Wheelset Structural Flexibility and Track Flexibility in Vehicle-Track Dynamic Interaction

by

Nizar Chaar

Doctoral Thesis

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Preface and acknowledgements

This thesis summarizes the work I have carried out during my doctoral studies at the Division of Rail Vehicles, Department of Aeronautical and Vehicle Engineering, at the Royal Institute of Technology (KTH).

This work, which is part of the SAMBA\textsuperscript{1} research programme, investigates the influence of the wheelset structural flexibility and track flexibility on the vehicle-track dynamic interaction.

In this context, I would like to express my sincere gratitude to my supervisor Prof. Mats Berg for his encouragement, help, assistance, and support throughout these years. I also would like to thank Prof. Evert Andersson for his support and valuable advice as well as my colleagues at the department for the friendly atmosphere they maintained during these last few years.

I also would like to thank the staff at the Marcus Wallenberg Laboratory for Sound and Vibration Research (MWL) for their help and assistance during the experimental modal analysis, Ingemar Persson at DEsolver AB for his advice and support with the software GENSYS, Interfleet Technology and Banverket for their help and assistance they provided with the on-track, respectively track flexibility and irregularity measurements.

Likewise, I am very grateful for the support and assistance I received from all the members of the project reference group and for the financial support from Banverket (Swedish National Rail Administration), Bombardier Transportation (Sweden), Interfleet Technology (Sweden), Tägoperatörerna (Association of Swedish Train Operators) and SL (Stockholm Transport).

Last but not least I would like to thank my parents for their support and my wife for her help and understanding especially during the last year.

Stockholm, April 2007

Nizar Chaar

\textsuperscript{1} SAMBA, in Swedish, is an abbreviation for “Samverkan Fordon-Bana” or, in English, Vehicle-Track Interaction.
Abstract

This thesis investigates the influence of wheelset structural flexibility and track flexibility on the vehicle-track dynamic interaction, mainly in terms of wheel-rail forces up to 200 Hz, using simulations and measurements.

The previous knowledge in this field is first reviewed and summarized, then two case studies are selected for investigation. The first case study involves a locomotive running on a tangent track section at a speed of 140 km/h, while the second one deals with a newly designed motor coach running at two adjacent and tangent track sections with different track components and at speeds up to 280 km/h.

For the locomotive case study, the wheelset dynamic properties are first investigated through experimental modal analysis (EMA) for a frequency range of 0-500 Hz, assuming free boundary conditions. The EMA results showed relatively low wheelset eigenfrequencies. A three-dimensional finite element (FE) model, which also includes the wheelset gear-box, is then developed and validated against the measurements for frequencies up to 200 Hz with good agreement. The FE results displayed a significant influence of the wheels’ flexibility on the wheelset’s total structural flexibility.

In order to assure proper representation of the track flexibility the vertical and lateral dynamic track properties at a sleeper are measured through a special vehicle at standstill, and measured track irregularities are used. In the numerical simulations, the wheelset structural flexibility is introduced using the calculated eigenmodes above while so-called moving track models are used to model the track flexibility. The simulated wheel-rail forces are then validated against measured ones obtained from corresponding on-track measurements. Results from the simulations highlight the importance of proper track flexibility modelling and track data and also show a significant influence of the wheelset structural flexibility on the lateral track forces.

For the motor coach case study, the wheelset dynamic properties are determined through numerical modal analysis using a rather simple FE model and a number of eigenmodes are then introduced in the simulations. The vertical and lateral track dynamic properties at selected track sections are measured using the standstill technique but rolling stiffness measurements, where the vertical track flexibility in the frequency range 5-50 Hz is measured continuously along the track, are also included. The track flexibility is introduced through moving track models. Measured track irregularity and vertical track roughness are also considered.

Basic numerical simulations, where the calculated track forces are compared to measured ones, are first performed and followed by a set of parametric studies. The results display a significant influence of the track flexibility on vertical wheel-rail forces for frequencies above 80 Hz, with higher forces for the stiffer track (but weaker rails). The effect of wheelset structural flexibility on the lateral force is also confirmed. The parametric studies highlight the importance of track flexibility modelling and show that modifications of the vertical track receptance, motivated by uncertainties in the pertinent measurements, can improve the simulated forces.

Keywords: Vehicle-track dynamic interaction, wheelset structural flexibility, experimental modal analysis, track flexibility, moving track models, track irregularity, track roughness, measurements, simulations, wheel-rail forces, parametric studies.
Outline of the thesis

This thesis consists of an introduction and four papers. The introduction reviews the current research field and summarizes the present work. The four papers are:

**Paper A**

**Paper B**

**Paper C**

**Paper D**
Chaar N and Berg M: *Dynamic wheel-rail force measurements and simulations of a high-speed train running on two tracks with different flexibility and irregularities*, submitted for publication.

Chaar carried out the experimental and numerical modal analyses of the wheelsets, specified and evaluated the track flexibility measurements, performed the on-track simulations and wrote the appended papers.

Berg supervised the work and reviewed the documentation.

The track flexibility measurements were performed by Banverket while Interfleet Technology carried out the on-track measurements.

The locomotive multibody dynamics model with rigid wheelsets was obtained from Interfleet Technology. The model of the Regina coach was developed by Bombardier Transportation and remodelled according to recent bogie modifications by Chaar.
Thesis contributions

This thesis provides the following contributions to the present research field:

- A literature review on methods used in measuring and modelling the wheelset structural flexibility and track flexibility is provided. Also, the influence of these flexibilities on the overall vehicle-track dynamic interaction is included.
- The wheelsets investigated are neither symmetrical nor axisymmetrical, a circumstance that is very rare in previous studies.
- The dynamic properties are investigated for two different wheelsets: The first wheelset pertaining to a locomotive is rather slender and heavy due to its large wheel diameter and heavy gear-box. In contrast, the second wheelset pertaining to a motor coach has a smaller wheel diameter and a lighter gear-box.
- A detailed finite element model of the locomotive wheelset, including traction gear and axle-boxes, is developed and validated against measurements for the frequency range of 0-200 Hz.
- Vertical and lateral dynamic track properties are measured for given preloads at different track sections having different track components and at a frequency range of 0-200 Hz. This enables a proper assessment of the influence of the track flexibility and track components on the wheel-rail forces.
- The vertical track properties are also investigated using the rolling stiffness technique in the frequency range of 5-50 Hz.
- A set of moving track models are developed and used in the vehicle-track dynamic simulations. The track model data are chosen so as to get good agreement with the measured track receptances above.
- The sleeper passing effect is represented in the simulations by varying the track parameters along the track.
- Measured track irregularities are considered in the simulations over a wide wavelength interval, also including track roughness.
- The influence of the wheelset structural flexibility and track flexibility on wheel-rail forces is investigated in the frequency range of 0-200 Hz for two case studies. This range stands in contrast to traditional vehicle-track interaction analysis, which typically cover frequencies up to 20 Hz only, and studies detailing the track behaviour with the main interest on frequencies in the range of 500-1500 Hz.
- The calculated track forces are validated against forces from on-track measurements enhancing the credibility of the present work.
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Paper A to D
1 Introduction

1.1 Vehicle-track dynamic interaction

The last decades have witnessed a substantial development of high-speed trains such as TGV in France, ICE in Germany, Pendolino in Italy, Shinkansen in Japan and X2000 in Sweden. Moreover, current research activities are focusing on new methods to increase train speed and capacity and, in particular, on improving the vehicle-track dynamic interaction.

Vehicle-track interaction is a broad and multi-disciplinary subject that deals with ride comfort and safety, vehicle stability, wheel-rail forces, wheel-rail corrugation, wheel out-of-roundness, noise propagation, etc., and is influenced by a variety of factors. While numerical simulations are considered a fast and inexpensive tool for investigating the main sources and causes of the vehicle-track dynamic interaction, on-track measurements are still seen as the most reliable approach for approval of vehicles under various operational conditions. These measurements are regulated through specific standards, e.g. UIC 518 [13] and EN 14363 [5], which in turn present some unresolved topics requiring continuous and further review. An example of such an unsettled issue is the 20 Hz cut-off frequency of the low-pass filtering which may be appropriate for examining the ride safety and comfort, but might be too forgiving for wheel-rail forces and pertinent damage mechanisms like fatigue.

In this context, the wheelset structural flexibility and track flexibility are two important factors known to influence the vehicle-track interaction forces at frequencies above 20 Hz. The following sections present a brief description of a conventional railway wheelset and track, provide an overview over the wheelset structural flexibility, track flexibility and track irregularity, and give an outline of the current study.

1.2 Wheelset structural flexibility

A conventional wheelset consists of two conical wheels rigidly connected to a common wheelset axle, axle-boxes mounted on the two axle ends through roller bearings, and often axle-mounted or wheel-mounted brake discs. In case of a powered wheelset, an axle gear wheel is fixed to the wheelset and a gear-box is mounted on the wheelset axle through a roller bearing.

The wheelset forms an important entity of a railway vehicle. It helps carrying the vehicle through the primary suspension, supports the vehicle at rolling, guides the vehicle, transfers the longitudinal forces at braking and in case of a powered wheelset, at traction. A general review of the wheelset behaviour and functions can be found in [3], [7], [10], [12] and [14]. Since wheelsets are more prone to show fatigue and stress related problems, several guidelines were specifically developed to design the wheelsets and their components. Some of these codes are reviewed in [30].

