VERTICAL RAILWAY TRACK DYNAMICS: FROM MEASUREMENTS TO NUMERICAL MODELLING

CHARACTERISTIC FREQUENCIES AND RAIL-RAILPAD-SLEEPER INTERACTION
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FROM MEASUREMENTS TO NUMERICAL MODELLING
CHARACTERISTIC FREQUENCIES AND RAIL-RAILPAD-SLEEPER
INTERACTION

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Maider OREGUI ECHEVERRIA-BERREYARZA

Ingeniero Superior Industrial, Universidad de Navarra (Spain) geboren te Donostia-San Sebastian, Spanje.
To my family
Aita, Ama, Dorleta eta Leire,
zuek gabe, ezinezkoa
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SUMMARY

Railway deterioration has an immediate impact on our daily life. The correct functioning of trams, metros and trains is required to regularly transport hundreds of millions of passengers and tons of goods. A key aspect to guarantee the massive passenger and freight transport is to prevent or, at least, slow down railway deterioration.

The main goal of this dissertation is to have a better understanding of track vibrations and interactions between components. In turn, such deeper understanding will enable us to slow down the deterioration through new, optimized track designs and maintenance measures, thus extending the service life of railway tracks and, consequently, lower their life-cycle costs. Therefore, we focus on measuring and modeling the vertical dynamics of railway tracks.

The first part of the dissertation focuses on the in-depth analysis of extensive field hammer test measurements. Hammer tests are simple and inexpensive, yet they provide valuable information about the characteristic frequencies of tracks. To assess the potential of hammer tests to be employed for track deterioration investigation, we designed and conducted a feasibility study on insulated rail joints (IRJs) in the field. First, a reference dynamic response is defined and then it is compared to the response of three different damaged IRJs. Three characteristic frequency bands related to the damaged IRJs are derived independently of the type of damage. In view of these promising results, a Frequency Response Function (FRF)-based statistical method is proposed to identify characteristic frequencies of railway track defects. The method compares a damaged track state to a healthy state, which, in this case, is defined following the concepts of control charts employed in process monitoring. The FRF-based statistical method is tested at squats of different severity in two tracks types. For both squats and damaged IRJs, the identified characteristic frequencies agree with those found with an extensively validated vehicle-borne detection system (i.e. Axle Box Acceleration (ABA) system). This means we are indeed able to identify characteristic frequencies of defects using hammer tests.

In the second part of the dissertation, a three-dimensional finite element (3D FE) model of tracks with monoblock sleepers is developed to study in-detail track vertical dynamics. This sleeper type is used worldwide, but its dynamic behavior is often not accurately considered in track models. To study the track dynamics, hammer tests are numerically reproduced applying an Implicit–Explicit FE procedure. First, the equilibrium state of the track is calculated and then the response of the track to hammer excitation is simulated in the time domain. Next, our 3D FE model and field measurements are combined by fitting simulations to measurements, so that (1) the model is validated and the accuracy to reproduce measurements is determined, (2) the in-service track parameters are derived, and (3) insight is gained into the contribution of components to track dynamics and the effect of simplifications in modeling.
After validating our 3D FE model with measurements and performing a comprehensive analysis, we find that the frequency response between 300 and 3000 Hz is defined by seven characteristic features and vibration modes. This is an important result for understanding track dynamics, given that the tracks with biblock sleepers only have four such features. The bending modes of monoblock sleeper and the stronger coupling between the two rails cause two of the additional features. The third additional feature occurs in the frequency range dominated by the rail-railpad-sleeper interaction. With the 3D FE model as basis, the influence of the representation of this interaction into the numerically calculated vertical dynamics is investigated. For this purpose, four fastening representations are developed: (1) commonly used spring-damper pair, (2) area covering spring-damper pairs, (3) solid railpad connected to the rail, and (4) solid railpad in frictional contact with the rail and fixed to the support by preloaded springs, which represent the clamps. Their comparison shows that the overall numerical reproduction of the measurements improves the more realistic is the representation of the fastening. If the accuracy of the fastening models is quantified, the model with solid railpads and clamps reproduce the seven characteristics at a maximum frequency difference of 6%, whereas for the conventional model, the difference can be as high as 27%. The lateral and longitudinal dimensions of the railpad, and the lateral and longitudinal constraints between the rail and support applied by the fastening, are both relevant aspects to consider when modeling fastenings.

By examining field measurements and numerical models, useful information is gained for track design and for the development of maintenance measures. Regarding measurements, if field hammer tests are analyzed by employing the FRF-based statistical method, characteristic frequencies of track defects can be identified and can become valuable input data for the development of vehicle-born detection systems, such as ABA systems. Concerning the 3D FE model, a deep insight into the vertical dynamics of tracks with monoblock sleepers is obtained and the comprehensive study of fastening representation has indicated the need to model this track component more realistically, so that the track dynamics at high frequencies can be correctly reproduced. This finding should be considered in vehicle-track models to improve the reproduction of vehicle-track high frequency dynamics. A more realistic fastening modeling is especially required to represent tracks with rail defects since the defect’s development may be related to the condition of the fastening. Thus, the advanced fastening models presented in this research are expected to significantly contribute to the investigation of fastening degradation and track defects. In addition, the evolution of in-service track parameters (i.e. stiffness and damping of railpad and ballast) identified during the fitting process can be monitored and it may become valuable input data for the planning of maintenance. Overall, our combination of measurements and numerical models contributes to the understanding of track dynamics. Furthermore, we have provided models and tools that can be used for the investigation and monitoring of track deterioration.
De conditie van de spoorconstructie heeft een directe impact op ons dagelijks leven. Het foutloos functioneren van Tram, metro en treinverkeer is nodig om dagelijks honderden miljoenen passagiers en tonnen goederen te vervoeren. Om een vlot verloop te garanderen is het van groot belang dat aftakeling van het spoor wordt voorkomen of ten minste wordt vertraagd.

Het hoofddoel van dit promotieonderzoek is om tot een beter inzicht te komen van de trillingen in het spoor en de interacties tussen de verschillende componenten in het spoor. Op zijn beurt zal een dieper inzicht leiden tot nieuwe geoptimaliseerde spoornetworken en onderhoudsstrategieën en ons in staat stellen de aftakeling van het spoor te vertragen. Zo wordt de levensduur van het spoor verlengd en worden de totale kosten verlaagd.

Het eerste deel van het proefschrift richt zich op de diepgaande analyse van talrijke hamerexcitatieproeven. Hamerexcitatieproeven zijn eenvoudig en goedkoop, maar geven toch waardevolle informatie over de karakteristieke frequenties van het spoor. Om het potentieel van de hamerexcitatieproeven voor spoorafktaking vast te stellen, is een haalbaarheidsstudie uitgevoerd naar Electrische Scheidingslassen (ES-lassen) in het veld. Eerst wordt een dynamische referentieresponsie gedefinieerd, waarna dit wordt vergeleken met de dynamische responsie van drie verschillende beschadigde lijmlassen. Ten aanzien van de beschadigde lijmassen zijn, onafhankelijk van het soort schade, drie karakteristieke frequentiebanden te onderscheiden. Gezien de veelbelovende resultaten, wordt, gebaseerd op een statische methode, een frequentieresponsie functie (Frequency Response Function, FRF) geponeerd om de karakteristieke frequenties van spoorbeschadigingen te identificeren. De methode vergelijkt een beschadigd spoor met een onbeschadigd spoor, welke in dit geval wordt gedefinieerd volgens het concept van controle kaarten in het proces beheer. De FRF gebaseerde statistische methode is getest op squats (een veel voorkomende vorm van schade in de rijspiegel van de rail) van verschillende grootte en op twee verschillende types spoor. Zowel voor de squats als voor de lijmlassen komen de gevonden karakteristieke frequenties overeen met de frequenties die gevonden worden met het uitgebreid gevalideerde, in een meettrein ingebouwde aspotversnellingsysteem (Axle Box Accelerations, ABA). Hieruit blijkt dat het inderdaad mogelijk is om karakteristieke frequenties van spoorbeschadigingen te identificeren door middel van hamerexcitatieproeven.

Om de verticale dynamiek van het spoor te bestuderen wordt, in het tweede deel van het proefschrift, een driedimensionaal eindig elementen model (Finite Elements, FE) van een spoor met mono-bloc dwarsliggers ontwikkeld. Hoewel dit type dwarsligger wereldwijd gebruikt wordt, wordt in spoormodellen haar dynamische gedrag vaak niet nauwkeurig meegenomen. Om het dynamische gedrag van het spoor te bestuderen worden de hamerproeven numeriek gereproduceerd met een impliciete-expliciete FE-procedure. Eerst wordt de evenwichtstoestand van het spoor berekend en vervol-
gens wordt de responsie van het spoor als gevolg van de hamerexcitatie gesimuleerd in het tijddomein. Daarna worden de simulaties van het 3D FE model en de veldmetingen dusdanig gecombineerd en gefit dat (1) de modellen gevalideerd worden en hun precisie wordt bepaald, (2) de spoorparameters van een in bedrijf zijnd spoor worden verkregen, (3) Inzicht wordt verkregen in het aandeel dat iedere component in de spoordynamica heeft en (4) welke effecten het vereenvoudigingen van het model kunnen hebben.

Na validatie van het 3D FE model met de veldmetingen en een uitgebreide analyse, is gevonden dat de frequentiekarakteristiek tussen 300 en 3000 Hz bepaald wordt door zeven karakteristieke kenmerken en trillingsmodi. Dit is een belangrijk resultaat voor het begrijpen van de spoordynamica, daar bi-bloc dwarsliggers slechts vier van zulke kenmerken hebben. De buigmodi van de mono-bloc dwarsliggers en de sterkere koppeling tussen de twee spooralven geven twee extra kenmerken. Het derde extra kenmerk bevindt zich in een frequentiebereik dat gedomineerd wordt door de spoornaafonderlegplaat-dwarsligger interactie. Met het 3D FE model wordt de invloed van de verticale spoordynamica bestudeerd, afhankelijk van de wijze waarop deze interactie wordt voorgesteld in het model. Vier verschillende modellen voor de spoornaafbevestiging worden vergeleken: (1) een veel gebruikt veer-demper paar, (2) veer-demper paren verspreid over een oppervlak, (3) een vaste onderlegplaat bevestigd aan de spoornaaf en (4) een vaste onderlegplaat met wrijving tussen de onderlegplaat en de spoornaaf, bevestigd aan de dwarsligger met voorgespannen enen, die de klemmen voorstellen. De vergelijking toont aan dat de metingen beter worden gereproduceerd naar mate de voorstelling van de bevestiging realistischer is. Wanneer de precisie van de modellen wordt gekwantificeerd, blijkt het model met vaste onderlegplaten en klemmen de zeven kenmerken te reproduceren met een maximum verschil in frequentie van 6%, terwijl bij het klassieke model met veer en demper de verschillen kunnen oplopen tot 27%. Relevante aspecten bij het modelleren van bevestigingen zijn enerzijds de laterale en longitudinale afmetingen van de onderlegplaat en anderzijds de laterale en longitudinale randvoorwaarden van de bevestiging tussen de spoornaaf en de dwarsligger.

Door veldmetingen en numerieke modellen te onderzoeken, wordt waardevolle informatie verkregen voor het spoornontwerp en de ontwikkeling van onderhoudsmaatregelen. Ten aanzien van veldmetingen kan worden gesteld dat wanneer hamerexcitatieproeven worden geanalyseerd door middel van de statische FRF methode, karakteristieke frequenties van spoornbeschadigingen geïdentificeerd kunnen worden met een maximum verschil in frequentie van 6%, terwijl bij het klassieke model met veer en demper de verschillen kunnen oplopen tot 27%. Relevante aspecten bij het modelleren van bevestigingen zijn enerzijds de laterale en longitudinale afmetingen van de onderlegplaat en anderzijds de laterale en longitudinale randvoorwaarden van de bevestiging tussen de spoornaaf en de dwarsligger.
van de spoorbevestiging en andere spoorbeschadigingen. Bovendien kan de ontwikkeling van gefitte spoorparameters van een bereden spoor (d.w.z. stijfheid, demping van de onderlegplaten en ballast) worden gevolgd en zo een belangrijke bijdrage leveren aan de planning van het onderhoud. In het algemeen kan worden gesteld dat de combinatie van metingen en numerieke modellen in dit proefschrift bijdragen aan het beter begrijpen van spoordynamica. Ook zijn modellen en instrumenten ontwikkeld die kunnen worden gebruikt voor onderzoek naar- en monitoring van spoordegradatie.
1.1. **Railway Transport**

Railway trains are a major mean of transport worldwide. Millions of passengers travel by railway daily. For instance, 1.1 million people are daily transported in The Netherlands by trains such as the one shown in Figure 1.1.

In middle distance journeys (i.e. less than 800 km), high speed trains compete with the aerospace industry connecting cities, such as the important Madrid-Barcelona connection in Spain. In shorter distances between urban areas, the number of intercity travels and commuter trips is yearly increasing supported by the good integration with other forms of transport (bicycle, buses, tramway, metro). For instance, bicycle renting services are available outside many main stations in European cities, such as Berlin, Amsterdam and Bilbao.

![Figure 1.1: An intercity train in The Netherlands](image)

Traveling in big and congested urban areas is often only possible thanks to the mass transport of trams and metros. Although the infrastructure is expensive, the large number of passengers transported and its independence from road traffic make the railway transport essential for many public transport systems. In addition, railway transport offers a high degree of automation, reliability and safety. In Europe, the transit networks of Paris, Madrid and London carry more than 1 billion passengers a year. As an Asian example, approximately 10 million people travel by metro every weekday in Beijing.

In addition to passenger transport, the freight transport is capable of moving cargo in an energy efficient way. An example of long-range distance journey is the transport of coal from mines to ports in South Africa using 4-km-long heavy haul trains. In medium-range distances such as usually happens in Europe, the transport of goods by freight trains is being promoted to (1) relieve the roads, often congested, from heavy traffic and (2) improve the environmental conditions by reducing CO₂ emissions because railway is significantly more energy efficient than road.

In summary, hundreds of millions of passengers use regularly rail transport, and hundreds of millions of tons of goods are carried by rail. Within this framework, the rail-
way transport is continuously evolving; passengers demand for shorter travel times and freight transport for higher capacities. These demands take the railway infrastructure often close to its operational and structural limits.

1.2. **Deterioration of Railway Tracks**

The severe working conditions accelerate the deterioration of the track structure (see the main track components in Figure 1.2). The dynamics of the vehicle/track interaction can affect the railway bridges and nearby buildings [1–3]. Furthermore, the substructure layers (i.e. ballast, subballast and subgrade) deteriorate under the vehicle/track interaction [4, 5]. The subgrade can suffer from settlements so that tracks lose their geometry. With time, the ballast deteriorates so that its vertical, lateral and longitudinal stiffness change. The movement of ballast stones can also cause voids under sleepers, which worsens the transmission of the loads to the ballast and subgrade layers and also, larger track displacements happen. White dust on the ballast stones is often a symptom of the extra movement of the sleeper within the ballast bed, which may be the case shown Figure 1.3a. When trains travel over degraded substructure layers, the rolling condition worsens to the extent that severe deterioration might lead to vehicle rolling instability, and sometimes to derailment.

The sleeper, fastening and rail, which form the superstructure layer, deteriorate as well [6–9]. The bearing capacity of sleepers changes if cracks appear in the sleeper, see for instance Figure 1.3b. In the fastening system, the bolts can become loose or the railpads can slip away, as shown in Figures 1.3c and 1.3d, respectively. In both cases, the function of the fastening system as a rail fixing mechanism and vibration damper changes. The consequences may not immediately result in detrimental effects, but these conditions significantly accelerate the degradation of the track.

The rail top suffers wear, plastic deformation and cracks under the high vehicle/track contact forces [10–12]. These forces are especially high at discontinuities or material property changes of the rail, such as insulated rail joints (Figure 1.3e) and welds (Figure 1.3f). Furthermore, isolated or periodic defects develop on the rail surface, see a squat in Figure 1.3g and short pitch corrugation in Figure 1.3h. The presence of these types of damage spurs the deterioration of the track worsening the rolling conditions. In
some cases, such as rails with cracks, safety may be threatened.

The worsened rolling condition causes increased rolling noise, which is a nuisance for the people living nearby railway tracks [13, 14].

1.3. **Maintenance of railway tracks**

To guarantee safety and acceptable noise levels, maintenance measures are taken. Some examples are tamping of the ballast bed, grinding of the rail top or rail replacement. Furthermore, the maintenance measures need to be performed in a shorter time because the railway schedules are becoming more saturated to offer a more complete service to the customer. Less maintenance can lead to faster degradation, higher life cycle costs, and more unexpected interruptions. In summary, tracks are subjected to more demanding service conditions while the resources to keep the desired performance are reduced.

To evaluate the condition of the track, measurements are regularly performed, such as rail inspection using ultrasound [15] or track inspection using visual cameras [16]. However, problems often arise. For instance, the time-frame between measurements is sometimes too long so that damage occurs and develops to a severe state. Another problem is that some measurement methods detect the damage at a late state so that expensive immediate maintenance measures are required.

In some networks, preventive maintenance measures are adopted as part of their annual maintenance planning. The objective is to take actions at relatively fixed intervals so that the damage does not develop into a state that requires time-consuming and expensive maintenance. Two example of preventive maintenance measures are the cyclic grinding of the rail top that is performed in some metro and train networks and the cyclic tamping carried out in many mainline networks. In this manner, the unwanted vibrations and noise are kept under undesired levels, and crack formation may be prevented.

An intermediate option between the preventive maintenance and immediate maintenance is the early-state maintenance, which consists of preventing the track from reaching an irreversible damaged state. This means that the maintenance actions remove the damage that is still reversible, so that the track can be used to its optimum. For this purpose, the early-state maintenance relies on frequent or continuous monitoring. For instance, optical fiber sensors, such as fiber bragg gratings, can be installed to monitor the condition of railway subcomponents [17]. An example of vehicle-borne inspection systems are strain-gauge-instrumented wheelsets [18] or Axle Box Acceleration (ABA) systems [19–22]. With frequent monitoring, the deterioration of tracks can be tracked, and the maintenance measures are carried out when the deterioration crosses certain thresholds. Consequently, tracks stay in service and fulfill safety conditions as long as possible. Thus, the service life of railway tracks can be extended by applying condition based maintenance and predictive power to anticipate problems.

These three maintenance frameworks (i.e. periodic, preventive and early-state maintenance) could be improved with a better understanding of the deterioration of the vehicle/rail/track system. Gaining insight into the interactions and vibrations occurring in the complex railway system may help to develop improved and new monitoring and maintenance methods, and also, to develop new designs. Better understanding and new developments can lead to custom-made maintenance schedules and to delay in deterioration. Consequently, the high life-cycle costs can be lowered.
1.3. MAINTENANCE OF RAILWAY TRACKS

(a) Loose ballast
(b) Broken sleeper
(c) Loose bolt
(d) Displaced railpad
(e) Damaged rail joint
(f) Damaged weld
(g) Squat
(h) Short pitch corrugation

Figure 1.3: Different types of track deterioration
1.4. **Vehicle/Wheel-Rail/Track System**

The vehicle/wheel-rail/track interaction is a complex system. Separately, the vehicle as well as the track have complex dynamics, see for instance [23] and [24] respectively. Together, the degree of complexity increases due to the wheel-rail contact, an interaction that has extensively been studied [25]. In view of the difficulty to understand the whole system all at once, the vehicle, the wheel-rail contact and the track are often investigated separately as sub-systems. By gaining insight into the dynamic behavior of the sub-systems, the complex vehicle/wheel-rail/track system can be tackled on a solid base.

1.4.1. **Vehicle Dynamics**

Vehicle dynamics are optimized following three main criteria. (1) The first criterion is passenger comfort because it is a key factor for the increase in the number of railway passengers in the last years. The vehicle parameters, mainly the secondary suspension and train car design, are optimized to reduce as much as possible the inside-cabin noise and vibrations. (2) The second criterion is rail and wheel wear, which are minimized by reducing the slip between wheel and rail. In this manner, profiles last longer and maintenance measures can be taken at longer periods; consequently, maintenance cost are reduced. However, the reduction in slip is obtained by providing a better steer in curves which also means that the vehicle is less stable. Therefore, the optimization of the suspensions parameters considers the trade-off between rail and wheel wear and vehicle stability. (3) The third criterion is vehicle/wheel-rail/track forces for frequencies lower that 50 Hz. Low frequency forces affect track components such as ballast, subballast and subgrade. If the forces are too large, ballast shifts and excessive settlements can occur changing significantly the track geometry, which may cause derailment.

Regarding the second and third criteria, the vehicle dynamics are commonly studied by employing the multi-body approach [23, 26, 27]. In the models, the vehicle is defined consisting of masses connected with linear or non-linear spring and dampers. The stiffness and damping values of the suspensions are optimized to minimize rail and wheel wear and the vehicle/wheel-rail/track forces for frequencies lower that 50 Hz are calculated.

To study the inside-cabin noise, three methods are generally used [28–30]. In the finite element-boundary element method, the vehicle is defined with finite elements and the air with boundary elements. In the statistical energy analysis method, the train car is divided into subsystems of similar characteristics and the energy flows are calculated. In the transfer function method, transfer functions of different parts of the train car are experimentally defined and then, combined. Depending on the accuracy needed and available time, one of the three methods is used to study the inside-cabin noise.

1.4.2. **Wheel-Rail Contact**

Wheel-rail contact forces are required to investigate vehicle dynamics employing multi-body models (Section 1.4.1). Also, slip and stress distributions in the contact path are of interest because these distributions relate to the wear rate of rails and wheels, and the profile change rate of rails and wheels directly affects their life-cycle.

The wheel-rail contact problem is divided into the normal and tangential contact
problems which are studied using different approaches (see [31] and the reviews [25, 32]). For the normal contact, hertzian, multi-hertzian or non-hertzian methods are implemented. For the tangential contact, look-up tables, Kalker FASTSIM or Kalker CONTACT approaches are commonly used. The methods chosen depend on the type of contact investigated (e.g. conformal vs. non-conformal, one-point contact vs. multi-point contact [33, 34]), the accuracy required and the time available. The main challenge of the models is their validation because, to our best knowledge, there are no devices available that can measure the high contact stress and slip distribution on the contact patch.

1.4.3. TRACK DYNAMICS

Track dynamics are studied to examine the load distribution in the track. If the loads can be spread in a more optimized way, the track can last longer and the life-cycle costs can be reduced. Also, if critical components are identified, new designs or maintenance measures can be developed to slow down their deterioration. In addition to load distribution, the track structure damps rolling noise and ground vibrations. A maximum attenuation of noise and vibrations is desired because they are a nuisance for people living close to railways [13, 14].

To study the track dynamics, models are combined with experimental work such as impact or sinusoidal excitation or train passages. Fitting simulations to measurements is a common practice to obtain parameters of track components (e.g. ground material) [24, 35–39]. For modeling, finite element, boundary element or analytical models are generally used, see a review of track models in [40].

1.5. EXAMINING TRACK DYNAMICS BY HAMMER TESTS

Hammer test measurements are often employed in railway to gain insight into the dynamics of track systems [7, 41–43]. The test consists of exciting a track component and measuring the response at the location of interest. For the superstructure layer, the main interest lays on the vertical dynamics of the track so that hammer tests are performed by vertically exciting the rail top and measuring its response. Then, the measured signal is transformed into the frequency domain and the frequency response function is calculated. A field hammer test measurement is shown in Figure 1.4.

Hammer tests are a simple and inexpensive measurement system, yet valuable information about the track can be obtained. The analysis of hammer test measurements gives an insight into the dynamic behavior of tracks because the characteristic frequencies can be identified [7, 41–43]. The resonances and antiresonance (i.e. peaks and dips, respectively) of the track occur at frequencies where the track tends to vibrate. Often, damping measures are designed to attenuate vibrations at these frequencies so that the track does not get into resonance, which would accelerate the deterioration of the track. In Section 1.6, the characteristics of vertical track dynamics are presented.

In addition, hammer test measurements and numerical modelling are often combined so that, (1) track models can be validated [37, 44, 45], (2) track parameters (i.e stiffness and damping of the railpad and ballast) can be derived by varying the model parameters until the model response agrees with the field hammer test measurements [24, 35–39], and (3) insight can be gained over the contribution of a track component to the
Field hammer tests are commonly performed at nominal (i.e. healthy) locations. From the measured dynamic response, characteristic modes and frequencies of tracks are identified. According to the literature, the vertical dynamic response of tracks is characterized by five resonances and anti-resonances. The characteristic modes are defined by the vibration of track components and their interaction, and the characteristic frequencies are determined by the parameters of the track components.

The dynamic response of the track can be divided into three frequency ranges depending on the contribution of track components [47]. In the low frequency range (0-40 Hz), the substructure layer is the dominant contributor to the track response. In the middle frequency range (40-400 Hz), the ballast and sleeper are the two key components that define the dynamic response of tracks. In the high frequency range (400-3000 Hz), the superstructure layer provides the major contributions. Based on this classification and complementary research [36, 37, 41, 42, 44, 48], the five characteristics and the components of the track are related as follows:

**Full track resonance** (40-140 Hz) The superstructure layer of the track vibrates on the flexibility of the ballast. Although the mass of the sleeper and rail influence the location of this resonance, the ballast properties mainly determine the full track resonance frequency.

**Sleeper anti-resonance** (100-400 Hz) The sleeper vibrates between the railpad and ballast, while the rail barely moves. This mode mainly depends on sleeper properties, such as its mass, and the railpad and ballast stiffness.
**Rail resonance** (500-1100 Hz) The rail vibrates in anti-phase with the sleeper. The rail and railpad properties, such as stiffness and damping, mainly determine the rail resonance frequency.

**Pin-pin resonance** (800-1200 Hz) The rail vibrates with its nodes on the sleepers. This characteristic is mainly determined by the sleeper distance, and to a minor degree by the rail and railpad properties.

**Second order pin-pin anti-resonance** (2600-3000 Hz) The rail vibrates with its nodes on the sleeper at half wavelength of the pin-pin resonance. As in the case of the pin-pin resonance, the sleeper distance and rail and railpad properties determine the frequency of this characteristic.

This description matches the dynamics of tracks with biblock sleepers [39, 41], as it is shown in a measured frequency response function in Figure 1.5. In the case of monoblock sleeper tracks, however, the measured track response differs considerably from the frequency distribution described in the literature. The dynamic response of the track shows a different behavior between 450 and 1000 Hz as shown in Figure 1.5 [24, 49]. This frequency range corresponds to the dynamic behavior of the sleeper and it includes frequencies related to track defects, such as squats and short pitch corrugation [10, 11].

![Figure 1.5: Measured frequency response functions: — track with monoblock sleepers and — track with biblock sleepers. Four characteristic vibration modes are shown (1) sleeper anti-resonance, (2) rail resonance, (3) pin-pin resonance, and (4) second order pin-pin resonance](image-url)

### 1.7. Modeling vertical track dynamics

To investigate the vertical track dynamics and derive track parameters by fitting simulations to measurements, different models have been developed (see, for instance, a summary in [40, 50]). In the models, the track components are represented differently. Some modeling simplifications determine the applicable frequency range of the model. In this
section, different representations of the rail, sleeper, fastening and ballast are summarized.

1.7.1. **Rail**
The applicable frequency range of a track model is closely related to the modeling of the rail, major contributor to the dynamic behavior of tracks at frequencies above 500 Hz [24, 48, 51]. Depending on the representation of the rail, the models can be divided into four main classes.

**Class I** The first class includes the models that represented the rail as a beam, either Euler or Rayleigh-Timoshenko, continuously [41, 52, 53] as well as discretely supported [45, 51, 54]. The advantage of this rail representation is the short calculation times. The disadvantage is that the application of these models is limited to frequencies lower than 1500 Hz because the cross-sectional deformation of the rail is not considered; this deformation significantly influences the track dynamics above 1500 Hz [55].

**Class II** The second class considers the relative movement between foot and web of the rail by defining the head, web and foot of the rail as connected parts. One combination represented the head as a beam while the web and the foot were modeled as plates [52, 55]. In a second variant, the head as well as the foot were represented as beams [42]. The head was connected to the foot with elastic coupling. Although foot flapping was taken into account, the models that had the rail defined as a combination of parts showed reasonable agreement with the measurement only up to 1500 Hz [24, 52] which was also the upper frequency limit in the first class. As for Class I rails, the computational time of these models is short.

**Class III** The third class of models represent the rail with its real cross-section of the rail. First Knothe et al. developed a simplified rail cross-section [56] that was later defined more realistically by Gry [44, 57]. In both models, the vibration modes of the rail cross-section were calculated, and later introduced in a full track model by means of mode superposition in which some modes were preselected and others excluded. As in the previous two classes, the models could fit the measurements up to 1500 Hz. Between 1500-3000 Hz, the measured frequency distribution was not properly reproduced except for the so-called second order pin-pin anti-resonance (2700-3000 Hz). As the calculations are performed in the frequency domain, the computational time is short. However, the accuracy of the results is dependent on the number of rail cross-section modes included.

**Class IV** In the forth class, the rail is represented with its real cross-section using the Finite Element (FE) approach in which the vibrations are calculated in time domain [58, 59]. The advantage of this approach is that vibrations in the track structure are automatically considered without the need to preselect any modes for superposition. The drawback is the long calculation time.
1.7.2. **Sleeper**

Sleepers strongly influence the vertical track dynamics in the frequency range of 300-1000 Hz [24, 41, 51]. The dynamic behavior of sleepers has numerically and experimentally been studied by employing modal analysis. Based on the results, the sleeper has been represented differently in track models. These two aspects are further explained in the following sections.

**Modal analysis of the sleeper**

The natural frequencies and vibration modes of monoblock sleepers have been identified by conducting modal analysis. Experimentally, the sleeper is often hung on soft springs so that the natural frequencies are derived under the free-free condition [60]. In the free-free condition, the sleeper is completely free to move because there are no other track elements. In some experiments, the modal testing is conducted on a sleeper lying on ballast so that the interaction between the sleeper and ballast is considered, the so-called on-foundation condition [5, 6, 61]. The difference in derived frequencies from the two test conditions is considerable only for the first bending mode (i.e. 15% difference) [43]. For higher bending modes, the difference is a small 2%.

Numerically, sleeper modal analysis has been studied under different boundary conditions varying from realistic track conditions [62] to a lack of boundaries [63]. Besides the free-free and on-foundation boundary condition, the in-situ boundary condition has been investigated, in which the sleeper interacts with the fastening and ballast.

The monoblock sleeper has typically been modeled with Timoshenko beams. By defining the sleeper as a Timoshenko beam, the natural frequencies of sleepers were analytically studied in both free-free and on-foundation conditions [63]. The main conclusion was that the translation, rotation, and low bending eigenmodes of the sleeper changed when including the ballast in the model. Using a 2D finite element analysis and an equivalent Timoshenko beam whose properties were averaged over its entire length, sleeper dynamics were analyzed under in-situ conditions [62]. The same conclusions were obtained as for the free-free and on-foundation conditions, namely that the first three eigenfrequencies changed depending on the boundary condition because the presence of other track components increased the stiffness of the sleeper by limiting its movements.

In summary, the modal analysis of monoblock sleepers resulted in (1) the identification of the natural frequencies and (2) the quantification of differences in natural frequencies for different testing and modeling conditions.

**Reproducing the contribution of the sleeper**

In track models, sleepers are usually represented as rigid masses [41, 45] or as beams with either a constant section [41, 54, 64], an equivalent section [62], or a combination of sections [51]. These representations provided a considerably good correspondence between the modelling and measurements of tracks with biblock sleepers [39, 44]. However, the use of biblock sleepers is decreasing. Instead, monoblock sleepers are widely used as its bearing capacity is considered higher than for other sleeper types [65]. For instance, in the Netherlands, monoblock sleepers are used in new lines and renewals since 1990 [66].
The track models have not been able to reproduce the peaks and dips in the receptance function of tracks with monoblock sleepers between 450 and 1000 Hz, a range that corresponds to the dynamic behavior of the sleeper [24, 51, 67]. This disagreement is not caused by the simplification of the sleeper-ballast interaction because this interaction is relevant for frequencies lower than 300 Hz [68–71]. The difference may be related to the representation of the sleeper-fastening-rail interaction.

Furthermore, the frequency range 450-1000 Hz includes frequencies related to track defects, such as rolling contact fatigue defects [72, 73] or short pitch corrugation [74]. In addition, the appearance and growth of squats are related to the condition of the support according to field observations [11] and numerical results [59]. The relation between short wave defects and support condition also points to studying the representation of the sleeper-fastening-rail interaction.

1.7.3. FASTENING
The fastening is a key contributor to the dynamic behavior of the track for frequencies higher than 500 Hz [24, 48, 51, 75]. The fastening system consists of two main components:

**Railpad** The railpad is a solid resilient material that is placed between the rail and the sleeper. By contributing with vertical stiffness and damping distributed in the rail seat area, the railpad isolates and damps vibrations and adds flexibility to the track.

**Clamps** The clamps ensure that the rail remains connected to the sleeper. To guarantee that the rail is properly placed in any working condition, the clamps are preloaded with a toe load. Thus, the clamps limit the relative vertical, lateral and longitudinal movements of the rail with respect to the sleeper.

Although the main elements of the fastening are the railpad and the clamps, studies found in the literature mostly focus on the railpad. The railpad is the main target because of two main reasons. First, the dynamic loads between the rail and the concrete sleeper caused by the passing of vehicles, the rolling noise and the growth of defects on the rail top strongly depend on the properties of railpads [51, 76–79]. Second, modeling the clamps is beyond the capacity of some simulation tools, and when possible, the clamps would significantly increase the complexity of track models.

Obtaining broad and precise information about the behavior of different railpads in the working conditions can help to estimate the deterioration of tracks. From laboratory testing, the key factors that influence the stiffness and damping of railpads have been identified and are summarized in the following section. These fundamental aspects have been considered differently when modeling the railpad as it is shown in the next section.

**Railpad key factors**
Extensive laboratory tests have been performed to gain insight into the behavior of railpads so that the four most relevant factors which influence the railpad properties are identified. The four factors are as follows:

**Preload** The stiffness of the railpads shows a non-linear behavior in the static load-deflection curve [37, 80–82]. The parameters of the railpad are subjected not only to momentary preload condition but also to preload history [48].
1.7. **Modeling Vertical Track Dynamics**

**Frequency dependency**  The stiffness of the railpad increases for higher frequencies [47, 80, 81, 83, 84]. The opposite happens to the damping.

**Temperature**  The stiffness of the railpad increases significantly for lower temperatures, whereas the damping decreases [85].

**Aging**  Contradictory observations were reported about the development of the railpad properties after aging. Whereas fatigue tests indicated an increase in stiffness for a larger number of cycles [85], the comparison of worn railpads taken from the field revealed a decrease in stiffness with the increase in MGTM (Million Gross Ton Mile) [86].

Test procedures that consider differently the effects of preload, frequency and temperature have been developed to investigate the railpad behavior and derive its parameters (i.e. stiffness and damping). In some railpad testers, the railpad is placed between plates that are tied together so that the railpad is subjected to the toe load of the clamps [9, 48, 75]. In other testers, one support (i.e. rail, fastening and sleeper) is employed [87]. Depending on the test set-up, either one plate or the rail is vertically excited and the response to an impact force is measured. The stiffness and damping of the railpad are calculated by fitting a model to the measured resonances. In this manner, the influence of the preload is investigated.

Other railpad testers included an harmonic loading so that the frequency dependent behavior of the railpad is analyzed, besides the dependency on preload [47, 81, 82]. The maximum frequency reached is limited by the robustness of the structure of the test set-up. A very robust test set-up is required to withstand the large loads at high frequencies that often occur in railway tracks, specially at high-speed train lines.

The influence of the temperature is studied by performing the tests under temperature control conditions, such as climate boxes [85].

The number of key factors that are considered when modeling railpads depends on the representation of the fastening system. Different models of the fastening are discussed in the following section.

**Railpad in track models**

In the literature, if the track is modeled as continuously supported, the railpad is represented as a continuous viscoelastic layer [41, 55]. In contrast, if the discreteness of the support is considered, the railpad is mainly represented as one pair of a linear spring and a viscous damper in parallel (see, for instance, [41, 45, 64]).

