Use of Railway Track Vibration Behaviour for Design and Maintenance

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Summary

Components applied in a railway track structure (e.g. rails, fastening systems and supports) are significantly influencing the vibration behaviour of this so-called 'superstructure'. Superstructures for trams, metros and trains can be classified on the impact that the vibration behaviour has on the wheel-rail interaction and on sound radiation. Methods for recording, simulation and analysis have been developed to perform an assessment, which supplies track managers with arguments on track design and track maintenance.

Keywords: railway track, vibrations, dynamics, recording, design, assessment

1. Introduction

A railway track structure forms the physical mean, which is necessary for the support and the guidance of a railway vehicle. The use of a railway track structure by a vehicle initiates dynamic loading, which gives vibrations of the interacting systems of the train and the track. The vibration behaviour of the railway track structure in the mid- and high-frequency range (40-1500 Hz) can act as an indicator for the performance of the track structure with respect to sound radiation, vibration sensitivity and wheel-rail interaction forces. These aspects deserve the attention of railway track managers: a well qualified and - even better - a well-quantified performance indicator is a suitable guiding instrument in track design and track maintenance. The research project Dynatrack, performed at TU Delft under the auspices of the foundation of applied research in infrastructure in the Netherlands CROW, has resulted in the development of such an instrument. It is however necessary that the impact of the mid- and high-frequency vibration behaviour on the performance is accurately determined and estimated, and also placed in perspective to other issues of the track.

2. Railway structures and components

The variety in railway track structures is very large for numerous reasons: historical, technical, economical, local, operational, etc. Railway track structures have either the appearance of rails simply mounted on (transverse) sleepers in ballast, or the appearance of rails mounted on a steel bridge or a concrete slab by means of a more ingenious fastening system. This complies with the ordinary distinction between two groups: ballasted track and slab track. For a closer look into the vibration behaviour of track structures, the group of continuously supported 'embedded rail' tracks takes a special position, apart from all other track structures, which are discretely supported.

The designs of railway track structures and the properties of the applied components are very diverse. A suitable mechanical model of a railway track structure is however a prerequisite in order to determine the vibration behaviour and the performance. In the past numerous analytical and numerical models have been formulated, which all incorporate design and component parameters of track structures, but to a certain extend, see e.g. Figure 1 for a 2D model. The parameters in terms of the dimensions of the track structure and the applied components (e.g. type of rail, fastener, sleeper, etc.) are generally constant along a track section, but tolerances in construction and maintenance works do exist. The component properties, in particular those related to elasticity and resilience, are sensitive towards numerous aspects. Thus, vibration behaviour should be seen against the background of a limited knowledge of the exact structural conditions.

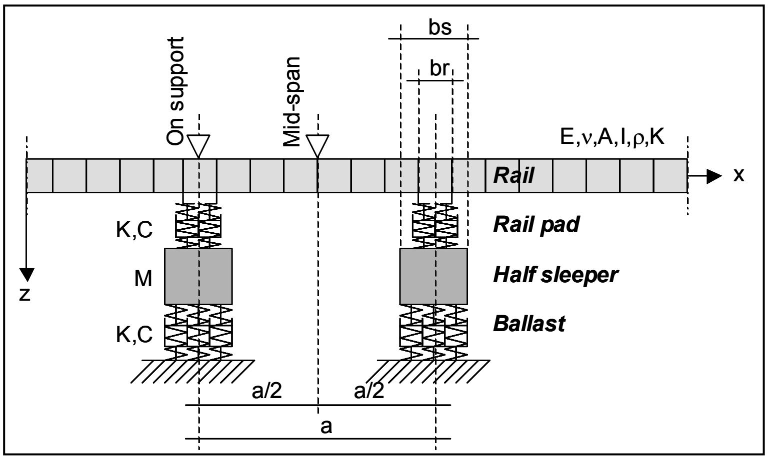

Fig. 1 Ballasted track model with components, properties and dimensions and two reference positions ('on support' and 'mid-span') for typical dynamic track behaviour

3. Contact stiffness and sound radiation

The here-investigated vibration behaviour of track structures pays attention to two performance-based issues: contact stiffness and sound radiation (which might be considered as noise). The first issue relates the vibration behaviour of the track to the vertical forces of wheels loading the track. Dynamic amplification of these interaction forces has several causes, i.e. wheel and rail surface irregularities and track alignment in relation to running speed. Figure 2 gives a schematic impression of the mechanism. Many researches have indicated the effects of surface irregularities, track alignment and running speed on dynamic amplification and subsequent wave phenomena. Few researches however have investigated the influence of track components on these issues [2]. The stiffness, which applies to the contact point of wheel and rail, is a frequency-dependent equivalent stiffness, covering the properties of the wheel and – more important - the properties of the track. Balancing the contact stiffness over the frequency range, decreasing the contact stiffness in general and avoiding interferences at resonant frequencies are three of the most practical tips to reduce dynamic amplification and - by doing so - to reduce wear of both the rails and the wheels [1,3].