According to Popp et al. [66], the wheelset is by far the most interesting component of the vehicle in the frequency range of 50-500 Hz as a result of its running behaviour and elastic deformation. Like all mechanical structures, wheelsets have structural flexibility and natural frequencies. The wheelset is deformed due to its dead weight and when
subjected to external forces via its axle-boxes, brake discs and axle gear wheel as well as through the rails (track). The dynamic contributions to the wheel-rail forces mainly originate from wheel and track irregularities. In some designs a local wheelset flexibility is introduced as for resilient wheels or independently rotating wheels, see also [30]. This kind of local flexibility is not considered in this study. An example of wheelset structural flexibility, showing the wheelset’s first bending mode at free boundary conditions, is illustrated in Figure 1-1.

![Example of wheelset structural flexibility; first bending mode](image)

Several studies have investigated the influence of the wheelset structural flexibility on vehicle-track dynamic interaction. In short, the wheelset elastic deformation can affect the vehicle lateral stability, cause fluctuations in the wheel-rail forces, contribute to the development of wheel-rail corrugation and wheel out-of roundness, and lead to noise propagation in tight curves. Yet researchers, such as Popp et al. [66], agreed that the knowledge regarding the influence of the wheelset structural flexibility in the frequency range 50-500 Hz is rather poor. See also Chapter 2.

### 1.3 Track flexibility

Railway tracks play also an important role in supporting and guiding the rolling vehicles in a safe and economic manner. Tracks can be divided into two types: ballasted and non-ballasted (slab tracks). The track superstructure components can also differ significantly with regards to their geometrical, material and physical properties, see also [3].

A typical ballasted track, shown here in Figure 1-2, is composed of two steel rails mounted on concrete or wooden sleepers through fasteners. Plastic or rubber rail pads are usually located between the sleepers and the rails. The sleepers are embedded in a ballast layer which in turn is resting on subballast and subgrade.
Likewise, the track dynamic properties are of great interest since the vertical and lateral track flexibilities are known to influence the wheel-rail forces and induce oscillations of the wheels and rails [3]. In this context, the types of track components play an important role for the overall track dynamic properties, [96]. For instance, soft rail pads isolate the high frequency vibrations and their transmission down to sleeper and ballast, while stiff rail pads give more direct transmission of axle load, including the high-frequency vibration, [91] and [92]. Therefore, measurement of track flexibility is quite important and useful since it can quantify weak spots of the tracks, verify newly built tracks, identify places with excessive sources of vibrations, etc., [88].

Track flexibility is generally defined as the ratio between the track displacement and the pertinent load and can be assessed through rolling or standstill measurements. These techniques are illustrated and reviewed by Berggren [87], and briefly described in Chapter 3 of this thesis. For the standstill technique, one way to investigate the dynamic properties of a railway track is to first preload the track vertically and then load the track dynamically through its rails with a sinusoidal force. At frequencies up to about 200 Hz, this can be done by using hydraulic cylinders. The results are often presented in the form of track receptance, or the complex-valued ratio of dynamic track displacement over the dynamic force. Due to non-linear track properties such as load-dependent behaviour of rail pads and ballast, the receptance is not only dependent on the excitation frequency, but also on the static preload and dynamic amplitude. Figure 1-3 exemplifies the vertical and lateral receptance magnitudes measured for two static preloads, showing a higher receptance (lower stiffness) for the lowest preload. The lateral track receptance is higher (lower stiffness) than the vertical one, and has a clear resonance at 80 Hz.

In addition, the track properties can be dependent on the season, a frozen track being stiffer than a non-frozen one, [3].

Due to the varying subgrade conditions and ballast compaction, the track dynamic properties differ along the track, even between two adjacent sleepers. A special variation in track flexibility along the track arises through the discrete positions of the sleepers. This relatively small additional rail deflection between any two sleepers can give rise to large wheel-rail force fluctuations. This phenomenon is referred to as sleeper passing effect and the corresponding frequency is termed sleeper passing frequency, see also [3] and [103]. Variations in track dynamic properties also arise at switches and turnouts [91], and at tunnels and bridges due to a stiffer foundation.
Introduction

Figure 1-3 Example of measured vertical and lateral track receptances as magnitude, at 50 and 90 kN static preload per rail and at the same dynamic amplitude “50%”, [89].

Once the track is loaded by the vehicle, system resonance frequencies occur. An example of such a resonance is the so-called P2 resonance where the wheelset vibrates vertically on the track, see also [41]. The vehicle is subjected to different sources of excitation, one being the variations of track flexibility along the track described above. Track irregularities and roughness present another major excitation source, see below.

1.4 Track irregularity and roughness

Track misalignments often constitute the main source of excitation for vehicle-track systems. According to [91] and [3], four different types of geometric errors, or irregularities can be identified. These errors are the lateral, vertical, gauge and cant misalignments, see Figure 1-4. Long wavelength irregularities cause nuisance and discomfort whereas short wavelength irregularities, often called roughness, result into high levels of noise and vibration.

Figure 1-4 Vertical, lateral, cant and gauge irregularities, [3].
Measurement of track irregularities is an important indicator for condition assessment and monitoring of railway tracks, [87]. The track irregularities are usually measured via special measurement vehicles [3], and relevant standards are developed to regulate these measurements and set the required limit values for different wavelength intervals. The suggested interval in the code UIC 518 [13] is $\lambda = 3-25$ m, which eliminates the high-frequent track forces and hence may be forgiving with regards to wheel and rail damage. For instance, for a speed of $v = 100$ km/h (27.8 m/s) the frequency range is $f = 1.1-9.3$ Hz, whereas for $v = 275$ km/h (76.4 m/s) the frequency range is $f = 3.1-25.5$ Hz, see Figure 1-5.

![Figure 1-5](image)

*Figure 1-5  Excitation frequency as a function of track irregularity wavelength for three different speeds ($f = \nu/\lambda$). The UIC 518 wavelength range 3-25 m is indicated.*

1.5 This thesis

This thesis investigates the influence of the wheelset structural flexibility and track flexibility on wheel-rail forces in the frequency range of 0-200 Hz using simulations and measurements.

Chapter 2 reviews wheelset structural flexibility measurements and models and highlights the major effects of this flexibility on the vehicle-track interaction.

Chapter 3 briefly surveys track flexibility measurements methods and results, illustrates different types of track flexibility models, and examines the influence of the track flexibility on the vehicle-track dynamic behaviour.

Chapter 4 introduces the author’s work and gives a summary of the four appended papers.

Chapter 5 presents the conclusions and suggestions on future work.

Finally, a reference list is included.
Introduction
2 Wheelset structural flexibility

This chapter briefly reviews measurement methods and results of wheelset structural flexibility, common wheelset structural flexibility models and the major effects of this flexibility on the vehicle-track dynamic interaction. These topics were also examined in detail in [30].

2.1 Measurements of wheelset structural flexibility

Experimental modal analysis: Free boundary conditions

Experimental modal analysis (EMA) is generally used in assessing the wheelset modal parameters (eigenfrequencies, eigenmodes and relative damping) through measured frequency response functions (FRF) and assuming a linear, time-invariant system, [4] and [8]. During the experiments, “Free” or “Supported” boundary conditions can be implemented. Free boundary conditions can be achieved by suspending the wheelset freely through soft springs. This section reviews some cases where the wheelset modal parameters were extracted through EMA and lists the major findings.

EMA involving several types of non-powered wheelsets was carried out by Tassilly and Vincent, [77]. During the measurements the axle was connected to the bogie and the bogie was held on specific supports so that the excited wheelset did not contact the rails. The wheelset was excited by an electro-magnetic shaker using stepped-sine ranging for 20-400 Hz. Several driving points were selected: two points in the vertical direction to study the influence of the contact patch position, one point in the transverse direction and one point in the longitudinal direction, Figure 2-1. The authors confirmed a strong coupling between vertical and transverse directions at the contact point. On the other hand, the coupling between the longitudinal direction and the two other directions was very weak. The resulting eigenfrequencies and eigenmodes are shown in Figure 2-2 and summarized in Table 2-1.

![Image](image.png)

Figure 2-1 Vertical and lateral driving points of a non-powered wheelset investigated in [77].
Wheelset structural flexibility

<table>
<thead>
<tr>
<th>Mode shape</th>
<th>First torsion</th>
<th>First bending</th>
<th>Second bending</th>
<th>First umbrella</th>
<th>Second umbrella</th>
<th>Second torsion</th>
<th>Wheel deformation with 2 nodal diam.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Frequency range [Hz]</td>
<td>50-90</td>
<td>50-90</td>
<td>100-160</td>
<td>250-400</td>
<td>300-500</td>
<td>-----</td>
<td>300-600</td>
</tr>
</tbody>
</table>

**Figure 2-2**  **Natural frequencies and mode shapes of non-powered wheelsets, [77].**

The radial, lateral and longitudinal receptances of several wheelsets with axle-mounted gear wheel, were measured by Grassie et al. in [40] using hammer excitations. The radial response was determined by mounting an accelerometer on the wheel tread and hitting the tread with the instrumented hammer at a location opposite to the accelerometer. Lateral excitation was performed by placing the accelerometer on the face of the wheel rim and striking the rim laterally. Then, the accelerometer was rotated by 90 degrees and mounted on the tread to measure the longitudinal response. In the frequency range of 0-500 Hz, two torsional modes were detected. In the first mode the two wheels rotate out-of-phase, while in the second torsional mode the wheels rotate in phase but out-of-phase with the axle gear wheel, see Figure 2-3. Results obtained from these measurements, are also listed in Table 2-1.

**Figure 2-3**  **First and second torsion modes of a powered wheelset, [40].**

The modal parameters of a non-powered wheelset with Swedish SJ57H wheels were extracted in [51]. In order to imitate free boundary conditions in the measurements, the wheelset was suspended through its roller bearings by very flexible “polymeric bungees”. The wheelset was then excited by an electro-magnetic shaker using stepped-sine excitation. The measured damping ratios for the first torsion and first and second bending modes (Table 2-1) are less than 2%. 