With the one spring-damper railpad model, studies have been performed to investigate the influence of the railpad parameters on the track dynamics and noise. Soft railpads are favorable for loading sleeper and ballast [88–91], but lowered wheel-rail contact forces is not always noticed [51]. Concerning noise, by reducing the railpad stiffness, the noise radiation from the sleeper is reduced but that from the rail is increased, and vice versa [76]. However, by employing soft railpads, the displacements of the track are larger which accelerates the fatigue of track components. In the case of short pitch corrugation, some studies have demonstrated that the wheel/rail contact forces are significantly influenced by the stiffness of the railpad [41, 51, 77]. Stiffer railpads cause an increase in the corrugation growth [78, 92, 93].
The railpad consists of multiple spring-damper pairs when the rail seat is defined as an area or line instead of one connecting point between the rail and the sleeper. Studies show that considering the longitudinal and/or lateral dimensions of the rail seat significantly influence the dynamic response of the track [44, 94]. For instance, the dominant pin-pin resonance (i.e. when the rail vibrates with the nodes on the sleepers) becomes a significantly smaller peak if the longitudinal dimension of the railpad is considered [94]. Since the track dynamics are affected when the railpad is modeled covering an area, the vehicle-track dynamic response is influenced too. The magnitude and position of the characteristic contact forces significantly change depending on the configuration of the fastening system [59].

For both single and multiple linear springs, only the dependency of the railpad to preload is commonly considered. To obtain the railpad parameters, the railpads are tested under different preload conditions which resemble the toe load applied by the clamps, as in, for example, [48, 81, 82, 95]. Then, the stiffness that corresponds to the toe load of interest is introduced in the track model.

To account for the increase in stiffness with the increase in frequency, the railpad can be modeled as (1) a spring in series with a spring and damper [83] or (2) a spring in parallel to a pair of spring and damper in series [80, 96]. However, these models transform the track into a significantly more complex model because the railpad becomes a non-linear material. Consequently, the solution schemes required, such as time domain integration, are complex (see, for instance, [97]) and therefore, time consuming. This is the reason why these models are barely used.

Regarding the key factor temperature, no investigation has been found in which the influence of this factors on the vertical track dynamics is analyzed. Nevertheless, the principle of the time-temperature superposition for viscoelastic materials could be employed [98, 99]. The physical meaning behind the principle is that the dynamic behavior at low temperature resembles the dynamic behavior at high frequency. Thus, the dynamic behavior at different temperatures could be derived from the dynamic behavior at different frequencies employing, for instance, the Williams-Landel-Ferry (WLF) model [100].

1.7.4. Ballast

The ballast influences the dynamic behavior of the track for frequencies lower than 250 Hz [41, 101], whereas below 50 Hz, the subgrade is the dominant contributor [102]. Modeling the ballast as a halfspace or as a viscoelastic foundation affects the numerically calculated receptance function on the frequencies under 250 Hz [103]. To numerically reproduce measurements in the low frequencies, the preload in the foundation caused by the wheels should be considered [101, 104] and the contact in the sleeper-ballast interaction should be accounted for [68–70, 105].

1.8. The Gaps in Vertical Track Dynamics

The success of identifying characteristic frequencies from measurements and deriving unknown track parameters by fitting simulations to measurements strongly depends on the measurements performed and on the capacity of the model to reproduce measure-
ments. Gaps in vertical track dynamics related to measurements and modeling are discussed in Section 1.8.1 and Section 1.8.2, respectively.

1.8.1. Measurements
Field hammer test measurements are commonly performed at one nominal location. However, only one measurement does not consider sources of variability present in the track, such as differences between sleeper bays. For instance, the clamping force may vary between consecutive supports, so that the railpad stiffness shows differences because the stiffness is preload dependent [48, 81, 82, 95]. Differences in railpad stiffness cause variations in some characteristic frequencies of the dynamic response of the track. Consequently, the track parameters required to fit the simulations to the measurements are different. These differences in characteristic frequencies and identified track parameters belong to the intrinsic variability of the track and are not considered with the current approach of only one nominal measurement.

In addition to the issue of field hammer test variability, another point in question is the limitation on measurement use. From the three main purposes of identifying characteristic frequencies, validating track models and deriving track parameters, field hammer test measurements are mainly used in combination with numerical models to obtain in-service track parameters. The derived stiffness and damping of the railpad and ballast can be used to assess the condition of the track. However, this process has two main drawbacks (1) the accuracy of the results depends on the capacity of the model to reproduce the measurements, and (2) the study is mainly limited to the degradation of railpad and ballast because the models are fitted to the measurements by varying railpad and ballast parameters. If measured signals of field hammer tests were directly used, the fitting process of the simulation to the measurements would be avoided. Furthermore, the study of the degradation would not be limited to the railpad and ballast; the deterioration of the whole track could be investigated.

In summary, there are two main gaps related to measurements. (1) There is lack of knowledge of the variability of field hammer measurements. Thus, the influence of the intrinsic variability of the track in its dynamic response should be investigated. (2) The use of field hammer test measurements is mainly limited to the combination with numerical simulations. However, measurements could contain valuable information on their own and they should be examined without numerical simulations. The studies may provide interesting and useful insights which could be employed in the analysis of track degradation.

1.8.2. Modeling
One of the main factors that determines the modeling of the track is the calculation time. The track components are modeled so that the main track dynamics are considered and the calculation time is minimized. Commonly, the rail is modeled with beam elements, the sleeper with beam or mass elements and the ballast and fastening with spring-damper pairs (for details see Section 1.7). However, two main problems arise from this modeling approach when investigating tracks with monoblock sleepers, whose use is increasing due to their higher bearing capacity in comparison to other sleeper types [68].
First, track models have not been able to reproduce the characteristic track resonances and antiresonances that most probably correspond to the dynamic behavior of monoblock sleepers (see Figure 1.5). Moreover, the frequency range (i.e. 450-1000 Hz) includes frequencies related to track defects, such as squats [11].

Second, the models are limited to frequencies of 1500 Hz because the cross-sectional deformation of the rail is not considered; this deformation significantly influences the track dynamics above 1500 Hz [55]. Attempts have been made to overcome the 1500 Hz limitation by modeling the rail differently (see Section 1.7.1). However, the models can only partially reproduce the track dynamics at frequencies between 1500 and 3000 Hz. This disagreement could be caused by (1) the fastening is modeled at a point or a line instead of in an area, or (2) a constant stiffness of the railpad is used when it is frequency dependent.

In summary, there is a gap in understanding the sleeper-fastening-rail interaction because (1) the dynamics of monoblock sleepers are not yet reproduced and (2) the effects of simplified modeling of fastening systems are unknown. Furthermore, the need to have a more realistic representation of the track components and their interactions has arisen because investigations show that rail surface degradation is related to the condition of the support [59, 77–79, 106]. A better understanding of the interactions between track components and their dynamics is required to gain insight into the deterioration of the track. Moreover, the developments in the computer industry have reduced the calculation times and open the opportunity to more complex track models.

### 1.9. PROBLEM STATEMENT

The analysis of the current knowledge of vertical track dynamics has shown that a better understanding of the vertical track dynamics is required to investigate the deterioration and design of tracks. Thus, our main research question is as follows:

What more can we learn about vertical track dynamics so that the investigation into the deterioration and design of tracks can be tackled on a solid base?

The gaps in the knowledge are to certain degree related to the limitations of the field hammer test measurements and track models available. On the one hand, the measurements do not consider the intrinsic variability of the track. Also, the use of the measurements has been limited to the combination with numerical models so that the results depend on the capacity of the model to reproduce the measurements. On the other hand, the numerical models significantly simplify the modeling of the sleeper-fastening-rail interaction so that the dynamics of monoblock sleepers are not yet reproduced, and the consequences of simplifying the fastening are unknown. The simplification also limits the study of fastening deterioration to the railpad deterioration. Based on this analysis, the main research question is divided into the following sub-questions:

**Q1** How can we account for the intrinsic variability of the track in its measured dynamic response?

**Q2** What and how can we extract more from field hammer test measurements?
Q3 What kind of model considers the dynamic behavior of monoblock sleepers better?

Q4 What kind of model reproduces the vertical track dynamics in a wider frequency range?

Q5 What can we learn from fitting the new model to the field hammer test measurements?

Q6 What are the consequences of simplifying the sleeper-fastening-rail interaction?

1.10. Our approach

We propose to investigate hammer test measurements, simulations, and their combination. On the one hand, sets of field hammer test measurements are examined at healthy and damage conditions to derive characteristic frequencies of damaged tracks. On the other hand, state-of-the-art models for the sleeper-fastening-rail interaction are developed to perform a comprehensive study of fastening modeling. Based on the valuable and insightful knowledge obtained by investigating measurements and simulations separately, numerical simulations are fitted to measurements to complete the study of vertical track dynamics. This combination is essential to validate the models, derive in-service track parameters, and investigate the contribution of components and the effect of simplifications in modeling. The models presented and the insight gained can be used to study the deterioration and design of tracks.

1.10.1. Measurements: Identify Signature Tunes

For considering the variability of tracks (Q1), we propose to define a healthy state based on a set of measurements combined according to a process-monitoring technique called control-charts [107]. In this manner, the baseline state considers small differences in the track structure, unlike the one nominal measurement commonly used.

In addition to the analysis of nominal locations, we propose to carry out hammer tests at deteriorated locations in tracks. The goal is to identify characteristic frequencies of damage states (also called signature tunes) (Q2). This information is employed to detect damaged locations in signals measured with vehicle-borne dynamic-response-based monitoring systems, such as strain-gauge-instrumented wheelsets [18] and ABA systems [19–22]. These systems are the basis for early-state maintenance measures. By frequent or continuous measurements track deterioration can be monitored so that maintenance measures can be taken before the damage becomes irreversible. In this manner, the service life of tracks can be stretched as long as possible under safe conditions.

By employing hammer tests, the damages that could be investigated are in the superstructure level (i.e. rail, fastening system and sleeper), because hammers are generally too light to fully excite the substructure layers (i.e. ballast, subballast and subgrade). Thus, the damages studied could be (1) track geometry problems of short-wave (i.e. less than 80 mm [10]), such as corrugation and squats [11, 41], (2) fastening damage, such as loose bolts or severely degraded railpads [8] and (3) sleeper damage, such as cracked
sleepers or sleeper voids (if the hammer is heavy enough to excite the sleeper-ballast interaction) [6, 35].

In our work, we first investigate one of the weakest components of the track structure: insulated rail joints (IRJs) (Chapter 2). By comparing healthy and damaged IRJs, the capacity of employing hammer tests to identify signature tunes is assessed. In view of the promising results from the IRJs study, we present the application of a statistical method adapted to railway tracks for the purpose of identifying signature tunes of damage conditions (Chapter 3). The method consists of statistically comparing the dynamic response of damaged tracks to healthy (i.e. non-damaged) tracks in the frequency domain (related to Q2).

An alternative way to obtain signature tunes of damage stated is by vehicle-track field measurements. The advantage of vehicle-track field measurements is that real rolling conditions occur during the testing, whereas hammer tests are performed under unloaded conditions (i.e. there is no vehicle in the track). The track loading condition may influence identified signature tunes below 500 Hz [106]. The disadvantage of vehicle-track field measurements is that the tests are expensive, time consuming, complex and sometimes safety threatening (i.e. if damage track conditions are tested), unlike the easy and quick hammer tests. A vehicle-track experimental investigation to evaluate bolt tightness condition at rail joints by an ABA system is presented in Appendix A.

1.10.2. Simulations: 3D finite element models
Based on the analysis of the track models available in the literature (see Section 1.7) and our main focus on investigating the sleeper-fastening-rail interaction, the track components of our model will be represented as follows:

Rail
We choose for Class IV rail representation (i.e. rail represented with its nominal cross-section using FE approach) for three main reasons. First, the Class IV does not suffer frequency limitations due to the nature of the elements employed (i.e. there is no 1500 Hz limitation). In this manner, the new model covers a wider frequency range. Second, non-linear materials and area covering railpads can be defined. This may be required because the disagreement between Class III models and measurements in the high frequency range may be caused by (1) the railpad is modeled at a point or a line instead of in an area [44], or (2) a constant stiffness of the railpad is used when it is frequency dependent [48, 81, 82]. And third, by applying Class IV rail representation, a complete study was performed about squats growth process from prediction [11] to validation [108] so that an early detection maintenance measure could be developed [22]. Therefore, we are encouraged to apply the Class IV rail representation in our model to examine the vertical dynamics of tracks with monoblock sleepers.

Sleeper
We propose to model 3D sleepers with solid elements so that there are no frequency limitations introduced by the nature of the element type, and the dynamic behavior of sleepers is included (Q3). In addition, the modeling of 3D sleepers offers the possibility to study the sleeper-fastening-rail interaction. By combining the 3D rails and 3D sleepers, the rail seat encompasses an area as in the field so that the simplification of the
railpad from an area to a line or single point can be investigated. Furthermore, the finite element approach has the potential to model the fastening realistically (e.g. solid railpads).

**FASTENING**

We propose to examine the sleeper-fastening-rail interaction first with the fastening models found in the literature, and next, by developing more complex systems (Q6). We focus on the two main components of the fastening system: the railpad and the clamps. We investigate the simplification of a solid railpad to discrete elements, and the influence of the lateral and longitudinal dimensions of railpads. Moreover, we study the fixing function of the clamps limiting the relative vertical, lateral and the longitudinal movements of the rail with respect to the sleeper.

**BALLAST**

The main focus of our study is on the sleeper-fastening-railpad interaction, which contributes to the dynamic behavior of the track at frequencies higher than 300 Hz. Thus, we are interested in higher frequencies than the ones influenced by the ballast representation as mentioned in Section 1.7.4. Furthermore, we are developing a model for nominal conditions, which for the sleeper means that a sleeper is placed in a recently tamped ballast in which it is partly buried. The sleeper and the ballast are in contact so that the ballast applies vertical, lateral and longitudinal restrictions on the sleeper. Thus, we model the ballast with multiple linear spring and viscous damper pairs to account for the vertical stiffness of the ballast and the need to define lateral and longitudinal stiffness is investigated. The spring-damper pairs are homogeneously distributed under the sleeper.

**Our 3D FE models**

In summary, we develop 3D FE models (Q4). The rail is modeled with its nominal cross-section with solid elements; the monoblock sleeper is modeled with its nominal geometry with solid elements; the fastening is represented with multiple spring-damper pairs, solid elements, or solid elements and clamps; and, the ballast is modeled with multiple spring-damper pairs.

**1.10.3. FITTING SIMULATIONS TO MEASUREMENTS**

Simulations are fitted to field measurements by varying track parameters (i.e. stiffness and damping of railpad and ballast). The fitting has three main outcomes (Q5). (1) Models are validated. (2) Track parameters that account for the intrinsic variability of the track are derived. (3) Insight is gained into the dynamic behavior of the track, such as the contribution of components and the effect of simplifications in modeling.

**1.11. MAIN CONTRIBUTIONS AND OUTLINE OF THIS DISSERTATION**

We investigate the vertical dynamics of tracks with monoblock sleepers with field hammer test measurements, numerical models and their combination.
From the examination of measurements, contributions are made in these three aspects. First, a method that considers the intrinsic variability of tracks is proposed to define the baseline state of tracks. Second, characteristic frequencies of damaged insulated rail joints are identified. Third, a statistical method to identify the characteristic frequencies of damaged railway tracks is presented. As a feasibility study, the rail surface defect squats are investigated in two ballasted tracks. The identified characteristic frequencies are expected to be useful in the development of early-detection vehicle-borne inspection systems.

The numerical analysis of the 3D FE models results in three contributions. First, insight is gained into the main characteristics that define the dynamic behavior of tracks with monoblock sleepers. Second, a 3D FE model is developed for tracks with monoblock sleepers to derive track parameters. Third, the influence of the fastening representation in the vertical dynamic behavior of tracks is extensively investigated.

The outline of the dissertation is shown in Figure 1.6. In Chapter 2 and 3, field hammer test measurements are examined. In Chapter 2, the characteristic frequencies of damaged insulated rail joints (IRJs) are identified by combining hammer and hardness tests. In this feasibility study, it is assessed if characteristic frequencies of damaged tracks can be identified by comparing hammer tests from healthy and damaged locations. In view of the positive results, in Chapter 3, a statistical method to identify characteristic frequencies of damaged railway tracks is presented and a case study of track at squats is shown. To strengthen the statistical reliability, the repeatability of hammer tests is investigated and the baseline state is defined according to a method that considers the intrinsic variability of tracks. The identified characteristic frequencies of both IRJs and track at squats agree with the frequencies derived with an extensively validated vehicle-borne ABA monitoring system.

In Chapter 4 and 5, the numerical models and the baseline state defined in Chapter 3 are combined by fitting the simulations to the measurements. In Chapter 4, a nominal-geometry-based 3D finite element model is presented. Half- and whole-track models are calibrated by fitting the simulations to the measured baseline state. By comparing the half- and whole-track models and performing numerical modal analysis, the origin of six out of the seven main characteristic of tracks with monoblock sleepers are identified. With this 3D FE model as starting point, in Chapter 5, the influence of the fastening representation is examined by comparing different fastening models to the field measurements. A new 3D FE model that closely reproduces hammer test measurements is identified.

In Chapter 6, the main conclusions are summarized and recommendations for future work are given.

REFERENCES


[2] N. Triepaischajonsak, D. Thompson, C. Jones, J. Ryue, and J. Priest, Ground vi-
 REFERENCES

Ch. 2
Experimental investigation into the condition of insulated rail joints by impact excitation

Ch. 3
Identification of characteristic frequencies of damaged railway tracks using field hammer test measurements

Ch. 4
An investigation into the vertical dynamics of tracks with monoblock sleepers with a 3D finite element model

Ch. 5
An investigation into the modeling of railway fastening

Ch. 6
Conclusions and recommendations

Figure 1.6: Outline of the dissertation


This paper presents a feasibility study to determine if the health condition of Insulated Rail Joints (IRJs) can be assessed by examining their dynamic response to impact excitation. First, a reference dynamic behavior is defined in the frequency domain of 50-1200 Hz based on field hammer test measurements performed on an IRJ without visible damage. Then, measurements on IRJs with different damage states are compared to the reference IRJ response. Three cases of IRJs are analyzed: a IRJ with a broken fastening, a IRJ with a damaged insulation layer and a IRJ with a rail top with plastic deformation. Combining hammer test measurements, hardness measurements and pictures of the IRJs, three frequency bands were identified as characteristic for damaged IRJs. In the identified high frequency band (1000-1150 Hz), the measured dynamic response with both a vehicle-borne health monitoring system and hammer tests shows a clear difference between the damaged IRJs and the reference IRJ. Furthermore, different damage types can be identified by examining the dynamic responses of the investigated IRJs in the identified mid-frequency band (420-600 Hz). Further analysis over a larger number of IRJs may complete and support the significant differences observed between different damaged states so that the information can be employed for the condition assessment and monitoring of IRJs.
2.1. **INTRODUCTION**

Insulated Rail Joints (IRJs) are railway track components where impact and high dynamic wheel-rail contact forces occur due to the rail discontinuity, see an example of an IRJ in Figure 2.1. The frequent high impacts reduce up to five times the service life of IRJs with respect to continuously welded rails [1]. Although other methods to connect rails are more economical to maintain, IRJs are still needed because they are a fundamental component of non-GPS-based positioning systems. In those systems, the railway network is divided into electrically insulated sections defined between IRJs so that sections can be “occupied” by a train (red light) or “free” of trains (green light). Thus, IRJs are safety-critical components because its malfunction can lead to failures of the signaling system and originate accidents. The timely monitoring of the health condition of IRJs is important because about 50% of the track circuit failures are originated at IRJs [2]. Preventive maintenance of IRJs can guarantee safety and reduce the expensive costs of complete renewal of IRJs. For instance, in China, half of the total replacing of rails were IRJs in 2005 [3]. Yet there is still not effective condition monitoring methods for assessment of the dynamic performance of IRJs. The mostly used method is visual inspection, by which it is often impossible to pinpoint early invisible degradations.

![Figure 2.1: Insulated rail joint and its components](image)

Research conducted to understand the wheel-IRJ dynamic interaction can be employed to better understand the degradation of IRJs, so their service life can be extended with a timely maintenance. Most of the studies have approached the problem from the numerical point of view; two of the exceptions are the field trial of different IRJ prototypes [4] and the visual and geometric monitoring of IRJs [2]. Contrary to field measurements, the advantage of modeling is the flexibility to investigate the influence of different parameters under controlled simulations. There are different factors influencing the degradation of IRJs. For instance, regarding the vehicle, higher speed and higher axle loads accelerate the degradation of IRJ [3]. With respect to the track, the misalignment between the rail ends causes higher wheel-rail impact forces [5–8]. Furthermore, the deflection between the two rails should be kept as small as possible so that the wheel-rail contact forces and the stresses in the bonding material between the rails and the plates are as low as possible [9–14]. The properties of the insulating material between the rails, called end-post, also plays a role in the wheel-rail contact stresses close to the discontinuity; a lower Young's modulus of the end-post results in a less uniform pressure distri-
2.2. FIELD MEASUREMENTS AT IRJS

These studies have given an insight into the wheel-IRJ interaction. However, the disadvantage of the numerical approach is that the results are limited to the applicability of the models.

The vehicle/wheel-IRJ/track interaction is a complex system. In most of the existing models, simplifications are made when representing the track, such as not accounting for the contact between rail and plates [3] or simplifying the track to only one rail section [7]. However, little is known about the consequences of these simplifications because few measurements are available for the validation of IRJ models. Even fewer measurements are found for IRJs with different damage states in the literature. Most of the existing studies focus on the initiation of the deterioration [2]. However, useful information can be obtained from the analysis of degraded IRJs. For instance, characteristic frequencies related to the deterioration of IRJs could be derived, which might be employed for assessment and monitoring of the condition of IRJs.

In this paper, an experimental investigation of the vertical dynamic behavior of IRJs is presented. The contributions of this paper are threefold (1) to gain a better insight into the dynamic behavior of IRJs so that the complex vehicle/wheel-IRJ/track system can be tackled on a solid base, (2) to provide a measured nominal state that can be used for validation of models of tracks with IRJs and for assessment of health conditions, and (3) to study the feasibility to assess the condition of IRJs by examining their dynamic response. For this purpose, field hammer test measurements have been performed at different IRJs with two hammer sizes so that a wide frequency range is covered. First, a non-damaged IRJ is investigated and a reference dynamic behavior is defined. Then, IRJs with different damage conditions are compared to the reference case. The investigation is complemented with hardness measurements of the rail top, visual inspection (i.e. pictures) and dynamic-response-based vehicle-borne measurements (i.e. from an Axle Box Acceleration (ABA) system).

2.2. FIELD MEASUREMENTS AT IRJS

2.2.1. CHARACTERIZING THE DYNAMICS OF THE RAILWAY TRACK BY HAMMER TEST MEASUREMENTS

In railway engineering, hammer test measurements have been used for two main purposes. First, the dynamic behavior of the track or track components can be characterized and insight can be gained by examining the frequency response function obtained [16–18]. Second, parameters of the track that are difficult to directly measure can be derived. The parameters that are usually unknown are the stiffness and damping of the railpad, which is the resilient component between the rail and the sleeper, and the stiffness and damping of the ballast, which is the stone bed of the track (see the components in Figure 2.1). The operational and environmental conditions deteriorate these components, resulting in stiffness and damping values different from the nominal values. The in-service parameters of the railpad and ballast are typically derived by numerically reproducing the response of the track to impact excitation (i.e. hammer test) [19–22].

Different hammer and hammer tip types are used depending on the frequency range of interest. The low frequencies are investigated with heavy hammers and soft tips, whereas the high frequencies are analyzed with light hammers and hard tips [22].
this paper, two hammer sizes have been used. For the high frequency, a Bruel & Kjaer 8206 hammer with a hard plastic tip is used [23] (called the small hammer). For the low frequency, a 086D50 ICP impact hammer with a hard plastic tip is employed [24] (called the big hammer). A hard tip is used in the big hammer so that the frequency ranges of the two hammers overlap. As a result, the frequency range is defined between 50 and 3000 Hz [22], which includes the frequencies of interest [9, 12]. A hammer test with the small hammer and one with the big hammer are shown in Figure 2.2a and 2.2b respectively.

![Field hammer test measurements](image)

**Figure 2.2**: Field hammer test measurements with (a) the small hammer and (b) the big hammer.

In the field hammer test measurements presented in this paper, the same procedure was followed for all the tests. A rail section was vertically excited on the top multiple times $F_i(t)$ and the response was measured with an uni-directional accelerometer that was placed on the rail top with a magnet, $a_i(t)$, see a schematic representation of an example in Figure 2.3. The input and output data were recorded with a sampling frequency of 20000 Hz.

The excitation and measurement locations were the same for hammer tests carried out with the small hammer. Four configurations were investigated, which corresponded to the rail above the four bolts. If the bolts are numbered from one to four in the direction of the traffic, the excitation configurations are called P1, P2, P3 and P4 respectively from now on. As an example, the P2 configuration for the impact $i$ is shown schematically in Figure 2.3. When performing the hammer test measurements with the big hammer, the rail could not be both excited and measured above a certain bolt because of the large diameter of the hammer. Therefore, it was decided to measure on the rail top above the bolt of interest and excite the rail between the two bolts of that half of the IRJ. For instance, the rail was excited between the first and the second bolts and the response was measured above the first bolt for the P1 configuration and above the second bolt for the P2 configuration.

The post-processing of the measured data is schematically shown in Figure 2.4. In each configuration, five impacts were applied. By matching the maximum force values of the five impacts (time shift $\tau_i$), the average force and acceleration ($F(t)$ and $a(t)$, re-
Field Measurements at IRJs

2.2. Obtaining Information About the Wheel-Rail Contact Forces from Hardness Measurements

Performing hardness measurements of the rail top is an indirect way of obtaining information about the impact and dynamic contact forces between the wheel and the rail. High wheel-rail contact forces lead to high stresses in the rail, which result in plastic deformation and hardening of the rail top if the stresses are higher than the yield stress of the steel. Thus, a wave pattern in the hardness measurements may indicate a dominant frequency of the wheel-rail interaction.

In the field monitoring presented in this paper, the hardness of the rail top was measured every 5 mm along three lines in the running band, in the cases were the wavelength were not visible. A DynMIC 34247 was used in combination with a self-developed ruler-guide as it shown in Figure 2.5a. An example of the three hardness measurement lines denoted by inner, middle and outer is shown in Figure 2.5b.
2. EXPERIMENTAL INVESTIGATION INTO IRJs

2.2.3. TRACK SITE

The field measurements were performed at a track site of a main line in Weert, the Netherlands. The traffic at the test track site was one-directional with a maximum allowed speed of 140 km/h. In the track, UIC54 rails with 1/40 inclination were supported by NS90 monoblock sleepers every 0.6 m except in the proximity of IRJs. As it is shown in Figure 2.1, the IRJs were supported with sleepers to reduce the deflection of the joint. Furthermore, the wooden sleepers introduce damping to the system, which helps in the absorption of vibrations caused by the impact when a wheel rolls over the discontinuity. A set of different IRJs with the same IRJ (including track) structure and traffic were examined. Over this set, the most representative cases are analyzed in this paper. There

Figure 2.4: Post-processing procedure of the measured data

\[ F(t) = \frac{1}{N} \sum_{i=1}^{N} F_i(t - \tau_i) \]

\[ a(t) = \frac{1}{N} \sum_{i=1}^{N} a_i(t - \tau_i) \]
2.3. Defining the reference IRJ response

To define the reference IRJ response, an IRJ with no damage to the naked eye was examined (see Figure 2.6). The track components did not have visible cracks or defects and the end-post seemed to be in a good condition. Furthermore, the IRJ was properly supported as the stones in the neighborhood of the supporting wooden sleepers were not white. At loosened supports, the sleeper gains freedom of movement in the ballast bed and consequently, the interaction between the stones and the sleeper increases resulting in worn stones and white stone dust.

2.3.1. Hardness tests for the IRJ without visible deterioration

The hardness of the rail top immediately after the discontinuity was measured and it is shown in Figure 2.7. The values of the three hardness measurements considerably differed for the same position along the rail. The scatter of the hardness values is in
agreement with the hardness measurements at the beginning of the monitoring of new IRJs presented in [2]. The presence of a wave pattern in the hardness measurements would indicate that the wheel-rail contact forces are high enough to cause plastic deformation and hardening of the rail top, and consequently, even higher wheel-rail contact forces [12]. Under these conditions, the deterioration of the IRJ would accelerate and the service life would shorten. Thus, an IRJ with a hardness wave pattern on the rail top after the gap means that the IRJ is already captured in an accelerated deterioration process. For the IRJ under study, however, the development of the wave pattern had just started as it is only visible immediately after the discontinuity (see the first 30 mm in Figure 2.7b). Thus, in view of the absence of damage and of a clear wave pattern, this IRJ is considered to be in good condition in terms of rail head deformation.

2.3.2. HAMMER TESTS FOR THE IRJ WITHOUT VISIBLE DETERIORATION
Defining the reference characteristic dynamic behavior of IRJs is not straightforward. Although the structure is symmetric with respect to the discontinuity, the service conditions (i.e. one-direction traffic) may influence the IRJ differentiating the dynamic behavior of the track before and after the discontinuity. As trains always run in the same direction on the track of this case study, only the rail after the discontinuity is subjected to impact forces, which may affect the track components such as the rail, the interaction between the rail and the connecting plates, and the condition of the support where the rail end is supported and the consecutive supports. The symmetry of the dynamic behavior of IRJs with respect to the discontinuity is studied in Section 2.3.2.1.

Furthermore, the IRJ is an heterogeneous structure. The distance between the measured response and the end-post may significantly influence the resulting dynamic response of the track. Also, the presence of the bolt holes may affect the measured vibrations of the IRJ. The effect of the structure of the IRJ on its dynamic behavior is investigated in Section 2.3.2.2.

2.3.2.1. SYMMETRY OF IRJ WITH RESPECT TO THE DISCONTINUITY
The symmetry of the vertical dynamics of IRJs with respect to the discontinuity is studied by comparing measurements of symmetric hammer tests.

The measured receptance functions of the P1 and P4 configurations for the small hammer are shown in Figure 2.8a. The frequency response functions differ in the frequency ranges of 1200 and 1800 Hz, and of 2100 and 2600 Hz. The two measured recep-
2.3. **Defining the Reference IRJ Response**

![Graph](image)

Figure 2.7: Measured hardness after the gap at the IRJ without visible deterioration (a) along the inner (○), middle (♦) and outer lines (×) indicated in Figure 2.5b, (b) the average of the three measurements (△)

According to these observations, the IRJ is not dynamically symmetric at high frequency. The two halves of the IRJ show different dynamic behaviors in the frequency range of 1200-3000 Hz because the passages of wheels over the IRJ already caused small changes in the rail and railpad, which dominate the dynamic response of the track in this frequency range [16, 26]. In summary, to define a reference receptance function based on this IRJ, the reference frequency range should be limited up to 1200 Hz.

The receptance functions with the P2 and P3 configurations disagree in a third frequency range, which is between 750 and 950 Hz, besides the approximated ranges of 1200-1800 Hz and 2100-3000 Hz. The interaction between rail ends, end-post and connecting plates is most probably the influencing factor between 750 and 950 Hz since the difference is only seen between the two configurations close to the discontinuity. The
tance functions of the configurations P2 and P3 show a different behavior between 1200 and 1800 Hz as well (see Figure 2.8b). This suggests that the impact and high wheel-rail contact forces had already caused the initial changes in the IRJ. Regarding the different behavior of the receptance functions in the high frequency range of 2100-2600 Hz in Figure 2.8a, the two curves of Figure 2.8b also show a disagreement that starts at approximately 2100. In this case, the difference is seen up to 3000 Hz.
impact and high contact forces between the wheel and the rail after the discontinuity altered the second half of the IRJ. Thus, the reference receptance functions should be defined based on the measurements performed with the P2 configuration because the first half of the IRJ is most probably closer to the healthy condition of the IRJ.

The receptance functions obtained with the big hammer are shown in Figure 2.9. The receptance functions of the P1 and P4 configurations show the biggest difference between 300 and 450 Hz. In this frequency range, a peak is displayed at 430 Hz for P1, whereas the peak is shown at 350 Hz for P4. These peaks share most probably its origin and the shift from high to lower frequency is related to the sleeper because it is a dominant contributor to this frequency range [16]. When a IRJ is placed in the track, the
ballast is often tamped in the neighborhood of the IRJ so that the IRJ is properly supported. However, the IRJ was starting to change from the nominal condition as it was also seen from the receptance functions obtained with the small hammer. The high impact forces between the wheel and the rail may have loosened the support by making the ballast beneath the sleeper less compact. Consequently, the sleeper has more freedom of movement so that vibration modes that are closely related to the sleeper may shift to lower frequency.

![Graph](image)

**Figure 2.9:** Measured receptance functions after exciting the IRJ without visible deterioration with the big hammer at different configurations: (a) (—) P1 and (—) P4, and (b) (—) P2 and (—) P3

The same peak shift is observed in the measurements of the P2 and P3 configurations (see Figure 2.9b). In this case, the frequency range of influence of this phenomenon is
wider (i.e. 300-550 Hz) than for the P1 and P4 configurations because the response of the rail is measured just on top of the sleeper (see the locations of the bolts with respect to the sleeper in Figure 2.6). For the P1 and P4 configurations, some vibrations of the sleeper are too weak to be captured.

From the analysis of the small and big hammer measurements, it appears that the dynamic behavior of the IRJ is not symmetric with respect to the discontinuity. Although the structure is geometrically symmetric, the fact that the track is one-directional leads to different dynamic behaviors of the two ends of the rail joint. Nevertheless, in the frequency range of 50-1200 Hz the receptance functions of the symmetric excitation configurations agree or the slight differences are understood (see Figures 2.8 and 2.9). The reference receptance functions should be defined based on the measurements carried out on the first half of the IRJ (i.e. P1 and P2 configurations) because the impact should have a smaller effect on the components of the first half than on the ones of the second half. A reference IRJ is assumed to be symmetric geometrically and dynamically, this is, the P1 configuration is equivalent to the P4 configuration, whereas the P2 configuration is equivalent to the P3 configuration. For the reference case, the configurations are denoted by the letter N instead of P. Thus, from now on, the N3 and N4 will be used as the reference configurations based on the P2 and P1 configurations, respectively.

To build the reference receptance functions, the measurements of the two hammers are combined by considering the receptance function obtained with the big hammer for lower frequencies and the one with the small hammer for higher frequencies. The dividing frequency corresponds to the frequency at which the two curves meet, which is in the frequency range of 400-500 Hz. The resulting reference receptance functions are shown in Figure 2.10.

Figure 2.10: Measured receptance functions of the reference IRJ built by combining the big and small hammer test measurements at (——) the N3 configuration and (—) the N4 configuration
2.3.2.2. Influence of the Structure of the IRJ

The influence of the structure of the IRJ on the measured dynamics is investigated by comparing the measured receptance functions of the N3 and N4 configurations. The reference IRJ response is practically identical for frequencies lower than 500 Hz as shown in Figure 2.10. The subgrade, ballast and sleeper are the track components that dominate the dynamic behavior of the track at frequencies lower than 500 Hz [16, 27]. At higher frequencies, the rail and railpad become the key track components that define the dynamic response of the track [16, 26]. The two configurations have a different distance to the rail end, the plates ends and the support. Further, the rail contains bolt holes. Thus, the different frequency response functions at frequencies higher than 500 Hz are expected in Figure 2.10.

Therefore, contrary to what was suggested in [6], only one frequency response function is not sufficient to characterize the dynamic behavior of the reference IRJ because above 500 Hz, the dynamic response depends on the excitation configuration. This is also the case in a discretely supported track, which depends on the excitation configuration when defining its characteristic behavior [19–21]. Typically, two characteristic curves are defined and are known as the on-support and mid-span frequency response functions. Numerically reproducing the on-support and mid-span curves is a widespread method to validate track models and to derive track parameters by fitting simulations to the measurements. The same concept could be applied for IRJs; this is, the N3 and N4 curves shown in Figure 2.10 could be used for the validation of a reference IRJ model and for deriving the parameters in the proximity of a reference IRJ.