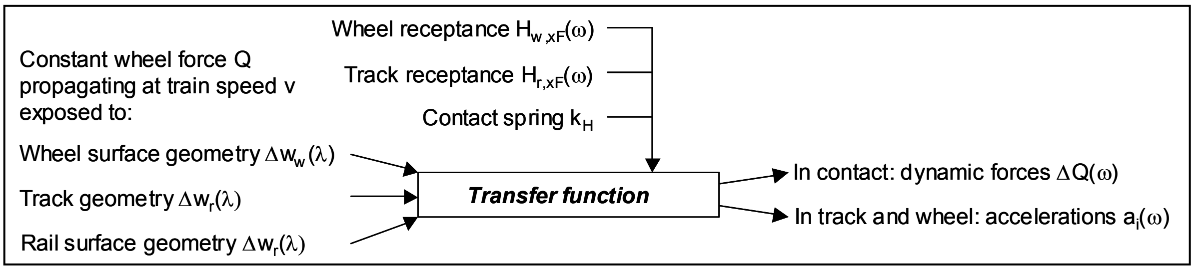

Fig. 2 Interaction model for dynamic track and wheel behaviour

Sound radiation of a track structure, which is only a part of the total sound level, is quantitatively determined by the load (input) of the train and the response (output) of the track structure. Both are expressed in general by spectral density functions. The response can also be expressed by a sound level value in dB(A). The conditions of input and transfer parameters (e.g. position and properties of the transferring media) should be described accurately to enable a systematic comparison of track structures, see e.g. Figure 3. Sound radiation is best explicitly investigated by simplifying the input to a single impact force and by avoiding radiation due to wheel and rail surface irregularities and running speed. It demonstrates the sound performance of an ideally constructed and maintained track.

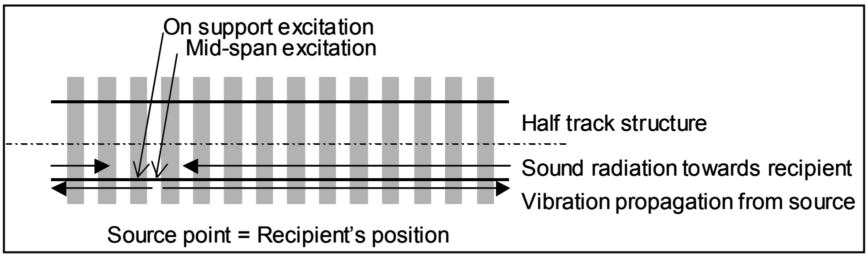

Fig. 3 Propagation of vibrations and radiated sound interception

4. Examples and results

An inventory of component properties, which are significantly influencing the vibration behaviour of track structures between 40 and 1500 Hz, has been performed, followed by an implementation of component properties in models on vibration behaviour. Field recordings have contributed to an understanding of the simulation model results and vice versa. Although the transverse dimensions of the track have been neglected in the model, the recordings show that the sleepers in ballasted track are influencing the rail vibrations, but only in typical frequency ranges. These aspects are clearly illustrated in a comparison of field recorded data and model simulation results as shown in Figure 4 and Table 1.