8
In addition, EMA of a wheelset assuming free boundary conditions were performed in the following studies:

- In [59], where the case studied involved a wheelset pertaining to an ICE-1 coach with four axle-mounted brake discs, see corresponding results in Table 2-1.
- In [80] where the modal parameters of a non-powered wheelset with two axle-mounted brake discs pertaining to a Swedish X2000 train were extracted.

Table 2-1 lists some mode shapes and pertinent eigenfrequencies obtained from EMA assuming free boundary conditions. Frequencies as low as 50 Hz can be identified.

**Table 2-1 Examples of measured wheelset eigenmodes and eigenfrequencies, [Hz].**

<table>
<thead>
<tr>
<th>Ref.</th>
<th>Wheelset type</th>
<th>First torsion</th>
<th>First bending</th>
<th>Second bending</th>
<th>First umbrella</th>
<th>Second umbrella</th>
<th>Second torsion</th>
<th>2 nodal diam.</th>
</tr>
</thead>
<tbody>
<tr>
<td>[35]</td>
<td>ETR500, no brake discs</td>
<td>----</td>
<td>79</td>
<td>130</td>
<td>----</td>
<td>----</td>
<td>----</td>
<td>----</td>
</tr>
<tr>
<td>[40]</td>
<td>Powered transit wheelsets</td>
<td>50-70</td>
<td>60-80</td>
<td>----</td>
<td>----</td>
<td>290-500</td>
<td>----</td>
<td>----</td>
</tr>
<tr>
<td>[51]</td>
<td>Non-powered SJ57H wheelset</td>
<td>----</td>
<td>84</td>
<td>152.5</td>
<td>249</td>
<td>397</td>
<td>----</td>
<td>----</td>
</tr>
<tr>
<td>[59]</td>
<td>ICE, 4 axle-mounted discs</td>
<td>96</td>
<td>82</td>
<td>162</td>
<td>----</td>
<td>----</td>
<td>289</td>
<td>----</td>
</tr>
<tr>
<td>[77]</td>
<td>Non-powered wheelsets</td>
<td>50-90</td>
<td>50-90</td>
<td>100-160</td>
<td>250-400</td>
<td>300-500</td>
<td>----</td>
<td>300-600</td>
</tr>
</tbody>
</table>

In some situations, the natural frequencies and mode shapes of a single railway wheel were measured. Examples of such studies are Cervello et al. [29] and Kämmerling [48]. In [48] the wheel was placed horizontally with its outer side resting on two car tyres, see Figure 2-4. This elastic support gives, according to the author, only a small influence on the intended free wheel vibrations.

![Figure 2-4 Experimental modal analysis of a free wheel, [48].](image)
**Experimental modal analysis: Supported wheelset**

An example of EMA performed on a supported wheelset can be found in [49]. The measurements, performed at the Royal Institute of Technology, involved a cargo wheelset placed on rails and having a wheel diameter of 1 m.

The natural frequencies of a wheelset mounted on stiff and soft tracks were examined by Diana et al., [36]. Results obtained from these measurements are listed in Table 2-2. The authors noted that only the P2 resonance frequency was mostly affected by the change of track stiffness.

**Table 2-2 Measured wheelset natural frequencies on stiff and soft tracks, [36].**

<table>
<thead>
<tr>
<th>Mode of vibration</th>
<th>Stiff track</th>
<th>Soft track</th>
</tr>
</thead>
<tbody>
<tr>
<td>First bending</td>
<td>83</td>
<td>55</td>
</tr>
<tr>
<td>Second bending</td>
<td>140</td>
<td>148</td>
</tr>
<tr>
<td>Third bending</td>
<td>252</td>
<td>----</td>
</tr>
<tr>
<td>First wheel mode</td>
<td>297</td>
<td>291</td>
</tr>
</tbody>
</table>

Natural wheel frequencies were investigated assuming supported conditions in Edgren [37] and Schneider and Popp [69]. In [37], the wheelset was placed on the rails and the wheel was excited axially using an impact hammer equipped with force transducer as shown from the experimental set-up below, Figure 2-5.

**Figure 2-5 Experimental set-up for EMA of wheel supported on rail, [37].**
On-track measurements: instrumented wheelsets and codes/regulations

On-track measurements can describe the dynamic behaviour of the vehicle on the track. These measurements are carried out using instrumented wheelsets and are often linked to relevant codes such as UIC 518 [13], see also Chapter 1. Measured data such as wheel-rail forces and axle-box accelerations are gathered through strain gauges usually placed on the wheel web and accelerometers mounted on the axle-boxes. For proper results and placement of the strain gauges, see Figure 2-6, the instrumented wheelsets have S-shaped wheels and possible wheel-mounted brake discs are removed. More information regarding instrumented wheelsets can be found in [3], [19] and [80].

![Figure 2-6  Principal design of an instrumented wheelset for force measurements, [19].](image)

The code UIC 518 [13] has set the rules that must be met when conducting dynamic behaviour tests in connection with safety, track fatigue and running behaviour. The specifications have set the cut-off frequency for the low-pass filtering of the track forces. As mentioned in Chapter 1, the cut-off frequency suggested is 20 Hz which means that the load variations due to the sleeper passing effect and the main wheelset vibrations will be filtered out, even though such variations can be very high.

Figure 2-7 illustrates an example of measured vertical wheel-rail forces of an X2000 power unit on fairly rough track at high speed curving. The figure depicts two filtering procedures and clearly shows major force differences for the two cut-off frequencies of 30 and 90 Hz.
Wheelset structural flexibility

Figure 2-7  Measured vertical wheel-rail force of an X2000 power unit at high-speed curve negotiation, [19].

2.2 Modelling of wheelset structural flexibility

Most researchers agree that the wheelset structural flexibility must be considered in the mid- and high-frequency ranges (i.e. 50-500 Hz and 500 Hz to 20 kHz [66]), since this flexibility can influence the vehicle-track dynamic behaviour. A rigid wheelset model can be appropriate for the low-frequency range (below 50 Hz), where the ride comfort and stability are of main interests, [35], [66], [68] and [71]. The introduction of the wheelset structural flexibility allows also for the determination of different quantities such as strains and stresses [18], and in some cases it is used to avoid numerical problems in the simulations due to a rigid connection between the axle and the track, [20].

According to [66], the methods used in modelling the wheelset structural flexibility are: continuous, finite element (FE) and lumped models. These methods are detailed and reviewed in [27]. The wheelset flexibility is then usually considered in the numerical simulations through eigenmodes derived from the respective models or from other
reduced wheelset models, see also [1]. The choice of the wheelset flexibility model and number of degrees of freedom (dof) depends on the frequency range of interest and on the number of wheelset mode shapes in that range.

**Continuous models**

In these models, the wheelset’s geometry is represented by means of different continua possibly with additional discrete elements. For an appropriate representation of the wheelset flexibility in the mid- and high-frequency ranges, continuous models should include among others, a realistic distribution of mass and inertia along the wheelset, the three-dimensional deformation of the axle including bending and torsion, the out-of-plane wheel deformation, and gyroscopic effects [66].

A two-dimensional continuous wheelset model was developed by Szolc, [71]. The model accounts for the wheelset bending, torsional and lateral vibrations in the frequency range of 30-300 Hz. The model is shown in Figure 2-8 and consists of a wheelset axle represented by an axially rigid and torsionally deformable beam, whereas the wheels and the brake discs are represented by rigid rings attached to the axle through a massless, elastically isotropic membrane.

![Figure 2-8](image)

**Figure 2-8 Two-dimensional continuous wheelset model developed by Szolc, [71].**

A two-dimensional continuous wheelset model was also developed by Meywerk in [58]. In this model the wheelset axle and the wheel rim are modelled as one-dimensional Timoshenko beams while the wheel disc is assumed to be clamped at the hub and its deflections obey Kirchhoff’s plate theory, Figure 2-9.
Wheelset structural flexibility

Three-dimensional finite element models

Finite element models allow for more accurate representation of the wheelset geometry and flexibility and are now common with the development of high-speed computers. A 3-D FE wheelset model consisting of two wheels and two bearings connected to a wheel axle was developed by Fingberg [38]. Due to symmetry only half of the wheelset was modelled. The axle is discretised using Timoshenko beam elements taking into account the bending, torsion and longitudinal deformations. The wheels are meshed with special annular shell elements. The calculated mode shapes and corresponding frequencies are shown in Figure 2-10.

![Figure 2-10 3-D FE model developed by Fingberg, [38]. The wheelset axle is modelled using beam elements and the wheels are represented by special annular shell elements.](image)

The gyroscopic properties of an ICE wheelset with four wheel-mounted brake discs were examined in [55] using the 3-D FE model shown in Figure 2-11. In this model, the wheels, brake discs and the wheelset axle are represented by solid elements. Model reduction based on Guyan’s method [9], was then performed.
Wheelset Structural Flexibility and Track Flexibility in Vehicle-Track Dynamic Interaction

Figure 2-11  3-D FE model of an ICE wheelset with four axle-mounted brake discs, the wheels and the axle are meshed with solid elements [55].

A FE wheelset model with two axle-mounted brake discs was developed by Kaiser and Popp [46] and [47], Figure 2-12. The eigenmodes and eigenfrequencies of this model are shown in Figure 2-12.

Figure 2-12  Mode shapes and natural frequencies of the FE model developed in [46].

Finite element modelling of a non-powered wheelset was also performed in [51]. In this model, the wheels were discretised using quadratic tetrahedron elements and the wheelset axle was modelled using Timoshenko beam elements, Figure 2-13. The axle-boxes were represented by rigid masses lumped to the wheelset axle.
Finally, 3-D FE wheelset models were formulated in [16] to [18], [21], [43], [62] and [80].