2.4. Damaged IRJs

Apart from the IRJ without visible deterioration, different IRJs were examined in the track site located in Weert, The Netherlands. For the analysis, three damaged IRJs were considered. In two of the IRJs, damage on some of the track components were visible (Figures 2.11 and 2.14). In the third IRJ, the only sign of deterioration was the plastic deformation of the rail top after the discontinuity (Figure 2.17).

2.4.1. IRJ with a Broken Fastening

The first damaged IRJ examined had a fastening broken in the supporting sleeper after the discontinuity, as it is shown in the right close-up of Figure 2.11. A broken fastening means a relaxation of the restraints on the rail. The looseness of the end-post was also visible at the rail foot. The end-post was slightly thinner than the gap between the rail ends and it was popping out (see left close-up of Figure 2.11).

2.4.1.1. Hardness Measurement

The presence of a wave pattern was found by the hardness measurements, which are shown in Figure 2.12. Contrary to the reference case (Figure 2.7a), almost the same hardness value was measured in the first 40 mm immediately after the discontinuity along the three measurement lines, see Figure 2.12a. The very good agreement between the three measurement lines for the first wave is caused by the impact immediately after the gap which widened the running band. The average of the three hardness measurements
shows peaks at 20 mm and 55 mm, so that the wavelength of the wave pattern is 35 mm (see Figure 2.12b).

At a further distance, the agreement between the three measurements is less evident because the running band is narrower, which means the stresses drop quickly out of its center. The hardness measurement at the outer line shows the wave pattern the clearest (see × in Figure 2.12a). Between 20 and 140 mm, the wavelength varies between 35 to 45 mm. The scatter of the wavelength should mainly be due to the 5 mm interval of the hardness measurement. This wavelength is similar to those of corrugation [21] and of the wave pattern caused by squats [28] in the Dutch network.

At 140 mm, the three measurements show almost the same hardness value, which possibly indicates the effect of the broken fastening at 140 mm. This change in stiffness in the structure of the IRJ caused high wheel-rail contact forces that resulted in high stresses and consequently, plastic deformation and hardening of the rail top.

The comparison of the average hardness profiles of the IRJ without visible deterioration with that of the IRJ with the broken fastening (see Figure 2.12b) shows that all the hardness values are below 300 HB for the IRJ without visible deterioration, whereas almost all the hardness values are above 300 HB for the IRJ with the broken fastening. Based on the difference in the average hardness, one may assume that the IRJ without visible deterioration is relatively new with respect to the IRJ with the broken fastening.

The fluctuation of hardness value is also significantly larger for the IRJ with the broken fastening than for the IRJ without visible damage; the difference between the largest and smallest hardness values are approximately 35 HB and 15 HB respectively. A larger varying hardness value means larger stresses on the rail top, which were caused by larger dynamic wheel-rail contact forces. The increase in the dynamic contact forces were induced by the deterioration of the IRJ, such as the wave pattern on the rail top and the broken fastening.

2.4.1.2. HAMMER MEASUREMENT
To further investigate the dynamics of the IRJ, the hammer test measurements were analyzed. The reference IRJ response and the response of the IRJ with the broken fastening are shown in Figure 2.13. The two frequency response functions show two main differences if the rail top is excited and measured according to the N3 configuration (see
Figure 2.12: Measured hardness after the discontinuity at the IRJ with the broken fastening (a) along the inner (o), middle (+) and outer lines (x) indicated in Figure 2.5b, (b) the average of the three measurements (△) and the average of the IRJ without visible deterioration (□) from Figure 2.7b

First, the damaged IRJ has a peak at 270 Hz (B1) that the reference IRJ response does not show. This change in the receptance function is most probably related to the sleeper, which is one of the dominant components in defining the track response at frequencies lower than 500 Hz [16, 27]. The interaction between the sleeper and the rail changed due to the broken fastening so that additional vibration modes that involve rail-sleeper interaction may occur.

Second, a dominant peak is observed at 1040 Hz for the IRJ with the broken fastening that is not observed in the reference IRJ response (see B2 in Figure 2.13a). This frequency corresponds to the pin-pin resonance of the test track site; in this mode the rail vibrates with its nodes on the supports. It is believed that the occurrence of the wide spread rail defect short pitch corrugation is directly related to the pin-pin phenomenon [29]. In the case of the IRJ with the broken fastening, this frequency may also be related to the occurrence of the wave pattern on the rail top. The wavelength of the measured hardness profile is between 35 and 45 mm (Figure 2.12b), which combined with the nominal train speed of 140 km/h, results in vibrations at the frequency range of 865-1111 Hz.
Figure 2.13: Measured receptance functions (---) at the reference IRJ and (—) at the IRJ with the broken fastening for (a) the N3 configuration and (b) the N4 configuration.

The receptance functions measured with the N4 configuration also show the differences at 270 and 1040 Hz (see B1 and B2 in Figure 2.13b). For the N4 configuration, a peak at approximately 570 Hz gain prominence if the fastening after the discontinuity is broken (compare B3 and A3 in Figure 2.13b). This type of damage affects the interaction between the sleeper and the rail, which is one of the dominant phenomena defining the dynamic response of the track in the medium frequency range [16, 26].
2.4.2. IRJ WITH A DAMAGED INSULATION LAYER

The second damaged IRJ examined showed a gap between the rail ends that is significantly wider than the width of the end post as it is shown in the close-up of the left in Figure 2.14. Furthermore, the fastening before the discontinuity was broken as it is shown in the close-up of the right in Figure 2.14. Notice that in this case the traffic is from right to left. Also, the ballast bed in the proximity of the IRJ showed a white color, indicating a loosened support.

IRJs are often prefabricated but this IRJ was most probably assembled in the site. Some signs of the site-assembling were the tape that was applied at the end of the plates to keep the insulating material between the rail and the plates, and the extra washers on the bolts to improve the tightening of the bolts to the plates. Moreover, extra holes were made in the plates to fit to the existing holes in the rail. These indicate that the IRJ assembly underwent a major maintenance. However, this maintenance action did not stop the degradation of the IRJ and the whole IRJ should soon be replaced to guarantee the electrical insulation between the track sections.

2.4.2.1. HARDNESS MEASUREMENT

The hardness measurements on the rail top are shown in Figure 2.15. In this case, the measurements were performed over 100 mm along the rail. The comparison of these hardness profiles with the ones of the IRJ with the broken fastening in Section 2.4.1 shows that the difference between the largest and smallest hardness values is unexpectedly considerably smaller at the IRJ with the damaged insulation layer than at the IRJ with the broken fastening, 20 HB and 30 HB respectively (see Figure 2.15b). A similar or larger hardness difference was expected because the damage is more severe at the IRJ with the damaged insulation layer than at the IRJ with the broken fastening and consequently, the deflection between the rail ends is larger leading to larger wheel-rail contact forces and therefore, to more plastic deformation and hardness on the rail top. However, the hardness fluctuation at the IRJ with the damaged insulation layer is 33% smaller. This difference may be explained by the grinding marks on the rail top. The fact that the grinding marks were still visible and the measured wave pattern showed the typical wavelength after grinding of 10-20 mm [29] (see Figure 2.15b), indicate that the rail top was ground some time near the measurement campaign. Thus, a layer of the rail top was
removed clearing away hardened material.

![Graph](image-url)

Figure 2.15: Measured hardness after the discontinuity at the IRJ with damaged insulation layer (a) along the inner (○), middle (×) and outer lines (×) indicated in Figure 2.5b, (b) the average of the three measurements (△), and the average at the IRJ with the broken fastening (□) from Figure 2.12b

2.4.2.2. HAMMER MEASUREMENT

The responses of the damaged and the reference IRJs were compared between 50 and 1200 Hz. The receptance functions for the N3 and N4 configurations are shown in Figure 2.16a and 2.16b respectively. The peak that occurs at 195 Hz for the reference IRJ (A4) moves to 255 Hz for the damaged IRJ (C4). However, a shift to lower frequencies was expected, such as the shift of the receptance function in the frequency range of 350-450 Hz, because a loosened support was identified due to the presence of white stones in the track. Below 500 Hz, the ballast, sleeper and their interaction mainly determine the dynamic response of the track [16, 27] and a gain in freedom usually results in lowering the characteristic frequencies, such as the bending modes of sleepers [30]. However, the opposite happens for the peak at 195 Hz; investigating the origin of this frequency shift is part of future research.

The frequency response function of the damaged IRJ shows a significant increase in the receptance value at frequencies higher than 500 Hz for the N3 configuration (see Fig-
2.4. **DAMAGED IRJs**

This means that the rail vibrates with larger amplitudes at these frequencies for the same input force. Thus, one should expect larger wheel-rail vibrations immediately after the gap when a wheel rolls over a IRJ with a damaged insulation layer than over a good IRJ.

Furthermore, the peak at 1055 Hz becomes dominant for the damaged IRJ (see C2 in Figure 2.16a). Contrary to the IRJ with a broken fastening, the peak is only seen in one of the two receptances (there is no C2 in Figure 2.16b). This may be because the wave pattern is starting to develop from the gap excited by the wheel-rail impact and with the wavelength being determined by the eigen characteristics of the track. In the IRJ with the broken fastening, however, the wave pattern was already developed so that the B2 vibra-
tion was strong enough to be visible in both N3 and N4 configurations, (see Figure 2.13a and 2.13b, respectively). For the IRJ with the damaged insulation layer, the vibrations at approximately 1055 Hz can be measured close to the gap with the N3 configurations, but the N4 configuration is too far, and therefore the vibrations are too weak to be captured.

The receptance function of the IRJ with the damaged insulation layer shares one characteristics with the IRJ with the broken fastening for the N4 configuration. The frequency response functions show a peak at approximately 570 Hz (see B3 and C3 in Figures 2.13b and 2.16b respectively). This feature has larger receptance values for the two damaged IRJs, than for the reference IRJ response.

2.4.3. IRJ with plasticity

The third damaged IRJ examined is shown in Figure 2.17. Neither the track components nor the end-post seemed to be damaged to the naked eye. The sleepers were properly supported by the ballast as the stones in the proximity of the IRJ were not white. The only deterioration sign was the plasticity of the rail top after the gap (see the close-up in Figure 2.17).

![Figure 2.17: An insulated rail joint with plasticity](image)

2.4.3.1. Hammer measurement

As the wave length was clearly visible on the rail top, hardness measurements were not necessary for this IRJ. Hammer test measurements, however, were carried out. The measured receptance functions are shown in Figure 2.18. As in the case of the IRJ with the damaged insulation layer, the peak of the reference IRJ at approximately 195 Hz moves to a higher frequency of 255 Hz for the damaged IRJ (see A4 and D4 respectively in Figure 2.18). The origin of the shift will be examined in future studies as for C4 in Figure 2.16. The vibration at 255 Hz reaches larger amplitudes for the IRJ with the damaged insulation layer than for the IRJ with plasticity since the receptance value is larger for the former than for the later (see C4 in Figure 2.16a and D4 in Figure 2.18a respectively). The difference in the receptance value may be related to the deterioration state of the support. The deterioration of the support may also have already started for the IRJ with plasticity, even though the interaction between sleepers and stones is still relatively small.
so that not like the IRJ with the damaged insulation layer, here no white stones are observed. The significant change D4 in the receptance function suggests that monitoring systems that are based on analyzing the dynamic behavior of the track might be able to detect the deterioration of the ballast in the proximity of the supporting sleepers before it is visible.

Figure 2.18: Measured receptance functions (---) at the reference IRJ and (---) at the IRJ with plasticity for (a) the N3 configuration and (b) the N4 configuration

In Figure 2.18, the damaged IRJ shows a distinctive peak at approximately 1010 Hz (D2), which is close to the frequency of the vibrations that rise when a train runs at a nominal speed of 140 km/h over the approximate wavelength of 34 mm derived from the photos (see the close-up in Figure 2.17). The peak is observed in the receptance functions for the N3 and N4 configurations, similar to the IRJ with the broken fastening
(see Figure 2.13). The peaks B2 and D2 occur at approximately the pin-pin resonance frequency of the track. The small frequency difference between the two damaged IRJs is explained by analyzing the track parameters that influence the pin-pin mode. The pin-pin phenomenon is a consequence of the discrete nature of the track support and therefore, it is closely related to the support distance. However, investigations have shown that the pin-pin resonance also changes its frequency, although slightly, depending on the stiffness value of the railpad [30]. Thus, the reason why the frequency of the peaks of the two damaged IRJs do not match precisely is most probably related to the condition of the railpad. In the absence of information about the “age” of the IRJs examined, the railpads may not have deteriorated exactly in the same way resulting in a small difference in the pin-pin frequency.

The receptance values at the pin-pin peak are smaller for the IRJ with plasticity than for the IRJ with the broken fastening (see B2 (1040 Hz) in Figure 2.13 and D2 (1010 Hz) in Figure 2.18 respectively). This means that the vibration at approximately 1040 Hz is larger for the IRJ with the broken fastening. This might be the consequence or the cause of the breaking of the fastening. Since the IRJ with plasticity also shows a high frequency peak and its fastenings were still in good condition, one could assume that the broken fastening should be a consequence of the phenomenon occurring at 1040 Hz. One could thus adventure to predict damaged fastenings in the IRJ with plasticity in the future. One might even predict how soon the fastening would break by observing the closeness of D2 to B2 in magnitude and frequency.

The receptance function of the damaged IRJ in Figure 2.18b does not show the prominent peak at 570 Hz as the other two damaged IRJs (see B3 and C3 in Figures 2.13b and 2.16b respectively). However, the receptance function shows a dip (D5) at approximately 445 Hz in Figure 2.18a. The receptance functions of the IRJ without visual deterioration showed this same different behavior in the frequency range of 300-550Hz between the receptance functions measured before and after the gap (see Figure 2.9). This difference is most probably the result of the initial changes in the structure of the IRJs due to the wheel-rail impacts.

2.5. DISCUSSION

2.5.1. REFERENCE IRJ RESPONSE VERSUS DAMAGED IRJ RESPONSE

Independent of the type of damage, the damaged IRJs have three characteristic frequency bands in common in the frequency range of 50-1200 Hz.

180-320 Hz: This frequency band is dominated by the condition of the support in the ballast [16, 27]. In the proximity of IRJs, the supports, especially the sleepers close to the gap, transmit to the ballast the vibrations due to the impact, a phenomenon that bears some similarity with Jenkins et al. [9]. Based on the field observations, the ballast was loosened at the proximity of some of the damaged IRJs allowing extra freedom of movement of the sleepers, which accelerated the degradation of the ballast. Unexpectedly, a characteristic peak occurred at higher frequencies for damaged IRJs than for the reference IRJ (see C4 and D4 with respect to A4 in Figures 2.16 and 2.18 respectively). The origin of this frequency shift will be investigated in future studies. The presence of a broken fastening also influenced this frequency band because the connection of the
Table 2.1: Characteristics of the receptance function of the damaged IRJs in the identified frequency bands

<table>
<thead>
<tr>
<th>Damage types of IRJs</th>
<th>180-320 Hz</th>
<th>420-600 Hz</th>
<th>1000-1150 Hz</th>
</tr>
</thead>
<tbody>
<tr>
<td>Without visual deterioration</td>
<td>Peak at 195 Hz</td>
<td>Dip at 440 Hz</td>
<td>-</td>
</tr>
<tr>
<td>Broken fastening</td>
<td>Peaks at 195 Hz and 270 Hz</td>
<td>Larger peak at 570 Hz (B3 in Figure 2.13b)</td>
<td>Peak at 1040 Hz (B2 in Figure 2.13)</td>
</tr>
<tr>
<td>Damaged insulation layer</td>
<td>Shift peak from 195 to 255 Hz (C4 in Figure 2.16)</td>
<td>Larger peak at 570 Hz (C3 in Figure 2.16b)</td>
<td>Peak at 1055 Hz (C2 in Figure 2.16a)</td>
</tr>
<tr>
<td>Plastic deformation</td>
<td>Shift peak from 195 to 255 Hz (D4 in Figure 2.18)</td>
<td>Dip at 440 Hz (D5 in Figure 2.18b)</td>
<td>Peak at 1010 Hz (D2 in Figure 2.18)</td>
</tr>
</tbody>
</table>
sleeper with the rail affects the dynamic behavior of the sleeper [30, 31]; a new peak was observed at 270 Hz in the measured receptance functions (see B1 in Figure 2.13).

**420-600 Hz:** Between 420 and 600 Hz, a characteristic (A3) is noticed in the receptance function with the hammer test configuration in which the top of the rail is excited above the fourth bolt. Since the peak is observed when the rail is excited close to the end of the plates, one may assume that the phenomenon is related to the interaction between the rail and the plates. Further investigation of the origin of these characteristics is part of future work. The resonance reaches higher values for damaged IRJs (see B3 and C3 with respect to A3 in Figures 2.13b and 2.16b respectively). Another damaged IRJ, with a less severe deterioration, shows a predominant dip instead of a peak in this frequency band (see D5 in Figure 2.18b).

**1000-1150 Hz:** In this frequency band, the receptance functions of the damaged IRJs show a dominant peak which is clearly visible if the rail is excited above the third bolt (see B2, C2 and D2 in Figures 2.13a, 2.16a and 2.18a respectively). For some damage types, the peak is also visible when the rail is excited above the fourth bolt (see B2 and D2 in Figures 2.13b and 2.18b respectively). In these cases, the frequency of the peak is approximately the pin-pin resonance frequency of the track site. Furthermore, the wavelengths on the rail tops measured with hardness tests or pictures corresponds to vibrations in this frequency range for the nominal train speed of 140 km/h.

The main characteristic of the measured receptance functions of the damaged IRJs in the three frequency bands are summarized in Table 2.1. This analysis shows that it seems feasible to detect some damage conditions of IRJs by analyzing their frequency response. Furthermore, the comparison between different damaged IRJs suggests that monitoring the prominence of the dominant peak in the frequency range 1000-1150 Hz may help to predict the condition of IRJs.

In Section 2.5.2, it is investigated if the information obtained from the hammer test can be employed for the assessment and monitoring of the condition of IRJs with vehicle-borne monitoring systems which are based on analyzing the dynamic behavior, such as axle box acceleration systems [32–35] and strain-gauge-instrumented wheelsets [36].

### 2.5.2. COMPARISON TO A VEHICLE-BORNE MONITORING SYSTEM

#### 2.5.2.1. THE AXLE BOX ACCELERATION (ABA) SYSTEM

ABA systems consist of accelerometers mounted on axle boxes, a GPS antenna for positioning and the recording of the vehicle speed [32–35]. The ABA system is a monitoring system that measures the response of the axle boxes to changes in the wheel-track structure. Depending on the origin of the response, different characteristic frequencies (also called signature tunes) can be found in the measured signals. Thus, by post-processing the measured accelerations in combination with the position and the vehicle speed, defects can be identified and their deterioration state can be assessed if the characteristic frequencies are known.

For instance, squats, which are a short wave defect on the rail top, can be automatically detected at an early stage by employing this method [37]. The automatic detection of track defects can be performed by using scale averaged wavelet power (SAWP). The SAWP measures the localized variations of wavelet power spectrum in a certain frequency bands and it is defined as the weighted sum of wavelet power spectrum over
2.5. Discussion

scales $s_{j_1}$ to $s_{j_2}$ [38]:

$$W_n^2 = \frac{\delta j \delta t}{C_\delta} \sum_{j=j_1}^{j_2} \left| W_n(s_j) \right|^2 \tag{2.3}$$

where $\left| W_n(s_j) \right|^2$ is the wavelet power spectrum, $\delta j$ is a scale step, $\delta t$ is time step, $C_\delta$ is an empirically derived constant for each wavelet function.

2.5.2.2. Potential to Monitor IRJs

ABA systems can be a suitable monitoring system for IRJs because impact between the wheel and the rail occur due to the presence of the discontinuity. Furthermore, ABA systems would examine the dynamic behavior of IRJs which changes depending on their damage state, as it shown in Section 2.4.

To develop a dynamic-response-based vehicle-borne monitoring system for IRJs, characteristic frequencies of damaged IRJ are required. For this purpose, the information obtained in this paper could be used and further completed, if the difference in the loading of the track (i.e. presence of a vehicle in the track) between the hammer test measurements and the ABA system does not influence the frequency bands related to damaged IRJs. The loading condition of the track may affect the identified characteristic frequency bands because some track components behave non-linearly when the track is loaded [39]. Thus, in the following section it is investigated if the frequency bands identified with the hammer test measurements can be employed to examine ABA signals for monitoring of IRJs.

2.5.2.3. Measured ABA Signals at the Damaged IRJs

The four IRJs investigated were examined with an extensively validated ABA system [37]. The SAWP of the measured vertical ABA signals was calculated for the three frequency bands of 180-320 Hz, 420-600 Hz and 1000-1150 Hz that were identified in the analysis of hammer test measurements. The SAWP values are shown in Figure 2.19a, 2.19b and 2.19c respectively. Each figure displays the SAWP of the ABA measurements of the four investigated IRJs in this paper. The distance shown covers the length of the plates, from 340 mm to 340 mm, where 0 mm is the center of the end-post. Although the gap of the IRJ shown in the picture in Figure 2.19 is centered with respect to the supports, the IRJ with the broken fastening and the IRJ with the damaged insulation layer were not centered (see Figures 2.11 and 2.14, respectively). For each damaged IRJ investigated, the maximum SAWP values of each frequency band are summarized in Table 2.2.

The largest difference between the damaged IRJs and the IRJ without visible deterioration is found in the frequency band of 1000-1150 Hz (Figure 2.19c). Whereas the ABA measurements at the IRJ without visible deterioration have SAWP values lower than 0.05, the three damaged IRJ show local maxima larger than 0.15. This significant difference in maximum SAWP value is related to the presence of a dominant peak at approximately 1040 Hz in the receptance function of the damaged IRJs (see B2, C2 and D2 in Figures 2.13a, 2.16a and 2.18a respectively).

When analyzing the SAWP values between 420 and 600 Hz (Figure 2.19b), the difference between the damaged IRJs and the IRJ without visible damage lays on the location of the local maximum. The IRJ without visible damage shows the maximum after the
Figure 2.19: SAWP of the ABA signals along the IRJs: (—) the IRJ with no visible deterioration, (—) the IRJ with the broken fastening, (—) the IRJ with the damaged insulation layer and (—) the IRJ with plasticity, in the frequency bands of (a) 180-320 Hz, (b) 420-600 Hz and (c) 1000-1150 Hz.
2.5. Discussion

Table 2.2: Maximum SAWP values of ABA measurements over the damaged IRJs in the identified frequency bands

<table>
<thead>
<tr>
<th>Damage types of IRJs</th>
<th>180-320 Hz</th>
<th>420-600 Hz</th>
<th>1000-1150</th>
</tr>
</thead>
<tbody>
<tr>
<td>Without visual deterioration</td>
<td>≈0.2</td>
<td>&gt;0.9</td>
<td>&lt;0.05</td>
</tr>
<tr>
<td>A broken fastening</td>
<td>≈0.2</td>
<td>&lt;0.8</td>
<td>&gt;0.15</td>
</tr>
<tr>
<td>Damaged insulation layer</td>
<td>&lt;0.1</td>
<td>&lt;0.8</td>
<td>&gt;0.15</td>
</tr>
<tr>
<td>Plastic deformation</td>
<td>&gt;0.5</td>
<td>&gt;0.9</td>
<td>&gt;0.15</td>
</tr>
</tbody>
</table>

support (190 mm). The damaged IRJs, however, display the peak at approximately the discontinuity (0 mm).

To compare the SAWP values to the receptance functions, one should keep in mind that the reference IRJ response was defined based on the first half of an IRJ without visible deterioration. The second half of the IRJ without visible deterioration showed a different behavior with respect to the reference IRJ response in the frequency band of 420-600 Hz (see Section 2.3.2.1). By comparing the maximum SAWP values and the main characteristics of the receptance functions of the IRJs in this frequency range (see the third columns of Tables 2.1 and 2.2), the IRJs can be divided into two groups. In the first group, the IRJ without visible damage and the IRJ with plasticity show a dip at approximately 445 Hz and a maximum SAWP value larger than 0.9. In the second group, the IRJ with a broken fastening and the IRJ with a damaged insulation layer displayed a larger peak at 570 Hz in their receptance functions and a maximum SAWP value lower than 0.8.

The IRJ with damaged insulation layer shows significantly low SAWP values and at difference frequencies with respect to the other damaged IRJs, also in the frequency band of 180-320 Hz. The different dynamic behavior was also noticed when analyzing the hammer test measurements and it is most probably caused by the difference in structure. The condition of the isolating material may be of significant importance because it may not bond the rail and the plates.

Regarding the frequency band of 180-320 Hz (Figure 2.19a), no trend was found between the different damage states and the maximum SAWP values. The IRJ with plasticity reaches the largest SAWP maximum, whereas the IRJ with damaged insulation layer shows the smallest SAWP maximum. Furthermore, there was no clear relation between the characteristics of the receptance function and the SAWP values (see the second columns of Tables 2.1 and 2.2). This absence of direct relation between the characteristics identified with the hammer tests and the SAWP of the ABA signals may be caused by the load condition of the track because, in the identified frequency band of 180-320 Hz, the ballast and the subgrade are the dominant track components [16, 27], which are excited by a train running over the IRJs, whereas the hammers are too light to fully excite them.

In summary, the SAWP values of the ABA measurements calculated in the frequency ranges identified from the hammer test measurements give highly useful information about (1) the relation between hammer test measurements and ABA signals and (2) the condition of IRJ. Table 2.3 summarizes the information that can be employed in the
monitoring, prediction and prognosis of IRJ based on the presented analysis. First of all, the SAWP value in the frequency band of 1000-1150 Hz could be used to detect damaged IRJs. Second, for identifying the damage state, the SAWP signal in the frequency band of 420-600 Hz could be employed by analyzing both the value and the location of the local maximum. It indicates that different damage types may be identified by looking at their respective characteristics of SAWP and positions along the IRJs in the different frequency bands. Investigating the manner to combine such information and pin-point the damage types and state is part of future work.

Table 2.3: Relations between characteristics of the receptance function, maximum SAWP values and types of damage of IRJs

<table>
<thead>
<tr>
<th>Frequency band</th>
<th>Hammer tests</th>
<th>ABA</th>
<th>Possible damage type of IRJs</th>
</tr>
</thead>
<tbody>
<tr>
<td>420-600 Hz</td>
<td>dip at 440 Hz</td>
<td>&gt;0.9</td>
<td>light deterioration (e.g. plasticity on the rail top)</td>
</tr>
<tr>
<td></td>
<td>large peak</td>
<td>&lt;0.8</td>
<td>severe deterioration (e.g. broken fastening damage insulation layer)</td>
</tr>
<tr>
<td></td>
<td>at 570 Hz</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1000-1150 Hz</td>
<td>peak at approx. 1040 Hz</td>
<td>&gt;0.15</td>
<td>damaged (i.e. not-healthy)</td>
</tr>
</tbody>
</table>

2.6. CONCLUSIONS

A feasibility study is presented to assess if the health condition of Insulated Rail Joints (IRJs) can be determined by analyzing their response to impact excitation. The impact excitation is performed with hammers of small and large size and by a wheel of trains hitting the rail at the gap. The response was picked up by accelerometers on the rail head for the former case, and by accelerometers on the axle box of the train for the latter.

The condition of IRJs was investigated by comparing the vertical dynamic behavior of damaged IRJs to a reference IRJ response. For this purpose, hammer test measurements were performed at IRJs of different damage states. Moreover, the hardness of the rail top immediately after the gap was measured, photos were taken and vehicle-borne acceleration measurements were performed.

First, a IRJ without visible deterioration was studied. The analysis showed that the vertical dynamic behavior of the IRJ was not symmetrical to the discontinuity because the one-directional traffic changed considerably the receptance function at high frequency. For the case studied, the frequency range in which the frequency response was almost not changed and therefore can be used as reference, was identified between 50 and 1200 Hz, which was built by combining the measurements performed with a small and a large hammer. This study also showed that IRJs should be dynamically represented with two receptance functions obtained from exciting the rail above the third bolt and above the fourth bolt. By fitting numerical models to the measured reference curves,
2.6. Conclusions

Numerical IRJ models can be validated and parameters of the track in the proximity of a reference IRJ can be derived.

Second, three IRJs at different damage states were examined. By comparison to the reference IRJ response, three characteristic frequency bands related to the damaged IRJs were derived independent of the type of damage. At the lowest frequency range, the condition of the sleeper in the ballast significantly influences the receptance function. At the mid-frequency range, two characteristics are identified related to damaged IRJs. At the highest frequency range, a prominent characteristic in the receptance function appears; this phenomenon fixes the dominant wavelength of the wave pattern on the rail top. In the IRJs investigated, the three frequency bands are 180-320 Hz, 420-600 Hz and 1000-1150 Hz.

The feasibility study was completed by employing the frequency ranges derived from the hammer test to examine vehicle-borne axle box acceleration measurements which were performed at the same IRJs as for the hammer test. By comparing the scale-averaged wavelet powers (SAWP) of the measured signals, it was found that the damaged IRJs differed significantly from the reference IRJ in the identified high frequency range. This difference in the maximum SAWP value was related to the presence of the prominent characteristic at approximately the pin-pin resonance in the receptance function of damaged IRJs. In the identified mid-frequency band, the differences in SAWP observed between the damaged IRJs matched the presence of either one or the other of the two characteristics identified in the receptance functions of the hammer tests so that this information may be used to assess the type of damage. In the identified low frequency band, the relation between the SAWP values and the characteristics of the receptance functions was not straightforward. The loading condition of the track changes in a significant manner the response of the track measured with hammer test and with the ABA system so that further investigation is required to relate the responses of the two measurement systems in the low frequency band.

In summary, the presented work shows the potential to monitor and assess the condition of IRJs based on the analysis of the response to impact excited by hammers at track side or by wheels of trains hitting the gap of IRJs. The results of the study were threefold: (1) damaged IRJs can be detected, (2) different types of damage may be identified by analyzing the response of the IRJ in the characteristic frequency bands and, (3) the monitoring of the dynamic response of IRJs may lead to the diagnosis and prognosis of their condition.

To complete and support the results, the extension of the current study to a larger set of IRJs is part of future work. In addition, further investigation into the vibration modes occurring in the three characteristic frequency ranges and the contribution of the track components to these modes may be of interest to pin-point locations and types of damage. To obtain a better understanding of the dynamic behavior of IRJs, measurements and numerical simulations could be combined. This information may be useful for improved design and development of IRJs as well as for the development of monitoring systems and maintenance measures.
REFERENCES


In this paper, the feasibility of the Frequency Response Function (FRF)-based statistical method to identify the characteristic frequencies of railway track defects is studied. The method compares a damaged track state to a healthy state based on non-destructive field hammer test measurements. First, a study is carried out to investigate the repeatability of hammer tests in railway tracks. By changing the excitation and measurement locations it is shown that the variability introduced by the test process is negligible. Second, following the concepts of control charts employed in process monitoring, a method to define an approximate healthy state is introduced by using hammer test measurements at locations without visual damage. Then, the feasibility study includes an investigation into squats (i.e. a major type of rail surface defect) of varying severity. The identified frequency ranges related to squats agree with those found in an extensively validated vehicle-borne detection system. Therefore, the FRF-based statistical method in combination with the non-destructive hammer test measurements has the potential to be employed to identify the characteristic frequencies of damaged conditions in railway tracks in the frequency range of 300-3000 Hz.

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3.1. INTRODUCTION

Extending the service life of railway tracks is a challenge for infrastructure managers as tracks have to withstand harder service conditions for longer times and at lower costs. Regarding service conditions, railway transport is moving towards faster and heavier trains, which accelerate the degradation of the track. The high wheel/rail contact forces increase even more when wheels roll over short-wave rail defects such as poorly insulated joints, poor welds, corrugation and squats [2–4]. Consequently, vehicle and track components deteriorate, sometimes suffering detrimental defects. To guarantee safety, high-cost maintenance measures must be taken, such as grinding of the rail top or rail replacement. However, these measures conflict with the actual objective of reducing maintenance costs. Furthermore, maintenance measures need to be performed in a shorter time because the railway schedules are becoming more saturated in order to offer a more complete service to the customer. In summary, tracks are being subjected to more demanding service conditions while the resources available to maintain the desired performance are reduced.

To evaluate track conditions, measurements are regularly performed (see for instance [5–8]). However, problems often arise. For instance, the time frame between measurements is sometimes too long allowing damage to occur and develop, reaching a severe state. Another problem is that some measurement methods detect the damage at a late state when immediate maintenance measures are needed.

In some cases, preventive maintenance measures have been adopted. The objective is to take actions at relatively fixed intervals so that the damage does not develop into a state that requires time-consuming and expensive maintenance measures. An example of preventive maintenance measures is the cyclic grinding of the rail top that is performed in some metro and train systems [9]. In this manner, unwanted vibrations and noise are kept below undesired levels, and crack formation is prevented.

An intermediate option between preventive maintenance and severe damage removal is early-state maintenance, which consists of preventing the track from reaching an irreversible damaged state. This means that maintenance actions remove the damage that is still reversible, so that the track can be used to its optimum. For this purpose, early-state maintenance relies on frequent or continuous monitoring. In this manner, the deterioration of the system can be tracked, and maintenance measures are carried out when the deterioration crosses a certain threshold. Consequently, the track stays in service and fulfills safety conditions as long as possible. Thus, the service life of railway tracks can be extended by applying early-state maintenance measures.

For early-state maintenance, a continuous or frequent monitoring system should be used to effectively shorten measurement and maintenance times. Such systems could either be mounted on railway vehicles or installed in tracks operating without direct human assistance. For instance, optical fiber sensors, such as fiber Bragg gratings, can be installed to monitor the condition of railway subcomponents [10]. An example of vehicle-borne inspection systems is the Axle Box Acceleration (ABA) system [8, 11–13]. The damage detection method used by the ABA system consists of scanning the measured ABA signal in search of signature tunes (i.e. characteristic frequencies and amplitudes). If the signature tunes are detected, the measured ABA values are compared to a threshold to inform maintenance decisions. This detection and identification process
based on the ABA inspection and detection system can be applied to different damaged conditions if the signature tunes of the damaged conditions of interest are known.

This paper presents the application of a statistical method adapted to railway tracks for the purpose of identifying signature tunes of damaged conditions. The method is based on non-destructive field hammer test measurements and consists of statistically comparing the dynamic response of damaged tracks to healthy (i.e. non-damaged) tracks in the frequency domain. The healthy state is defined based on the quality control tracking measure known as control charts, which is employed in process monitoring. The statistical method is reliable in terms of objectivity, as the system automatically decides whether differences with respect to the healthy state are caused by variability or deterioration of the system.

As the method is based on hammer tests, the damage conditions that could be investigated are in the superstructure level (i.e. rail, fastening system and sleeper), because hammers are generally too light to fully excite the substructure layers (i.e. ballast and subgrade). Thus, the damages studied could include (1) track geometry problems at short wavelengths (i.e. less than 80 mm [14]), such as corrugation and squats [15, 16]; (2) fastening damage, such as loose bolts or severely degraded railpads; and (3) sleeper damage, such as cracked sleepers or sleeper voids (if the hammer is heavy enough to excite the sleeper-ballast interaction).

In the paper, the repeatability of hammer tests in railway tracks is investigated first. Then, the proposed statistical method for a frequency-based identification of railway track damage is presented, followed by the application of the method to a case study of squats in two different ballasted tracks. To conclude, the results of the feasibility study are compared to characteristic frequencies of squats used in an extensively validated ABA system. Finally, conclusions are drawn.

