in case of rigid dampers from previous study [2]). The track performed much better in lateral direction after adding the dampers. Only track with rail profile 49E1 analyzed in vertical direction did show only slightly improvement in vibrations mitigation. Once again this is probably related to the need of the proper damper tuning (depending on the track properties e.g. rail profile).
Figure 7. Track Decay Rate results obtained from simulation for rail track with rail profile 49E1 in lateral direction.

Figure 8. Track Decay Rate results obtained from simulation for rail track with rail profile 49E1 in vertical direction.

Figure 9. Track Decay Rate results obtained from simulation for rail track with rail profile 60E1 in lateral direction.
4. Conclusions

Approach presented in this paper is a further development of methodology shown in the prior article [2]. The rail track vibrations were analyzed in both directions (vertical and lateral separately) by numerical simulations. Furthermore, the rail track with dampers was also simulated. Three different weights of damper were tested. Every damper was modeled as single degree of freedom oscillator connected to the rail (beam element) by viscoelastic joint. Simulation results were post processed according to the standard approach [4]. On the basis of the obtained results, following conclusions can be drawn:

- The same damper performs different when applied to the track with different rail profiles.
- Applying dampers with proposed parameters results in more efficient vibration and noise mitigation in lateral direction.
- The lightest damper performed the best. However, it is probably due to the better tuning rather than effect of applied weight. It should be further investigated.

Presented results and conclusions outline the scope of future work, which is:

- developing more sophisticated numerical model (e.g. increasing the complexity of damper model),
- obtaining damper parameters on the basis of actual shape and material properties, rather than assuming them arbitrarily,
- comprehensive parametric study of different rail damper designs in order to choose the most efficient solution for every track type.

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