**Two-dimensional finite element models**

In the following section, two examples of two-dimensional FE wheelset models are illustrated. The first model was developed by Meinke [57] and was used to investigate the influence of the dynamic and static imbalances of the wheelset on the development of wheel out-of-roundness (OOR) in high-speed trains. The flexibility of the wheelset axle was introduced through elastic beam elements whereas the two wheels and the four axle-mounted brake discs were modelled as rigid entities, Figure 2-14.

The second model, shown in Figure 2-15, was proposed by Diana et al. [35]. The model is based on simple discretization in which all parts of the wheelset are represented by beam elements, each node having three degrees of freedom (dofs) being vertical and lateral displacements as well as rolling rotation. The authors mentioned that such a model can give satisfactory representation of the axle deformation but enable only the flexural deformability of the wheels and can reproduce the wheelset behaviour up to 250 Hz.
Figure 2-15  2-D FE model developed in [35]. The wheelset is modelled through beam elements, each node having three degrees of freedom.

Lumped models

The wheelset structural flexibility can also be modelled using lumped or multibody models. The wheelset is then split into several parts interconnected through springs and dampers.

In [59] a lumped wheelset model was developed by Morys to investigate the enlargement of wheel OOR in high-speed trains. The studied wheelset pertains to an ICE-1 coach and consists of eight rigid bodies: four axle-mounted brake discs, two wheels and two axle ends interconnected by three dimensional springs and damper elements in order to introduce the bending and torsional flexibility, Figure 2-16. Numerical modal analysis was then performed using this model, cf. Table 2-3.

Figure 2-16  Lumped model of a wheelset with four brake discs, [59].

Another lumped wheelset model was generated by Zobory et al. [83], while investigating the self-excited vibrations in disc-braked running gear. The model consisted of a plane wheelset (axle and two wheels), three axle-mounted brake discs and two axle-boxes and was used to calculate the bending and torsional wheelset dynamics, see Figure 2-17.
Wheelset structural flexibility

A simple lumped model, including the first wheelset torsional and bending modes, was generated in [20] while investigating the lateral track forces at high-speed curving using radial self-steering bogies, see Figure 2-18.

Lumped wheelset models were also developed in [25], [34], [36], [42], [53], [54], and [70].

**Main findings and remarks**

*Calculated mode shapes and natural frequencies of a free wheelset:* The calculated mode shapes and natural frequencies recorded from several case studies are listed in Table 2-3. The lowest torsional and bending modes recorded occurred at around 67 and 84 Hz respectively.
Table 2-3 Calculated mode shapes and natural frequencies [Hz] for different types of wheelsets.

<table>
<thead>
<tr>
<th>Ref.</th>
<th>Wheelset type</th>
<th>Model</th>
<th>First torsion</th>
<th>First bending</th>
<th>Second bending</th>
<th>First umbrella</th>
<th>Second umbrella</th>
</tr>
</thead>
<tbody>
<tr>
<td>[38]</td>
<td>Wheelset + axle-boxes</td>
<td>FE model</td>
<td>67</td>
<td>89</td>
<td>141</td>
<td>235</td>
<td>308</td>
</tr>
<tr>
<td>[51]</td>
<td>Wheelset with SJ57H wheels</td>
<td>FE model</td>
<td>----</td>
<td>83.9</td>
<td>152.5</td>
<td>250.5</td>
<td>396.9</td>
</tr>
<tr>
<td>[55]</td>
<td>ICE wheelset, 4 axle-mounted brake discs</td>
<td>FE model</td>
<td>82.5</td>
<td>84.6</td>
<td>131.8</td>
<td>234.8</td>
<td>296.1</td>
</tr>
<tr>
<td>[59]</td>
<td>ICE wheelset, 4 axle-mounted brake discs</td>
<td>Lumped, rigid wheels</td>
<td>95.9</td>
<td>82.0</td>
<td>182.0</td>
<td>-----</td>
<td>-----</td>
</tr>
</tbody>
</table>

Modelling of wheels and brake discs: Up to frequencies corresponding to the first wheelset bending mode, the wheels and the brake discs can be modelled as rigid entities. However, wheels start to show considerable deformations at frequencies close to the second bending mode, [55]. Morys [59], stated that for frequencies up to 200 Hz, the wheels can be assumed as stiff and rigidly coupled to the axle. Szolc [75] noted that in the frequency range of 0-100 Hz, the planes of the wheels and brake discs are always perpendicular to the axis of the deformed wheelset which imply that the wheels can be modelled as rigid bodies in that frequency range.

Gyroscopic effects: The influence of gyroscopic effects was examined in [55]. For a speed of 300 km/h, the eigenfrequencies of the bending modes split up into an increasing branch, called forward whirl, and a decreasing branch, called backward whirl. The torsion mode is not influenced by the rotary frequency. Similar findings were reported by Szolc, [75].

Static and dynamic imbalances: According to [57], imbalances arise from small inaccuracies in the manufacturing process and cause a coupling between the wheelset bending and torsional vibrations leading to creepage and wheel OOR due to continuous wear. In [55] and [56] excitations caused by dynamic and static imbalances were studied. The static imbalances were represented by mass points whereas the dynamic imbalances were introduced via massless rigid bodies with products of inertia.

Simplified wheelset models: A fast algorithm for modelling and investigating the wheelset flexibility was developed by Hempelmann, [43]. The wheelset axle was modelled as beam elements including the bending, shear and longitudinal deformation while the wheels were modelled as shear plate discs, see Figure 2-19. This model comprised only 130 dofs and showed quite good accuracy in the results according to the author.
2.3 Influence of wheelset structural flexibility

The influence of the wheelset structural flexibility is generally examined through numerical simulations. This section surveys the influence of wheelset flexibility on different phenomena, such as the vehicle lateral stability, track forces, wheel-out-of-roundness, wheel/rail corrugation, and wear and noise propagation.

On lateral stability

The influence of the wheelset’s bending, torsional and umbrella modes on the lateral vehicle stability was examined by Kaiser and Popp, [46], [47] by comparing the running behaviour of rigid respectively flexible wheelsets. The authors mentioned that the strongest influence on the running behaviour is caused by the wheelset flexural displacements. The out-of-plane bending causes a deflection of the wheelset in the vertical and lateral directions and leads to additional creepage. The influence of wheelset flexibility on the lateral stability was also studied in [15].

On track forces and axle-box accelerations

In [16] the influence of wheelset flexibility on the longitudinal and normal track forces was investigated through numerical simulations involving a wheelset with rigid or flexible wheelset axle. The wheelset first torsional mode was excited by means of a sinusoidal corrugation on both rails with a corresponding wavelength of 0.66 m, the corrugation being out-of-phase by 180 degrees. Hence, for a running speed of 160 km/h (44.4 m/s) the excitation frequency around 67 Hz corresponds to the wheelset fundamental torsion mode. The authors stated that a flexible wheelset axle results in lower longitudinal force as compared to the rigid case, see Figure 2-20a, and reported a rather small influence of the wheelset flexibility on the normal contact force, cf. Figure 2-20b and [17]. In addition, Andersson et al. [17] noted that the influence of the wheelset flexibility on the vehicle lateral dynamics is believed to be stronger as a result of out-of-plane bending of the wheels.
The influence of the complexity of track and wheelset models on numerical results was examined in [26]. For that purpose, two track models were compared in the simulations: a so-called complete model, referring to a FE model with beam elements, and a rigid track model. Likewise, a FE wheelset model with 12 dofs accounting for the axle bending modes was compared to a rigid wheelset model with 5 dofs. Also, two types of wheel-rail contact models were evaluated: A single contact and a multi-Hertzian contact model. The vehicle, from a Pendolino train, was simulated on a track at a speed of 170 km/h. The results are summarised in Table 2-4. It can be shown that the values of the lateral axle-box accelerations and the limit cycle frequency change considerably as the complexity of the models decreases, leading to an over-estimation in the vehicle stability threshold. The effect of wheelset flexibility on the lateral axle-box accelerations and the limit cycle frequency can be seen by comparing the last two rows of Table 2-4. In fact the lateral acceleration in the flexible wheelset model is almost 1.7 times higher than that reported for the rigid wheelset model.

Table 2-4 Lateral axle-box acceleration and limit cycle frequency as functions of wheel-rail contact model, track model and wheelset model, [26].
The influence of the wheelset structural flexibility on wheel-rail forces and axle-box accelerations was also examined in [35], [62] and [77]. Additional studies concerning the effects of the wheelset deformation on vehicle-track dynamics in the mid-frequency range were also carried out by Szolc and illustrated in [71] to [76].

**On wheel out-of-roundness (OOR)**

Wheel OOR or polygonalization is an imperfection on the wheel tread that can lead to detrimental influence on both tracks and wheels, result in high impact loads and cause an increase in rolling noise, [44]. The classification of wheel imperfections and their influence on the vehicle-track dynamics were reviewed in [44]. The authors also surveyed the possible causes of the formation and development of wheel OOR which include, among other, the influence of wheelset structural flexibility.

A simple mechanism of wheel polygonalization by wear was examined by Brommundt, [25]. The author quoted that the wheelset axle torsional vibrations combined with the wheelset lateral dynamic vibrations lead to creep generation and to the formation and development of corragation wheel wear.

The enlargement of wheelset OOR was also investigated by Morys in [59]. The author stated that the normal force accelerates the wheel vertically causing a bending oscillation of the wheelset which in turn leads to lateral slip and material excavation. The material excavation due to longitudinal slip and spin plays a minor role in the OOR growth.

The growth of wheel OOR was also examined by Meywerk, [58]. The author defined two mechanisms behind the formation of wheel OOR. The author stated that the wheelset first and second bending modes play an important role in the formation and development of wheel OOR due to the lateral slip caused by the excitation of these two modes.

Finally, similar findings were reported in [82].