3.2. Hammer test measurements in railway tracks

Hammer test measurements are a non-destructive testing technique widely used in engineering to gain insight into the dynamic behavior of systems. In railway tracks, hammer tests are mainly used to derive the parameters of the track. The method consists of fitting simulations to measurements by varying track parameters, such as the stiffness of the railpad, and can be applied to whole tracks or individual track components. The excitation of individual components allows the possibility to study track components under controlled conditions in the laboratory; see, for instance, studies of fastening [17, 18] and sleepers [19, 20]. Laboratory test conditions should closely duplicate field conditions because the interactions between track components significantly influence the resulting dynamic behavior [21, 22]. Hammer tests have been performed by different authors on tracks in the field to investigate whole track dynamics and to obtain in-service track parameters [22–27]. The derived stiffness and damping of the railpad and ballast could be used to assess the condition of the track. In this work, instead, we propose to directly use the measured signals of field hammer tests. The advantage of our approach is that simulations need not be fitted to the measurements; thus, the accuracy of the results does not depend on the capacity of the model to reproduce the measurements. Furthermore, our approach is not limited to the study of the degradation of the railpad and ballast; the deterioration of the whole track can be investigated.
In railway tracks, hammer test measurements are carried out as follows. The rail head is excited with a hammer, and the dynamic response is measured at the location of interest. Depending on the frequency range of interest, different hammer weights and tip types are used. If low frequency is the target, heavy hammers and soft tips are usually used, whereas high frequencies are excited with light hammers and hard tips [17]. Typically, the excitation is applied vertically, either on a rail section located on the sleeper (on-support configuration) or on a rail section placed between two supports (mid-span configuration). An example of field hammer test measurements with the on-support configuration is shown in Figure 3.1. The response of the system is usually measured close to the excitation point with an accelerometer placed on top of the rail as shown in Figure 3.1. Then, the input force and output accelerations are converted via Fast Fourier Transform (FFT) into the frequency domain, and the Frequency Response Function (FRF) is calculated assuming linear behavior of the track. If required, hammer test measurements under loaded conditions can be performed, as some track elements may not behave linearly when exposed to unloaded vs. loaded track conditions [28].

For the measurements presented in this paper, a 280 gr Bruel & Kjaer 8202 hammer with a hard plastic tip is used, and the frequency range of interest is defined between 300 and 3000 Hz. Lower frequencies are not included, as the hammer is not able to excite the ballast and subgrade, which are the track components that dominate the dynamic behavior of the track at low frequency [29]; therefore, the measured dynamic response may be significantly different from that of the track under working conditions. The upper frequency is limited by the hammer-tip configuration used and is defined by analyzing the drop in frequency of the measured force autospectrum. A -3 dB drop separates the frequency range between very reliable and reliable data, and above a -10 dB drop, the recorded data are not considered sufficiently reliable. For our configuration, a -3 dB drop occurred at approximately 2300 Hz, and the -10 dB drop occurred at approximately...
3.3. REPEATABILITY OF HAMMER TEST MEASUREMENTS IN RAILWAY TRACKS

The -10 dB limit of 3000 Hz was considered in this paper to include a characteristic of ballasted tracks called second order pin-pin resonance, which is in the frequency range 2800-2900 Hz. The vertical response of the rail is measured at a sampling frequency of 20000 Hz with a single-axis DeltaTron 4526-001 accelerometer with a sensitivity of 10 mV/g. At each measurement point, five impacts are applied, and the average of the five measurements is calculated in order to minimize the random error. The length of the signals is 1.2 s. If the averaged measured signal is shorter, zeroes are added to the signal so that a frequency resolution less than 1 Hz is obtained in the FRF.

3.3. REPEATABILITY OF HAMMER TEST MEASUREMENTS IN RAILWAY TRACKS

The damage identification method for railway tracks we propose in this paper is based on the difference in the dynamic response between a damaged and a healthy track. The method differentiates between whether the disagreement between the frequency response functions is due to inherent variability or deterioration of the track. Regarding inherent variability, one way to account for the small differences between healthy locations, such as slight differences in the railpad preload or compactness of the ballast, is presented in Section 3.4.2. Additionally, one should keep in mind that the measured dynamic response of the track includes the stochasticities introduced by the test process. The locations of the input force and output acceleration deviate slightly during hammer test measurements in railway tracks. On one hand, it is difficult to guarantee that the excitations are applied exactly at the same spot. On the other hand, if different locations in the track are measured, the sensor will most likely not be placed precisely at the same place relative to the sleeper for all the measurement locations. Therefore, this section presents a number of tests to investigate how repeatable hammer tests are on railway tracks, with the objective of quantifying the variability introduced by the test process.

3.3.1. TEST SET-UP

The repeatability tests were performed on a track without damage visible to the naked eye. The tests were carried out on a rail section located over a sleeper. The on-support configuration was chosen because the clamps of the fastening system apply the toe-load at two points on the rail foot, not on the whole width of the rail on the support. This may lead to slightly different dynamic behaviors of rail sections along the width of the support. Thus, this study investigates whether the non-homogeneity of the support influences the measured frequency response of the system, as well as the relation between the excitation and measurement locations.

The tests were divided into a study of the variability introduced by the change in location of the excitation and of the sensor. In the first part of the repeatability tests, the effect of the location of impact was studied. For this purpose, the sensor was kept fixed in the middle of the support and the rail was excited at nine spots close to the sensor. As Figure 3.2a shows, an excitation spot mesh was defined on the top of the rail. In this manner, we could study the effect of varying the excitation location in both the lateral and longitudinal direction of the rail.

In the second part of the repeatability tests, the influence of the location of the sensor
was studied by placing the sensor at three locations along the rail on the support. The three measurement locations are denoted by circles and letters in Figure 3.2b. In all three cases, the distance between the sensor and the excitation point was kept constant. The effect of the lateral position of the sensor was not studied because the sensor is almost always placed in the middle of the rail head, which is also the running band. This is because the magnet that fixes the sensor to the rail needs to be placed on a flat surface to avoid unwanted vibrations due to possible tilting of the sensor. Therefore, the magnet is placed in the flat running band, which the magnet covers almost completely in the lateral direction of the rail. A picture of the test location is shown in Figure 3.3, where the placement of the sensor on the rail and the mesh spots for the repeatability study can be seen.
3.3. RESULTS OF THE REPEATABILITY STUDY

3.3.2.1. Changing the excitation location along the longitudinal direction of the rail

In Figure 3.4a, one can see the receptance functions in the frequency range of 300-3000 Hz when the excitation location is changed in the longitudinal direction of the rail. Only the results of the tests carried out in the middle line of the mesh (spots 2, 5 and 8 in Figure 3.2a) are shown. Unlike the upper and lower lines, the middle line is located on the running band where the excitation is always applied (see Figure 3.3 for the mesh configuration used).

The three measurements show a different frequency response between 1000 and 1700 Hz and at approximately 2800 Hz. Two main characteristics of tracks with monoblock sleepers are found in the frequency range 1000-1700 Hz: pin-pin anti-resonance at 1095 Hz and the fourth bending mode of the sleepers at approximately 1320 Hz [22]. Pin-pin anti-resonance is the vertical bending mode of the rail with the nodes on the sleepers [15]. There is also a second-order pin-pin anti-resonance, which is found at approximately 2800 Hz. Thus, three features of the vertical track dynamics are affected by a longitudinal change in the impact location. This is explained by the fact that the measured vibration modes related to these characteristics are influenced when the rail is excited farther from the sensor. If excitation is applied at spots 2, 5 or 8, the vibration modes that primarily consist of the movement of the rail are measured at slightly different locations. This measurement point change is reflected in the receptance function as a small difference in magnitude of the pin-pin anti-resonance at 1095 Hz. For the second-order pin-pin anti-resonance, the difference is greater, since a small frequency shift is observed in addition to the magnitude disagreement. Still, the shift in frequency between excitation at spot 2 and spot 8 is smaller than 4%, as the dips occur at 2795 and 2910 Hz, respectively.

The characteristic feature found at approximately 1320 Hz corresponds to the fourth bending mode of NS90 monoblock sleepers [22]. In the bending modes, the vibration of the sleeper along its length dominates but this behavior also implies minor vibrations in the sleeper cross-section, which may slightly influence the measured dynamic behavior if the excitation and measurement locations vary along the width of the sleeper. However, the effect is not significant. First, the greatest frequency difference is observed between excitation at spot 5 and spot 8, which show a dip at 1300 and 1350 Hz, respectively; this results in a frequency difference smaller than 4%. Second, the three measurements show a consistent trend in the change in magnitude of the receptance functions between 1300 and 1700 Hz and the receptance functions of spots 2 and 5 have similar shapes between 1300 and 1700 Hz. Since a relatively skilled person performing the hammer tests will almost always hit the rail between spots 2 and 5, the resulting difference in measurements will be negligible.

3.3.2.2. Changing the excitation location along the lateral direction of the rail

Figure 3.4b shows the receptance functions of the measurements if the excitation location is varied along the lateral direction of the rail. The receptance functions correspond to excitation locations 1, 2 and 3, which are the closest spots to the sensor, so that the
3. IDENTIFICATION OF CHARACTERISTIC FREQUENCIES OF DAMAGED RAILWAY TRACKS

Figure 3.4: Measured receptance functions based on changing the excitation location (a) along the longitudinal direction of the rail: (---) spot 2, (-----) spot 5, and (----) spot 8; and, (b) along the lateral direction of the rail: (---) spot 1, (-----) spot 2, and (----) spot 3

As shown in Figure 3.4b, the dynamic behavior of the track is different between 1000 and 1600 Hz. Thus, the pin-pin anti-resonance at approximately 1095 Hz and the fourth bending mode of the sleepers at approximately 1320 Hz are affected by the lateral change in excitation location. The same trend is apparent for the two characteristic features; the frequency increases if the excitation is applied away from the center of the track (spot 1), and decreases if the rail is excited closer to the gauge corner (spot 3). Further investigation is needed to fully understand the interaction between track components that lead...
to this trend, which is beyond the scope of this paper. From the perspective of hammer test repeatability, the measured differences are of interest.

The maximum differences in frequency occur between excitation applied at spot 1 and spot 3. For pin-pin anti-resonance, the frequencies differ by 3%, as the corresponding receptance dips are at 1085 and 1115 Hz, respectively. For the fourth bending mode of the sleepers, the frequency difference is 10%; the corresponding frequencies are 1260 and 1388 Hz. This variation is slightly higher than those found in Section 3.3.2.1, which may be caused by some lateral component of excitation when the rail is excited off-center. Nevertheless, this comparison corresponds to the two extreme cases spot 1 and spot 3, which are close to the edge of the rail head (see Figure 3.3), while the impact will always be applied at the middle of the rail head in field hammer test measurements. Therefore, the expected deviation will be considerably smaller. If the middle case (spot 2) is compared to the extreme cases (spot 1 and spot 3), the frequency difference of the fourth bending mode of the sleepers is smaller than 5%, as the corresponding receptance dip at spot 2 is at 1320 Hz.

3.3.2.3. Changing the Measurement Location Along the Longitudinal Direction of the Rail

Figures 3.5 displays the receptance functions obtained by varying the location of the sensor along the longitudinal direction of the rail. The three measurements have the same dynamic behavior between 300 and 3000 Hz. This confirms the conclusions drawn in Section 3.3.2.1; the differences observed in the receptance functions in Figure 3.4a are caused by the distance between the excitation and measurement locations. Thus, if the rail is excited close to the measurement location fixed in the running band, the track exhibits the same dynamic response within a 30 mm range from the center of the support in the longitudinal direction of the rail, which is the distance range investigated, as shown in Figure 3.2b.

Figure 3.5: Measured receptance functions obtained by changing the measurement location along the longitudinal direction of the rail: (---) A, (---) B, and (---) C.
3.3.3. DISCUSSION: REPEATABILITY OF HAMMER TEST MEASUREMENTS IN RAILWAY TRACKS

Three studies were carried out to assess the repeatability of hammer test measurements in railway tracks. The outcome of the first study is that different excitation locations along the longitudinal direction of the rail introduce variation in three features of the receptance function (i.e. the first- and second-order pin-pin anti-resonances and the fourth bending mode of the sleepers). The deviation between measurement frequencies, which is smaller than 4%, is caused by the use of different points when measuring the track vibration modes. The results of the second study show that applying excitation to the rail with a lateral offset with respect to the rail’s center line influences the dynamic behavior of the track at the pin-pin anti-resonance and fourth bending mode of the sleeper. Future work should investigate the relation between lateral variations of the excitation location and changes in these two features. For the repeatability study, the focus is not on the origin of the variability but on its quantification, and for this purpose, the maximum frequency differences are less than 5%. The third study leads to the conclusion that the track’s dynamic behavior does not change when the response is measured within 30 mm distance from the center of the support in the longitudinal direction of the rail.

Therefore, one can conclude that the variability introduced by the hammer test procedure in railway tracks is negligible if the vertical dynamic behavior of interest is between 300 and 3000 Hz. To perform the tests, two aspects should be taken into account. First, the sensor should be placed on the running band and close to the center of the support or the mid-span in the longitudinal direction of the rail (i.e. within 30 mm from the center of the support or mid-span). Second, the rail should be excited at the middle of the rail head, as close as possible to the sensor. In this manner, the variability introduced in the characteristic frequencies of the frequency response of the track between 300 and 3000 Hz is less than 5%.

3.4. FRF-BASED STATISTICAL METHOD APPLIED TO RAILWAYS

The characteristics of damage conditions may be identified based on the principle that small changes in track structures can cause different vibration responses. This is, the measured dynamic responses of tracks under damaged conditions contain information about the damage state. To obtain this information, we propose the Frequency Response Function (FRF)-based statistical method [30, 31]. As a non-parametric time-series method, it offers a number of advantages over parametric time-series methods. First, there is no need for a numerical model to fit to the measured response, which avoids the difficulty of developing a track model that accurately reproduces the dynamic response of the track in the frequency range of interest. Second, decisions related to the assessment of the deterioration state are made by applying statistical tools, removing subjectivity introduced by human operators from the decision-making process [32]. Furthermore, the statistical tools used account for inherent uncertainty, such as measurement repeatability, which was examined in detail in Section 3.3.
### 3.4.1. FRF-BASED STATISTICAL METHOD

The FRF-based statistical method uses FRF magnitudes to detect damage. First, the FRF magnitudes are evaluated via Welch estimates. Let us denote the function $\hat{H}(j \omega)$ for the FRF estimate at frequency $\omega$ (with $j$ being the imaginary unit). For the structure under study and the healthy structure, the FRF estimates are denoted by $\hat{H}_u(j \omega)$ and $\hat{H}_0(j \omega)$, respectively.

Second, the structure’s damage can be detected by applying a binary composite hypothesis test to the FRF magnitude [30]. For this purpose, the FRF magnitude of the current state of the structure ($|\hat{H}_u(j \omega)|$) is compared to the healthy (i.e non-damaged) structure ($|\hat{H}_0(j \omega)|$). If the current state of the structure significantly deviates from that of the healthy structure, the structure is considered damaged. In contrast, if the statistical deviations are not significant, the structure is considered healthy.

As the FRF estimators approximately follow a Gaussian distribution [33], the difference between the FRF estimators also follows a Gaussian distribution:

$$\delta \equiv |\hat{H}_u(j \omega)| - |\hat{H}_0(j \omega)| \sim \mathcal{N}(0, 2\sigma_0^2(\omega))$$  \hspace{1cm} (3.1)

A method to define the FRF of the healthy state of a railway track ($|\hat{H}_0(j \omega)|$) and its variance ($\sigma_0^2(\omega)$) is proposed in Section 3.4.2.

Based on the difference between the damaged and healthy FRF magnitudes and the variance of the healthy structure, the $Z$ statistic can be defined for each $\omega$:

$$Z(\omega) = \frac{|\delta|}{\sqrt{2\sigma_0^2(\omega)}} \hspace{1cm} (\forall \omega)$$  \hspace{1cm} (3.2)

Then, the $Z$ statistic can be examined at the selected $\alpha$ risk level, which is the probability of having a false alarm. The statistical test is defined as follows:

$$Z(\omega) \leq Z_{1-\frac{\alpha}{2}} \hspace{1cm} (\forall \omega) \Rightarrow \text{Healthy structure}$$
$$\text{else} \hspace{1cm} \Rightarrow \text{Damaged structure} \hspace{1cm} (3.3)$$

with $Z_{1-\alpha/2}$ designating the critical point of the standard normal distribution at $1 - \alpha/2$.

### 3.4.2. ASSESSMENT OF A HEALTHY STRUCTURE FOR RAILWAY TRACKS

Data describing a healthy structure state are necessary for the application of the FRF-based statistical method. The healthy state would be easy to assess at a recently built railway track where the components follow the design requirements, or in a full-scale laboratory set-up where the conditions can be controlled. However, most railway tracks have already been in service for some time, and use and environmental conditions deteriorate the track structure. As these structural changes may be not visible, a perfectly healthy location cannot be guaranteed.

Therefore, we propose to define a baseline state as a combination of reference locations. This principle is applied on the process-monitoring technique called control charts to define the limits that track the quality of the monitored process [34]. In the case of process-monitoring, the mean and variance of a process are usually unknown. They are estimated on the basis of $m$ preliminary data via sample mean and sample variance.
If this concept is adapted to the study of railway tracks via hammer test measurements, the baseline state of railway tracks can be defined as an average of hammer test measurements at locations without damage to the naked eye (i.e. reference locations). In the future, a method that selects reference locations with a higher certainty of their healthy state could be used.

To select the reference locations, the rational subgroup concept is proposed [34], which consists of grouping a number of consecutive locations at distanced track sections (e.g. 40 sleeper spans). In this manner, the likelihood of differences among the subgroups is maximized, such as differences in subgrade, whereas the likelihood of differences within the subgroups is minimized, such as variability between consecutive sleeper bays. Thus, the differences within and among the subgroups are considered in the sample variance, which is employed in the calculation of the \( Z \) statistic (see Equation (3.2)). In this manner, the difference between different sleeper bays is considered in this study.

Once the reference locations are defined, the sample mean and sample variance are calculated according to the following respective equations:

\[
\hat{H}_0(j\omega) = \frac{1}{m} \sum_{i=1}^{m} \hat{H}_i(j\omega)
\]

\[
\hat{\sigma}_0^2(\omega) = \frac{1}{m-1} \sum_{i=1}^{m} (|\hat{H}_i(j\omega)| - |\hat{H}_0(j\omega)|)^2
\]

where \( m \) is the number of reference locations. For the baseline state to be statistically representative, at least 20 reference locations should be included based on process-monitoring experience [34].

3.5. Feasibility Study: Identifying the Characteristic Frequencies of Squats

3.5.1. Squats and their detection via the ABA system

Squats are rolling contact fatigue defects found on the tops of rails [4, 35]. Squats can be light, moderate or severe; one example of each severity type is shown in Figure 3.6. This short-wave rail surface defect can be found in railways around the world in all types of tracks [36–38]. The wavelength range (25-35 mm) gives rise to vibrations at damaging frequencies for the vehicle-track system [38]. Furthermore, in the case of moderate and severe squats, the defect involves cracks in the rail. These cracks, together with a high impact, load could lead to a catastrophic rail break. To guarantee safety, rail-top grinding is employed as a maintenance measure to address light and moderate squats, whereas for severe squats, the costly replacement of the rail is usually inevitable [39]. Consequently, the maintenance costs are high and disturbances in network operation are common.

An extensive study of squat growth was performed to develop an early-stage detection method [16, 38]. The goal was to detect light squats, which can be completely removed by grinding, so that more expensive maintenance measures can be avoided. In
3.5. **Feasibility study: identifying the characteristic frequencies of squats**

3.5.1. **Ease of Study**

Identifying the characteristic frequencies of squats involves

![Figure 3.6: Degrees of severity of squats: (a) light (b) moderate (c) severe](image)

this manner, high maintenance costs and possible disturbances in the network could be reduced; moreover, safety could be increased.

Based on the previous extensive study, an ABA inspection and detection system was developed [13, 40]. The system consists of a combination of accelerometers mounted on axle boxes, a GPS antenna and train speed data. By post-processing the measured signals, the location of squats can be detected and their severity assessed [40]. To extract this information from the ABA measurements, squats are recognized in the measured signals by combining the magnitude of the measured signals with the characteristic frequencies of squats. Thus, identifying the characteristic frequency of squats is a fundamental factor in this detection method. To find this frequency, a wide and detailed but also time-consuming investigation has been carried out combining 3D Finite Element (FE) simulations and ABA measurements [40, 41]. In this paper, we propose a faster method to identify the characteristic frequencies of squats in order to contribute to and hasten the development of vehicle-borne detection systems.

We apply the FRF-based statistical method presented in Section 3.4 to locations with squats at two track sites with different sleeper types. The study of a track with monoblock sleepers is presented in Section 3.5.2, and the study of a track with biblock sleepers is presented in Section 3.5.3. Then, the results are compared to the results obtained in a previous extensive squat study [40, 41] in Section 3.6.

3.5.2. **Test site with monoblock sleepers**

As input for the FRF-based statistical method, hammer test measurements were performed at a test site with monoblock sleepers (see Figure 3.7). At this track site, the repeatability tests of Section 3.3 were also carried out. The ballasted track consisted of continuously welded UIC 54 rails on concrete NS90 monoblock sleepers with a Vossloh fastening system and FC9 railpads. The track site had squats of different severities, as well as long track sections without damage conditions detectable by visual inspection.

3.5.2.1. **Baseline state**

For the assessment of the baseline state, hammer test measurements were performed at 21 locations, which were divided into three groups of seven consecutive supports.
These locations did not have defects visible to the naked eye, such as corrugation or squats, on the rail surface. The structure also did not show damaged conditions, for instance missing bolts and railpads, broken or loose clips, or broken sleepers. At each of the 21 locations, hammer test measurements were performed with the on-support configuration and with the successive mid-span configuration in the direction of traffic.

Figure 3.8 displays the receptance functions of the 21 locations as gray lines. Figure 3.8a corresponds to on-support excitation, whereas Figure 3.8b corresponds to mid-span excitation. The two figures also include the average FRF of the 21 reference locations ($\hat{H}_0(j\omega)$) and lines indicating one standard deviation with respect to the mean (i.e. $\hat{H}_0(j\omega) \pm \hat{\sigma}_0(\omega)$). That is, 68% of the values fall between the two standard deviation lines if a normal distribution of the measurements is assumed. The mean and the variance were calculated according to Equations (3.4) and (3.5), respectively.

At the test site, hammer test measurements were also performed at locations with squats. The measurements were carried out at locations with squats of different severity. The study included five severe, five moderate and seven light squats. At each location with a squat, hammer test measurements were performed at the closest support and mid-span to the squat; that is, the squat was located between the on-support and the mid-span excitations.

The hammer tests at the reference locations and squats were performed during the same month, in which the weather conditions did not drastically change (during the day and in dry conditions). Although the measured receptance function may change under different temperatures because the properties of the railpad are temperature-dependent [42], it is assumed that the change in the dynamic behavior of the track is smaller than the variation observed between different sleeper spans. The influence of temperature on the receptance function will be investigated in future work.
3.5. Feasibility study: identifying the characteristic frequencies of squats

3.5.2.2. FRF-based statistical method applied to the track with monoblock sleepers

The FRF-based statistical method, with a baseline defined by hammer test measurements at reference locations, was applied to the hammer test measurements at locations with squats in the track with monoblock sleepers. The chosen \( \alpha \) risk value was 0.02, which implies a 2% probability for a false alarm to occur. This value of \( \alpha \) corresponds to a \( Z \) value of 2.32 (see (3.3)), which is the statistical limit \( Z_{lim} \) and is shown as a horizontal line in Figure 3.9. The gray lines in Figure 3.9 correspond to the \( Z \) statistic of the measurements at the locations with squats. Graphically, the statistical decision-making
consists of checking whether \( Z \) exceeds \( Z_{lim} \). Thus, if a gray line crosses the red line, that location is damaged. One of the advantages of the FRF-based statistical method is that the study is not limited to the detection of damage. In the case of damage, the frequencies at which \( Z \) crosses \( Z_{lim} \) are visible so that the frequency related to the damage condition can be identified.

Figure 3.9 consists of eight graphs with four rows and two columns. The first row corresponds to the application of the FRF-based statistical method to the baseline. All measurements are below the statistical limit \( Z_{lim} \), except for one measurement in the mid-span configuration, which significantly differs from the average FRF below 400 Hz (see Figure 3.8b).

The second, third and fourth rows correspond to the application of the FRF-based statistical method to one of the three squat severity types. The left column corresponds to on-support excitation and the right column corresponds to mid-span excitation. As Figure 3.9c shows, most light squats cannot be detected by hammer test measurements on the support. For almost all seven cases, \( Z \) statistic stays below the statistical limit \( Z_{lim} \). At a few frequencies, such as 1300 and 2686 Hz, one gray line crosses the horizontal line, but the frequencies are not considered representative, as they are isolated phenomena. For mid-span excitation, a few measurements exceed the limit at approximately 1000 Hz. Such repetition suggests that this frequency may be related to squats. For confirmation, the measurements at locations with more severe squats are investigated.

For moderate and severe squats, the statistical limit is consistently crossed between measurements, with such events concentrated in the same frequency ranges (compare Figure 3.9e with 3.9g, and Figure 3.9f with 3.9h). Furthermore, more measurements exceed \( Z_{lim} \), with larger \( Z \) values at severe squats than at moderate squats. This is expected, as a more damaged state differs more from the baseline state resulting in a larger \( Z \) value.

Histograms are constructed to provide a better overview of the frequency ranges at which the statistical limit \( Z_{lim} \) is passed when moderate and severe squats are present in the track. Histograms for on-support and mid-span excitations are shown in Figure 3.10. To emphasize the prominent frequencies, the number of observations that exceed the statistical limit \( Z_{lim} \) are normalized by the number of defects of each type. For instance, none of the hammer test measurements exceeded \( Z_{lim} \) at 400 Hz, while 60% (i.e. the normalized number of observations is 0.6) crossed \( Z_{lim} \) at 2000 Hz for moderate squats with on-support excitation (see Figure 3.10a).

A frequency range is assumed to be significant when the normalized number of observations is 0.4 or larger. Thus, four frequency ranges are identified for moderate squats for on-support excitation; these are 360-380 Hz, 565-615 Hz, 790-1055 Hz and 1640-2020 Hz (see Figure 3.10a). However, if the excitation is applied in the mid-span, only the frequency range 790-1055 Hz has a significant normalized number of observations. This suggests that the other three frequency ranges (i.e. 360-380 Hz, 565-595 Hz and 1640-2020 Hz) are related to phenomena occurring in the support, which are too far away and therefore too weak to be measured by the mid-span configuration.

In the case of severe squats, more frequencies have a normalized number of observations of 0.4 or larger than for moderate squats (see Figure 3.10). These frequencies clus-
Figure 3.9: $Z$ test at the baseline for (a) on-support excitation and (b) mid-span excitation, at light squats for (c) on-support excitation and (d) mid-span excitation, moderate squats for (e) on-support excitation and (f) mid-span excitation, and severe squats for (g) on-support excitation and (h) mid-span excitation, with $Z_{lim}$ and $Z_{measurments}$ at the locations with squats.
ter around the frequencies identified for moderate squats, so that the frequency ranges become wider for severe squats. Figure 3.11 shows an overview of the evolution of the identified frequency ranges to facilitate the comparison between frequencies identified from hammer test measurements at moderate and severe squat locations. Figure 3.11a and 3.11b correspond to the on-support and mid-span excitations, respectively. For the two excitation configurations, the frequency ranges corresponding to moderate squats are shown with shaded green regions and black arrows followed by the letter B; for severe squats, the shaded regions and arrows are gray, and the letter C is used. For instance, there are two identified frequency ranges at approximately 800 Hz for moderate squats with on-support and mid-span excitations. These two frequency ranges have been merged into a wide frequency range for a more advanced damaged state (i.e. severe squat). The same trend of merging and widening can be seen for the other frequency ranges. This confirms that the frequencies identified by applying the FRF-based statistical method to locations with squats are indeed related to squat severity because the frequency ranges evolve according to the damage process of the squats.

Analyzing the histogram of severe squats in Figure 3.10b, some frequencies have a large normalized number of observations (i.e. 0.8 or more). Under mid-span excitation,
only the frequency range of 2480-2510 Hz is found to have such a large value. This frequency range was not identified for moderate squats, thus it is most likely related to phenomena occurring during the late stage of squat growth, such as widening of the running band and growth of cracks. Under on-support excitation, two frequency ranges, 830-910 and 1835-2030 Hz, have a normalized number of observations of 0.8 or higher and are also present in the histograms of moderate squats (see Figure 3.10a). In conclusion, these frequency components are strongly related to squats. Moreover, because they are measured in the on-support configuration and not in the mid-span configuration, the vibrations at these frequencies are most likely caused by phenomena originating in
the support. Consequently, one can venture to affirm that the occurrence and growth of squats is related to the conditions of the support. This is in agreement with field observations, in which 74% of the squats in a correlation study were found on the rail part over the sleeper [16]. Furthermore, numerical work performed using a validated model that accounts for the vehicle-track interaction at squats [43] shows that the condition of the fastening influences the dynamic contact forces at squats. The dynamic loads at a squat are higher in tracks with damaged fastenings than in tracks in good conditions. Consequently, the probability for squat growth is higher in deteriorated tracks than in good tracks. This possible relation between squats and support condition is further investigated in Section 3.5.2.3, where the identified significant frequencies are related to characteristics of the receptance function and therefore to track components.

3.5.2.3. RELATING IDENTIFIED CHARACTERISTIC FREQUENCIES TO TRACK VIBRATIONS

Figure 3.11 shows the baseline reference receptance function ($\hat{H}_0(j\omega)$) and the identified frequency ranges for on-support and mid-span excitations. The frequency response of the track has two peaks, corresponding to the in-phase and antiphase rail resonances [22], which occur at 644 and 937 Hz, respectively, under the on-support configuration. In these modes, the rail and the sleeper vibrate in anti-phase, and the two rails are in-phase or in anti-phase. The rail resonances depend on rail and railpad properties; the latter also influences the coupling between the rails. Therefore, changing the rail and railpad properties directly affects the in-phase and antiphase rail resonances. For instance, moderate and severe squats have cracks in the rail, which decrease rail stiffness. Furthermore, the fastening deteriorates due to high wheel/rail impact forces; the toe load may decrease, lowering the railpad stiffness [17, 28, 44], which results in a more flexible track and a weaker coupling. These factors can change significantly the measured receptance function with respect to the baseline at frequencies lower than the two rail resonance frequencies. These changes were identified by employing the FRF-based statistical method at the frequency ranges of approximately 520-620 Hz and 700-900 Hz, which are displayed as gray shading in Figure 3.11.

The pin-pin resonance of the averaged reference receptance function is clearly visible at 1040 Hz under mid-span excitation. This mode occurs when the rail vibrates with its nodes on the sleepers. Thus, sleeper distance is one of the most important parameters besides the rail bending stiffness and the railpad properties. Cracks in the rails change the rail bending stiffness, and deteriorated railpads deviate from their nominal stiffness and damping values. These changes can cause alterations in the receptance function around the pin-pin frequency, as in the case of the in-phase and anti-phase rail resonances. Deterioration is captured by the FRF-based statistical method by identifying the frequency range of 970-1055 Hz in the on-support configuration (Figure 3.11a).

A shaded frequency range is also seen at frequencies below the feature at approximately 2000 Hz in the on-support configuration. The origin of this feature has not been found yet. However, some studies indicate that it may be related to the fastening system [22].
3.5.3. **TEST SITE WITH BIBLOCK SLEEPERS**

Hammer test measurements were also performed at a test site with biblock sleepers (see Figure 3.12). The rail was UIC54, which was attached to the sleeper with Vossloh fastenings, and the railpad type was FC9. As with the track with monoblock sleepers, squats at different degrees of severity were found. The track also contained long track sections without damage visible to the naked eye.

![Figure 3.12: Test site with biblock sleepers](image)

3.5.3.1. **BASELINE STATE**

Similar to the measurements performed at the track with monoblock sleepers, for biblock sleepers, the rail was excited using on-support and mid-span configurations at 21 locations. The locations were selected from three groups of seven consecutive supports. The chosen sections did not exhibit damage conditions detectable by visual inspection.

Figure 3.13a and Figure 3.13b show the measured receptance functions at the reference locations for the on-support and mid-span configurations, respectively. The receptance functions of the 21 locations are displayed as gray lines. Their mean and variance were calculated according to Equations (3.4) and (3.5), respectively. The average FRF ($\hat{H}_0(j\omega)$) is shown with a red line, whereas one standard deviation with respect to the mean (i.e. $\hat{H}_0(j\omega) \pm \hat{\sigma}_0(\omega)$) is displayed with blue lines.

Hammer test measurements were also carried out at locations containing squats of different severity. The study included six severe and seven light squats. Moderate squats were not found at the track site. The same measurement procedure was followed for each location with a squat. First, the rail was excited at the support closest to the squat, and then, at the mid-span so that the squat was located between the two excitation configurations.

As in Section 5.2, the hammer tests at the reference locations and at the squats were performed during the day and in dry conditions over the period of one month.
3. IDENTIFICATION OF CHARACTERISTIC FREQUENCIES OF DAMAGED RAILWAY TRACKS

Figure 3.13: Baseline state for (a) on-support excitation and (b) mid-span excitation with (—) the average FRF \( \hat{H}_0(j\omega) \), (—) one standard deviation with respect to the mean \( \hat{H}_0(j\omega) \pm \sigma_0(\omega) \), and (—) reference location measurements.

3.5.3.2. FRF-BASED STATISTICAL METHOD APPLIED TO THE TRACK WITH BIBLOCK SLEEPERS

The FRF-based statistical method was applied to the hammer test measurements at the track with biblock sleepers in the same way as at the track with monoblock sleepers. The calculated \( Z \) statistics were compared to the statistical limit \( Z_{lim} \), which was also 2.32 because the same \( \alpha \) risk value of 0.02 was chosen. The \( Z \) statistics are shown in six figures distributed in three rows and two columns (see Figure 3.14). The left column corresponds to on-support excitation, and the right column corresponds to mid-span excitation. The first row displays the results of the FRF-based statistical method applied to the
baseline. Except for one measurement, which significantly differs from the average FRF at approximately 420 Hz (see Figure 3.13b), the $Z$ statistics of the reference locations are below the statistical limit $Z_{\text{lim}}$. The second row corresponds to the FRF-based statistical method applied to light squats and the third row to severe squats. As Figure 3.14d shows, the hammer test measurements with the mid-span configuration present $Z$ statistics below the statistical limit $Z_{\text{lim}}$ for almost all seven cases. At a few frequencies, such as at 1720 and 2040 Hz, $Z_{\text{lim}}$ is crossed, but these frequencies are not considered representative because they are isolated phenomena (i.e. only one measurement crosses the line). For on-support excitation, a few measurements exceed $Z_{\text{lim}}$ between 430 and 810 Hz (see Figure 3.14c), which suggests that this frequency range may be related to squats. To support this hypothesis, the presence of this frequency range at locations with severe squats is investigated.

More measurements exceed $Z_{\text{lim}}$, with a higher $Z$ value, at severe squats than at light squats (compare the second and third rows in Figure 3.14). This increase in $Z$ is expected as a more damaged state differs more from the baseline state.

For severe squats, the crossing of the statistical limit $Z_{\text{lim}}$ is concentrated at certain frequency ranges, as shown in Figure 3.14e and 3.14f. Histograms are constructed to provide a better overview of the frequency ranges (see Figure 3.15a and 3.15b for light and severe squats, respectively). As in the case of the track with monoblock sleepers, the histograms exhibit the normalized number of observations that exceed $Z_{\text{lim}}$ for each frequency between 300 and 3000 Hz. A value close to one at a certain frequency means that the frequency component is present in most of the measurements and can therefore, be related to squats. In our case, a frequency range is assumed to be significant when the normalized number of observations is 0.4 or larger.

The evolution of the identified frequency ranges is summarized in Figure 3.16. The frequency range related to light squats is shown by a shaded green region and a black arrow followed by the letter A. The frequency ranges corresponding to severe squats are shown with shaded regions and arrows in gray and are denoted by the letter C. The frequency ranges displayed for light and severe squats tend to widen as squats deteriorate, which is in agreement with the analysis of the track with monoblock sleepers in Section 3.5.2.2.