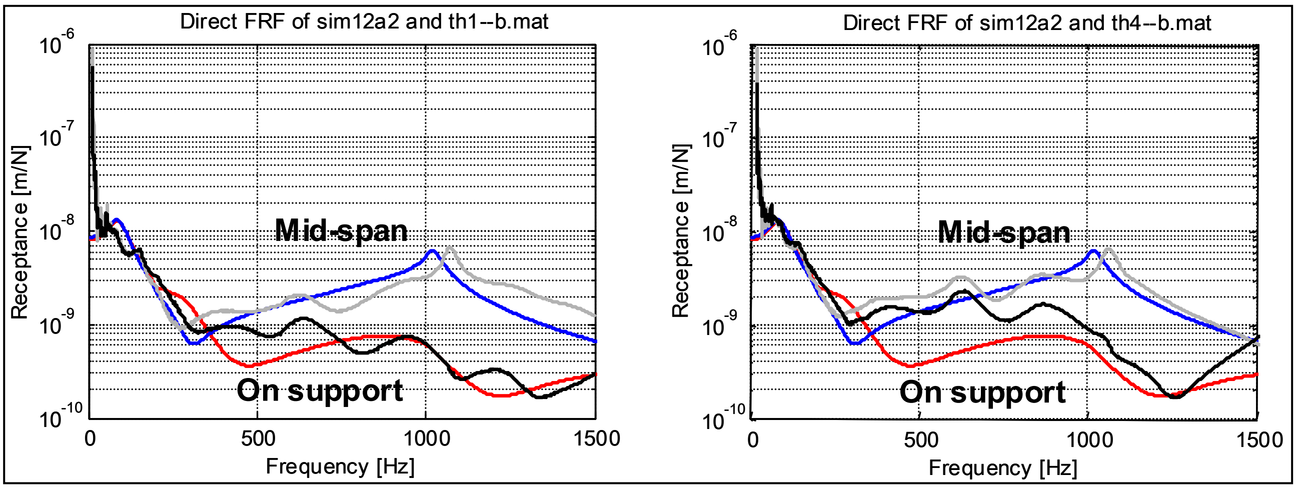

Fig. 4a,b Track receptance functions illustrating the vibration behaviour of ballasted track before (a) and after (b) rail pad replacement. Black and grey lines for field recordings (changed conditions) and red and blue lines for model simulations (constant data).

Table 1 Parameters used for the simulation of the track vibration behaviour in Figure 4

Code	Rail profile	Rail pad K (MN/m)	Rail pad C (kNs/m)	Sleeper M (kg)	Ballast K (MN/m)	Ballast C (kNs/m)	Spacing a (m)
A2	UIC54	1120	120	145	60	57	0.6

The lines of recordings and simulations correspond better in Figure 4a than in Figure 4b, in particular in the frequency range 400-1200 Hz. This demonstrates that the replacement of rail pads has been effective to the track receptance. It was known that the new rail pads were having more resilient properties in the same conditions, but not to what extend. This has been confirmed by laboratory tests on the exchanged and the newly inserted rail pads: a difference in resilience of more than 20% was established. The installation conditions and fatigue can change the resilience of rail pads instantly or over a long period of in-service life.

Both flattened peaks between 400 and 700 Hz are due to sleeper vibrations. The effects of sleepers at rail level are indirectly recorded and thus sensitive to rail pad properties.

The resonant peak at 1050 Hz corresponds to the pin-pin vibration mode, which is typical for discretely supported track. The difference between recorded and simulated results is due to small model simplifications. There is no effect of rail pad replacement observed in this vibration mode.

Finally the mid-frequency vibration behaviour 40-400 Hz is dominated by the sleepers support. The modelling by means of a distributed elastic support under every single sleeper is only suitable for a qualitative impression.

The reading of track receptance functions of ballasted track in Figure 4 is easily transferred to slab track structures. The recorded data or simulated results need to be interpreted in order to determine the appropriateness of the track design or the actual track condition. Numerous examples with alternative track components have shown that tracks can be assessed and optimised to minimum variation in wheel-rail forces and/or to minimum sound radiation [4]. Table 2 gives data for three alternative High-Speed Line tracks: ballasted track, slab track and embedded rail.

Table 2 Parameters used in several simulations on track vibration behaviour

Code	Rail profile	Rail pad K (MN/m)	Rail pad C (kNs/m)	Sleeper M (kg)	Ballast K (MN/m)	Ballast C (kNs/m)	Spacing a (m)
A6	UIC60	300	80	145*	60	57	0.65
B6	UIC60	60	57	12**	500	20	0.65
C6	UIC60	92	88	0***	500	20	N/A

*: sleeper mass: rail is mounted with a resilient rail fastening system on a sleeper