**On wheel/rail corrugation and wear**

The causes, characteristics and treatments of the rail corrugation were investigated by Grassie and Kalousek in [41]. Rail corrugation is a phenomenon whereby more or less periodic undulations appear on the running rail surface and gives rise to large dynamic forces between wheel and rail, noise propagation and ballast deterioration, [41]. The authors also stated that the mechanism of corrugation formation consists of two steps: First a wavelength fixation mechanism followed by a damage mechanism. Based on the wavelength fixation mechanism, they defined six types of rail corrugations. Booted sleepers and rutting are two types (causes) of rail corrugations where the wavelength fixation mechanism corresponds to the wheelset flexural and torsional modes respectively. The influence of the wheelset second torsion mode on corrugation formation was also reported by Grassie et al. [39] and by [40] while investigating the formation of corrugation on a North American transit system.

In [53] and [54], Matsumoto et al. studied the mechanism of rail corrugation formation on curved track. The authors indicated that the axle torsional stiffness has an influence on the rail head wear index. The larger the axle torsional stiffness, the smaller the wear index is. However, the axle torsional stiffness does not influence the wavelength of corrugation which is mainly influenced by the vertical system dynamics rather than by the axle torsional oscillation.
The first torsional mode of a powered wheelset was introduced in [70] while studying the formation and development of rail wear. The author mentioned that the lateral vehicle motion coupled with the wheelset first torsional mode initiate the formation of irregular, abrasive wear due to stick-slip phenomenon.

A linear wheel-track model aiming at investigating the rail corrugation at narrow curves for ballasted and slab tracks was developed by Tassilly and Vincent, [77]. According to the authors the corrugation formation on the leading axle on a ballasted track is significant in the frequency range of 60-80 Hz, at proximity of the wheelset first bending mode, where a slight fluctuation in the vertical force can cause large transverse sliding. For the trailing axle, a longitudinal wear function is highest at around 55 Hz, which can be related to the wheelset first, highly undamped, torsion mode. This phenomenon is more likely to occur on narrow curves rather than on a straight track.

Similar findings and conclusions were drawn by Meinders [55] and Hempelmann et al. [43]. The latter indicated that the primary influence of the wheelset flexibility on formation of corrugation is caused by the coupling of its vertical and lateral dynamics through axle bending.

On noise propagation

Thompson [78] indicated that the major source of noise from railways at high speeds originates from the wheel-rail region. The wheel-rail rolling noise is attributed to the structural vibrations of the wheels and the rails excited from the contact patch area by the wheel and rail surface undulations.

Thompson and Jones reviewed in [79] the theoretical models that have been developed to predict the wheel/rail noise generation. The authors divided the wheel-rail noise into three categories: Rolling noise which occurs on straight track and is caused by vertical dynamics induced by undulations (roughness) on the wheel and rail surfaces, impact noise due to discontinuities at the wheel-rail level, and squealing noise which occurs at narrow curves and is induced by lateral excitation mechanisms. The authors quoted that noise prediction is nowadays generally based on finite element modelling of the wheels. According to [79], the wheel natural frequencies with one-nodal circle are most important in rolling noise propagation. The axial mode with zero nodal circles has a predominantly lateral motion at the wheel-rail contact point and hence it is important in curve squeal.

The propagation of squealing noise in narrow curves was investigated by Fingberg [38] and a corresponding wheel-rail model was generated. The author stated that the generation of squealing noise can be assigned mainly to the natural frequencies of the wheel whereas the sound radiation from the rail may be neglected. Likewise, [69] indicated that squealing noise in narrow curves is related to the bending vibration of the wheel discs excited by the stick-slip in the wheel-rail contact.

In [67], Remington reviewed and presented the current state of knowledge of wheel-rail rolling noise. The author stated that the sound radiation can be caused by a variety of factors such as vertical and horizontal rail vibrations, axial and radial wheel tread vibrations, etc. He further indicated that in the frequency range of 315-630 Hz all the above mentioned factors contribute to sound radiation. From 800 to 2500 Hz, the sound radiation due to vertical rail vibration dominates, whereas above 2500 Hz sound radiation originating from the radial wheel vibrations tends to dominate.
Finally, in the state-of-the-art survey presented by Vincent, [81], a series of actions were proposed in order to reduce and control the rolling noise at the source. These actions include, among others, wheel damping, i.e. installation of damping material on the wheel to control the wheel noise, resilient wheels and shielding of the wheel sides by means of damped steel plates.

Table 2-5 presents a list of wheelset structural flexibility (and track flexibility) models. Aim of the study, frequency range of interest and modelling methods are also specified.

Table 2-5 Review of structural flexibility models for wheelsets.

<table>
<thead>
<tr>
<th>Ref.</th>
<th>Aim of the study, frequency range</th>
<th>Wheelset model</th>
<th>Track model</th>
</tr>
</thead>
<tbody>
<tr>
<td>[16]</td>
<td>Vehicle-track dynamic interaction, up to 1000 Hz</td>
<td>Axle: Beam elements, incl. bending and torsion Rigid wheels</td>
<td>Beam on discrete supports</td>
</tr>
<tr>
<td>[18]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>[20]</td>
<td>Lateral track forces at high-speed curving</td>
<td>Lumped wheelset model: First torsion mode only Rigid wheels</td>
<td>Moving track model Lateral track stiffness included</td>
</tr>
<tr>
<td>[25]</td>
<td>Polygonalization of wheels by wear</td>
<td>Lumped wheelset model: First torsion mode only Rigid wheels</td>
<td>not specified</td>
</tr>
<tr>
<td>[34]</td>
<td>Interaction between track superstructures and vehicles</td>
<td>Lumped wheelset model: Torsion mode only</td>
<td>Beam on discrete supports</td>
</tr>
<tr>
<td>[35]</td>
<td>Vehicle-track dynamic interaction at high speeds</td>
<td>Two-dimensional FE model, modelled by beam elements</td>
<td>Beam on discrete supports</td>
</tr>
<tr>
<td>[38]</td>
<td>Model of wheel-rail squeal noise at curving</td>
<td>Axle: Timoshenko beam elements Wheel: Annular shell elements</td>
<td>Beam on discrete supports</td>
</tr>
<tr>
<td>[39]</td>
<td>Rail corrugation mitigation for transit traffic</td>
<td>Lumped wheelset model: Torsional modes only Rigid wheels</td>
<td>Beam on discrete supports</td>
</tr>
<tr>
<td>[40]</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>[42]</td>
<td>Improve the design of a locomotive bogie, 0-100 Hz</td>
<td>Axle: Lumped model, incl. first torsion and bending modes Rigid wheels</td>
<td>not specified</td>
</tr>
<tr>
<td>[43]</td>
<td>Modelling of wheelset-track dynamic interaction, 0-3500 Hz</td>
<td>Axle: Beam elements, incl. bending, torsion and elongation Wheel: Plate elements</td>
<td>Beam on discrete supports</td>
</tr>
<tr>
<td>Ref.</td>
<td>Aim of the study, frequency range</td>
<td>Wheelset model</td>
<td>Track model</td>
</tr>
<tr>
<td>------</td>
<td>---------------------------------</td>
<td>----------------</td>
<td>-------------</td>
</tr>
<tr>
<td>[46] [47] [28]</td>
<td>Interaction between elastic wheelset and elastic rails</td>
<td>FE model, solid elements</td>
<td>Beam on discrete supports  Moving track model, two dofs</td>
</tr>
<tr>
<td>[51]</td>
<td>Identification of a finite element wheelset model</td>
<td>Axle: Beam and solid elements  Wheels: Brick elements</td>
<td>not specified</td>
</tr>
<tr>
<td>[53] [54]</td>
<td>Rail corrugation on curved track</td>
<td>Lumped wheelset model: First torsion mode only  Rigid wheels</td>
<td>not specified</td>
</tr>
<tr>
<td>[55] [56]</td>
<td>Modelling of wheelset as a rotating elastic system and irregular wear</td>
<td>FE model  Axle: Solid elements  Wheels: Solid elements</td>
<td>not specified</td>
</tr>
<tr>
<td>[57]</td>
<td>Wheel tread OOR by static &amp; dynamic imbalances</td>
<td>Axle: Elastic beam elements  Rigid wheels &amp; brake discs</td>
<td>not specified</td>
</tr>
<tr>
<td>[58]</td>
<td>Polygonalization of railway wheels</td>
<td>Continuous model  Axle: 1-D Timoshenko beam  Wheel: 2-D Kirchoff’s plate</td>
<td>1-D beam element</td>
</tr>
<tr>
<td>[59]</td>
<td>Enlargement of wheel OOR on high-speed trains, 0-300 Hz</td>
<td>Axle: Lumped model incl. torsion and bending  Rigid wheels and brake discs</td>
<td>Beam on discrete supports</td>
</tr>
<tr>
<td>[60]</td>
<td>Prediction of short-pitch corrugation</td>
<td>Axle: Timoshenko beam, Wheel: Rotational symmetric shells</td>
<td>Beam on discrete supports</td>
</tr>
<tr>
<td>[62]</td>
<td>Short-pitch wheel-rail corrugation</td>
<td>FE model followed by Craig-Bampton reduction</td>
<td>Beam on discrete supports</td>
</tr>
<tr>
<td>[68]</td>
<td>Simulation of high-frequency vehicle-track interaction</td>
<td>Finite element model</td>
<td>Beam on discrete supports including ballast mass</td>
</tr>
<tr>
<td>[69]</td>
<td>Noise propagation due to wheel-rail contact forces</td>
<td>Wheel: FE model, ring elements based on Mindlin’s plate theory</td>
<td>not specified</td>
</tr>
<tr>
<td>[70]</td>
<td>Study of wheel-rail corrugation with flexible wheelsets</td>
<td>Axle: Lumped model incl. first torsion  Rigid wheels &amp; brake discs</td>
<td>Beam on discrete supports</td>
</tr>
<tr>
<td>[71] to [76]</td>
<td>Vehicle-track interaction in the mid-frequency range</td>
<td>Continuous-discrete wheelset axle  Rigid wheels and brake discs</td>
<td>Beam on discrete supports</td>
</tr>
<tr>
<td>[80]</td>
<td>Wheel-rail forces at high frequencies</td>
<td>FE model</td>
<td>not specified</td>
</tr>
<tr>
<td>[83]</td>
<td>Vibrations in disc-braked running gear</td>
<td>Lumped wheelset model incl. bending and torsion  Rigid wheels</td>
<td>not specified</td>
</tr>
</tbody>
</table>
Wheelset structural flexibility