Four frequency ranges are identified for severe squats at mid-span configuration: 300-380 Hz, 460-660 Hz, 1260-1375 Hz and 2860-2870 Hz (see Figure 3.16b). For the on-support configurations, two frequency ranges are identified (see Figure 3.16a). The first frequency range is defined between 300 and 730 Hz, which covers the 300-380 Hz and 460-660 Hz frequency ranges identified for the mid-span configuration. The second frequency range is defined between 1190 and 1530 Hz, which also contains the identified frequency range of 1260-1375 Hz for the mid-span configurations. The observation that the identified mid-span frequency ranges are within the identified frequency ranges for the on-support configuration indicates that the phenomena causing significant changes in the receptance function most likely originate in the support. This is in agreement with the analysis of squats in the track with monoblock sleepers in Section 3.5.2.2, field observations [16] and numerical results that account for the vehicle-track interaction [43]. The vibrations are fully captured with the on-support configuration, whereas some of the vibrations are weak or not excited when the hammer test is on the mid-span. The
Figure 3.14: $Z$ test at the baseline for (a) on-support excitation and (b) mid-span excitation, at light squats for (c) on-support excitation and (d) mid-span excitation, and severe squats for (e) on-support excitation and (f) mid-span excitation, with (—) $Z_{lim}$, and (—) $Z_{measurements}$ at the locations with squats.

possible relation between the condition of the support and the occurrence and growth of squats is investigated through a comparison of the identified frequency ranges and track vibration modes in Section 3.5.3.3.

3.5.3.3. Relating Identified Characteristic Frequencies to Track Vibrations

The identified frequency ranges and the baseline receptance functions ($\hat{H}_0(j\omega)$) overlap in Figure 3.16 for the track with biblock sleepers. The receptance functions exhibit the characteristic track modes of sleeper anti-resonance and rail resonance at approximately 490 and 860 Hz, respectively (see the dip and the peak in the on-support receptance function in Figure 3.16a, respectively) [15]. In sleeper anti-resonance, the sleeper vibrates between the ballast and the rail, whereas in rail resonance, the rail vibrates on
3.5. Feasibility study: identifying the characteristic frequencies of squats

Figure 3.15: Histograms of the exceeded frequencies at (a) moderate squats and (b) severe squats for (-) on-support excitation and (---) mid-span excitation

The sleepers [15]. For the two vibration modes, the rail and sleeper properties are key aspects. Additionally, the properties of the fastening that connects the rail and the sleeper have a significant influence on vibration. Thus, the presence of cracks or a decrease of the railpad stiffness may alter the receptance functions near the sleeper anti-resonance and rail resonance features, which are identified as the frequency range of 300-730 Hz by employing the FRF-based statistical method.

By applying the FRF-based statistical method to locations with squats in a track with biblock sleepers, a second main frequency range was identified. However, the possible characteristic phenomena occurring from 1190-1530 Hz are still unknown. For a track with monoblock sleepers, the fourth bending mode of the sleeper dominates this frequency range [22]. For a track with biblock sleepers, further investigation is required.
Figure 3.16: Characteristic frequencies of (green + A) light and (gray + C) severe squats according to the FRF-based statistical method and the baseline state receptance function \( \hat{H}_0(j\omega) \) for (a) the on-support excitation and (b) the mid-span excitation.

3.6. **Assessment of the FRF-based statistical method for identification of characteristic frequencies in railway tracks**

3.6.1. **Introduction**

We have gained insight into the frequencies related to the evolution of squats by employing the FRF-based statistical method. The number of squats observed was, nevertheless, small. Furthermore, the basis of the method are hammer test measurements...
that were performed under unloaded conditions (i.e. there was no vehicle on the track), and some track parameters behave non-linearly when changing from an unloaded track to a loaded track [28], which may influence the identified characteristic frequencies.

To build conclusions on a solid base and to investigate whether the track’s loading condition affects the identified frequencies, the results of the FRF-method are compared to the results of an extensive study based on both numerical simulations and Axle Box Acceleration (ABA) measurements [40]. The FRF-based statistical method and the ABA system measure the dynamic response, which for the ABA system is the response to the real load from trains rolling over squats. The two measurement systems are compared by analyzing the frequency ranges identified from measurements that were performed at the same defects, which were located on the track with biblock sleepers presented in Section 3.5.3.

3.6.2. Comparing frequencies identified with the FRF-based statistical method and the ABA system

In the numerical and ABA study, two main frequency ranges were identified related to moderate and severe squats [40]. One frequency range included frequencies lower than 600 Hz, and the other frequency range covered 1000 to 2000 Hz. More specifically, the maximum power spectral density of the measured ABA signals was found between 1000-1300 Hz.

We found these two frequency ranges in practice by way of the FRF-based statistical method (see Figure 3.16). The difference between the high frequency ranges can be explained by examining the two systems studied. The ABA system is a complex system including vehicle-track interactions in which the resulting vibrations are measured at the axle. Many vibrations are included in the final measured signal, which is far from the origin of the impact. With hammer test measurements, however, the response of the track is effectively measured at the excitation location with barely any noise. In this manner, the measured response contains information only about the track and therefore, the identified frequency ranges are more accurate. In our case, the derived frequency range of 1190-1530 Hz is concentrated over the maximum frequency range of the power spectral density of the measured ABA signals.

In the low frequency range, the ABA system identifies an upper frequency limit of 600 Hz, whereas the FRF-based statistical method in combination with hammer test measurements defines upper frequency limits at 660 Hz and 730 Hz for the mid-span and on-support configurations, respectively. This small frequency difference is also related to the complexity of the system. In the ABA system, the measured signal is far from the origin of the impact, so some track vibrations may be lost in the transmission to the axle.

In summary, the characteristic frequencies derived based on numerical simulations and ABA measurements and the characteristic frequencies identified with the FRF-based statistical method largely overlap. Therefore, the FRF-based statistical method based on field hammer test measurements performed on an unloaded track is a suitable method to derive characteristic frequencies of squats in railway tracks in the frequency range of 300-3000 Hz. These promising results suggest that the FRF-based statistical method could be employed to identify the signature tunes of other damage conditions so that the application of the ABA detection system could be extended faster.
3.7. **DISCUSSION**

In this paper, an FRF-based statistical method is presented to identify the characteristic frequencies of damage conditions, so that the information can be used in the development of dynamic-response-based train-borne (DRTB) systems. Through a feasibility study, the characteristic frequencies of squats were identified. The results are promising as the identified frequencies agree with those identified using DRTB systems. To employ this method for the development of DRTB systems throughout an entire network, some further sources of variability should be investigated (see Section 3.7.1). Nevertheless, the general method presented and the results obtained are already useful. Two applications are presented in Section 3.7.2 and 3.7.3.

3.7.1. **FROM FEASIBILITY STUDY TO PRACTICAL APPLICATION**

To implement the FRF-based statistical method for a wider set of tracks and conditions, the influence of some types of variability should be investigated to define the applicability region of the baseline state. A different healthy state should be defined for different track types because the dynamic response of the track is determined by the dynamic behavior of its components and their interaction, so that different track components lead to different track responses. As each track type has a characteristic response, at least one healthy state should be defined for each track type.

When defining different track types, some of the parameters that should be considered include sleeper type (see Figures 3.8 and 3.13), sleeper distance [15] and rail type [15]. Regarding the fastening and ballast types, their influence on the dynamic response of tracks should also be investigated. The method could also be applied to non-ballasted tracks because the FRF-based statistical method is based on the statistical comparison of measurements at the location of interest to the characteristic response of the track investigated. An example of this is the feasibility study in which characteristic frequencies were identified from two significantly different receptance functions.

In addition to the track type, other aspects that may influence the dynamic response of the track should be further investigated, such as track aging and temperature. When those conditions change drastically, a new baseline state may be required because the railpad and ballast (if any) degrade under environmental and service conditions, which may change the dynamic response of the track over time. Additionally, the properties of the railpad are influenced by the temperature and consequently, the dynamic response of the track may change. Further investigation is needed to gain understanding into the change in the response of the track over time and under different temperatures.

These changes in the dynamic response of the track make it difficult to implement the detection algorithm used in DRTB condition monitoring systems, such as the ABA system and strain-gauge-instrumented wheelsets [45], in a railway network with different track structures. The adaptation to the entire network can be made easier by employing the FRF-base statistical method as it is relatively easy and quick.

Regarding the expansion of DRTB systems to different isolated damaged conditions, the first step would be to apply the FRF-based statistical method to a larger number of squats so that the characteristic frequencies can be identified by the statistically reliable accumulation of crossings over the statistical limit \( Z_{lim} \). Then, the FRF-based statistical method could be applied to other damage types such as loose bolts, severely degraded
railpads, cracked sleepers or sleeper voids (if the hammer is heavy enough to excite the sleeper-ballast interaction). To be statistically reliable, more than 20 measurements of each defect type should be collected [34]. The damage investigated should be in a moderate/severe state so that the dynamic response of the track significantly changes with respect to the baseline state and the FRF-based statistical method captures the differences. The information obtained could be employed in the development of DRTB systems for automatic detection.

3.7.2. APPLICATION OF THE BASELINE FOR STRUCTURAL HEALTH MONITORING

The work presented in this paper can be employed to investigate not only isolated damaged conditions, such as squats, but also the general deterioration of railway tracks. The baseline of a track can be defined based on field hammer test measurements at locations without visible damage to the naked eye (see Section 3.4.2). This average condition changes over time due to environmental conditions and traffic, meaning that, at a certain point, the track may not attain the required performance. By periodically defining the average condition of the same track, the condition of the track could be monitored using quality control tools such as control charts [34]. This information about the general deterioration of a track may be useful for the assessment of track condition and for planning maintenance measures.

3.7.3. RELATING TRACK COMPONENTS TO SQUATS

In addition to providing useful information and methods for monitoring general deterioration and isolated damage in railway tracks, this work also indicates possible track components related to the occurrence and/or growth of squats. By examining the characteristics of the receptance function of the tracks in the frequency ranges identified by the FRF-based statistical method, the vibration modes of the track related to squats can be determined. Then, by examining the vibration modes that are defined by track components and their mutual interaction, the track components related to squats may be identified.

Different characteristic frequencies related to squats were derived for the tracks with monoblock and biblock sleepers (see Sections 3.5.2.2 and 3.5.3.2, respectively). This difference is attributed to the influence of the track structure on the occurrence and/or growth of squats. For the two track types, the frequency ranges identified were close to certain features of the receptance function that depend on the properties of the rail and the fastening (see Section 3.5.2.3 and 3.5.3.3).

On one hand, the difference between the baseline state and measured receptance functions at locations with squats could be attributed to cracks in the rails. Light squats, which might not contain cracks, could barely be detected with the FRF-based statistical method, whereas moderate and severe squats, which contain cracks, were detected. However, Figure 3.9c and 3.14c show that some of the calculated $Z$ statistics of measurements at light squats are close to the statistical limit $Z_{lim}$. A lower limit may detect light squats, but at the expense of increasing the probability of false positives.

On the other hand, some identified frequency ranges were present for 80% or more of the measured locations with squats, but only for the on-support configuration. Ad-
ditionally, the frequency ranges were wider for the on-support configuration. These observations suggest that either the deterioration of the support condition contributes to the evolution of squats or squats contribute to the deterioration of the support. This is in agreement with (1) a correlation study [16] in which, along 122 km, 74% of the squats were found at the half-sleeper-span rail section supported on the sleeper, and (2) a numerical work with a validated model that considers the vehicle-track interaction at squats; the results show that the probability of squat growth is higher for a track with damaged fastenings than for a track in good condition because the dynamic wheel/rail contact forces are higher in the former than in the later [43].

In summary, both cracks in the rail and the condition of the fastening affect the growth process of squats; the current study could not further explain the initial development of squats.

3.8. CONCLUSIONS
The feasibility of the FRF-based statistical method to identify characteristic frequencies of damage conditions in railway tracks was investigated. To employ this method to assess the condition of railway tracks, two inputs were required: non-destructive field hammer test measurements and healthy state data. For the former and in view of the statistical decision-making process, a repeatability test was performed to study the variability introduced by the test process. It is shown that the repeatability of hammer tests in railway tracks is high enough for our purpose with negligible variability in the characteristic frequencies of the dynamic response of the track between 300 and 3000 Hz. Regarding the healthy state and faced with the obstacle of not knowing the nominal receptance function, it is shown that a baseline can be defined as an approximately healthy state by choosing a statistically reliable number of locations without visible damage.

The feasibility study examined squats of different severity to assess the applicability of the FRF-based statistical method to identify characteristic frequencies of damage conditions in railway tracks. Two tracks with different sleeper types were investigated. It is shown that the identified frequency ranges related to squats were consistent with the increase in severity. Furthermore, the identified characteristic frequency ranges related to squats with the FRF-based statistical method correspond to the frequency ranges excited by trains rolling over squats, because these identified frequency ranges largely agree with the identified frequency ranges used for detection of squats with an extensively validated train-borne system. In view of the promising results, the FRF-based statistical method has the potential to be used to identify the characteristic frequency of damage conditions in railway tracks based on field hammer test measurements in the frequency range of 300-3000 Hz. To implement the proposed method for a wide set of tracks and conditions, sources of variability, such as temperature and aging of materials, should be further investigated.

The frequency ranges identified in the case study with squats were close to characteristic values of receptance functions that are dependent on rail and fastening properties. These features occur at different frequencies for the two track types studied because the type of sleeper influences rail-railpad-sleeper interactions. The agreement between the track vibration modes and the identified frequencies suggests a possible relation between squat growth and rail and fastening properties.
The structural health monitoring of railway tracks can be improved based on the investigation presented. On one hand, the application of the FRF-based statistical method to identify frequencies of isolated damaged conditions in railway tracks can significantly facilitate and speed up the development of dynamic-response-based train-borne monitoring systems such as ABA systems and strain-gauge-instrumented wheelsets, as well as their adaptation to an entire network. On the other hand, the baseline state could be employed to monitor the general deterioration of railway tracks.

REFERENCES


AN INVESTIGATION INTO THE VERTICAL DYNAMICS OF TRACKS WITH MONOBLOCK SLEEPERS WITH A 3D FINITE ELEMENT MODEL

This paper investigates the vertical dynamic behavior of railway tracks with monoblock sleepers. Whole- and half-track Finite Element (FE) models are presented in which the rails and sleepers are represented with their nominal 3D geometry using solid elements. The railpad encompasses the rail seat area modeled as multiple spring-damper pairs. The 3D FE models are employed for three purposes. First, the stiffness and the damping of the railpad and ballast are derived by fitting the simulations to a set of field hammer test measurements. Second, the origins of six of the seven main characteristics are identified. Third, the influence of the railpad representation on the track's dynamic response is studied. The results show that, in contrast to the 3D FE half-track model, the 3D FE whole-track model reproduces six of the seven main vertical track characteristics with a maximum deviation of 10% from the measured frequencies. The seventh characteristic is reproduced at approximately the measured frequency when the frequency-dependent stiffness of the railpad is considered. This model can be used to derive track parameters that will aid in the study of track degradation.

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4.1. INTRODUCTION

Railway trains, from metro systems to high speed lines, are a popular means of transportation worldwide. The severe working conditions, such as high speed and large tonnage, accelerate the deterioration of the track structure from the foundation of railway bridges [1, 2] to the rail top surface [3, 4]. To guarantee safety and acceptable noise levels, high cost maintenance measures, such as periodic tamping of the ballast and grinding of the rail top, are taken. A better understanding of the interactions and physics occurring in the complex vehicle-wheel-rail/track system will help slow down the deterioration, and consequently lower the life-cycle costs.

To gain insight into the dynamics of a system, hammer test measurements and numerical models are often combined in many engineering fields (see, for instance, [5–8]). In railway tracks, by numerically reproducing hammer tests, (1) track parameters (i.e. stiffness and damping of the railpad and ballast) can be derived by varying the model parameters until the model response agrees with the field hammer test measurements [9–12] or (2) insight can be gained on the contribution of a track component to the dynamic response of the track [13]. The main components of the track structure are schematically shown in Figure 4.1.

Figure 4.1: Main components of the track structure

Whereas previous studies focused on studying the ballast-sleeper interaction [14–16], our main focus lays on the sleeper-railpad-rail interaction. Our main focus is the sleeper-railpad-rail interaction. We concentrate on tracks with monoblock sleepers, which are becoming more popular due to their higher bearing capacity [17]. For instance, monoblock sleepers have been used in new lines and repairs in the Netherlands since 1990. Although studies have investigated the dynamic behavior between the monoblock sleeper, railpad and rail [13, 18, 19], the track models have not been able to reproduce the characteristic peaks and dips in the receptance function between 450 and 1000 Hz [20, 21]. This frequency range corresponds to the dynamic behavior of the sleeper and it includes frequencies related to track defects, such as squats [4, 22]. Therefore, a track model that can reproduce the dynamics of monoblock sleepers is needed to help understand the deterioration of the superstructure (i.e. sleeper, railpad and rail) of railway tracks with monoblock sleepers of many worldwide networks.

The new model should cover a wide frequency range so that an extensive understanding of the vibrations and interactions of tracks with monoblock sleepers can be obtained. The applicable frequency range of the model is closely related to the modeling of
the rail and railpad, major contributors to the dynamic behavior of tracks at frequencies above 500 Hz [20, 21, 23]. The rail and the sleeper are often defined as beams [21, 24, 25], while the railpad is usually defined as one spring-damper pair; an exception in which the railpad is defined as a line of spring-dampers is [26]. By representing the rail as a beam, the models are limited to frequencies of 1500 Hz because the cross-sectional deformation of the rail is not considered; this deformation significantly influences the track dynamics above 1500 Hz [27].

To overcome the 1500 Hz limitation, some models represent the rail as a combination of parts [27, 28]. Other models calculate the vibration modes of the rail cross-section that are subsequently introduced in the track model [29, 30]. In both cases, the models represent the railpad as three groups of springs located in a line of the rail section on which the support is assumed to act, rather than by a single point. Except for the second-order pin-pin resonance which is related to the discrete nature of the model, the models cannot reproduce the track dynamics at frequencies between 1500 and 3000 Hz [20, 31]. This limitation is caused by (1) the railpad being modeled as a point or a line instead of as an area or (2) a constant railpad stiffness being used instead of one that is frequency-dependent [23, 32, 33].

In this paper, a 3D FE model capable of reproducing the dynamics of tracks with monoblock sleepers over a wide frequency range of 300-3000 Hz is developed. For this purpose, the dynamic behavior of monoblock sleepers is included by modeling sleepers in 3D. The nominal rail cross-section is also modeled to consider cross-sectional deformations. By modeling 3D rails and 3D sleepers, the railpad encompasses an area in which the railpad is defined with multiple spring-damper pairs. In this manner, we investigate the effect of simplifying the railpad from an area to a line. We also study the effect of using a frequency-dependent stiffness of the railpad on the vertical track dynamics.

Using these rail and railpad representations and the Explicit Finite Element (FE) approach, a complete study was performed on the squat growth process from prediction [22] to validation [34] so that early detection and maintenance measures could be developed [35]. In light of these results, we are encouraged to employ the Explicit FE approach to study whether the new model can overcome the 1500 Hz limitation. Furthermore, with this approach, non-linear materials, such as those with frequency-dependent stiffness, can be defined.

In summary, we investigate the vertical dynamics of tracks with monoblock sleepers using a 3D FE model. The major contribution of this work is threefold: (1) to gain insight into the main characteristics that define the dynamic behavior of tracks with monoblock sleepers; (2) to develop a 3D FE model for tracks with monoblock sleepers to derive track parameters; and (3) to investigate the influence of the railpad representation on the vertical dynamic behavior of tracks.

The paper is divided as follows. In Section 4.2, nominal-geometry-based 3D FE half- and whole-track models are developed for hammer test reproduction. In Section 4.3, the field hammer test measurements used for model validation are presented, and the track parameters are summarized. In Section 4.4, the simulations are performed, the models are validated using hammer test field measurements, and the characteristic modes of tracks with monoblock sleepers are identified. The influence of the railpad representation on the vertical track dynamics is studied in Section 4.5. Finally, the main findings
and guidelines are presented in Section 4.6, and the main conclusions are drawn in Section 4.7.

4.2. A NOMINAL-GEOMETRY-BASED 3D FE MODEL

A nominal geometry-based 3D FE track model is developed to reproduce hammer test measurements and to explore the dynamic behavior of tracks with monoblock sleepers (see an overview in Figure 4.2). The model is 24 sleeper bays long. Numerical experiments indicated that this modeled track length is sufficient for the loading cases and frequency range studied here because the same results were obtained with a model that is 52 sleeper bays long. By employing the shorter model, the calculation time decreases significantly.

![Figure 4.2: Overview of the 3D finite element model and close-ups of the track components](image)

The rails are modeled using solid elements with their nominal cross-sectional geometry. As illustrated in the rail close-up in Figure 4.2, the rail cross-section is meshed heterogeneously. To reduce the computational time required for the model, small-size elements are only defined in two particular areas: the rail top, so that the high-frequency phenomena caused by the hammer impact are captured, and the rail foot, whose flapping is a relevant vibration in the frequency range of interest [27]. The realistic representation of the rail is completed by considering the rail inclination. The rails are defined as elastic, which is suitable for the loads applied. In the 3D FE model, the rail ends are fixed in the three directions. The impact wave reflection at the rail ends does not influence the hammer test numerical reproduction because the same results were obtained with non-reflective boundary conditions applied at the rail ends.

The sleepers are modeled using solid elements according to their nominal geometry (see the sleeper close-up in Figure 4.2). Although sleepers are composed of reinforced concrete, a non-prestressed homogeneous elastic material is assumed here because Gustavson and Gylltoft [16] concluded that linear elastic material properties can
be defined for non-cracked concrete sleepers under train passages.

A sleeper is considered under nominal conditions when it is partly buried in a recently tamped ballast. The sleeper and the ballast are in contact at a few points so that the ballast applies vertical, lateral and longitudinal restrictions on the sleeper. Our investigation has shown that, besides the vertical stiffness of the ballast, the lateral stiffness should be considered when reproducing hammer tests. The lateral restriction is considered in the model by fixing the nodes of the sleeper connected to the ballast in the lateral direction (i.e. \( u_x \neq 0 \) except for \( u_{Rx} = 0 \), see the R nodes in Figure 4.2). Although this is a simplification of the nominal field condition, it is a suitable representation for vertical hammer test reproduction because it considers the flexibility of the sleeper and the lateral (x-direction) stiffness of the ballast.

In the modeling of 3D rails and 3D sleepers, the rail seat encloses an area that the railpad covers entirely for a nominal case. In reality, the railpad is accompanied by clamps that form the fastening system that fixes the rail to the support (see Figure 4.1). Under this configuration, downward forces compress the railpad and upward forces are resisted by the clamps. Although this behavior is not linear, the fastening system is commonly modeled using one linear spring and damper because, up to now, it has shown to be a suitable simplification. In this paper, the railpad is represented with multiple discrete spring and damper pairs that consist of one linear spring and one viscous damper (see the railpad close-up in Figure 4.2). The rail seat is divided into nine rows (the lateral direction of the rail is the x direction in Figure 4.2) and nine columns (the longitudinal direction of the rail is the z direction in Figure 4.2) to define 81 fastening elements. The upper nodes of the spring-damper pairs are connected to the rail and the lower nodes to the sleeper. The stiffness and damping of the railpad are derived by fitting simulations to field hammer test measurements (see Section 4.4.1).

The ballast is modeled with multiple pairs of one linear spring and one viscous damper (see the ballast close-up in Figure 4.2). The spring-damper pairs are defined homogeneously under the sleepers. The upper ballast nodes are connected to the sleeper, and the lower nodes are fixed in all three directions. As for the railpad, the stiffness and damping of the ballast are derived by fitting simulations to measurements (see Section 4.4.1).

4.2.1. Half-Track

To complete the study, a half-track model is also constructed. This model is typically used when modeling hammer tests [24, 25] or vehicle/wheel-rail/track interactions [26, 28]. The purpose of this model is to determine the effect of the half-track simplification when numerically reproducing track dynamics and investigate if whole-track models are required to numerically investigate tracks with monoblock sleepers. The geometrical symmetry of the track with respect to its center line is applied such that the half-track model consists of one rail, and half of the sleepers, railpads, and ballast. The symmetry condition is defined in the model by allowing only in-plane displacements of the sleeper nodes in the symmetry plane.
4.2.2. **Time Domain - Frequency Domain Approach**

Hammer test measurements are numerically reproduced by applying an Implicit-Explicit FE approach using an ANSYS-LS-Dyna sequential solution. First, the equilibrium state is calculated with ANSYS Mechanical. Small displacements are assumed, and the preconditioned conjugate gradient (PCG) solver is used [36]. Subsequently, the response of the system to hammer test excitation is simulated in the time domain with LS-Dyna using the central difference integration scheme [37]. Calculation in the time domain may be required to further improve the model to (1) consider non-linear materials, which may be necessary as the stiffness of the railpad is frequency-dependent [23, 32, 33] and (2) define frictional contact between the rails and solid railpads for the purpose of studying the deterioration of the fastening system.

The impact of the hammer is modeled as a force applied on the top of the rail, where the element size is 1 mm. Based on measured hammer tests, a triangular force is defined with a zero magnitude at $t_0$, a maximum magnitude at $t_1$ and a vanishing magnitude at $t_2$, as shown in Figure 4.3a. In this manner, a small set of easily obtainable parameters is required to define the excitation force. The approximation of a measured impact as a triangular function has been numerically determined to not affect the results of this study.

The response of the track is studied when the excitation is applied on the rail over a support (on-support) and on the rail between two supports (mid-span). The response of the system is measured at a rail section 2 cm away from the excitation application area based on the hammer-sensor configuration in the field, as shown in Figure 3.1. The element size in the measured area is also 1 mm. The small time step of $1.22 \times 10^{-7}$ s guarantees that Courant’s convergence criterion [38] is fulfilled during the Explicit calculation for the size of the elements.

The numerically computed signals (i.e. force $F(t)$ and acceleration $a(t)$) are transferred to the frequency domain via the Fast Fourier Transform (FFT), and the receptance function of the track (displacement per unit force) is calculated according to (4.1). By employing (4.1) to estimate receptance functions, the effect of the noise is minimized at the output because ambient random noise is removed via the averaging process of the
4.3. HAMMER TEST MEASUREMENTS

In this section, the field hammer test measurements are shown. A set of hammer tests are combined to define a baseline state according to the approach presented in Section 4.3.1. The track site and its measured baseline state are presented in Section 4.3.2.

4.3.1. BASELINE STATE

The numerical simulations are compared to a baseline state of a track with monoblock sleepers [41]. The baseline state represents the characteristic response of a track at a certain moment, which may differ from the nominal characteristic response because tracks deteriorate under service and environmental conditions. The baseline state is defined as a combination of reference locations that do not show damage to the naked eye. The reference locations are chosen employing the rational subgroup concept and combined according to the process-monitoring technique called control charts [42]. In this manner, the small differences between sleeper spans are considered. The sample mean \( \hat{H}_0(f) \) and the sample variance \( \hat{\sigma}_0^2(f) \) are calculated according to equations (4.2) and (4.3),

\[
\hat{H}_0(f) = \frac{1}{m} \sum_{i=1}^{m} \hat{H}_i(f) \\
\hat{\sigma}_0^2(f) = \frac{1}{m-1} \sum_{i=1}^{m} (|\hat{H}_i(f)| - |\hat{H}_0(f)|)^2
\]

where \( m \) is the number of reference locations, \(|•|\) is the absolute value of the complex number, and \( \hat{H}_i(f) \) is the receptance function of measurement \( i \).

In this paper, to guarantee that the baseline state is statistically representative [42], 21 reference locations were included. These locations were divided into three groups separated by more than 24 m (i.e. at least 40 sleeper spans). Each group consisted of 7 cross-spectrum [39].

\[
H_d(f) = \frac{1}{(2\pi f)^2} \frac{S_{aF}(f)}{S_{FF}(f)} = \frac{1}{(2\pi f)^2} \sum_{n=1}^{N} \sum_{m=1}^{N-m-1} a[m+n]F[m]e^{-j2\pi fn} \sum_{n=1}^{N} \sum_{m=1}^{N-m-1} F[m+n]e^{-j2\pi fn}
\]

(4.1)

where \( f \) is frequency, \( S_{aF} \) is the cross-spectrum between the force and the acceleration, and \( S_{FF} \) is the autospectrum of the force. Receptance functions show the response of tracks with monoblock sleepers to vibrations as a function of frequency, and they are commonly used in railways to present frequency response functions, such as in [24, 26, 28, 40].

In this paper, the receptance function is used to identify the characteristic frequencies of the track at the resonances and antiresonances (i.e. peaks and dips, respectively) in Section 4.4.2. The fit of the simulations to the measurements is assessed by comparing the measured and numerically calculated receptances in Section 4.4.3. The receptance functions are also used to examine the influence of the railpad representation on the vertical track dynamics in Section 4.5.
consecutive supports. In this manner, the baseline state considers differences between consecutive supports and between track sections.

4.3.2. Track site
The track site investigated contained UIC54 rails on monoblock sleepers with a sleeper distance of 0.6 m. The Young’s modulus of the rail is 210 GPa, the density is 7800 kg/m, and the Poisson’s ratio is 0.3. The monoblock sleeper was of type NS90. The sleeper has a density of 2480 kg/m³, and the concrete type is C50/60. According to [43], uncracked C50/60 concrete has a Young’s modulus of 39 GPa and a Poisson’s ratio of 0.2. The stiffness and damping values of the railpad and the ballast are derived by fitting simulations to the measurements in Section 4.4.1.

At each reference location, five impacts were applied using a Bruël & Kjær 8202 hammer with a plastic tip, and the response was measured with a unidirectional accelerometer fixed on the rail with a magnet (see Figure 3.1). The input and output signals were averaged for each location to minimize random error. The derived average values of the excitation parameters for the on-support and mid-span excitations are as follows: \( t_0 \) is 0, \( t_1 \) is 0.25 ms and \( t_2 \) is 0.4 ms for the two configurations, and \( F_{\max} \) is 3097 N for the on-support excitation and 3302 N for the mid-span excitation. The small difference of 7% between the mid-span and the on-support excitation force is due to the variability introduced by performing the tests manually.

To transform from the time domain to the frequency domain, the transform function of the measured signals was calculated via FFT. The frequency range to be considered, 300-3000 Hz, was determined by the mass of the hammer and the material of the tip used. The lower frequency range (i.e. below 300 Hz) was excluded because the light hammer with the hard plastic tip used did not excite the track components that dominate the response of the track at low frequencies, such as subgrade or ballast [44]. Furthermore, to numerically reproduce measurements at frequencies lower than 300 Hz, the ballast should not be simplified to spring-damper pairs [44], as the loading condition (i.e. with or without a vehicle) influences the results [45] and the sleeper-ballast interaction must be modeled in detail [14, 16]. The upper frequency range (i.e. above 3000 Hz) was also not evaluated, as the hammer used provided reliable results (i.e. drop of the autospectrum of the force > −10 dB) only up to 3000 Hz [23].

Figure 3.8 displays the mean (\( \tilde{H}_0(j\omega) \)) and two standard deviations with respect to the mean (i.e. \( \tilde{H}_0(j\omega) \pm 2\tilde{\sigma}_0(\omega) \)) in the frequency range of 300-3000 Hz. If a normal distribution of measurements is assumed, 95% of the measured receptance functions at the reference locations are between the two standard deviation lines that quantify the variability in the receptance function between different sleeper bays.

4.4. Results
In this section, the numerical simulations are compared to the baseline state. In Section 4.4.1, the track parameters are derived by fitting the simulations to the mean of the baseline state. In Section 4.4.2, the main characteristic of a track with monoblock sleepers are identified, and their origin is investigated by comparing the receptance functions
of the half- and whole-track models and employing modal analysis. In Section 4.4.3, the numerical reproduction of its main characteristics using the 3D FE whole-track model is analyzed.

### 4.4.1. Deriving Track Parameters

The numerically-calculated receptance functions of the 3D FE model were compared with the baseline state of the ballasted track with monoblock sleepers. A parameter analysis of the stiffness and damping of the railpad and ballast was performed to fit the numerical simulations to the track measurements. The resulting stiffness and damping values of the railpad are 1560 MN/m and 67.5 kNs/m, respectively. The stiffness and damping values of the ballast are 45 MN/m and 32 kNs/m for the half-track model, and 90 MN/m and 64 kNs/m for the whole-track model. Hence, half of the whole-track model corresponds to the half-track model. The closest match of the numerical receptance functions to the measured baseline state is shown in Figure 4.5.

### 4.4.2. Identifying Characteristic Modes of the Track

According to the literature, the vertical dynamics of a ballasted track are characterized by four features in the frequency range of 300-3000 Hz [24]: sleeper anti-resonance, rail resonance, pin-pin anti-resonance and second-order pin-pin anti-resonance. This description matches the dynamics of tracks with biblock sleepers, but, in the case of monoblock sleeper tracks, the measured track response differs considerably from the dynamic response described in the literature (see Figure 4.5). The origin of the six characteristics is investigated in this section.

#### 4.4.2.1. Pin-Pin Resonance

The pin-pin resonance (M4) is the track mode in which the rail vibrates with its modes on the sleepers, and it mainly depends on the sleeper distance and rail properties [24]. The pin-pin resonance is numerically obtained at the same frequency (i.e. 1050 Hz) for
4.  **VERTICAL DYNAMICS OF TRACKS WITH MONOBLOCK SLEEPERS**

![Graph](image)

**Figure 4.5**: Influence of the track symmetry simplification on the receptance function of the track: (—in) the measured mean ($\hat{H}_0(f)$), (■) two standard deviations with respect to the mean ($\hat{H}_0(f) \pm 2\hat{\sigma}_0(f)$), — simulation with the whole-track FE model, and — simulation with the half-track FE model. M stands for the measurements, W for the whole-track model and H for the half-track model.

The whole- and half-models for the mid-span case (see W4 and H4 in Figure 4.5b), but in the on-support receptance case, the pin-pin anti-resonance occurs at a slightly higher frequency in the half-track model than in the whole-track model (W4 occurs at 1182 and H4 at 1250 Hz in Figure 4.5a). W4 is closer to M4 than H4. This result is related to the weight of the sleeper that participates in the vibration mode. When the impact is applied on the rail top located on the support, more than half of the sleeper contributes to the vibration. Because the half-track model includes only half sleepers, the excited
mass is slightly less than the excited mass in the whole-track model. The reduced weight is reflected in the response of the system vibrating at the pin-pin mode at the higher frequency. In the mid-span case, the impact is applied too far away from the support, and is thus too weak to noticeably excite more than half of the sleeper.

The second-order pin-pin resonance (M6) occurs when the wavelength is half of the wavelength of the pin-pin resonance. The whole- and half-track models reproduce this dip at the approximately measured frequency (compare W6 and H6 to M6 in Figure 4.5).

In summary, the half-track simplification has no influence on the pin-pin resonances at mid-span. For on-support excitation, the half-sleeper should be modeled slightly heavier in a half-track model than in a whole-track model.

4.4.2.2. Sleeper Bending Modes

The measured baseline state shows two characteristics at approximately 530 and 1300 Hz (M1 and M5 in Figure 4.5, respectively). These characteristics are reproduced by the whole-track model, but not by the half-track model. As the track properties and simulation parameters are identical for the two models, the inability of the half-track model to reproduce the characteristics is caused by the application of the symmetry with respect to the center line of the track. Thus, a whole-track model should be employed.

The first characteristic occurs in the frequency range of 500-700 Hz, in which the sleeper is one of the dominant contributors to the track dynamic behavior [20, 21]. To confirm that the characteristic of the track was closely related to the sleeper, a numerical modal analysis of the sleeper was performed. The vertical interaction of the sleeper with the railpad and the ballast were considered, as these interactions significantly influence the vibration modes of the sleeper [18]. The derived values of the stiffness and damping of the railpad and ballast were used. The lateral stiffness of the ballast was considered in the same way as in the 3D FE model, this is, the sleeper nodes connected to the sleeper were fixed in the lateral direction ($u_{Rx} = 0$).