**: baseplate mass: rail and baseplates are mounted on a continuously supported slab

***: rail is mounted on a slab: both are continuously supported

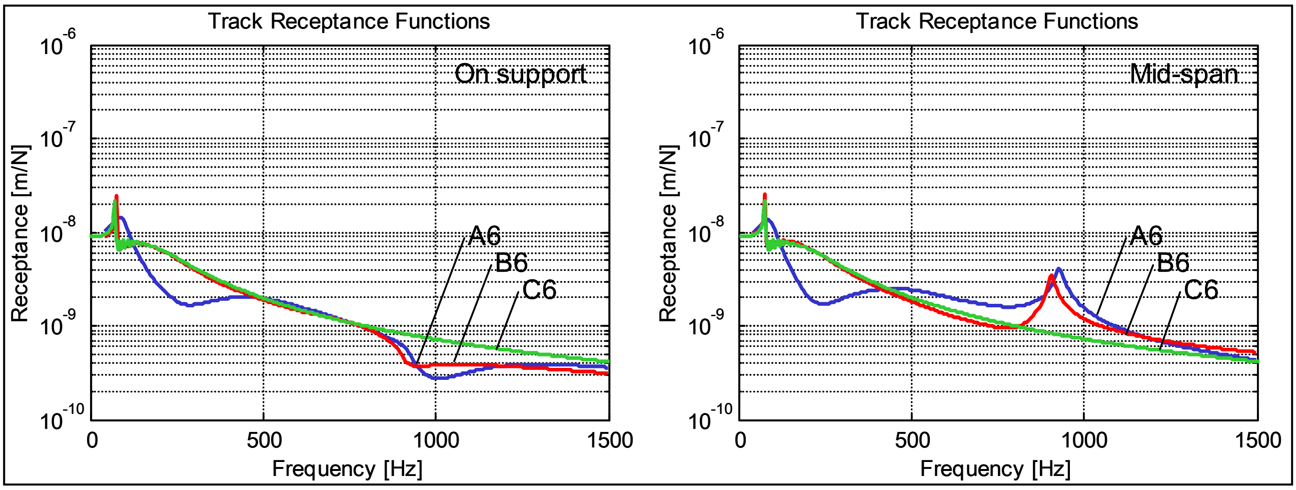

Fig. 5a,b Receptance functions expressing the vibration behaviour of three alternative HSL tracks

Half of the slab weighs 1250 kg per meter and the equivalent portion of the soil mass is 1000 kg per meter. These values only apply to the slab track B6 and to the embedded rail system C6.

According to Figure 2, the variation in the dynamic forces ΔQ is a result of the surface of wheel and rail, the running speed and the vibration behaviour of the wheel-track system. When taking the influences of the geometry relative to the variation in dynamic forces, the contact stiffness is found. A graph is shown in the lower left corner of Figure 6 for track structure A6. Quantitative data is listed in Table 3 and gives the average equivalent wheel stiffness level in two frequency ranges. In fact this contact stiffness is changing between two rail supports: therefore the variation in contact stiffness is also listed.

Table 3 Results on contact stiffness and sound radiation of several simulated tracks

Code	**Contact stiffness relative to Hertzian stiffness (%)***		**Radiated sound (dB re 1e-5 Pa)** of rail, sleeper and slab**
	40-400 Hz	**400-1500 Hz**	
A2	31.8 +/- 3.8	45.4 +/- 18.3	100.7
A6	20.5 +/- 1.4	42.4 +/- 8.7	100.0
B6	13.8 +/- 1.3	44.3 +/- 6.1	100.2
C6	14.0	43.1	99.8

*: average value and variation for on support and mid-span positions with wheel properties M_{wheel}=675 kg, R_{wheel}=0.45 m, P_x=10 tonnes: Hertzian spring stiffness 1657 MN/m