Also, the structural flexibility of wheelsets was considered in the following studies:

In [22] while studying the influence of plastic deformation of corrugated rails.
In [23] while investigating the performance of wide-band dampers in wheels and their efficiency in reducing wheel/rail wear and cutting noise propagation.
In [24] while assessing the dynamic stiffness properties of bogie components, including wheelsets, on the design of high axle-load bogies.
In [44] while reviewing the mechanisms of irregular wear on wheel and rail surfaces.
In [50] while studying a wheelset eigenmodes and natural frequencies under gyroscopic effects.
In [52] while designing and developing a new freight wheelset for 30 tonne axle loads.
In [82] while examining the formation of wheel OOR due to unhomogeneous wear of the wheel treads.
In [83] while studying the self-excited vibrations in disc-braked running gears and in brake-pad suspension system.

The wheel structural flexibility was considered in the following investigations:

In [29] while designing and analysing a new method to reduce and cut the noise radiation from wheels via visco-elastic layers, and in [37], [48], [69], and [78].
3  Track flexibility

Track flexibility has for a long time been regarded as an interesting parameter for condition assessment and monitoring of the track, [87]. Tracks are seldomly laid on homogeneous subgrades. As a result, the track stiffness varies along the track and this variation is experienced by the train and the track, [90]. Furthermore, the properties of the track components, such as rails, rail pads, fastenings, etc., may also differ and contribute to the track flexibility variation and lead to fluctuations in track forces and noise propagation, [96].

This chapter briefly describes measurements methods and results of track flexibility as well as modelling methods of track flexibility. It also reviews the influence of this flexibility on vehicle-track dynamic interaction.

3.1  Measurements of track flexibility

Standstill techniques

As implied by the name, the standstill techniques measure track flexibility without any moving vehicle at one track section at a time, which makes them time consuming and expensive. As mentioned by Berggren [87], a simple way of measuring the track stiffness can be achieved by attaching a displacement transducer at a certain track section and measuring the response due to a train passage. The stiffness can be derived if the static axle load is known.

*Impact hammer* is another way of measuring track stiffness. Here the impact hammer is equipped with a force transducer while an accelerometer is attached to the rail head or at a sleeper. Measurements are taken from several hits with the hammer on the rail and the transfer function between the impulse force of the hammer and the acceleration at the rail is obtained. This method is suitable for problems associated with high frequencies, above at least 200 Hz, such as noise and vibrations, [87].

*Falling Weight Deflectometer* (FWD) is a fast method and can cover shorter distances of track according to [87].

*Track Loading Vehicle* (TLV) can apply a static, vertical preload on the track through hydraulic jacks to measure track stiffness. Usually the rail heads are loaded but there is also a possibility to preload a sleeper. A typical example of such a vehicle is Banverket’s Track Loading Vehicle illustrated in Figure 3-1, see also [86] and [87].

The TLV shown in the figure below is equipped with three hydraulics shakers, two acting vertically and one laterally and is capable of measuring the lateral and vertical track flexibilities in the frequency range of 1-200 Hz. The response is measured through accelerometers placed on the rail heads or on the sleeper.
Figure 3-1  Example of track loading vehicle capable of measuring the vertical and lateral track flexibility in the frequency range of 1-200 Hz, [89].

Rolling stiffness techniques

In rolling stiffness measurements the vertical track stiffness is usually assessed continuously while the vehicle is rolling. These methods can provide information on the vertical track stiffness over longer distances within relatively shorter time as compared to the standstill approaches.

An example of a rolling stiffness measurement vehicle is illustrated in Figure 3-2. The vehicle, developed by the Swedish National Rail Administration (Banverket), is a rebuilt two-axle freight wagon equipped with two 4000 kg vibrating masses above one of the axles and is capable of measuring the vertical track stiffness continuously over long track sections in the frequency ranges of 5-50 Hz via forces and accelerations recorded at the axle-boxes. The vertical motion of the masses is generated and monitored by means of two hydraulic cylinders. The imposed static axle load is about 180 kN. Additional information regarding this vehicle can be found in [88]. Additional examples of rolling stiffness vehicles were reviewed by Berggren in [87]. An example of a rolling stiffness vehicle is also illustrated in [85].

Figure 3-2  Example of rolling stiffness vehicle capable of measuring vertical track flexibility continuously along the track for frequency range of 5-50 Hz, [88].
Results and findings

Vertical and lateral track dynamics

According to Dahlberg [91], several well damped resonances can be found in a track structure. Sometimes, when the track is built on soft subgrade, a vertical resonance may typically appear in the frequency range of 20-40 Hz. This resonance occurs when the track and a great deal of the track substructure vibrates on a layered structure of the ground. A second track resonance usually occurs in the frequency range of 50-300 Hz where the track superstructure (rails and sleepers) vibrates vertically on the ballast bed. The rails and the sleepers provide the “mass” and the ballast provides the “spring” for this resonance vibration. The ballast also provides a large amount of damping, so this resonance is very well damped. A third resonance can often be found in the frequency range of 200-600 Hz where the rail is bouncing on the rail pads. The rail pad acts as a spring inserted between the rail and the sleeper. Also here, the ballast provides most of the damping. Another resonance frequency referred to as pinned-pinned frequency occurs at around 1000 Hz. The corresponding peak is narrow, indicating that the resonance at this frequency is very lightly damped. The pinned-pinned frequency occurs when the wavelength of the bending waves of the rail is twice the sleeper spacing. In this case the bending vibration of the rail has nodes at the supports, i.e. at the sleepers. See Figure 3-3a-c.

The resonances in the lateral direction often occur at frequencies around 50, 150, 550 and 1500 Hz according to [89]. At around 50 Hz, the rail and the support move in phase, while at 150 Hz they move out-of-phase. At around 550 Hz, the lateral pinned-pinned mode takes place and finally, close to 1500 Hz, the rail head and rail foot vibrate out-of-phase, cf. Figure 3-3d-f. The latter mode is not shown in Figure 3-3.

Figure 3-3 Vertical and lateral track resonances in the frequency range of 0-1000 Hz based on [43] and [100].
Examples of track flexibility measurement results

This section describes and illustrates some examples of track flexibility measurements and corresponding results. In [40] the vertical and lateral receptances were assessed by mounting accelerometers on the rail head and hitting it with an instrumented hammer. Parts of the results are illustrated in Figure 3-4. The measured vertical receptance magnitude displays a drop in the frequency range of 0-60 Hz and a peak at around 200 Hz.

![Figure 3-4](image)

**Figure 3-4** Measured vertical receptance obtained in [40]. Calculated receptance in solid line.

Figure 3-5 illustrates a measured vertical receptance of a ballasted track, [43] and [100]. In the measurements, the track has been preloaded by a vehicle whose closest wheel is at the distance shown in Figure 3-5. The results show a decrease in the track receptance in the frequency range of 0-80 Hz and three peaks at around 100, 500 and 870 Hz respectively. According to [100], in the first resonance the superstructure vibrates on the ballast while in the second resonance the rails and sleepers vibrate on the rail pads and the third resonance corresponds to the vertical pinned-pinned mode.

An example of measured vertical track receptance for a subsoil of soft clay and a static preload of 90 kN per rail is shown in Figure 3-6, [87]. The measurements were performed using the TLV shown in Figure 3-1. Berggren et al. [86], stated that for this particular case, a resonance occurs at only 5-8 Hz due to the soft soil and is followed by an anti-resonance at 9 Hz whereas the track is stiffer for higher frequencies.

Finally, Figure 3-7 illustrates measured vertical track stiffness obtained using the rolling stiffness vehicle shown in Figure 3-2, at a speed of 20 km/h [90]. The results display a continuous change in the vertical track stiffness along the three-kilometre section. They also show large variations in the vertical stiffness when passing over a bridge.
Figure 3-5  Measured vertical receptance of a ballasted track, [43] and [100]. Calculated receptance in solid line.

Figure 3-6  An example of vertical track receptance at subsoil with soft clay, [90].
Figure 3-7  **Vertical track stiffness obtained using the rolling stiffness technique, [90].**

Table 3-1 lists several cases where the vertical and lateral receptances were measured using the standstill or rolling stiffness techniques.

**Table 3-1** Example of studies where the track receptances were measured using the standstill or rolling stiffness techniques.

<table>
<thead>
<tr>
<th>Ref</th>
<th>General aim of the study</th>
<th>Measurement technique</th>
<th>Excitation, quantity of interest</th>
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<tr>
<td>[40]</td>
<td>Rail corrugation mitigation for transit traffic</td>
<td>Standstill</td>
<td>Hammer excitations, Vertical and lateral track stiffnesses</td>
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<td>Modelling of the vehicle-track dynamic interaction</td>
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<td>[84]</td>
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<td>Determination of track properties</td>
<td>Standstill</td>
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3.2 Modelling of track flexibility

Continuous track models
Continuous track flexibility models were reviewed in Popp et al. [66], which classified the track models into continuum track models for moving loads problems and finite element models. Knothe and Grassie [100], also presented a hierarchy of track flexibility models based on the track component modelling.