The numerically calculated natural frequencies of the sleeper are shown in Table 4.1. The table summarizes the rigid modes (i.e. translation and rotation) and the first five bending modes. Although sleepers have twisting modes in the frequency range of interest [13], they are not mentioned here because they were found to be negligible for the vertical receptance function. The sleeper vibrates in its second and fourth bending modes at 540 and 1337 Hz, which are approximately the frequencies of the M1 and M5 characteristics of the track. The second and fourth bending modes of the sleeper are asymmetrical modes that cannot be reproduced with half sleepers. Consequently, the half-track model does not exhibit the dips that correspond to the second (M1) and fourth (M5) bending modes of the sleeper, whereas the whole-track models show the two characteristics (see W1 and W5 in Figure 4.5).

In summary, the second and fourth bending modes of the monoblock sleeper are two main characteristics of the vertical track dynamics. As the two modes are asymmetric modes that cannot be reproduced with a half-track model, a whole-track model should be employed.

4.4.2.3. Rail Resonances

The numerically calculated receptances of the half- and whole-track models display different behaviors between 800 and 1000 Hz. Their disagreement is clearly observed in the
mid-span receptance in Figure 4.5b, in which the whole-track receptance peaks at 900 Hz (W3) and the half-track receptance dips at 920 Hz. According to the modal analysis of the monoblock sleeper, the third bending mode of the sleeper occurs at 937 Hz, and, because it is a symmetric mode with respect to the center line of the track, one would expect it to occur in both the whole- and the half-track simulations. However, because the half-track model is not able to reproduce M3 in Figure 4.5, we suspect that the characteristic of the track dynamics is related to other track vibrations than the sleeper’s third bending mode.

To gain better insight into the dynamic behavior of the track between 800 and 1000 Hz, a modal analysis was performed of the whole and half-track models to compare their vibration modes. At 890 Hz, the vertical mode shown in Figure 4.6a was observed only in the whole-track model. Each rail vibrates at the known rail resonance but the two rails are in antiphase, which is why this vibration mode cannot be reproduced with the half-track model. In contrast to tracks with biblock sleepers that weakly connect the dynamic behavior of the two rails, the combination rail-monoblock sleeper-rail results in a characteristic feature in the dynamic behavior of the track at 890 Hz. The rails vibrate in-phase at the rail resonance at 715 Hz (see Figure 4.6b). The track receptance displays a characteristic peak near this frequency (see M2, W2, and H2 in Figure 4.5).

In summary, tracks with monoblock sleepers have two rail resonances depending on whether the rails vibrate in-phase or in antiphase. Therefore, a whole-track model should be employed.

Figure 4.6: Rail resonance: (a) at 890 Hz with rails vibrating in antiphase and (b) at 715 Hz with rails vibrating in-phase.
4.4.3. Reproducing Vertical Track Dynamics with the Whole-Track Model

At frequencies lower than 1700 Hz, all of the main characteristics are reproduced with the whole-track model, although some are not reproduced precisely at the measured frequencies. In the on-support receptance function (Figure 4.5a), the pin-pin antiresonance provides the largest frequency difference on a percentage basis, with the simulation at 1182 Hz (W4) and the measurement at 1094 Hz (M4) (a small 8% difference). In contrast, if the response of the system is studied in the mid-span, the pin-pin resonance is reproduced almost precisely at the measured frequency of 1040 Hz (M4 and W4). The fact that the small disagreement is only observed in the on-support case suggests that there may be a phenomenon occurring in the support that is not properly modeled with the current representation of the support.

In the mid-span case (Figure 4.5b), the maximum frequency difference between the measured and simulated characteristics is determined by the dip related to the sleeper’s second bending mode (M1 and W1). The measured and simulated features differ by less than 10% (M1 occurs at 508 Hz and W1 at 556 Hz).

The model can largely reproduce the dynamic response of the track between 1700 and 3000 Hz. The shapes of the measured and numerically calculated receptances correspond well in the high frequencies; namely, the dominant second-order pin-pin antiresonance occurs at 2850 Hz for both the measurement (M6) and simulation (W6). At approximately 2100 Hz, however, the simulation exhibits a behavior different from the measurement. This difference is more noticeable in the on-support case than in the mid-span case, which again suggests that the support influences the response of the track around this frequency.

In the 3D FE model presented here, the railpad is simplified to a combination of springs and dampers that may not consider all of the phenomena in the frequency range of interest. Additionally, a constant railpad stiffness value was used over the entire frequency range, while laboratory measurements indicate that the stiffness of the railpad is frequency-dependent [23, 32]. The influence of the railpad modeling and the frequency-dependent stiffness of the railpad on the slight mismatch between the simulated and measured characteristics are investigated in Section 4.5.

4.5. The Railpad Representation

In this section, the influence of the railpad representation on the vertical track dynamics is investigated. In Section 4.5.1, different multiple spring-damper pair configurations of the railpad are examined. In Section 4.5.2, the frequency-dependent stiffness of the railpad is studied.

4.5.1. Representation of the Fastening

In this section, we investigate how the distribution of the multiple spring-damper pairs in the rail seat area influence the dynamic response of the track. Five different railpad configurations were studied (see Figure 4.7). Although it is expected that the use of more elements yields better accuracy, the studies allow us to quantify the difference between the cases. The dynamic response of the track to hammer excitation is calculated for the
different configurations and the results are analyzed. To identify trends in the numerically calculated frequency responses, the main characteristics are denoted by different symbols, as listed in Table 4.2.

### 4.5.1.1. Rail Seat Grid Density

This study investigates the influence of the number of spring-damper pairs that form the entire rail seat. Three fastening element configurations that encompass the entire rail seat in the lateral and longitudinal directions of the rail are defined. The differences between the models are attributed to the number of elements that compose the area, as shown in Figure 4.7a, 4.7b and 4.7c. The finest configuration contains a grid of 9x9 spring-damper pairs, whereas the coarsest configuration contains a grid of 3x3 spring-damper pairs, and the intermediate configuration contains a grid of 5x5 spring-damper pairs. The 1x1 configuration was not included because the inclined 3D rail is unstable if only one spring-damper pair is connected to the sleeper.

Figure 4.8 shows the receptance functions of the modeled track with different rail seat grid densities for the on-support and mid-span excitations. A comparison of the simulated receptance functions reveals that the models show significantly different behaviors, with the exception of the pin-pin characteristics (M4 and M6).

At high frequencies, the coarser the element grid in the rail seat, the larger the number of peaks and troughs there are in the receptance functions. This result is a reflection of the increase in the movement of the system when fewer fastening elements are defined. In the 9x9 rail seat grid, almost all of the nodes in the rail and the sleeper are connected, which limits the movements that may produce high frequency vibrations. Conversely, in the case of the coarse 3x3 rail seat grid, additional relative movement of
4.5. The railpad representation

Figure 4.8: Influence of the rail seat grid density on the track receptance functions: (—) the measured mean ($\hat{H}_0(jf)$), (Ⅲ) two standard deviations with respect to the mean ($\hat{H}_0(jf) \pm 2\hat{\sigma}_0(f)$), and simulated with different rail seat grid densities (—) 9x9, (——) 5x5, and (····) 3x3. The characteristics are the second bending mode of the sleeper (◦, M1), the in-phase rail resonance (*, M2), the antiphase rail resonance (×, M3), the pin-pin resonance (△, M4), the fourth bending mode of the sleeper (□, M5), and the second-order pin-pin resonance (∇, M6)

the rail occurs between the nodes connected to the fastening elements, which results in a large number of peaks and dips at high frequencies (Figure 4.8).

The sleeper resonances (M1 and M5) occur at lower frequencies in the receptance function with a coarse rail seat grid because the sleeper is less restricted. That is, the 3x3 configuration relaxes the restrictions applied to the sleeper, which causes a reduction in the natural frequencies of its bending modes. Consequently, the fit to the measurements
is better for the second bending mode of the sleeper (blue ○ and black ○). However, part of the sleeper’s vibration is lost in the rail seat with the coarse rail seat grid. The fourth bending mode of the sleeper (M5) is almost unreproduced by the 3x3 configuration (blue □), whereas the 9x9 configuration (green □) shows a dip in the numerically calculated receptance function at the measured frequency (black □). In summary, although a coarse rail seat provides a good fit to the second bending mode of the sleeper, the configuration is not able to completely transfer the dynamic behavior of the support to the rail, and consequently, the fourth bending mode of the sleeper is not reproduced as in the measurements.

The rail resonances (M2 and M3) are numerically reproduced at considerably lower frequencies as the rail seat coarsened. Similar to the case of the bending modes of the sleeper, the decrease in frequency is related to the flexibility of the system. Fewer fastening elements enable more relative movement which results in a more flexible system; consequently, the characteristics are evident at lower frequencies. The in-phase rail resonance produces a closer fit to the measurements (see red * and black *) but yields a worse fit to the antiphase rail resonance (see red × and black ×). This result occurs because the antiphase resonance is highly dependent on the coupling of the two rails through the sleeper, and this coupling is weakened for the coarse rail seats.

In summary, the 9x9 configuration gives the best overall fit to the measurements. Thus, if the railpad is represented with multiple spring-damper pairs, a fine grid should be defined.

4.5.1.2. NUMBER OF COLUMNS
In this study, the effect of the length of the railpad on the dynamic behavior of the track is investigated. Three models with fastening systems consisting of nine, five and one columns of spring-damper pairs in the longitudinal direction of the rail are developed (see Figure 4.7a, 4.7d and 4.7e). In each case, the columns cover the entire width of the rail seat in the lateral direction of the rail. The numerically calculated receptance functions for on-support and mid-span excitation are displayed in Figure 4.9.

At high frequency, the multiple-column models show a receptance function with few peaks and dips, while the one-column model shows many peaks and dips. These behaviors were also observed for models with coarse rail seat grids due to the freedom of movement of the rail between the fastening elements (see the 3x3 configuration in Figure 4.8). In the case of a variable number of columns, the difference is caused by the increase of numerical noise when changing from an area support to a line support. Under a perturbation such as a hammer’s excitation, the freedom of movement obtained by reducing the rail seat to a single column enables the system to vibrate at a high frequency, which differs from the field measurements. This finding highlights the importance of modeling the rail seat as an area to reproduce track dynamics with 3D FE models in a high frequency range.

The rail resonances (M2 and M3) shift to lower frequencies when the number of columns of spring-damper pairs decreases, which is similar to the result for a coarser rail seat grid density (see Section 4.5.1.1). Because the railpad covers a smaller area, a longer piece of rail has the freedom to move, which results in rail resonances at lower frequencies. The best total fit to the measurements is obtained with the 9x9 configuration
4.5. **The Railpad Representation**

Figure 4.9: Influence of the number of column of the rail seat on the track receptance functions: (---) the measured mean ($\hat{H}_0(j\omega)$), (■) two standard deviations with respect to the mean ($\hat{H}_0(j\omega) \pm \Delta\sigma(\omega)$), and simulated with different number of columns (---) 9x9, (—) 9x5, and (····) 9x1. The characteristics are the second bending mode of the sleeper (◦, M1), the in-phase rail resonance (⋆, M2), the antiphase rail resonance (×, M3), the pin-pin resonance (△, M4), the fourth bending mode of the sleeper (□, M5), and the second-order pin-pin resonance (▽, M6).

Because the rail resonances (green ⋆ and ×) are reproduced with 9% and 5% differences in frequency, respectively. The difference in frequency between the entire area rail seat and the line rail seat are 16% and 17% for the in-phase resonance (M2) and the antiphase rail resonance (M3), respectively.

Reducing the rail seat in the longitudinal direction of the rail caused the rail length between the supports to increase, and consequently, lower pin-pin frequencies (M4 and
M6) are obtained. The maximum difference in frequency of 6% corresponds to the limit cases (i.e. 9x9 and 9x1).

The sleeper resonances (M1 and M5) are marginally affected by the variation in the number of columns of spring-damper pairs in the longitudinal direction of the rail. The exception is the line rail seat, which shows significant variation in frequency due to the loss of stability when changing from an area support to a line support and the loss of vibration of the support transferred to the rail. For the area rail seat configurations, the differences in the frequencies of the sleeper bending modes are less than 4%. This frequency difference is minor because the second and fourth bending modes of the sleeper influence the dynamic behavior of the track in the vertical and lateral directions of the rail but not in the longitudinal direction. Thus, the width of the rail seat in the longitudinal direction of the rail does not affect the bending modes of the sleeper. The lateral configuration of the rail seat and the stiffness value of the railpad are key aspects of the complete transfer of the dynamic behavior of the support to the rail.

In summary, the 9x9 configuration is closest to the measured main characteristics. Therefore, the railpad should encompass the rail seat.

4.5.2. FREQUENCY-DEPENDENT STIFFNESS OF THE RAILPAD

One factor that causes the slight mismatch between the simulated and measured characteristics is the use of a constant railpad stiffness value over the entire frequency range. This argument is based on the observation (during the fitting process) that softer railpads led to a closer fit in the mid-frequency range, whereas stiffer railpads improved the numerical reproduction of the high-frequency response of the track. This behavior is in agreement with the frequency-dependent stiffness of railpads, in which the stiffness increases for higher frequencies [23, 32, 33]. Therefore, in this section we assess if the frequency-dependent stiffness of the railpad should be considered when numerically reproducing receptance functions.

For this purpose, numerical simulations with the 9x9 railpad configuration were run with different constant railpad stiffnesses and then combined to form an overall receptance function, \( H(f) \). The numerically calculated receptance functions were combined by employing a multimodel approach [46]. Each individual model uses a stiffness value that better fits the measurements in different frequency ranges. In this case study, three models are defined, where \( H_A(f) \), \( H_B(f) \) and \( H_C(f) \) are the numerically calculated receptance functions and \( k_A, k_B \) and \( k_C \) are different values of the railpad stiffness. To estimate the overall receptance function, the following piecewise function is defined:

\[
H(f) = \begin{cases} 
H_A(f), & \text{if } f \leq f_A^2 \\
H_{T1}(f), & \text{if } f_A^2 < f \leq f_B^1 \\
H_B(f), & \text{if } f_B^1 < f \leq f_B^2 \\
H_{T2}(f), & \text{if } f_B^2 < f \leq f_C^1 \\
H_C(f), & \text{if } f_C^1 < f 
\end{cases} 
\]  

(4.4)

where \( f \) is frequency, and \( H_{T1}(f) \) and \( H_{T2}(f) \) are the transition receptance functions calculated using cubic spline interpolations in MATLAB. \( f_A^2, f_B^1, f_B^2 \) and \( f_C^1 \) are the frequencies that define the frequency ranges of the transition receptance functions \( H_{T1}(f) \)
and $H_{T2}(f)$.

After a parameter analysis of the stiffness of the railpad, the best fit of the overall receptance function to the measurement is shown in Figure 4.10, and the respective parameter values are summarized in Table 4.3. The values of $f_{A1}$, $f_{A2}$, $f_{B1}$ and $f_{B2}$ were chosen so that no relevant characteristic of the receptance function was close, and a significantly good overall fit between the measured and simulated characteristics was obtained when using the multimodel approach.

The maximum difference in frequency between the characteristics of the numerical simulations and measurements is less than 8% for the on-support and mid-span configurations (see Figure 4.10a and 4.10b, respectively). Five of the six characteristics are reproduced within a 4% frequency difference with respect to the measurements. Thus, a significant improvement of the fit is obtained in the frequency range of 300-3000 Hz if a frequency-dependent railpad stiffness is employed instead of a constant stiffness. Furthermore, the numerical simulation is capable of reproducing the dominant peak at approximately 2000 Hz if a stiffer railpad stiffness value is applied, (see the values of $k$ and $k_C$ in Table 4.3).

Table 4.3: Railpad stiffness values for Study 5

<table>
<thead>
<tr>
<th>Constant stiffness</th>
<th>Frequency-dependent stiffness</th>
<th>Frequency limits</th>
</tr>
</thead>
<tbody>
<tr>
<td>$k$</td>
<td>$k_A$</td>
<td>$k_B$</td>
</tr>
<tr>
<td>1.56 GPa</td>
<td>0.78 GPa</td>
<td>1.56 GPa</td>
</tr>
</tbody>
</table>

In summary, this study shows that the frequency dependency of the stiffness of the railpad should be included in numerical models to reproduce the dynamic response of tracks with monoblock sleepers in the frequency range of 300-3000 Hz. As further research, the optimal number of frequency ranges could be calculated to obtain the closest fit and to smooth the transition receptance functions. Part of future research could also be to account for the frequency-dependent stiffness of the railpad in the model.

### 4.6. Major findings and guidelines

#### 4.6.1. Vertical dynamic response of tracks with biblock sleepers and tracks with monoblock sleepers

In the frequency range of 300-3000 Hz, tracks with monoblock sleepers and tracks with biblock sleepers display different dynamic behaviors (see Figure 4.11 for the measured average receptance functions calculated according to Section 4.3.1). The dynamic behavior of tracks with biblock sleepers is characterized by four main features: sleeper antiresonance (B1), rail resonance (B2), pin-pin resonance (B4), and second-order pin-pin resonance (B6), which are denoted by gray dots. The contribution of the track components to the vibration modes is well-understood. Although the rail resonance (M2), pin-pin antiresonance (M4), and second-order pin-pin antiresonance (M6) can be observed in tracks with monoblock sleepers, the information available is insufficient to completely define the dynamic behavior of these tracks without the additional consid-
4. VERTICAL DYNAMICS OF TRACKS WITH MONOBLOCK SLEEPERS

Figure 4.10: Influence of the frequency-dependent stiffness of the railpad on the track receptance functions: (—) measured and simulated with (—) \( k_A \), (—) \( k_B \), and (—) \( k_C \) in (a) on-support and (b) mid-span. The characteristics are the second bending mode of the sleeper (○, M1), the in-phase rail resonance (∗, M2), the antiphase rail resonance (×, M3), the pin-pin resonance (△, M4), the fourth bending mode of the sleeper (□, M5), and the second-order pin-pin resonance (▽, M6). The improvements of the reproduction of the characteristic at approximately 2000 Hz is indicated with (·) operation of a minimum of two additional features.

First, the second bending mode (M1) and fourth bending mode (M5) of the monoblock sleeper should be included. These two antisymmetric bending modes of the sleeper contribute to the dynamic response of the track with two clearly visible dips (see Figure 4.11). The bending modes of the monoblock sleeper contribute to the receptance
function, changing it with respect to the frequency response of tracks with biblock sleepers, due to the geometry of the sleeper. The monoblock sleeper is a solid concrete beam with a variable section that vibrates as a single body. However, the biblock sleeper consists of two concrete masses connected by a flexible pipe that largely decouples the interaction of the masses, making the bending modes less pronounced in the dynamic response of the track. In light of these results, monoblock sleepers dominate a wider frequency range (up to 1400 Hz) than the range from 400 to 1000 Hz assumed in the literature [20, 21].

Second, the antiphase rail resonance (M3), which is caused by the interaction be-
between the two rails through the monoblock sleeper, should be considered. For a half-track model with monoblock sleepers, the rail resonance is the antiphase movement between the rail and sleepers. However, the definition must be extended for a whole-track model with monoblock sleepers because two rails are present and the rail resonance can occur with the two rails in-phase (M2) or in antiphase (M3) (see Figure 4.6 for the mode shapes). The reason why the antiphase rail resonance is found in tracks with monoblock sleepers and not in tracks with biblock sleepers is also related to the geometry of the sleepers. Due to the relative freedom of movement between the two masses of the biblock sleeper, one rail is not affected by the other if hammer testing is performed. In contrast, because monoblock sleepers are considerably stiffer, relevant interactions occur between the two rails. This new feature broadens the concept of the known rail resonances.

4.6.2. **Modeling the Vertical Dynamic Behavior of Tracks with Monoblock Sleepers for Track Parameter Derivation**

To reproduce the vertical dynamics of tracks with monoblock sleepers in the frequency range of 300-3000 Hz, 3D FE whole-track models with monoblock sleepers should be used. The frequency range of application of the model is not limited by the nature of the elements when the rail is modeled with solid finite elements instead of with beam elements. Furthermore, in contrast to a half-track model, a whole-track model reproduces the antiphase rail resonance and the second and fourth bending modes of sleepers, which lead to characteristics in the receptance function of the track (see Section 4.2.1). With the exception of the 2000 Hz characteristic, a fit to the measurements can be obtained with a maximum of 10% difference between the measured and simulated frequencies of the characteristics M1, M2, M3, M4, M5 and M6 by performing a parameter variation of the railpad and ballast stiffness and damping (see Section 4.4.1). Therefore, the presented 3D FE model can be used for parameter derivation under unloaded conditions. Because some track components show nonlinear behavior from unloaded to loaded conditions, hammer test measurements can be performed and simulated in loaded tracks, if required, so that track parameters can be derived under loaded conditions.

4.6.3. **Railpad Configuration on the Rail Seat**

Based on the results, the railpad should be modeled with a fine grid of spring-damper pairs covering the entire rail seat for a nominal case. If the fastening system is modeled as a line instead of as a complete rail seat area, four of the six characteristics of the vertical track dynamics are reproduced at considerably lower frequencies for the same railpad and ballast parameter values of a nominal condition. In the cases studied, the difference was up to 17%. By increasing the number of spring-damper pairs in the longitudinal direction of the rail, the freedom of movement is reduced because more rail elements along its length are connected to the sleeper, and the characteristics consequently occur at higher frequencies.

The number of spring-damper pairs in the lateral direction of the rail also significantly influences the numerical dynamic response of the track. The sleeper bending modes are primarily transmitted to the rail by the railpad elements in the lateral direc-
4.6. Major Findings and Guidelines

Table 4.4: Change of characteristic frequencies to different railpad configurations

<table>
<thead>
<tr>
<th>Characteristic</th>
<th>Rail seat grid density</th>
<th>Number of columns</th>
</tr>
</thead>
<tbody>
<tr>
<td>M1</td>
<td>coarser grid, ↓ frequency</td>
<td>no change</td>
</tr>
<tr>
<td>M2</td>
<td>coarser grid, ↓ frequency</td>
<td>less columns, ↓ frequency</td>
</tr>
<tr>
<td>M3</td>
<td>coarser grid, ↓ frequency</td>
<td>less columns, ↓ frequency</td>
</tr>
<tr>
<td>M4</td>
<td>no change</td>
<td>less columns, ↓ frequency</td>
</tr>
<tr>
<td>M5</td>
<td>coarser grid, ↓ frequency</td>
<td>no change</td>
</tr>
<tr>
<td>M6</td>
<td>no change</td>
<td>less columns, ↓ frequency</td>
</tr>
</tbody>
</table>

The frequency shifts observed between the rail seat configurations investigated have a potential practical application because the rail seat configurations resemble deterioration conditions. This information, which is summarized in Table 4.4, can be used in the development of condition monitoring systems.

In this paper, the baseline state was defined based on reference locations in which the state of the railpad was assumed to be nominal. However, other measurement locations may not be in a nominal state, as represented by the 9x9 model. Railpads may exhibit some degradation caused by service and environmental conditions. For instance, partially covered rail seats are representations of partially worn railpads. Consequently, the frequency response of the track measured at these locations would exhibit frequency shifts with respect to a nominal state and consequently, the vehicle/track interaction changes [47]. These shifts might be used to monitor the condition of the tracks by dynamic response-based train-borne systems in the future.

4.6.3.1. Information for the Development of Condition Monitoring Systems

4.6.4. Future Steps: Improve the Modeling of the Fastening

Research has revealed the need to obtain a better understanding of the rail-fastening-sleeper interaction because the condition of the support seems to influence the development of rail surface defects such as short pitch corrugation [48, 49], squats [41, 47] and ratcheted rail ends at rail joints [50]. One improvement of the railpad modeling would be to account for frequency-dependent stiffness of the railpad, as it is shown in Section 4.5.2. Another improvement in the modeling of the fastening could be a more realistic representation of the railpad and clamps. On the one hand, the rail seat with a solid railpad is a continuous support, not a discrete one. On the other hand, the two clamps on each side of the rail foot apply the toe load at specific points on the rail foot. By defining solid railpads and clamps, the behavior of the fastening can be modeled more realistically, with a downward force compressing the railpad and an upward force being resisted only by the clips. Although these new models are more complex and time consuming than the linear spring-damper assumption, a more realistic representation
of the fastening system may contribute to a better understanding of track deterioration, such as the initiation and growth of rail surface defects.

4.7. **Conclusions**

The vertical dynamic behavior of tracks with monoblock sleepers was investigated (1) to gain an understanding of the vibrations and interactions occurring in the track and (2) to develop a suitable model to study the deterioration of this track type. A nominal geometry-based 3D finite element model was presented and validated with a set of field hammer test measurements. Good agreement was found between the numerical simulations and the measurements in the frequency range of 300-3000 Hz. In our case, the difference in frequency between the numerically calculated and measured characteristics was less than 10%. Furthermore, the influence of the modeling methods of the railpad on the vertical dynamics of tracks with monoblock sleepers was explored. From the presented work, the following conclusions can be drawn:

1. For tracks with monoblock sleepers, a whole-track model should be used because:
   
   (a) The second and fourth bending modes of the sleepers, which are two asymmetrical modes with respect to the center of the track, are the origin of two major characteristics in the receptance function of monoblock sleeper tracks.
   
   (b) The antiphase rail resonance, for which the two rails of the track vibrate in antiphase with respect to the sleeper and in antiphase with respect to one another, is the origin of a dominant characteristic of the vertical dynamics of tracks with monoblock sleepers.

2. Due to the fourth bending mode, the monoblock sleepers characterize the dynamic behavior of the track for frequencies up to 1400 Hz, in contrast to the 1000 Hz frequency limit reported in the literature.

3. If a nominal railpad is defined with multiple spring-damper pairs, the railpad should encompass the rail seat with a fine grid to ensure the following conditions:
   
   (a) The vibrations of the support (i.e. sleeper and ballast) are completely transferred to the rail, (e.g. nine rows by nine columns grid transmits the fourth bending mode of the sleepers for the track studied, but a three by three grid does not).

   (b) The strong coupling between the two rails through the sleeper can be considered. Thus, the characteristic related to this coupling is practically numerically reproduced at the measured frequency instead of at the considerably lower frequencies obtained with coarse grids.

   For degraded conditions, the railpad may be better simulated with spring-damper pairs distributed on a coarse grid or that only cover part of the rail seat.

4. The railpad should be modeled with frequency-dependent stiffness so that, contrary to a constant stiffness value, the characteristic at approximately 2000 Hz and the six characteristics of tracks with monoblock sleepers can be reproduced close to the measured frequencies in the frequency range of 300-3000 Hz.
The insight into the dynamic behavior of tracks with monoblock sleepers and the influence of track components on some dynamic characteristics, such as sleeper bending modes and rail resonances, may be used for the design of tracks and components. In addition, the presented 3D FE model provides a reasonable basis for deriving track parameters under unloaded conditions. The frequency shifts of the characteristics related to different deterioration states of the railpad might be used for condition identification and monitoring.

Future research should focus on a more realistic representation of the fastening system because the rail-fastening-sleeper interaction may be related to rail surface defect initiation and growth (e.g. squats). Some possible studies are (1) defining frequency-dependent stiffness of railpads, (2) investigating the effect of simplifying solid railpads to discrete elements, (3) defining solid railpads and clamps to account for the different behavior of the fastening system under upward and downward forces and (4) examining the fastening from the rail fixing mechanism perspective (i.e. vertical, lateral and longitudinal restrictions). A better insight into the interaction between track components can be useful for investigating track deterioration and for fastening design and optimization. By better understanding track deterioration, the maintenance schedule might be improved and maintenance costs might be lowered.

REFERENCES


An investigation into the modeling of railway fastening

In this paper, the influence of modeling the fastening with solid railpads on the vertical dynamics of railway tracks with monoblock sleepers is investigated. A 3D finite element (FE) model is presented with four different fastening representations: (1) commonly used spring-damper pair, (2) area covering spring-damper pairs, (3) solid railpad connected to the rail, and (4) solid railpad in frictional contact with the rail and fixed to the support by preloaded springs, which represent the clamps. The response of the four models to hammer excitation is simulated in the time domain, and the calculated response is transformed into the frequency domain to analyze how the models capture the seven main characteristics of tracks with monoblock sleepers. The numerical results show that the model with solid railpads and clamps reproduce the seven characteristics at a maximum frequency difference of 6%, while the conventional model with spring-damper pairs reaches only a 27%. In the improvement of the fit from multiple spring-damper railpad models to solid railpad models, the two key aspects identified are the Poisson’s effect and the damping of the ballast. Additionally, the railpad type investigated showed a frequency-independent behavior, at least with acceptable error. In view of the close fit, the models with solid railpads can be used for track and fastening design and to derive track parameters to, for instance, study the deterioration of tracks.
5. AN INVESTIGATION INTO THE MODELING OF RAILWAY FASTENING

5.1. INTRODUCTION

When trains roll over railway tracks, undesired vibrations and noise arise. On the one hand, the rolling noise, often high pitched, is a nuisance for the people living close to the railway tracks [2, 3]. On the other hand, the high wheel/rail contact forces and vibrations of the track components contribute to the deterioration of the track itself [4–8], vehicles [9] and sometimes, buildings and structures in the surroundings [10, 11].

To reduce the noise to acceptable levels and to delay or decelerate the deterioration of the track, vibration attenuation components are installed in the vehicle-track system. For instance, resilient wheels are often used in tram and metro lines [12, 13]. In the track, the railpad is a key component because it influences the rolling noise [14] by attenuating the vibrations due to the interaction between the rail and the sleeper [15, 16]. In addition, the stiffness of the railpad affects the growth of rail defects such as short pitch corrugation [17, 18], and the condition of the support (i.e. railpad and clamps) influences the growth of squats [19, 20]. Furthermore, impact forces at rail joints increase for stiff railpads [21] so that plastic deformation and ratchetting appears at the rail ends [8]. The relation between rail deterioration and fastening condition points to the need of closely examining the behavior of the support. The study of the fastening system may shed some light on the track deterioration process.

In the literature, numerous studies have been carried out to obtain information about the behavior of railpads under working conditions. The railpad is mainly represented as one pair of a linear spring and a viscous damper in parallel (see, for instance, [22–24]). By applying this approach, it was found that soft railpads are favorable to transfer the loads to the sleepers and ballast but wheel-rail contact forces are not always lowered [15]. Also, by reducing the railpad stiffness, the noise radiation from the sleeper is reduced but that from the rail is increased [14].

In other models in which the rail seat is defined as an area or line instead of one connecting point between the rail and the sleeper, the railpad consists of multiple spring-damper pairs. Studies show that considering the longitudinal and/or lateral dimensions of the rail seat significantly influences the dynamic response of the track [25]. For instance, the dominant pin-pin resonance (i.e. when the rail vibrates with the nodes on the sleepers) becomes a significantly attenuated resonance if the longitudinal dimension of the railpad is considered [26]. Since the track dynamics are affected when the railpad is modeled covering an area, the vehicle-track dynamic response is influenced too. The magnitude and position of the characteristic wheel-rail contact forces significantly change depending on the configuration of the fastening system [19].

These studies identify the lateral and longitudinal dimensions of the railpad as relevant parameters for accurately numerically reproducing measured data. One can expect that not only the dimensions but also the displacement restrictions in the lateral and longitudinal directions have implications in the vertical track dynamics because the fastening is a rail fixing mechanism. The rail is fixed to the support by clamps which constraint the displacement of the rail in the vertical, lateral and longitudinal directions. These constraints result in interactions between the rail and the railpad, and the rail and the clamps in the vertical, lateral and longitudinal directions. With a spring-damper pair railpad, only the vertical interaction is considered; this may have implications for the reproduction of the vertical track dynamics.
In this paper, a Finite Element (FE) track model with two types of 3D solid railpads is presented. In the first railpad model, the solid railpad is connected to the rail. This model is compared to a track model with railpads defined as multiple spring-damper pairs presented in [27] so that the effects of simplifying solid railpads to spring-damper pairs is investigated. In addition, the influence of the lateral and longitudinal restraints applied to the rail foot is studied (i.e. fix the rail in the lateral and longitudinal directions), whereas in [27], the effect of the lateral and longitudinal dimensions was analyzed. In the second model, the solid railpad is in frictional contact with the rail, and clamps are defined to fix the rail to the support. In this manner, the influence of the clamps as a rail fixing mechanism is investigated. To complete the study, a railpad representation commonly used in railway track models is also considered. The response of all the four models to hammer tests are numerically calculated in the time domain. Then, the numerically calculated signals are transformed into the frequency domain so that the main characteristics of the track are compared to field measurements.

In summary, by examining the frequency responses of the four fastening models and by comparing them to a set of measurements, the consequences of the different simplifications of the fastening system in the reproduction of the vertical track dynamics are analyzed and quantified.

5.2. REPRODUCING FIELD HAMMER TEST MEASUREMENTS

5.2.1. 3D Finite element models

To investigate the influence of the representation of the fastening in the vertical track dynamics, a 3D FE model is developed of a railway track with monoblock sleepers in a ballast bed (see Figure 5.1). Monoblock sleepers are reinforced concrete beams and their use is increasing because they endure larger loads in comparison to other sleeper types [28]. Instead of a half-track model, a whole-track model is required to consider the origin of some important characteristics in the vertical dynamics of tracks with monoblock sleepers; these are, the asymmetric bending modes of the sleepers with respect to the center of the track and the coupling of the two rails through the sleepers [27]. The rail ends are clamped and the track consists of 24 sleeper bays, which is a suitable model length for the reproduction of hammer test measurements [27].

The rails and sleepers are modeled with solid elements according to their respective nominal geometry. The rails are defined as elastic which is suitable for hammer loads which do not plastically deform the rail, unlike train passages, specially in the vicinity of rail surface defects [29, 30]. The sleepers are also defined as elastic, which is considered suitable for the loads investigated [31].

The ballast is modeled with multiple pairs of one linear spring and one viscous damper (see the ballast close-up in Figure 5.1). The spring-damper pairs are defined homogeneously under the sleepers. The upper ballast nodes, which are connected to the sleeper, are denoted by the letter R and are fixed in the lateral direction of the track (i.e. \( u_{Rx} = 0 \)) representing the lateral stiffness of the ballast, which is a suitable representation for reproducing hammer tests [27]. The lower nodes are fixed in all three directions. The sub-ballast and foundation layers are not considered because they are dominant contributors to the dynamic response of the track for frequencies lower than 250 Hz [32], whereas
Due to 3D rails and 3D sleepers, the rail seat encloses an area, which the railpad entirely covers for a nominal case. The railpad is represented with multiple spring-damper pairs or with solid elements (see the upper close-up in Figure 5.1). The railpad is part of the fastening system, which fixes the rail to the support. Four different models of representing the fastening are described in which railpads are modeled with spring-damper (SD) pairs or solid elements (see Figure 5.2).

5.2.1.1. **LINE-SD MODEL**
In railway track models, rail and sleeper are often represented as beams and the railpad is mainly represented as one pair of linear spring and viscous damper in parallel [22–24]. Defining the railpad as one spring-damper pair is not possible in the 3D FE model because the inclined 3D rail is unstable with only one spring-damper pair connected to the sleeper. Thus, the common railpad model is defined as nine spring-damper pairs in line which cover the width of the rail feet and are located in the middle of the support (see Figure 5.2a).

5.2.1.2. **AREA-SD MODEL**
In the second model, the rail seat is covered with multiple discrete linear spring and viscous damper pairs (see Figure 5.2b). The railpad consists of 81 spring-damper pairs divided into nine rows (lateral direction of the rail and the x direction in Figure 5.2b) and nine columns (longitudinal direction of the rail and the z direction in Figure 5.2b). The upper nodes of the spring-damper pairs are connected to the rail and the lower nodes to the sleeper. This model, presented in [27], can be interpreted as an intermediate step between the line-SD model and the solid railpad models (see 5.2.1.3 and 5.2.1.4) to differentiate the effect of considering the lateral and longitudinal dimensions, or the lateral and longitudinal constraints (i.e. fix the rail in the lateral and longitudinal directions).
Figure 5.2: Fastening models (a) in-line spring damper pairs, called line-SD (b) area-covering spring-damper pairs, called area-SD (c) solid railpad connected to the rail, called solid-connected (d) solid railpad in contact with the rail with clamping springs, called solid-contact.
5.2.1.3. SOLID-CONNECTED MODEL

In the third model, the railpad is represented with solid elements covering the entire rail seat (see the solid railpad in the upper close-up in Figure 5.1), and it is defined as elastic. To consider damping, Rayleigh stiffness-proportional damping is defined as \( C = \beta K \), where \( C \) is the damping matrix, \( K \) is the stiffness matrix and \( \beta \) is the stiffness proportional damping constant [33]. The upper surface of the railpad is connected to the rail and the lower surface to the sleeper (see Figure 5.2c). This means that there is no relative movement between the connected surfaces. Since there is no frictional contact defined, clamps are not required because the rail is fixed to the support through the railpad.