**: due to uniform load spectrum of 1e3 Ns between 40 and 1500 Hz

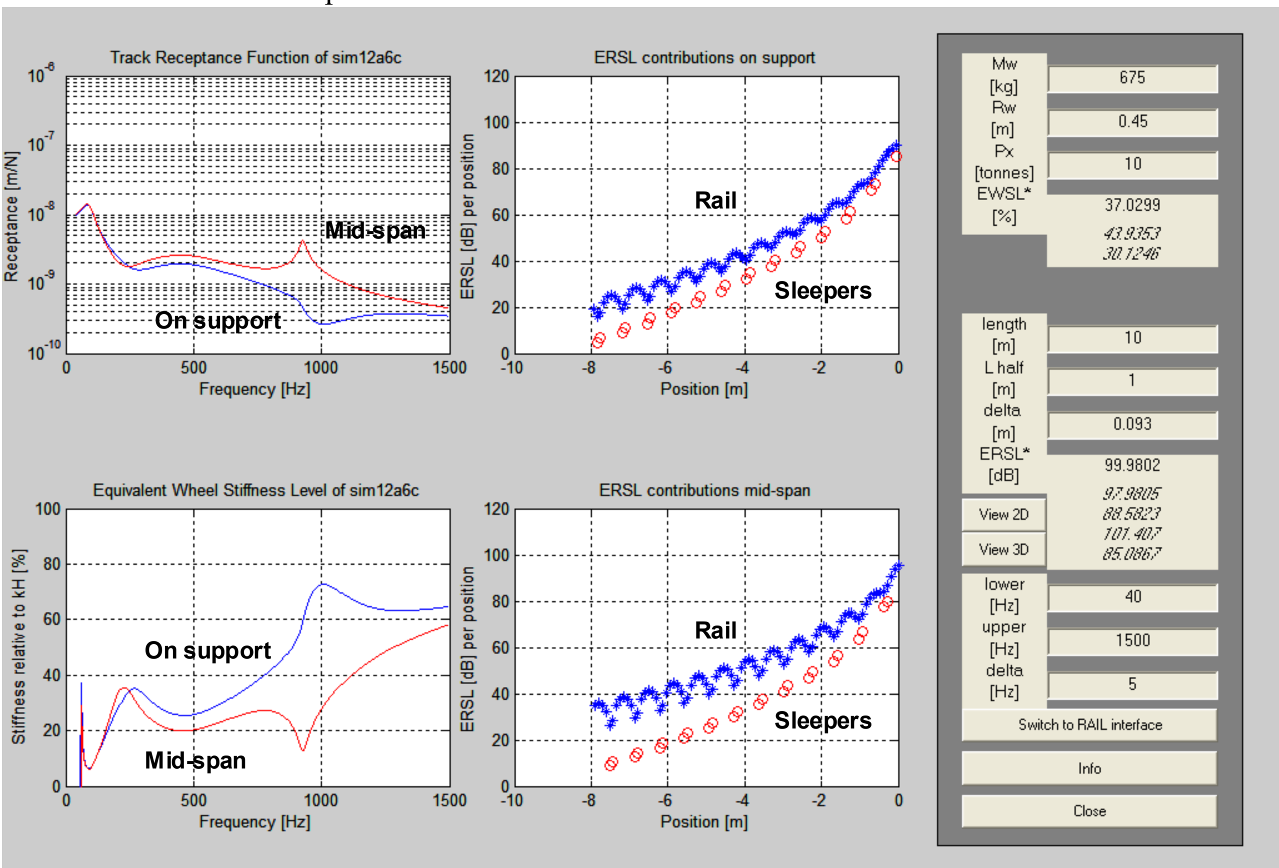

Fig. 6 Performance of alternative A6 on contact stiffness and sound radiation

Figure 3 has shown that vibrations propagate through the rail and radiate sound. The arbitrary calculation of sound radiation is performed by applying a unit loading and integrating all contributions of rail and sleepers relative to the recipient's position. The simulated vibration behaviour of a track structure displays these contributions per position as in the central part of Figure 6. The integral radiated sound levels are collected in Table 3.

5. Discussion and conclusions

The foregoing section has illustrated the capabilities of receptance functions to demonstrate the influences of track design, maintenance activities and e.g. exchange of components in terms of performance. The simulated receptance functions are suitable for design purposes. The recorded ones of existing track structures hold unique data, but require a systematic data collection method in order to qualify the collected data as representative. One should think of several recording sites and conditions.

The receptance functions of the three alternative HSL tracks are characteristic, and so are the values, which indicate their performance. For instance the average contact stiffness and its variation in the case of the ballasted HSL track is clearly lower than in case of the existing conventional ballasted track due to the application of different materials and modifying the design. However the performance is further improved in general terms for the slab track alternative and even more for the embedded rail alternative. Embedded rail systems demonstrate clear benefits for their homogeneity in contact stiffness thanks to the continuous rail support. Dynamic amplification of interaction forces is low and so are the wear rates. And this is very favourable with respect to rolling noise reduction.

Multi-layered track structures are beneficial to a step-wise reduction of vibration levels, which could not be demonstrated in this paper unfortunately. Sound radiation is dominated by rail vibrations and is effectively abated by reducing the free radiating surfaces and by applying resilient components, which absorb rail vibrations to a large extend [4]. Once more these results confirm that the design and the maintenance of track structures is a matter of balancing, also in the specific field of mid- and high-frequency vibrations.

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