Dahlberg [91] listed four types of continuous track models: Beam (rail) on continuous elastic foundation, beam (rail) on discrete supports, beam (rail) on discrete supports including ballast model, beams (rails) on sleepers embedded in continuum and including 3-D FE models. Track flexibility models reviewed below utilize this classification.

Beam on continuous elastic foundation
In these models, the rail is represented by a beam resting on a continuous elastic foundation which is modelled by an evenly distributed linear spring stiffness, Figure 3-8. This kind of track model, first introduced by Winkler, is rather simple and is used for fast-time simulations. The model shown in Figure 3-8 is only useful for static loading of a track since the dynamic track properties (mass, damping) are not included [91].

![Figure 3-8 Continuous track model on elastic foundation. The wheel static load is represented by the load P. The dashed line refers to the beam deflection under loading, [91].](image1)

Beam on discrete supports
This type of continuous track models allows for better representation of the track dynamic behaviour. Usually, the rail is modelled as a continuous beam and connected to a rigid mass (sleeper) via springs and dampers in parallel (fastenings and rail pads). The sleeper in turn is resting on elastic foundation representing the ballast and subballast stiffness, [91]. This type of track models has been widely in many studies, see also Table 2-5. An example of a track modelled as continuous beam on discrete supports is shown in Figure 3-9.

![Figure 3-9 Example of a continuous track model on discrete supports, [40].](image2)
**Beam on discrete supports including ballast mass**

For this type of track model, a new rigid mass representing the ballast is introduced. A typical track model is shown in Figure 3-10. Each rigid mass representing the ballast is connected to the subgrade through a spring and damper in parallel. Also, a deflection at one sleeper will influence the displacement at adjacent ones through cross-coupling model elements.

![Figure 3-10 Example of continuous track model on discrete supports including ballast mass, [68].](image)

**Beam on sleepers embedded in continuum, including 3-D FE model**

According to [91], these models present the most realistic track modelling but requires large computer capacity. The rails and sleepers are modelled as beams connected through springs and dampers. The sleepers are embedded in a continuous medium which requires the track bed to be modelled by three-dimensional finite elements, see Figure 3-11.

![Figure 3-11 Example of continuous track model embedded in continuum, [91].](image)

**Moving track models**

The track and its pertinent flexibility can also be modelled through so-called moving track models. In contrast to the common continuous track models above, moving track models are less complicated, have limited number of degrees of freedom and hence are less time consuming and can be used for long-distance simulations. Depending on the complexity level, such models can be appropriate for investigating the vehicle-track interaction for frequencies up to 200 Hz. Vertical static preload effects can also be accounted for in these models as well as the sleeper passing effect. Moving track models are situated under each wheelset and follow the vehicle with the same speed.
The concept of moving track model has been adopted in multibody dynamic softwares such as GENSYS [95], SIMPACK [97] and VAMPIRE [93], and has also been used in several studies such as [11], [20], [47], [98] and [99]. A series of moving track models with different levels of complexity was also developed and examined by Claesson [89]. Some examples of moving track models are illustrated in Figure 3-12.

The track model shown in Figure 3-12a has one degree of freedom (dof) in the lateral direction, while the track models illustrated in Figure 3-12b-c have one vertical and one lateral dof. The moving track model in Figure 3-12d has two levels of vertical stiffness and one level of lateral stiffness (4 dofs) and finally the two remaining models have one level of vertical stiffness and two levels of lateral stiffness (5 dofs).

Figure 3-12 Examples of moving track models from: (a) Andersson et al. [20], (b) SIMPACK [97], (c) Kaiser and Popp [46], (d) Iwnicki et al. [99], (e) GENSYS [95] and (f) Iwnicki [98].

3.3 Influence of track flexibility

The influence of track flexibility on vehicle-track dynamic interaction has been examined in different contexts. This section focuses on the effects of this flexibility on track forces and rail corrugation. The influence of track dynamics on noise propagation has been reviewed in Section 2.3.

On track forces
Ripke and Knothe [68] have compared simulated vertical contact forces at the leading wheels of an ICE coach at 250 km/h, using rigid and flexible models respectively. In the former model a rigid track and a rigid wheelset are considered, while in the latter the elastic properties of the track and wheelset were included. Figure 3-13 shows two-second time histories, corresponding to 140 m. The flexible model, referred to as SiRaGe model, provides the P2 resonance frequency of track and vehicle at around 66 Hz and the sleeper passing frequency at around 117 Hz (sleeper distance is 0.6 m) while the rigid model, referred to as MEDYNA model, lacked such information and covered only low frequency peaks corresponding to the vehicle rigid body motions. The influence of track modelling on simulations results was also investigated by Bruni et al. [26], see Table 2-4.

![Figure 3-13 Vertical contact force histories of the leading wheels (top) and corresponding Fourier spectrum (bottom) obtained using rigid (left) respectively flexible (right) wheelset-track model, [68].](image)

In addition, the influence of track dynamics on wheel-rail forces was investigated by: Morys [59], Szolc [73] and [76], Diana et al. [34] and [35].

**On rail corrugation**

The influence of the P2 resonance on the formation and development of several types of rail corrugation was discussed in [41]. The possible sources of rail corrugation, including the effects of track dynamics, were also examined in [91].
4 The present work

4.1 Introduction

In Chapter 2 and 3 the present knowledge in the research field at hand was reviewed. Despite that several studies have examined the effects of wheelset structural flexibility and track flexibility, several questions still have to be answered, for instance:

- What are the main effects of the wheelset structural flexibility on the wheel-rail forces in the frequency range of 0-200 Hz?
- How do the track dynamic properties influence the wheel-rail forces in this frequency range?
- How does the choice of track model and pertinent data affect the outcome of the simulations and what type of track model is most suitable?
- What is the influence of the track irregularities corresponding to a frequency range of 0-200 Hz and how to achieve a proper representation of these irregularities?

Paper A focuses on the experimental and numerical modal analyses of a locomotive wheelset.

In Paper B, the influence of the wheelset structural flexibility on wheel-rail forces in the frequency range of 0-100 Hz is investigated for the locomotive in Paper A. Results from simulated track forces are compared to measured ones.

In Paper C the influence of track flexibility on the wheel-rail forces in the frequency range of 0-150 Hz is assessed through simulations and measurements. Track flexibility measurements are now utilized in the locomotive case study. Moving track models are developed and used to model the track flexibility.

Paper D features a new case study, involving a newly designed high-speed train running on two adjacent tracks with different track components. Simulated track forces are validated against corresponding measurements in the frequency range of 0-200 Hz.

4.2 Summary of Paper A

In Paper A [31], a wheelset pertaining to a Swedish locomotive is selected and its dynamic properties in the frequency range of 0-500 Hz are investigated through experimental modal analysis (EMA) assuming free boundary conditions. The wheelset under consideration has been instrumented and used during on-track measurements. It has a wheel diameter of 1.3 m and its total mass amounts to 2960 kg. The wheelset shows relatively low fundamental eigenfrequencies: the first bending mode occurs at only 55 Hz and the second bending mode is at 82 Hz. The first and second umbrella modes, where the wheels deform in an umbrella-like shape, occur at 121 and 163 Hz respectively. The damping ratio for the first five eigenmodes is the less than 1%.

A detailed finite element (FE) model, including the gear-box and its components, is then developed and validated against the measurements for frequencies up to 200 Hz, cf. Figure 4-1. Results obtained from the FE model showed good agreement with the measurements. A parametric study revealed that for the present wheelset, not only the axle flexibility but also the wheels’ flexibility plays an important role in the dynamics of the dynamic interactions.
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the wheelset at frequencies below 100 Hz. In the numerical analysis, the gear-box is attached to the axle via linear springs in the longitudinal, lateral and vertical directions. The presence of the 680 kg gear-box and its components does not essentially influence the wheelset flexible modes for frequencies up to about 200 Hz. However, the presence of the gear-box does influence the wheelset rigid body properties.

Torsional excitation was not performed in the EMA. However, the torsional modes of the wheelset are calculated using a model comprising the two wheels and the axle gear wheel rigidly attached to the wheel axle. The first torsional mode occurs at only 47 Hz.

Figure 4-1  Five fundamental eigenmodes and eigenfrequencies derived from the finite element model in [31].

4.3  Summary of Paper B

In Paper B [32], the influence of the locomotive’s wheelset structural flexibility on the vehicle-track dynamic interaction is investigated in the frequency range of 0-100 Hz through simulations and measurements.

In the numerical simulations the wheelset structural flexibility is introduced via eigenmodes derived from the FE model in Paper A and displayed in Figure 4-1. The structural flexibility is considered for the wheelsets of the leading two-axle bogie and the gear-box is modelled as a separate and rigid entity attached to the wheelset in the longitudinal, lateral and vertical directions and to the bogie frame in the vertical direction through linear stiff springs and viscous damping. In contrast, the wheelsets of the trailing bogie are modelled as rigid bodies and the gear-box mass is distributed between the wheelset and the bogie frame.

A 780 m long tangent track section located near the Swedish city of Sala is selected for extensive on-track measurements. Along this section a constant test speed of 140 km/h is
essentially recorded. The track components consist of Swedish BV 50 rails mounted on concrete sleepers through Hambo fasteners. Plastic rail pads, of 5 mm in thickness, are placed between the rails and the sleepers. The sleeper distance is nominally 0.65 m. Measured track misalignments are introduced in the simulations. The wavelength interval considered in this study is $\lambda = 0.5$-100 m. The track flexibility is represented by a moving track model consisting of five degrees of freedom, see Figure 4-2, and the track flexibility data are derived from previous experience in this field. The sleeper passing effect is considered by varying the track stiffness when the wheelset is passing between any two sleepers.