Comparing this model to the area-SD model, the consequences of simplifying a solid railpad to multiple spring-damper pairs is investigated. Since the rail-railpad and railpad-sleeper connections are the same in the two models, any possible difference in the results comes from the lateral and longitudinal restrictions set by the solid railpad.

5.2.1.4. SOLID-CONTACT MODEL

In the fourth model, the railpad is also modeled with solid elements covering the entire rail seat and it is defined as elastic with Rayleigh stiffness-proportional damping, as in the solid-connected model. However, this model considers clamps and the contact between rail and railpad (see Figure 5.2d). Consequently, relative movement between the surfaces of the rail and the railpad may occur. This way, the effect of the clamps as a rail fixing mechanism can be investigated.

The interaction between rail and railpad is modeled using surface-to-surface algorithm employing the penalty method [34]. Coulomb friction with the friction coefficient \( \mu \) is defined between the two track components. The lower surface of the railpad is connected to the sleeper.

Representing the clamps, two springs are defined on each side of the rail. In each side, the two springs are separated a distance \( d_c \) in the longitudinal direction of the rail (see Figure 5.2d). This distance corresponds to the distance between the acting points of a clamp on the rail.

In the field, the clamps apply a toe load on the rail. In the model, this toe load is considered by defining an initial displacement of the springs. The initial displacement corresponds to the displacement difference \( \Delta l \) required to obtain the initial toe load \( F_{TL} \) according to (5.1).

\[
F_{TL} = k_{clamp} \Delta l
\]

(5.1)

where \( k_{clamp} \) is the stiffness of the clamp (\( k \) in Figure 5.2d). The toe load is divided equally between the two springs of a clamp.

The upper node of the spring (T) and its vertical projection on the sleeper (S) are coupled in the x, y and z directions (see Figure 5.2d). These couplings represent the fixing of the clamp to the sleeper. Also, the upper and lower nodes of the spring (T and Q, respectively) are coupled in the x direction of the rail. This coupling represents the lateral constraint that the base plate of the fastening system applies on the rail and railpad.

By defining these couplings and the frictional contact between rail and railpad, the fastening is the mechanism that attaches the rail to the support in the vertical, lateral and longitudinal directions. Thus, in the solid-contact model, the rail can move with
respect to the railpad (1) vertically under the constraint of preloaded springs and (2) longitudinally depending on the friction between rail and railpad.

### 5.2.2. Time Domain - Frequency Domain Approach

Hammer test measurements are numerically reproduced by applying an Implicit-Explicit FE procedure. First, the equilibrium state of the track is calculated using ANSYS (Implicit FE approach). Then, the response of the track to hammer excitation is simulated in the time domain with LS-Dyna (Explicit FE approach). The calculation in the time domain is required to account for the non-linearity of contact in the solid-contact fastening model (see Figure 5.2d). The Implicit-Explicit FE sequence can consider non-linear materials, which may be required when further investigating railpad degradation as its stiffness is frequency dependent [32, 35, 36].

The hammer test measurements are simulated as follows. As input, a triangular force is applied on the top of the rail representing the impact of the hammer. The starting, maximum force and finishing times of the triangular force (\(t_0\), \(t_1\) and \(t_2\) respectively), and the maximum force \(F_{\text{max}}\) are obtained from measurements (see Figure 5.3a). In this manner, the excitation force is defined with a small set of easily obtainable parameters. The simplification of a measured impact as a triangular function has been numerically determined to not affect the results of this study. As output, the response of the system is measured at a rail section 2 cm away from the excitation application area. This distance is defined based on the hammer-sensor configuration in the field measurements as shown in Figure 5.3b. By defining the element size of 1 mm in the excitation and measurement areas and the small time step of \(1.22 \times 10^{-7}\) s, the Courant’s convergence criterion [37] is fulfilled during the Explicit calculation. The central difference integration scheme was used for explicit integration.

![Figure 5.3: (a) Applied force, (b) Hammer test in the field](image)

The input force \(F(t)\) and output acceleration \(a(t)\) are transformed into the frequency domain by means of Fast Fourier Transform. Then, the accelerance function \(H_a(f)\) is calculated according to equation (5.2) so that the effect of the noise is minimized at the output [38].

\[
H_a(f) = \frac{S_{aF}(f)}{S_{FF}(f)} = \frac{\sum_{n=1}^{N} \sum_{m=1}^{N-m-1} a[m+n] F[m] e^{-j2\pi fn}}{\sum_{n=1}^{N} \sum_{m=1}^{N-m-1} F[m+n] F[m] e^{-j2\pi fn}}
\]  

(5.2)
where \( f \) is frequency, \( S_{aF} \) is the cross-spectrum between the force and the acceleration, and \( S_{FF} \) the autospectrum of the force. Finally, the receptance function \( H_d(f) \) is calculated as follows:

\[
H_d(f) = \frac{H_a(f)}{(2\pi f)^2}
\] (5.3)

Receptance functions are often used in railway to investigate track dynamics [22, 26, 29, 39] and it shows the response of tracks for vibrations in terms of displacement over force as a function of vibration frequencies. From the receptance function, characteristic frequencies of the track can be identified at the resonances and antiresonances by analyzing peaks and dips, respectively (see Section 5.2.3). Also, simulated receptance functions can be compared to measured ones so that the accuracy of the models on the vertical track dynamics can be analyzed (see Section 5.3).

5.2.3. Vertical dynamics of tracks with monoblock sleepers

In a track with monoblock sleepers, seven characteristics are identified in the receptance function between 300 and 3000 Hz, which is the valid frequency range of study. Frequencies lower than 300 Hz are not included because the light hammer (i.e. 280 gr) is not able to excite the ballast and subgrade, which are the track components that dominate the dynamic behavior of the track at low frequency [22, 40]. Therefore, the measured dynamic response may differ from that of the track under working conditions (i.e. vehicle on the track). Furthermore, to numerically reproduce measurements on the frequencies lower than 300 Hz, the ballast should be modeled such as a halfspace [41], the preload in the foundation caused by the wheels should be considered [40] and the contact in the sleeper-ballast interaction should be accounted for [31, 42]. Regarding the upper frequency limit, the measured data is not reliable for frequencies higher than 3000 Hz [43].

The characteristics are indicated with symbols in a baseline state based on a set of field hammer test measurements. The measurements were performed at 21 reference locations without visible damage to the naked eye and they were combined so that the small differences in the track structure are considered [20]. The sampling frequency of hammer tests was 20 kHz, and a low-pass filter of 10 kHz was applied to the measured data. Then, the measured signal were transformed into the frequency domain according to (2.1) and (2.2). Figure 5.4 shows the measured mean \( \hat{H}_0(f) \), a band representing two standard deviations with respect to the mean \( \hat{H}_0(f) \pm 2\hat{\sigma}_0(f) \), and the seven main characteristics. The on-support and mid-span configurations correspond to the cases when the excitation on top of the support (sleeper) or between two sleepers, respectively.

The dynamic behavior of the track is defined by the vibration of its components and their interaction. The seven characteristics and the components of the track are related as follows [22, 27]:

- Second bending mode of the sleeper (○, M1): the sleeper vibrates according to its second bending mode. This mode mainly depends on the geometry and material properties of the sleepers.
- In-phase rail resonance (●, M2): the rail vibrates in antiphase with the sleepers and the two rails of the track vibrate in-phase. The rail and railpad properties, such as stiffness and damping, mainly determine the in-phase rail resonance frequency.
5.2. Reproducing Field Hammer Test Measurements

The measured receptance function in the on-support configuration shows another characteristic at approximately 2000 Hz (\(\star\), M7). The origin of this peak is yet unknown, but it is significantly influenced by the stiffness of the railpad [27].

5.2.4. Track Parameters

The track investigated consisted of UIC54 rails, with inclination 1:40, supported on NS90 concrete sleepers. The nominal sleeper distance was 0.6 m, which results in a track model of 14.4 m. The Young’s modulus, density and Poisson’s ratio of the rail were 210
GPa, 7800 kg/m and 0.3, respectively. For the sleeper, the Young’s modulus, density and Poisson’s ratio were 39 GPa, 2480 kg/m, and 0.2, respectively.

For the spring-damper models, the stiffness and damping values of the fastening and the ballast were taken from [27]. These values correspond to the best fit obtained with the area-SD model (Figure 5.2b) using the field measurements. For the fastening, the stiffness was 1560 MN/m and the damping was 67.5 kNs/m. The stiffness value corresponds to a very stiff pad. This value is in agreement with measurements and studies performed in Dutch railway tracks, see [43] and [44]. For the ballast, the stiffness was 45 MN/m/half sleeper and the damping was 32 kNs/m. The stiffness and damping values were divided homogeneously among the number of spring-damper pairs forming the railpad.

For the models with solid railpads, initially the Young’s modulus of the railpad \( E \) was calculated according to (5.4), based on its stiffness \( k \) derived with the area-SD model.

\[
E = \frac{kl_0}{A_0}
\]  

where \( l_0 \) is the initial thickness and \( A_0 \) is the initial area of the railpad. In the 3D FE models with solid railpads, the railpad is 134 mm in width, 150 mm in length and 4.5 mm in thickness. However, the numerical results showed that the best fit between the measurements and solid railpad models was obtained with a different set of parameters (see Table 5.1); these differences are explained in Section 5.3.1.2 and 5.3.2. The FC9 railpad is made of rubber with cork particles inside and its Poisson’s ratio is 0.45 (value obtained from the manufacturer). The Rayleigh stiffness-dependent damping \( \beta \) is 0.02, which is obtained during the fitting process.

For the solid-contact model, the solid railpad is defined with the same material properties \( (E, \beta, \nu) \) as in the solid-connected model. The stiffness and nominal toe load of one clamp are 800 kN/m and 10 kN, respectively [45]. The properties are divided equally between the two springs that represent one clamp. The nominal toe load of 5 kN/spring is introduced in the model by defining an initial displacement of 12.5 mm in each of the two springs (see (5.1)). The distance \( d_c \) between the springs is 72 mm, based on field observations. Regarding the contact between rail and railpad, the friction coefficient is 0.75.

Table 5.1 summarizes the material properties of the railpad, ballast and clamp (if defined) employed to accurately reproduce the measurements with the spring-damper, solid-connected and solid-contact models.

Regarding the input impact data, the time and maximum force values are the same as in [27]. If the rail is excited with the on-support configuration, the maximum force \( F_{\text{max}} \) is 3097 N. If the rail is excited with the mid-span configuration, the maximum force \( F_{\text{max}} \) is 3302 N. The impact time values are the same for the two configurations and correspond to 0, 0.25 ms and 0.4 ms for \( t_0, t_1 \) and \( t_2 \), respectively.
5.3. **RESULTS: COMPARISON OF SIMULATIONS TO MEASUREMENTS**

The simulations were fit to the measurements by varying the stiffness and damping of the railpad and ballast. The values were distributed homogeneously among the elements forming the railpad or ballast. Other parameters, such as the clamping force and rail and sleeper parameters, were assumed constant. The numerically reproduced and measured receptance functions are shown in Figure 5.5 and 5.6. The comparison of the models with spring-damper railpads and solid railpads is presented in Section 5.3.1. The influence of the clamps and the contact between rail and railpad is investigated in Section 5.3.2. The frequencies of the simulated characteristics \( f_s \) are compared to the measured frequencies \( f_m \) according to the following expression:

\[
J_f = \left| \frac{f_s - f_m}{f_m} \right| \times 100
\]

(5.5)

The differences \( J_f \) are graphically displayed in Figure 5.7 to facilitate the comparison between models.

### 5.3.1. **SOLID-CONNECTED VERSUS SPRING-DAMPER PAIRS**

Figure 5.5 and 5.6 shows the best fit of the spring-damper models and the solid-connected model. The line-SD model reproduces two of the characteristic frequencies at a \( J_f \) frequency difference with respect to the measurements larger than 18% and the numerical reproduction at high frequency is far from the measurements (see Figure 5.5). With the area-SD model, six of the seven characteristics are reproduced with a maximum of 10% frequency difference with respect to the measurements. The exception is the 2000 Hz resonance which differs almost 300 Hz from the measured value, which is 14% lower. With the solid-connected model the characteristics are reproduced closer to the measured values. All the seven characteristics differ less than 10% from the measurements (see Figure 5.7). Furthermore, five out of the seven characteristics show a frequency difference smaller than 5% with respect to the measured characteristics. Thus, the nu-

### Table 5.1: Parameters sets for the best fit between simulations and measurements

<table>
<thead>
<tr>
<th></th>
<th>Spring-damper</th>
<th>Solid-connected</th>
<th>Solid-contact</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Railpad</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( k ) (MN/m)</td>
<td>1560</td>
<td>151</td>
<td>151</td>
</tr>
<tr>
<td>( c ) (kNs/m)</td>
<td>67.5</td>
<td>0.02</td>
<td>0.02</td>
</tr>
<tr>
<td>( \nu )</td>
<td>0.45</td>
<td>0.45</td>
<td>0.45</td>
</tr>
<tr>
<td><strong>Clamp</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( k ) (kN/m)</td>
<td>800</td>
<td>12.5</td>
<td>72</td>
</tr>
<tr>
<td>( \Delta l )</td>
<td>12.5 mm</td>
<td>72 mm</td>
<td></td>
</tr>
<tr>
<td>( d_c ) (mm)</td>
<td>72</td>
<td>72</td>
<td></td>
</tr>
<tr>
<td>( \mu )</td>
<td>0.75</td>
<td></td>
<td></td>
</tr>
<tr>
<td><strong>Ballast</strong></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>( k ) (MN/m)</td>
<td>45</td>
<td>45</td>
<td>45</td>
</tr>
<tr>
<td>( c ) (kNs/m)</td>
<td>32</td>
<td>64</td>
<td>96</td>
</tr>
</tbody>
</table>
Figure 5.5: Influence of the rail pad longitudinal dimension on the track receptance functions: (—) the measured mean (\( \hat{H}_0(f) \)), (■) two standard deviations with respect to the mean (\( \hat{H}_0(f) \pm 2\hat{\sigma}_0(f) \)), and simulated with (—) spring-damper pairs on a line, and (—) spring-damper pairs on an area. The characteristics are the second bending mode of the sleeper (●, M1), the in-phase rail resonance (∗, M2), the antiphase rail resonance (×, M3), the pin-pin antiresonance (▲, M4), the fourth bending mode of the sleeper (■, M5), the second-order pin-pin antiresonance (▼, M6) and the 2000 Hz resonance (∗, M7) [27]

merical reproduction of the vertical track dynamics with the solid-connected model is closer to the measurements than the receptance function of the spring-damper models. In summary, the constraints in the longitudinal and lateral directions introduced by defining the solid railpad connected to the rail and sleeper contribute to a better representation of the measurements.

To obtain the best-fits shown in Figure 5.6, two different parameter sets were em-
5.3. RESULTS: COMPARISON OF SIMULATIONS TO MEASUREMENTS

Figure 5.6: Influence of the rail pad modeling on the track receptance functions: (——) the measured mean ($\hat{R}_0(f)$), (●) two standard deviations with respect to the mean ($\hat{R}_0(f) \pm 2\hat{\sigma}_0(f)$), and simulated with (——) spring-damper pairs on an area, (—) solid railpad connected to the rail, (—) solid railpad in contact with the rail. The characteristics are the second bending mode of the sleeper (●, M1), the in-phase rail resonance (●, M2), the antiphase rail resonance (X, M3), the pin-pin antiresonance (▲, M4), the fourth bending mode of the sleeper (■, M5), the second-order pin-pin antiresonance (▼, M6) and the 2000 Hz resonance (★, M7).

ployed for spring-damper railpad models and solid railpad models (see Table 5.1). The two main differences between the set of parameters are the damping of the ballast (Section 5.3.1.1) and the stiffness of the railpad (Section 5.3.1.2).

5.3.1.1. DAMPING OF THE BALLAST
The damping of the ballast is doubled in the solid-connected model than in the area-SD model. The increase in the damping of the ballast was required to lower the recep-
5.3.1.2. Stiffness of the Railpad

The Young's modulus $E$ of the railpad material defined for the solid-connected model is significantly lower than the corresponding Young's modulus $E_k$ if the stiffness value of the railpad of the area-SD model is used in (5.4). This difference is related to the acting directions of the railpad. With the area-SD model, the railpad only acts vertically and the relative lateral and longitudinal displacement between rail and sleeper are not constrained. However, the solid railpad acts in all three directions limiting the relative movement between the rail and the sleeper.

The relation between a vertical compression and the corresponding extension in another direction is usually expressed as the Poisson's ratio. Analyzing the influence of the Poisson's ratio on the stress-strain relation of a material may explain why the partial constraint of the railpad (i.e. the nodes in the upper and lower surfaces of the railpad...
are connected to the rail and sleeper, respectively) results in a stiffer track in the vertical direction.

If a linear isotropic material is assumed, the generalized Hooke’s law is defined as follows [46]:

\[
\begin{align*}
\epsilon_x &= \frac{1}{E} (\sigma_x - \nu \sigma_y - \nu \sigma_z) \\
\epsilon_y &= \frac{1}{E} (\sigma_y - \nu \sigma_x - \nu \sigma_z) \\
\epsilon_z &= \frac{1}{E} (\sigma_z - \nu \sigma_x - \nu \sigma_y)
\end{align*}
\] (5.6)

where \(\sigma_x, \sigma_y, \text{ and } \sigma_z\) are the stress in the \(x, y\) and \(z\) directions respectively, \(\epsilon_x, \epsilon_y, \text{ and } \epsilon_z\) are the strain in the \(x, y\) and \(z\) directions respectively, \(\nu\) is the Poisson’s ratio and \(E\) is the Young’s modulus. In the case of railpads, the Young’s modulus is an unknown material property because the environmental and service conditions change its value with respect to the nominal one. Measurements can be performed to derive the in-service Young’s modulus.

If simple compression is applied on a non-constrained railpad, it can deform in all three directions and stress only arises in the direction of compression (i.e. \(\sigma_y \neq 0\) and \(\sigma_x = \sigma_z = 0\)). Thus, (5.6) results in:

\[
\frac{\sigma_y}{\epsilon_y} = E_s
\] (5.7)

Thus, the Young’s modulus \(E_s\) is the vertical stiffness for a non-constrained railpad under simple compression.

However, if the railpad is constrained in the \(x\) and \(z\) directions and compression is applied in the vertical \(y\) direction, the Poisson’s effect cannot occur because \(\epsilon_x = 0\) and \(\epsilon_z = 0\). Consequently, non-zero stresses arise in the \(x\) and \(z\) directions:

\[
\begin{align*}
\sigma_x &= (\nu \sigma_y + \nu \sigma_z) \\
\sigma_z &= (\nu \sigma_x + \nu \sigma_y)
\end{align*}
\] (5.8)

By substituting (5.8) in (5.6), the relation between the stress and strain in the vertical direction \(y\) results in:

\[
\frac{\sigma_y}{\epsilon_y} = E_c \frac{(1 - \nu)}{(1 + \nu)(1 - 2\nu)}
\] (5.9)

where \(E_c\) is the Young’s modulus for a constrained railpad. Thus, the stiffness in the vertical direction \(y\) depends on the Poisson’s ratio for a constrained railpad under compression.

To obtain the same stiffness in the vertical direction for a constrained and non-constrained railpad under compression, a smaller Young’s modulus is required if the railpad is constrained. This is shown by combining (5.7) and (5.9):

\[
\frac{\sigma_y}{\epsilon_y} = E_c \frac{(1 - \nu)}{(1 + \nu)(1 - 2\nu)} = E_s
\] (5.10)

In the solid-connected model, the railpad is not completely constraint in the \(x\) and \(z\) directions, only the upper and lower surfaces are constrained whereas the lateral and
longitudinal surfaces are free of constraints. This intermediate restriction means that the Young's modulus of the solid-connected railpad $E_{sc}$ is between the Young's modulus of the simple compression $E_s$ and the Young's modulus of the fully constrained compression $E_c$. In the case studied in this paper, if the Young's modulus of the simple compression is calculated based on the stiffness $k$ value of the area-SD model, $E_s$ is 349 MPa and $E_c$ is 92 MPa. By using these two values, the numerical results differed significantly from the measurements. By varying the Young's modulus $E$ to fit simulations to measurements, the $E_{sc}$ of 151 MPa was derived which is between the $E_s$ and $E_c$ as expected. With a Young's modulus of 151 MPa, the best fit between the solid-connected model to the measurements shown in Figure 5.6 was obtained.

5.3.2. SOLID-CONTACT VERSUS SOLID-CONNECTED

Figure 5.6 also shows the best fit of the solid-contact model, besides the spring-damper and solid-connected models. The solid-contact model reproduces the characteristics the closest to the measurements; the maximum difference in frequency is 6% (see the columns of solid-contact in Figure 5.7).

The numerically calculated receptance function closely follows the measured receptance function and is mostly within the band of measurements ($H_0(f) \pm 2\sigma_0(f)$) between 300 and 1500 Hz. Although an overall improvement of the fit is obtained in this frequency range, the receptance value at the rail resonance (M2) of the solid-contact model increases with respect to that of the solid-connected model and farther from the measurement. This peak becomes more prominent because the lateral displacement of the Q nodes of the rail are coupled to the sleeper (see Figure 5.2d), so that the connection between the rail and the support is stiffened. Although the damping of the ballast is increased for attenuating the amplitude of this vibration mode (see Table 5.1), the receptance value of the in-phase rail resonance (M2) is still slightly higher for the solid-contact model than for the solid-connected model.

For frequencies higher than 1500 Hz, the difference between the two models with solid railpads is larger than at lower frequencies (see Figure 5.6). The simulation shows high peaks and deep dips, unlike the blunt peaks and shallow dips of the measurement. These high frequency phenomena is not related to the friction coefficient between rail and railpad because barely changes were observed in the simulated receptance functions when investigating different friction coefficients between rail and railpad (i.e. from 0.3 to 1.4), yet it appears when contact is defined.

In the frequency range 1500-2100 Hz, the receptance function is strongly related to the relative longitudinal movement between rail and sleeper according to the results in Section 5.3.1. The 2000 Hz resonance shifts to higher frequencies if the longitudinal constraint increases. In that regard, the solid-contact model is the most restricted of the three models. In the spring-damper pairs railpad, the relative longitudinal displacement between rail and sleeper is not constrained, whereas it is significantly limited in the solid-connected railpad. In the solid contact model, the preload of the clamps makes the longitudinal displacement even more restricted and it is explained as follows. First, due to the small vertical impact force applied and the large toe load, one can assume that the tangential force is smaller than the frictional limit between rail and railpad so that there is no local slip there; this situation is equal to the solid-connected case. Sec-
ond, a toe load is applied on the rail which raises the stresses in the railpad making it stiffer. Consequently, the solid-contact railpad offers a larger opposition to the relative longitudinal displacement between rail and sleeper than the spring-damper pairs and solid-connected railpads.

Therefore, the 2000 Hz resonance of the spring-damper pairs model is reproduced at the lowest frequency, followed by the solid-connected model, and expecting that of the solid-contact model at the highest frequency. Thus, the peak at 2002 Hz of the solid-contact model corresponds to the measured 2000 Hz resonance. To support this line of reasoning, simulations were carried out with different railpad Young’s moduli. As shown in Figure 5.8, the 2000 Hz resonance follows the trend of shifting to higher frequencies with larger Young’s modulus, which means a more constrained relative longitudinal displacement between rail and sleeper.

![Figure 5.8: Influence of the Young’s modulus in the response of the solid-contact model at high frequency.](image)

The characteristics are the fourth bending mode of the sleeper (M5), the second-order pin-pin antiresonance (M6) and the 2000 Hz resonance (M7).

### 5.4. Discussion

#### 5.4.1. Solid Railpads

The longitudinal and lateral constraints introduced by defining solid railpads contribute to a closer numerical reproduction of the measurements in the entire frequency range of 300-3000 Hz. Such fit could not be achieved with spring-damper railpads because an improvement of the fit at low frequencies came at the expense of worsening the fit at high frequencies, and vice versa. Frequency-dependent stiffness of the railpad was therefore required to improve the reproduction of the measurements at both low and high frequencies [27]. However, for solid railpads presented above, one set of constant parameters ($E$, $\nu$) suffice, at least for the current pad type and for an engineering satisfactory error of 10%.
This means that for the railpad type being studied, the solid railpad model shows frequency-independency with acceptable error. The frequency-independency has two consequences. First, the tests to determine frequency-dependent behavior, which are difficult, time consuming and expensive, can be avoided. Second, the railpad can be modeled as elastic, instead of as complex non-linear material that considers frequency-dependent behavior. Whether this relation between solid-railpad modeling and frequency-independent behavior can be applied to other railpad types should be investigated in future work.

Thus, track dynamics are reproduced closer to measurements by modeling solid railpads. The lateral and longitudinal constraints introduced by defining solid railpads result in a stiffer support changing the numerically calculated track dynamics. Consequently, simulations of the vehicle/wheel-rail/track system may also change. Regarding wheel/rail contact forces, change in the contact dynamic forces is expected in amplitude and frequency for a stiffer support [15, 19]. Concerning track degradation, stiffer railpads accelerate the growth of rail surface defects such as short pitch corrugation [17, 18] and ratcheted rail ends at rail joints [8, 21]. Thus, as solid railpads result in a stiffer modeled support, a faster growth of defects would be predicted with a model with solid railpads than with a model with spring-damper pair railpads, if the same track parameters were used. In addition, increase of rail noise would be predicted with a model with solid railpads because, in a stiffer support, the rail is connected more strongly to the sleeper and substructure so that the rail foot vibrates with smaller amplitudes resulting in a decrease in decay rates [47].

5.4.2. Clamps and Contact
By defining clamps and contact between rail and railpad, a better fit can be obtained between the measurements and the simulations than without clamps. The improvement results from a more realistic fastening system, which means that a significantly more complex model was developed (see the solid-contact model in Figure 5.2d). By defining contact between rail and railpad, clamps needed to be defined, time-domain simulations were required (e.g. explicit FE integration), and longer calculation times were reached. Furthermore, large peaks and deep dips are found in the numerically calculated receptance function in the frequency range of 1500-3000 Hz, in contrast to the blunt peaks and shallow dips of the measured receptance function. Considering the contact between rail and railpad seems to amplify the high frequency phenomena. Nonetheless, the solid-contact model is a suitable model for investigating in-detail fastenings. The adverse problem of modeling the contact should be investigated and solved in further study.

5.4.3. Verification of the Findings with Field Observations
The evaluation of fastening modeling against hammer tests has shown that the relative movement between rails and sleepers influences the vertical track dynamics. The longitudinal and lateral movements are larger under service loading conditions than under hammer test, which may contribute to wear of the railpad, rail foot and clamps. The toe load of the clamps may decrease due to the wear of the clamps and rails. Consequently, the stiffness of the railpad decreases because its value is highly dependent on the toe
load, and the longitudinal and lateral constraints also relax which enables additional longitudinal and lateral displacements of the rail.

As an example, a rail foot worn due to fretting with the clamp is shown in the close-up photograph in Figure 5.9a. In this case the clamp does not fulfill the function of the longitudinal and lateral movements restriction any more, and the deterioration in the support accelerates. The deterioration can be as severe as the railpad shown in Figure 5.9b, which was found in a one-direction traffic mainline track. The railpad in Figure 5.9b shows variable wear along its length. A section of the pad that shows longitudinal flow of the material is observed between the most damaged parts and the undamaged parts. The railpad also shows different wear states along its width, although the transition is less clear than in the longitudinal direction.

Therefore, according to the analyses and the field observations in this paper, the longitudinal and lateral dimensions and interactions between the rail and the sleeper should be considered when reproducing vertical track dynamics between 300 and 3000 Hz.

![Figure 5.9: (a) Worn rail foot (b) Worn railpad with longitudinal material flow](image)

5.4.4. APPLICATION OF 3D FE MODELS WITH SOLID RAILPADS
The close fit between numerical simulations and measurements offers the possibility to derive track parameters and study the degradation of railpad and ballast parameters (i.e. stiffness and damping). The investigation can be extended to study the deterioration
of clamps so that the evolution of fastening health condition as rail fixing mechanism can be studied. Gaining a better understanding of the fastening deterioration (i.e. railpad and clamps) may help to understand the development and growth of rail defects that seems to be closely related to the support condition. Some of the defects are short pitch corrugation [17, 18], squats [19, 20] and ratcheted rail ends at rail joints [8, 21]. By examining derived track parameters and the relation between rail degradation and support condition, adapted or new maintenance measures can be developed so that the occurrence and growth of these defects could be avoided or delayed. As a consequence, the high maintenance costs could be reduced.

5.5. CONCLUSIONS

The influence of the modeling of the fastening system on the vertical dynamics of tracks with monoblock sleepers was investigated. For this purpose, a Finite Element (FE) model with 3D rails and sleepers was developed with four fastening models. The simplest fastening model corresponded to the commonly used spring-damper approach and the most complex model included clamps and frictional contact between rail and solid railpad. The seven main characteristics of measured vertical track dynamics were numerically reproduced with a frequency difference of 27% with the common spring-damper model. A great improvement of the fit was obtained with the most complex fastening system; the frequency difference was a small 6%. A drawback of the model was that the reproduction of the high frequency worsened when including contact. With an intermediate model of solid railpads connected to the rail, the high frequency disturbance was solved and a less-than-10% fit was obtained. This difference is acceptable for engineering purposes such as deriving track parameters for investigating the track deterioration.

The comparison of the fastening models showed that a significantly closer reproduction of the measurements was obtained in the frequency range of 300-3000 Hz with 3D FE models with solid railpads than with spring-damper pair railpads. The key aspects of the improvement obtained with solid railpads with respect to area-covering multiple spring-damper pairs railpads were:

1. The constraints in the longitudinal and lateral directions introduced by defining solid railpads. It means that the Poisson's ratio influences the resulting vertical stiffness of the fastening. Therefore, the longitudinal and lateral interaction between rails and sleeper should be considered when studying vertical track dynamics.

2. The damping of the ballast. The coupling between the two rails of the track and the rail-sleeper connection are strengthened due to solid railpads so that some vibration modes become more dominant. To attenuate these vibrations, the derived damping of the ballast was significantly larger for models with solid railpads than for models with spring-dampers. This difference in ballast was most likely required to maintain the overall attenuation capacity of the track.

3. Frequency independent stiffness of the railpad. For the type of railpad investigated, the solid-railpad modeling was equivalent to defining frequency-dependent
stiffness in spring-damper pairs so that the solid-railpad models showed frequency-independence. The frequency-independency has the advantage of avoiding difficult, time-consuming and expensive tests for obtaining the frequency-dependent behavior of railpads, and complex non-linear material models. Whether this equivalence is valid for other types of railpads should be studied in future work.

Further research will focus on degradation modeling and analysis of mechanical properties of fastenings and its influence on track dynamics. For instance, deteriorated conditions such as worn railpads or loose clamps may be examined to identify parameters to assess the performance of fastening systems. A better understanding of track degradation may contribute to the development of an optimized maintenance schedule and consequently, to lower maintenance costs.

REFERENCES


6

CONCLUSIONS AND RECOMMENDATIONS
6.1. Conclusion from Measurements

Our work has shown that hammer tests are a measurement method capable of significantly contributing to the investigation and understanding of railway tracks. A better understanding of both healthy (i.e. nominal) and damage states was obtained by examining hammer test measurements. In addition, characteristic frequencies of damage states (i.e. signature tunes) were identified which can be used for the development of vehicle-borne dynamic-response-based monitoring systems, such as Axle Box Acceleration (ABA) systems. By employing these monitoring systems, preventive maintenance actions can be taken.

Healthy Track

- The intrinsic variability of tracks is included in an approximate healthy state by defining a baseline state based on a statistically reliable number of locations without visible damage.
- For the reference Insulated Rail Joints (IRJs) investigated in this thesis, the dynamic response of the two halves was identified as almost identical in the frequency range of 50-1200 Hz.
- IRJs with four bolts should be dynamically represented with two receptance functions obtained from exciting the rail above the third bolt and above the fourth bolt.

Damaged Track

- The vertical dynamic behavior of IRJs does not evolve symmetrical to the gap of the joint in a one-directional traffic track.
- In the IRJs investigated, three characteristic frequency bands related to damaged IRJs were derived independent of the type of damage. The three frequency bands are 180-320 Hz, 420-600 Hz and 1000-1150 Hz:
  - At the lowest frequency band, the condition of the sleeper in the ballast influences the characteristic frequency.
  - At the mid-frequency band, two characteristic frequencies (i.e. 440 and 570 Hz) were identified related to damaged IRJs.
  - At the highest frequency band, the prominent characteristic in the receptance function fixes the dominant wavelength of the wave pattern on the rail top.
- A frequency response based statistical method can be used to identify characteristic frequencies of damaged railway tracks using field hammer test measurements.
- Characteristic frequency ranges related to squats and damaged IRJs identified from hammer tests corresponded to the frequency ranges excited by trains rolling over squats and damaged IRJs.
- The frequency ranges related to squats identified using hammer tests were consistent with the increase in severity.
• The frequency ranges identified in the case study with squats were close to the characteristic features in the receptance functions that are dependent on rail and fastening properties. The agreement between the track vibration modes and the identified frequencies suggests a possible relation between squat growth and rail and fastening properties.

6.2. CONCLUSIONS FROM MODELING

Further to the insight gained by examining field hammer test measurements, a deeper understanding of track dynamics was obtained by combining hammer test measurements and state-of-the-art numerical track models. The rail and sleeper were modeled with their nominal 3D geometry, the ballast was represented with multiple spring damper pairs and the fastening was modeled with four different representations: (1) commonly used spring-damper pair, (2) area covering spring-damper pairs, (3) solid railpad connected to the rail, and (4) solid railpad in frictional contact with the rail and fixed to the support by preloaded springs, which represent the clamps. By comparing the different models, the influence of sleeper and fastening modeling was thoroughly investigated.

REPRODUCING VERTICAL DYNAMICS OF TRACKS WITH MONOBLOCK SLEEPERS

• Tracks with monoblock sleepers should be modeled using whole-track models instead of half-track models because:

  – The second and fourth bending modes of the sleepers, which are two asymmetrical modes with respect to the center of the track, are the origin of two major characteristics in the receptance function of monoblock sleeper tracks.

  – The antiphase rail resonance, for which the two rails of the track vibrate in antiphase with respect to the sleeper and in antiphase with respect to each other, is the origin of a dominant characteristic of the vertical dynamics of tracks with monoblock sleepers.

• To model nominal track conditions, the movements of the sleeper should be restricted by the vertical stiffness of the railpad and ballast and by the lateral stiffness of the ballast. This result should be considered for the experimental modal analysis of monoblock sleepers as it is often performed on a free sleeper.

• Due to the fourth bending mode, the monoblock sleepers characterize the dynamic behavior of the track for frequencies up to 1400 Hz, in contrast to the 1000 Hz frequency limit reported in the literature.

FASTENING MODELING

Our 3D FE track models of four different fastening representations were compared and its capacity to reproduce the measured seven main characteristics of measured vertical track dynamics was assessed.

When modeling the railpad with multiple spring-damper pairs, the following aspects should be considered:
• The railpad should encompass the rail seat with a fine grid to ensure the following conditions:

  –The vibrations of the support (i.e. sleeper and ballast) are completely transferred to the rail, e.g. nine rows by nine columns grid transmits the fourth bending mode of sleepers for the track studied, but a three by three grid does not.