Simulated results showed fairly good agreement with the measured track forces. It was shown that the lateral track forces are more influenced by the wheelset structural flexibility than the vertical ones. From parametric studies it is concluded that the magnitude of the vertical track forces increases as the stiffness of the track is increased. Sleeper passage, short wavelength irregularities and increasing running speed also amplify the vertical track forces.

![Five degree of freedom moving track model used in [32]. (This model is called Model A in Paper C and D).](image)

**Figure 4-2**

**4.4 Summary of Paper C**

Paper C [33] studies the effects of the track flexibility on the wheel-rail forces in the frequency range of 0-150 Hz for the locomotive case study above. For that purpose track flexibility measurements are carried out for a track section, at a sleeper, along the present tangent track.

In this way the vertical and lateral receptances, in the frequency range of 0-200, are measured using Banverket’s track loading vehicle (TLV). In order to study the influence of static preloads and dynamic amplitude on the track dynamics, two vertical preloads of 50 and 90 kN per rail and two different amplitudes are applied.

Measured track irregularity and vertical track roughness, sampled for each 0.25 m respectively 0.01 m, are introduced in the simulations. The wavelength interval considered in this study is 0.25-100 m. Hence, at a speed of 140 km/h, the maximum excitation frequency is about 150 Hz.
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For a better representation of the track flexibility three moving track models, which differ in terms of complexity and number of dofs, are developed and tested. Track flexibility data are then extracted for each model based on the measured receptances above.

Simulated wheel-rail forces are again compared to measured forces sampled at 600 Hz obtained from on-track measurements performed by Interfleet Technology.

This paper reveals that the track flexibility influences the frequency content of the wheel-rail forces significantly. It also highlights the importance of reliable track input data and pertinent track modelling when simulating and investigating different aspects of the vehicle-track dynamic interaction. Track roughness contributes to the higher frequency content of the vertical track force and its inclusion in the simulations is important when predicting various damage mechanisms. The wheelset structural flexibility influences the lateral force significantly whereas its effect on the vertical force is small for this case study.

Figure 4-3a shows the measured and simulated power spectral density of the vertical force of the left leading wheel. In the simulations track Model A shown in Figure 4-2 is used.

The introduction of the “new data” (dark grey line) in Paper C has led to a major improvement in simulated results for virtually the whole frequency range studied as compared to the “old data” (light grey line) of Paper B. The influence of the choice of track flexibility model is illustrated in Figure 4-3b. The simulated vertical force obtained from a more advanced track model shows even better agreement with the measurements, especially for frequencies above 100 Hz.

Figure 4-3a shows the measured and simulated power spectral density of the vertical wheel-rail force of the left leading wheel: (a) using data from [32] and new data extracted from measurements together with track Model A, (b) using data from measurements and track Model C, [33].
4.5 Summary of Paper D

In Paper D a new case study involving a motor coach running on two adjacent tracks with different track components is examined for speeds up to 280 km/h.

Two two-kilometre long tangent track sections, referred to as “down-track” and “up-track”, are selected and investigated. Along these tracks, located near the Swedish village of Väring between Skövde and Töreboda, a high and essentially constant speed was recorded.

With respect to track components, the down-track is regarded as a modern track. The rails have a UIC 60 profile with an inclination 1:30 and are mounted on concrete sleepers through Pandrol fastenings. Rubber rail pads of 10 mm in thickness are placed between the rails and the sleepers. On the other hand, the up-track has Swedish BV 50 rails with an inclination 1:30 mounted on shorter concrete sleepers through Fist fasteners. Rubber rail pads of 5 mm in thickness are used. The sleeper distance is nominally 0.65 m for both tracks.

Rolling stiffness measurements, where the vertical track stiffness is measured continuously while a test vehicle is rolling, are carried out using Banverket’s rolling stiffness measurement vehicle (RSMV). The vehicle was used to measure the vertical track stiffness over the entire two two-kilometres track sections in the frequency range of 5-50 Hz. In contrast, the TLV was used to measure the lateral and vertical track dynamics at three discrete and parallel sections along the two two-km track sections for the frequency range of 1-200 Hz. The selected sections for the standstill measurements include, at each track, two sections at two adjacent sleepers and one section between the two sleepers. In order to study the influence of preloads on the track dynamics two vertical static preloads of 50 and 75 kN per rail are selected, the latter essentially corresponding to the present vehicle wheel load. Among other things, the results obtained from the track flexibility measurements show that the up-track is both statically and dynamically stiffer as compared to the down-track, see Figure 4-4.

![Figure 4-4](image)

*Figure 4-4 Magnitude of vertical track receptance at 75 kN preload per rail for the three sections investigated (a) Down-track, (b) Up-track.*
The present work

The track flexibility was introduced through the moving track models developed in Paper C. The speeds of interest are here essentially 220 and 275 km/h. For these speeds it is possible to compare the vehicle dynamics at both tracks and also investigate the influence of track flexibility and speed on the track forces.

Measured track irregularities and vertical track roughness are also introduced. The wavelength interval considered in this paper is 0.25-100 m. Hence at a running speed of 220 km/h, corresponding to the lowest speed considered, the maximum vertical excitation frequency is about 244 Hz.

From the track force measurements it can be concluded that the up-track vertical forces are significantly higher than those of the down-track, the track overall stiffness and sleeper passing effect being two explanations to this. With low-pass filtering at 20 Hz, like in UIC 518, the down- and up-track RMS-values of the vertical dynamic force is reduced to less than a half respectively a fifth.

Parametric studies confirm a significant influence of the track flexibility on the track forces for frequencies above 80 Hz, whereas the track irregularity difference of the two tracks mainly influences the vertical track force at frequencies around 50 Hz. The wheelset structural flexibility shows an influence on the lateral track force at the stiffer up-track, whereas its effect on the vertical force is rather small. The parametric studies also reveal that modifications of the vertical track receptance, motivated by uncertainties in the measurements, can improve the simulated forces, in particular the PSDs.
5 Conclusions and future work

The wheelset structural flexibility and track flexibility are two factors known to influence the vehicle-track interaction and contribute to higher frequency content of the wheel-rail forces. According to previous work in this field, the wheelset structural flexibility can also influence various aspects such as vehicle stability, wheel out-of-roundness, wheel-rail corrugation and noise propagation. Likewise, the track flexibility is associated with different phenomena such as fluctuations in wheel-rail forces, noise and vibration propagation, etc.

In this thesis two case studies were selected to investigate the influence of the wheelset structural flexibility and track flexibility on the wheel-rail forces. The first case study involves a locomotive running on a tangent track section at a speed of 140 km/h, while the second one deals with a newly designed motor coach running at two adjacent and tangent track sections with different track components and at speeds up to 280 km/h.

For the locomotive case study, the wheelset dynamic properties in the frequency range of 0-500 Hz were first investigated through experimental modal analysis showing comparably low eigenfrequencies. The corresponding results were also derived from a finite element model demonstrating a good agreement with measurements for frequencies up to 200 Hz, and displaying a significant influence of the wheels’ flexibility on the wheelset’s total structural flexibility.

In the on-track simulations, the wheelset structural flexibility was introduced using the calculated eigenmodes while moving track models were adopted to represent the track flexibility. The simulated wheel-rail forces highlight the importance of proper track flexibility modelling and track data. The simulations also showed a significant influence of the wheelset structural flexibility on the lateral track force but a smaller effect on the vertical force.

In the second case study, involving a motor coach running at two adjacent tracks, the wheelset dynamic properties were determined through numerical modal analysis using a rather simple finite element model. The vertical and lateral track receptances at selected track sections were measured using both standstill and rolling stiffness techniques. The results obtained from the track flexibility measurements mainly showed that the tracks have different vertical stiffness and that this stiffness increases significantly with frequency without any obvious resonance. From the track force measurements it was concluded that the vertical track force and also the sleeper effect are significantly higher at the vertically stiffer track.

Basic numerical simulations, where the calculated wheel-rail forces are compared to measured ones, were first performed and followed by a set of parametric studies. In these simulations, the track flexibility was again imitated through moving track models. Measured track irregularity and vertical track roughness were also considered. The wavelength interval for the vertical misalignment was 0.25-100 m. The simulated spectra at the up-track showed fairly good agreement with the measured one but displayed a smaller sleeper passing effect. On the other hand, the simulated and measured spectra at the down-track agreed well for frequencies up to 50 Hz, then the simulated spectra displayed higher forces but smaller sleeper passing effect.
The parametric studies showed that modifications imposed on the measured vertical receptance can significantly improve the simulated forces and lead to a good agreement over the entire frequency range. The studies also confirmed a significant influence of the track flexibility on the track forces for frequencies above 80 Hz, whereas the track irregularity difference between the two tracks mainly influenced the vertical track force at frequencies around 50 Hz. The wheelset structural flexibility showed an influence on the lateral force especially at the stiffer up-track, whereas its effect on the vertical force is rather small.

This thesis shows that lateral and vertical wheel-rail forces can be predicted by multibody dynamic simulations up to frequencies of at least the sleeper passing frequency. However, measurements of especially track flexibility plus track irregularities and roughness are important to the success of such simulations. Proper representation of wheelset structural flexibility is also important in predicting lateral wheel-rail forces. Further revisions of UIC 518 and EN 14363 should consider the potential of saving time and cost by using vehicle-track simulations. Also today’s upper frequency limit of 20 Hz for the wheel-rail forces in these codes should be raised to account for wheel and rail fatigue damage.

Further work should assess the influence of the wheelset and track flexibilities on wheel-rail forces at curve negotiation. Vehicle lateral stability should also be studied with respect to the flexibilities mentioned.
References

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References related to measuring, modelling and influence of track flexibility