  –The strong coupling between the two rails through the sleeper can be considered. Thus, the characteristic related to this coupling is numerically reproduced practically at the measured frequency instead of at the considerably lower frequencies obtained with coarse grids.

For degraded conditions, it may be more suitable to simulate the railpad with spring-damper pairs distributed on a coarse grid or that only cover part of the rail seat.

• The relative longitudinal movement between the rails and the sleepers should be enabled so that the pin-pin resonance resembles the sharp dominant peak of the measurements.

• The railpad should be modeled with frequency-dependent stiffness so that, contrary to a constant stiffness value, the characteristic at approximately 2000 Hz and the six characteristics of tracks with monoblocks sleepers are reproduced with reasonable accuracy compared to the measured frequencies in the range of 300-3000 Hz.

The key aspects to obtained a significantly closer reproduction of the measurements with 3D FE models with solid railpads than with spring-damper pair railpads are:

• The constraints in the longitudinal and lateral directions introduced by defining solid railpads. It means that the Poisson's ratio influences the resulting vertical stiffness of the fastening. Therefore, the longitudinal and lateral interaction between rails and sleeper should be considered when studying vertical track dynamics.

• The damping of the ballast. The coupling between the two rails of the track and the rail-sleeper connection are strengthened due to solid railpads so that some vibration modes become more dominant. To attenuate these vibrations, the derived damping of the ballast is significantly larger for models with solid railpads than for models with spring-dampers.

• Frequency independent stiffness of the railpad. For the type of railpad investigated, the solid-railpad modeling was equivalent to defining frequency-dependent stiffness in spring-damper pairs so that the solid-railpad models showed frequency-independence.
6.3. Conclusions from Fitting Simulations to Measurements

Fitting simulations to measurements was an essential step to perform the comparison between the fastening models because their accuracy was calculated to the same performance index related to real-life measurements. Furthermore, a valuable outcome of the fitting was that the in-service track parameters (i.e. stiffness and damping of the railpad and ballast) were derived for the track investigated.

By comparing our 3D FE track modelling to the measurements, it was found that:

- The seven main characteristics of measured vertical track dynamics were numerically reproduced with a frequency difference of 27% with the common spring-damper model.

- A great improvement of the fit was obtained with the most complex fastening system (i.e. solid railpad in contact with the rail and fixed to the support by preloaded springs); the frequency difference between measurement and simulation was a small 6%. A drawback of the model was that the reproduction of the high frequency shows disturbances when including contact between rail and railpad.

- With an intermediate model of solid railpads connected to the rail, the high frequency disturbance of the most complex model was solved and a less-than-10% fit was obtained. This difference is acceptable for engineering purposes such as deriving track parameters for investigating the track deterioration.

- The fit in characteristic frequencies between measurements and the multiple spring-damper pairs railpad model was of 14%.

- The derived stiffness and damping of the railpad and ballast vary depending on the fastening model used.

6.4. Recommendations on Future Research

As seen in this dissertation, despite the huge research efforts already put in understanding track deterioration, there are still some gaps to be filled. We believe that combining measurements and modeling is a suitable approach for solving some of these problems. More information about track deterioration can be obtained from examining measurements with other analysis tools or by applying the methods presented in this paper to other track defects. Yet, the focus of future research should be on numerical work because the investigation into track deterioration can be faster and easier based on flexible and relatively fast numerical models, than on time-consuming, expensive and often safety threatening field testing. Therefore, future work should target improving the numerical models and the combination between measurements and numerical simulations. In addition, we propose to investigate if it is possible to squeeze more information from measurements and to employ hammer test measurements in the development of vehicle-borne monitoring systems.
APPLICATION OF 3D FE MODELS WITH SOLID RAILPADS

The presented 3D FE model with solid railpads connected to the rail and sleeper provides a reasonable basis for deriving track parameters under unloaded conditions. For the track investigated, the difference in frequencies between the simulated and measured characteristic features was less-than-10%, and simulations were mostly within the variability of the measurements. As future research, three aspects that could be investigated are the following:

- **Railpads: frequency-dependent stiffness**
  
  The type of railpad studied showed frequency-independent behavior when modeled with solid railpads, whereas frequency-dependent stiffness was required to reproduce the high frequency track dynamics when modeled with spring-damper pairs. The frequency-independency has the advantage of avoiding difficult, time-consuming and expensive tests for obtaining the frequency-dependent behavior of railpads, and complex non-linear material models. Whether this equivalence is valid for other types of railpads should be studied in future work.

- **Fitting process between simulations and measurements**
  
  In this work, the simulations were fitted to the characteristic frequencies of the average baseline state. This criterion is quite conservative, in the sense that it tries to find solutions that follow an averaged behavior. A new fitting process should account for the uncertainties coming from real-life data, such as stochasticalities introduced by the sensors and the track itself. Thus, a new fitting criterion could be considering the deviation with respect to a set of measurements, instead of only the distance to an average. In addition to the conservative fitting criterion, the fitting process was performed manually which cost too much time. To speed-up the fitting process, computational intelligence based method could be used. Some examples are particle swarm optimization or genetic algorithms. Further research should analyze different optimization algorithms and study different ways to account for the intrinsic variability of tracks.

- **Fastening deterioration**
  
  The close fit between numerical simulations and measurements offers the possibility to derive track parameters and study the degradation of railpad and ballast parameters (i.e. stiffness and damping). The investigation can be extended to study the deterioration of clamps so that the fastening health condition as rail fixing mechanism can be studied. Gaining a better understanding of the fastening deterioration (i.e. railpad and clamps) may help to understand the development and growth of rail defects that seems to be closely related to the support condition. Some of the defects are short pitch corrugation [1, 2], squats [3, 4] and ratchetted rail ends at rail joints [5, 6]. By examining derived track parameters and the relation between rail degradation and support condition, adapted or new maintenance measures can be developed so that the occurrence and growth of these defects could be avoided or delayed. As a consequence, the high maintenance costs could be reduced.
CONTRIBUTING TO CONDITION MONITORING

The structural health monitoring of railway tracks can be improved based on the investigation presented. The starting point of condition monitoring is to define a healthy state and we have proposed a method to define a baseline state that accounts for track’s variability. Regarding damaged track, in our experimental and numerical work, characteristic frequencies of different deterioration states of railpads, squats and insulated rail joints have been identified from field and simulated hammer test measurements. Furthermore, these frequencies agree with the frequencies derived with an extensively validated ABA system.

- Baseline state

The baseline state can be employed to investigate the general deterioration of railway tracks. The average condition of tracks changes over time due to traffic and environmental conditions so that, at certain point, the track may not fulfill the required performance. To monitor the degradation, the average condition of the same track can periodically be determined by performing measurements. The periodicity of the measurements should be adapted to the usage and tonnage of the line of interest (e.g. shorter periods for a busy line than for a line with little activity). By periodically determining the baseline state, the condition of the track could be monitored using quality control tools such as control charts. Information about the general deterioration of the track may be useful for the assessment of the condition of the track and the planning of maintenance measures.

- Obtaining characteristic frequencies by hammer testing

The promising results of deriving squat and damaged IRJs characteristic frequencies point hammer tests as a complementary tool for developing a complete database of signature tunes. But first, to complete and support the results, a larger set of IRJs should be investigated in future work. In addition, to implement the FRF-based statistical method for a wide set of tracks and conditions, sources of variability, such as temperature and aging of materials, should be further investigated. The application of the frequency response function based statistical method to identify frequencies of isolated damaged conditions can significantly facilitate and speed up the development of dynamic-response-based train-borne monitoring systems such as ABA systems and strain-gauge-instrumented wheelsets, as well as their adaptation to an entire network.

REFERENCES


Evaluating bolt tightness of rail joints using Axle Box Acceleration Measurements

Rail joints are a weak component in railway tracks because of the large impact and wheel-rail contact forces. Every train passage contributes to the deterioration of rail joints, causing visible (e.g. battered rails) and invisible (e.g. loose bolts) damages. The invisible damage cannot be detected by the commonly performed visual inspection, which is labor intensive, unreliable, intrusive and unsafe. In this paper, a vehicle-borne monitoring system is used to automatically detect and assess the tightness condition of bolts at rail joints. The detection method is developed based on field Axle Box Acceleration (ABA) measurements using different bolt tightness conditions. The suitability of the method is assessed by bolt tightness prediction and verification of a set of rail joints in the tram network of Sheffield, United Kingdom. The results show that ABA system can be employed to evaluate bolt tightness conditions at rail joints. With this information better planning for selective preventive maintenance actions can be taken over rail joints.
A.1. INTRODUCTION
Rail joints (RJs) are an important component in many railway networks worldwide; an example of a RJ is shown in Figure A.1a. Although the use of continuously welded rail is nowadays more cost effective due to its significantly longer service life [1], many rails are still connected by rail joints. Furthermore, insulated rail joints (IRJs), in which the two rail ends are separated with an insulating material, are a fundamental component of the safety system in many railway networks.

![Figure A.1: Two examples of rail joints: (a) invisible damage: loose bolts, and (b) visible damage: missing bolt and battered rail surface](image)

When wheels roll over the discontinuity between rail ends, large impact and dynamic wheel-rail contact forces occur accelerating the deterioration of RJs [2–5]. Bolts get loose, rail joints ends become battered, and cracks develop in the rail. These damage conditions at severe state may lead to rail break, derailment and malfunctioning of the signaling system. In addition, impact noise is a nuisance for the people living nearby railway tracks. To guarantee safety and acceptable noise levels, cost-expensive maintenance measures are taken, usually at a late stage when the only solution is replacement of RJs.

The inspection of RJs is mostly performed visually, which is subjective and labor expensive, and often intrusive and unsafe, such as in 24/7 tram and metro networks. Furthermore, there is the additional challenge of assessing the real damage state of RJs. For instance, the RJ shown in Figure A.1b can be easily assessed: one bolt is missing and the rail ends are plastically deformed. Thus, this RJ can be tagged as "severely damaged". However, the condition of the RJ shown in Figure A.1a is visually unknown. Although the RJ seems to be in good condition to the naked eye, cracks in the rail web could be present or the bolts could be loose.

To overcome the limitations of visual inspection and improve safety conditions, vehicle-born monitoring systems have been developed in recent years (see a review in [6]). High quality videos and photos are taken from inspection vehicles, but the detection range of this system is limited to visible damage, such as missing fastenings [7]. Vehicle-borne ultrasonic measurements are also an alternative, used to detect surface and internal rail defects [8, 9], but its reliability is also limited [10]. Different types of deterioration, such as loose bolts or plastic deformation, cannot be detected with ultrasonic measurements.

A vehicle-borne monitoring system that may be able to detect different types of visible and invisible damage at RJs is the Axle Box Acceleration (ABA) system [11–14]. The damage detection method used by the ABA system is based on changes in the dynamic
behavior of vehicle-wheel/rail-track interaction. The response of a track is defined by its components and their interaction, so that if one component deteriorates, the track response changes in the signature tunes of the damage condition (i.e. characteristic frequencies and amplitudes). In the case of a RJ, vehicle-track dynamics are excited when the wheel impacts the rail at the discontinuity and the dynamic behavior of RJs changes for different visible damage states \[15, 16\]. In view of these promising results, we are encouraged to study if ABA systems can detect invisible deterioration.

In this paper, we assess the capacity of an ABA system for evaluating bolt tightness condition at RJs. For this purpose, a field test was carried out in which a vehicle with an ABA system run over a reference RJ with different bolt tightness conditions for calibration, and then over a set of other RJs for trial detection and verification. The paper is organized as follows. In Section A.2, the measurement devices are described. In Section A.3, a method for detecting bolt tightness condition is presented based on wavelet analysis and a designed test performed at a reference RJ, under controlled conditions. In Section A.4, a case study in Sheffield is presented to evaluate the detection method for two RJs. The results are discussed in Section A.5 and the main conclusions are summarized in Section A.6.

### A.2. RAILWAY TRACK MEASUREMENTS

To assess the feasibility to automatically detect bolt tightness variation at rail joints, two measurements types were performed: axle box accelerations and rail vertical-longitudinal profile. For the former, the ABA system used is presented in Section A.2.1 and for the later, the Railprof measurement system is presented in Section A.2.2.

#### A.2.1. ABA SYSTEM

ABA systems are dynamic-response-based vehicle-borne measurements [11–14]. As it is schematically shown in Figure A.2, the ABA system consists of three main components. First, accelerometers are mounted on axle boxes to measure acceleration. Second, the position of the vehicle is received in a GPS antenna. Third, the vehicle speed is usually obtained from a tacometer.

![Figure A.2: Schematic view of the ABA measuring and diagnosis system](image-url)

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The measured accelerations contain information of the wheel-track structure. When the track has deteriorated, changes with respect to healthy tracks measurements appear in the measured signals, which are called signature tunes. By detecting signature tunes in the response, deteriorated locations can be traced back by combining the measured position and vehicle speed. However, each track defect type has its own signature tunes, which needs to be determined in advance by defining the characteristic frequencies of interest. Thus, track defects can be detected only if their characteristic frequencies are known. After detection, deterioration state of defects can be assessed so that corrective maintenance measures can be planned according to the damage severity. For example, an ABA system can successfully be used to automatically detect both severe and early stage squats, which are a short wave defect on the rail top. [16].

In this paper, the ABA system was mounted on a regular Supertram tram with resilient wheels in Sheffield (see Figure A.3). Four accelerometers were mounted on the four axle boxes of a bogie to register acceleration measurement. The lower close-up in Figure A.2 shows a sensor mounted on the Supertram bogie. The accelerometers are of piezoelectric type (MEMS), and have a range of ±100g, a frequency bandwidth of 30 kHz, and a sensitivity of 50 mV/g. The accelerometer chosen is suitable for railway application as a similar instrumentation has been under extensive test in The Netherlands for mainline applications [14].

![Figure A.3: ABA system used with close-ups of the GPS antenna and an accelerometer](image)

The upper close-up in Figure A.3 shows the location of the positioning GPS antenna on the tram roof. The speed signal was obtained from the meter cabinet of the tram. The accelerations, speed and position were recorded in a data acquisition system on-board of the Supertram test tram.

### A.2.2. Rail Vertical Geometry

Rail vertical-longitudinal profiles were measured using the Railprof measurement device [17], see Figure A.4a. This measurement is used to better understand the condition of the RJs, specifically with respect to the status of the alignment of the rail ends.
Measurements of the rail profile at rail joints were taken every 5 mm along a length of 1000 mm. The accuracy is ±0.03 mm, if the deviation between the first and last measuring point in a straight line is less than 0.5 mm. An example of rail vertical-longitudinal profile at a rail joint is shown in Figure A.4b. The sharp deep indicates the location of the gap. In this case, the two rail ends are misaligned by 0.5 mm.

![Figure A.4: Rail vertical-longitudinal profile measurement: (a) field measurement and, (b) a measured profile](image)

**A.3. Evaluation Method for Bolt Tightness Condition Based on Wavelets**

To develop the detection method for bolt tightness condition at RJs, a controlled field test with different measurable bolt tightness levels is proposed at a reference RJ. Thus, by analyzing the frequency components under the different conditions, signature tunes and detection algorithms for bolt tightness can be developed. The number of reference RJ and the tightness condition applied depends on the accuracy required. At least three different conditions are recommended (loose, half-tight and tight) so that a clear trend of the signature tunes can be obtained. For the case study in this paper, five tightness conditions were considered at the reference RJ. Each of the five torque conditions were applied equally to all the four bolts except for one case as indicated in Figure A.5. The five tightness condition cases are:

- 320 Nm
- 270 Nm (nominal torque)
- 150 Nm
- 150 Nm - 0 Nm (see Figure A.5)
- 0 Nm
For each bolt tightness case, multiple measurements of acceleration were taken at the axle boxes by running the vehicle over the reference RJ with a speed of 21km/h. As the track studied was bidirectional, measurements in both directions A (i.e. Depot-Meadowhall) and B (i.e. Meadowhall-Depot) were obtained and analyzed separately. To analyze the alignment between rail ends, rail vertical-longitudinal profile at the reference RJ was also measured.

Figure A.5: Reference rail joint. The non-uniform 150 Nm - 0 Nm tightness condition and the A and B traffic directions are shown.

A.3.1. DATA ANALYSIS BASED ON WAVELETS
The wavelet transform analysis is used to extract signature tunes from the ABA measurements. With wavelet transform, the energy content of the accelerations can be obtained as a function of frequency and position along the track. One major advantage of wavelets is that the time-frequency representation is not dependent on the scale (or window size). Therefore, wavelet analysis is appropriate for investigation of non-stationary phenomena with local changes in the frequency components. For example, the frequency range and position of track irregularities, such as rail joints, can be identified.

Continuous wavelet transform (CWT) is a time-frequency analysis tool, where convolutions of the analyzed signal with a group of shifted and scaled wavelet functions are calculated [18]. CWT can be defined according to (A.1).

\[
W_x(s, \tau) = \int_{-\infty}^{\infty} x(t) \frac{1}{\sqrt{s}} \psi^* \left( \frac{t - \tau}{s} \right) dt
\]  

(A.1)

where \(x(t)\) is the analyzed signal, \(\psi(t)\) is a mother wavelet, \(\frac{1}{\sqrt{s}} \psi \left( \frac{t - \tau}{s} \right)\) is a family of wavelets deduced from the mother wavelet by different translations and scaling, \(\tau\) is a continuous variable for the translations, \(s\) is a wavelet scale with \(s > 0\), and \(^*\) indicates a complex conjugate. In this work, the Morlet function (A.2) is used as mother wavelet.

\[
\psi_0(\eta) = \pi^{-1/4} e^{i\omega_0 \eta} e^{-\eta^2/2}
\]  

(A.2)

where \(\omega_0\) is a non-dimensional frequency.

To analyze the most relevant frequency components in a signal, the wavelet power spectrum diagram (or scalogram) is calculated using the square of the wavelet coefficients \(W_x^2(s, \tau)\). An example of scalogram for ABA measurements over a rail joint is
shown in Figure A.6. Rolling distance is shown on the horizontal axis, whereas frequency is indicated on the vertical axis. In this case, a measurement along 1 meter is shown with the discontinuity of the RJ at 0 mm. The frequencies shown are in the range of 150-1200 Hz. The amount of energy concentrated at a certain frequency and location is indicated with different colors according to the colorbar. Blue means low energy concentration, whereas red means high energy concentration.

Figure A.6: Example of scalogram from ABA measurements at a RJ.

In this example, there is a high energy concentration between 150 and 400 Hz along 0.1 m after the discontinuity. This frequency range results to be the signature tune for bolt tightness analysis, which will be deducted in Section A.3. In addition there is a high frequency component at approximately 1000 Hz just after the discontinuity. Due to its short duration and the high frequency, it is most likely closely related to the wheel-rail impact. These frequency ranges are investigated in Section A.3.

The wavelet diagrams give a good overview of the energy distribution in frequency and location. However, similarities and differences in frequency between measurements are difficult to quantify. To facilitate the comparison, global wavelet spectra \( \overline{W}^2(s, \tau) \) are computed, which are defined as the wavelet spectrum averaged over the spectra [19]. The global wavelet spectrum is calculated in a discretized procedure, according to (A.3). The values of \( \overline{W}^2(s, \tau) \) estimates the power spectrum of a time series in an unbiased and consistent manner [20].

\[
\overline{W}^2(s, \tau) = \frac{1}{N} \sum_{n=0}^{N-1} |W_n(s, \tau)|^2
\]  

(A.3)

where \( N \) is the number of position points. The magnitude of the impact between wheel and rail fluctuates for different bolt tightness conditions, so that the global wavelet spectrum reaches different energy values. To facilitate the comparison between different cases, global wavelet spectra are normalized by its maximum value, see (A.4).

\[
\overline{W}_N^2(s, \tau) = \frac{\overline{W}^2(s, \tau)}{\max_{s, \tau} \{ \overline{W}^2(s, \tau) \}}
\]  

(A.4)
A.3.2. Reference rail joint analysis

Figure A.7 shows the scalograms of the measured accelerations when the tram was running in the B direction for the five tightness conditions proposed in the methodology. The signals were band-pass filtered between 150 and 1200 Hz. This was necessary to remove the frequency components lower than 150 Hz which had a high energy concentration due to the low vehicle speed (21 km/h) and which would otherwise have submerged the high frequency components. At very high frequency no relevant energy concentrations were observed.

Two tram passages are shown for each bolt loading conditions. The two measurements do not show exactly the same frequency distribution and energy magnitude. This slight disagreement is because the wheels may not run precisely along the same trajectory over the rail so that the wheel-rail contact and the stiffness of the system differ to certain degree each time. This means that each measurement collects the general characteristics of the system, in this case, for instance, the energy concentration between 150 and 600 Hz, as well as local characteristics which depend on the contact location. Figure A.7b corresponds to the wavelet power spectrum diagram of the nominal bolt tightness condition (i.e., 270 Nm). For the nominal condition, most of the energy is concentrated between the beginning of the gap at 0 m and the 3rd bolt at 0.05 m. The dominant frequency range covers between 150 and 600 Hz. Moreover, the measurement shows energy concentration at approximately 920 Hz immediately after the gap. No energy concentration is noticed at the end of the RJ at 0.20 m.

To facilitate the comparison of the responses at different bolt tightness conditions, the global wavelet spectra were calculated and the obtained values were normalized by their corresponding maximum value, as explained in the previous section. Figure A.7 shows the normalized global wavelet spectrum for the A and B traffic directions. First, all the measurements show a dominant peak between 380-420 Hz which can be identified as the characteristic frequency of the reference RJ. The maximum energy is concentrated on this peak, except when the plate is completely loose (i.e., 0 Nm in Figure A.7j). Thus, when the rail joint does not fulfill its purpose of connecting the two rail ends, the dominant frequencies changes with respect to the nominal conditions. Second, different running directions of the tram result in different normalized global wavelet spectrum diagrams. If bolts are looser, energy concentrates at low frequencies. If the tram travels in the A direction, loose bolts mean the appearance of a dominant frequency around 250 Hz. This trend is observed by comparing the 150Nm, 150-0Nm and 0Nm cases in Figures A.7h, A.7i and A.7j, respectively. Whereas if the tram travels in the B direction, loose bolts mean the appearance of one dominant frequency around 150 Hz and another at approximately 920 Hz.

The differences between the two traffic directions were further investigated by analyzing the measured rail vertical-longitudinal profiles. Figure A.8 shows that there was a misalignment between the rail ends. The misalignment seems not compatible with the traffic directions because traffic runs in both directions so that symmetric rail end geometry was expected. Rail end misalignment was most probably caused by the field test, in which the reference RJ was reassembled one week before the test, and during the test the tightness of bolts was changed from nominal value to being completely loose. This misalignment is therefore a disturbance to the local track system, and is causing
A.3. **Evaluation Method for Bolt Tightness Condition Based on Wavelets**

A detection algorithm is proposed based on the analysis of the measurements at the reference RJ. The decisions are taken according to the value of the normalized global wavelet spectrum \( \tilde{W}^{2}_{N}(s, \tau) \) from (A.4) at the identified three characteristic frequencies: (1) 250 Hz (A direction) or 150 Hz (B direction), (2) 380–420 Hz and (3) 920 Hz. The parameters of the algorithm are tuned according to the experimental analysis described in the previous section. The detection algorithm evaluates bolt tightness condition of half RJ after the discontinuity, which is the half RJ excited by the wheel-rail impact. So
for instance, bolts number 1 and 2 can be assessed when the tram travels in A direction; whereas bolts number 3 and 4 when the tram travels in B direction. The following simple expert system with three steps is proposed:

**Step 1** Is the half RJ tight?

- **A direction:**
  - If $W_N^2(s, \tau)$ at 250 Hz is $\geq 0.6$, then the half RJ is *loose or half-tight*.
  - Otherwise, if $W_N^2(s, \tau)$ at 250 Hz is $< 0.6$, then the half RJ is *tight*.

- **B direction:**
  - If $W_N^2(s, \tau)$ at 150 Hz is $\geq 0.6$, then the half RJ is *loose or half-tight*.
  - Otherwise, if $W_N^2(s, \tau)$ at 150 Hz is $< 0.6$, then the half RJ is *tight*.

**Step 2** If the half RJ is not tight,

- **A or B direction:**
  - If $W_N^2(s, \tau)$ around 380-420 Hz is $\approx 1$, then the half RJ is *half-tight*.
  - Otherwise, if $W_N^2(s, \tau)$ around 380-420 Hz is $< 1$, then the half RJ is *loose*.

**Step 3** Is there a misalignment between rail ends?

- **A or B direction:**
  - If $W_N^2(s, \tau)$ around 920 Hz is $\geq 0.1$, then there is misalignment.
  - Otherwise, if $W_N^2(s, \tau)$ around 920 Hz is $< 0.1$, then there is not misalignment.
A.4. CASE STUDY

The proposed methodology to evaluate bolt tightness condition was assessed by investigating two rail joints that were not used for tuning the detection algorithm. First the detection algorithm predicted their condition based on measured ABA signals, and later the bolt tightness condition was measured on the field. In this manner, the prediction was verified.

A.4.1. TRACK SITE

The test track site was the Meadow hall curve of the Supertram network in Sheffield. An overview of the test location is shown in Figure A.9a. The traffic runs in both directions. The BS-80A rail, with 1/40 inclination, was supported by wooden sleepers every 0.6 m. There were no railpads in the track. At rail joints, the nominal rail gap was 6 mm. The two rail joints chosen for validation (hereinafter denoted as RJ3 and RJ5) are on the outer rail and are shown in Figure A.10a and A.10e. The locations of the two rail joints with respect to the reference RJ is shown in Figure A.9b.

![Test track site](image)

**Figure A.9:** (a) Test track site, (b) location of the RJs

A.4.2. PREDICTION

The axle box responses to RJ3 and RJ5 were measured during the tests. The measured ABA signals were post-processed, and the scalograms and normalized global wavelet spectra calculated as shown in Figure A.10. For the two rail joints of validation, the RJ characteristic frequency is observed at 420 Hz in both travel directions, in contrast to the reference RJ which showed two different characteristic frequencies at 380 and 420 Hz depending on the rolling direction (see Figure A.7). Whereas RJ3 and RJ5 were in the track for a long time before the time of the measurements, the condition of the reference RJ was changed one week before the measurements. This altered the supports nearby the reference RJ so that there is a difference in characteristic frequency between the two running directions. In time, a convergence on 420 Hz is expected for the reference RJ.

Once the measured data is post-processed, the predictions of bolt tightness conditions were made employing the detection algorithm of Section A.3.3. For RJ3, the first half of the RJ was predicted as *half-tight* and the second half of the RJ was predicted as...
Figure A.10: For RJ3: (a) photo of RJ3, (b) scalogram in A direction for RJ3, (c) scalogram in B direction for RJ3, (d) normalized global wavelet spectrum diagram in A and B directions. For RJ5: (e) photo of RJ5, (f) scalogram in A direction for RJ5, (g) scalogram in B direction for RJ5, (h) normalized global wavelet spectrum diagram in A and B directions.

Figure A.11: (a) Prediction and (b) Verification

*loose*. For RJ5, the first half of the RJ was predicted as *loose*, whereas the other half of the RJ was predicted as *tight*. The prediction is graphically shown in Figure A.11a. Regarding misalignment between rail ends, misalignment is predicted in RJ5, unlike RJ3 where the rail ends are predicted to be aligned.
A.4.3. Verification
The prediction of the tightness conditions of RJ3 and RJ5 was made in Delft, while it was checked by the 5th author at Supertram on 23 September 2011 (one month after the ABA measurements). Between the ABA measurements and the verification, no maintenance measures were taken on the test site so that no significant change was expected for the RJs. The tightness of each bolt was checked using a torque of 100 Nm, 200 Nm and 270 Nm. In the first step, the four bolts of the two RJs used for validation tolerated the torque of 100 Nm. In the second step, for RJ5 all four bolts showed a torque smaller than 200 Nm, thus the four bolts are loose. Regarding RJ3, the bolt tightness condition was different between the two RJ halves. The bolts at the first half of RJ3 in the A direction tolerated a torque of 200 Nm, not being able to reach the nominal value of 270 Nm, thus their condition was considered half-tight. In the second half, the bolts did not tolerate a torque of 200 Nm, thus the two bolts are loose.

The comparison between the prediction and verification is graphically shown in Figure A.11b. The prediction for RJ3 and RJ5 agree with the verification for overall looseness condition of the rail joints. For each half of rail joints, the tightness prediction was correct for RJ3, but only for half of RJ5. Concerning misalignment between rail ends, no rail vertical-longitudinal profiles were measured at RJ3 and RJ5 for verification.

A.5. Discussion: Application range of ABA systems
ABA systems are under development, the analysis and diagnosis tools are improving. Those developments require a proper experiment design, so that under controlled conditions characteristics of the track can be correctly correlated with ABA measurements. In this paper, a method to evaluate invisible damage, such as loose bolts, was presented using an ABA system. In addition to the ability to detect invisible damage, the experimental analysis has given insight into the capacity of ABA systems regarding measurement speed and vehicle type.

First, measurements were performed at 21 km/h, which is significantly lower than the usual ABA measurement speeds of 80-100 km/h [23]. High speeds are commonly employed because wheel-rail impact is larger at small defects such as light squat, so that signature tunes are excited with more energy, and consequently, they are easier to find in measured signals. At rail joints, the opposite happens. Contrary to small rail defects, the discontinuity between rail ends is large (i.e. 6 mm gap) so that when the wheel rolls over the discontinuity large impact occurs, even at low rolling speeds. Thus, although high rolling speeds are employed to shorten measurement times, ABA systems are able to detect defects at low speeds.

Second, the ABA prototype system was installed in a tram. The vehicle was significantly lighter than the vehicles usually employed in mainlines. The vehicle weight could have affected the analysis because lighter vehicles cause smaller impacts than heavier vehicles and consequently, signature tunes are excited with less energy which may make the detection more difficult. However, the ABA detection system worked when mounted in the tram.

Third, the presence of flexible wheels (also called resilient wheels) did not influence the detection capacity of the ABA system. This types of wheel are used to prevent some of the track vibrations for reaching the vehicle and to reduce acoustic noise. In the case
of this paper, the resilient material layer between wheel threat and web did not eliminate the signature tunes related to bolt tightness and rail end misalignment. The influence of resilient wheels when detecting other type of defects should be investigated in the future.

In summary, the successful experimental investigation has shown that ABA systems could be employed in light rail systems, such as tram and metro lines, where rolling speeds are usually lower than in mainlines, vehicles are lighter and resilient wheels are often used.

A.6. CONCLUSIONS
The capacity of Axle Box Acceleration (ABA) detection and monitoring system for evaluating bolt tightness condition at rail joints (RJs) was experimentally investigated. First, the detection algorithm was developed based on controlled tests at a RJ with different bolt tightness conditions. Then, the bolt tightness condition of other RJs was used for verification of the method.

The experimental analysis has shown that:

- From ABA measurements, three states can be distinguished: tight RJ, intermediate loose RJ and completely loose RJ.
- ABA detection systems may be used for light rail systems, low speeds and resilient wheels.

Future work include the further extension of the ABA detection system to other types of damage at rail joints, such as cracks in the rail web, insulated rail joints, which are found in many networks worldwide, and broad range of speed, from metros to conventional railways. Developing numerical models is also part of future work. Although measurements are required for calibration and validation, the ABA system extension can be faster and easier based on more flexible and relatively fast numerical models than on time-consuming, expensive and safety threatening field testing.

ACKNOWLEDGMENTS
The project was partly financed by the EC-funded FP7 PMnIDEA project. This research was also partially supported by the Basque Government of Spain (Grant No. BFI10), the Dutch railway infrastructure manager ProRail, and the Dutch Technology Foundation STW, which is part of the Netherlands Organization for Scientific Research (NWO) and is partly funded by the Ministry of Economic Affairs. Tata Steel Rail and Cranfield University participated in the field work.

REFERENCES


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I owe thanks to Jan Moraal for his many days and nights performing precious hammer test measurements on railway tracks. Despite the unforgiving Dutch weather, he always did it with a smile. I enjoyed our long trips by ferry and train to Sheffield and Warsaw so that we could successfully perform measurements. Also thank you for your help with the laboratory tests, your valuable contribution to our discussions, and for talking Dutch with me! Additional thanks go to the other staff of Road and Railway Engineering: Jacqueline Barnhoorn, Jan-Willem Bientjes and Dirk Doedens. Special thanks go to Marco Poot, for his help in the laboratory, his endless patience when I wanted to practice Dutch and for helping translating the propositions and the summary.

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I would like to finish my “professional” acknowledgments by thanking three of my closest scientific partners, and also friends: Milliyon Woldekidan, Nico Burgelman and Alfredo Núñez. I would like to thank Milli - my guru of viscoelastic materials, his endless knowledge and patience when answering my hundreds of questions has significantly
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I am grateful for all the help and support I got from my friends and family. Many thanks go to my voluntary Dutch coaches; thank you, Sabine and José, for helping me loose my fear of speaking Dutch, and thank you, Elbard for discussing week after week so many interesting topics. Many thanks go to my “student” friends too, for keeping me young: Marc, Esteban and Eden, you are fun and great cooks too! Thanks also go to the Punch Dames basketball team and trainer Peter for enjoying together my favorite sport. I would also like to thank all the Punch players that run at my whistle, special thanks go to those who showed up for training on Mondays at 22.15, we really love this game! My thanks also go to my true friends in Spain who, although we do not talk very often, are always there: Ainara, thank you for always taking the time to talk to me; Leire, thank you for your energy boosts; and Mikel, many thanks for making me smile.

Vana, Yota and Regina, girls! Thank you so much for the weekly break with delicious Italian broodjes and friendly talk. Thank you, Regina, for introducing me to the board games, sauna and Christmas cookies. Thank you, Yota, for persuading me to wear high hills more often, we will be taller anyway! Thank you, Vana, for passing me on your good mood time and again during this long five years. Thank you very much, girls!

And the cheese & wine friends! Otto, Corina, Martin, Michiel, Gertjan, Alex F., Alexandru, Andra, Anca and Ana, it was always a great pleasure to share a (or more) glass(es) of wine with you! Thank you for the fun, the delicious cheeses, the tasty wines, the rainy but enjoyable trip to Spain, the geeky and not-geeky discussions, the laugher... All in all, thank you so much for your friendship, you made my stay in the Netherlands even more enjoyable. Special thanks go to the three “A”s. Thank you, Andra! I have enjoyed living together; hopefully, it was for you as much fun as it was for me! Thank you, Anca for making me laugh so much, for bringing me closer to cooking, and for the valuable advices. And thank you very much, Ana - my best friend, for always been there with whatever was needed: a smile, a tissue, a bike, a glass of wine, a basketball ball, a funny video, a good advice, a red pen, a push, a hug... thank you, chick!

My biggest “Thank you” goes to my family. Aita, Ama, Dorleta and Leire, although we were kilometers apart, I could always feel your unconditional support. Thank you so much for always believing in me. Mila esker familia, zuek gabe, ezinezkoa. Asko maite zaituztet. There is one more special person, to whom I am most grateful; Joan, thank you for bringing the wow effect into my life.
CURRICULUM VITÆ

Maider OREGUI ECHEVERRIA-BERREYARZA

17-10-1985 Born in Donostia-San Sebastián, Spain.

EXPERIENCE

2010–2015  PhD on Vertical railway track dynamics at Delft University of Technology, Delft (the Netherlands)
2010–2012  Researcher at PM’n’IDEA European Project at Delft University of Technology, Delft (the Netherlands)
2008–2009  Master thesis at HILTI AG, Schaan (Liechtenstein)

EDUCATION

2003–2009  Mechanical Engineering at Escuela Superior de Ingenieros de la Universidad de Navarra, Donostia-San Sebastián (Spain)
2007–2008  Erasmus exchange programme for one semester at KTH-Royal Institute of Technology, Stockholm (Sweden)
2001–2003  Secondary School degree with Cum laude distinction at La Salle, Donostia-San Sebastián (Spain)

LANGUAGES

Spanish  Native
Basque  Native
English  Full professional proficiency
Dutch  Limited working proficiency, Staatsexamen NT2-II (June 2011)

AWARDS & GRANTS

2014  UIC Young Researcher Award
2010–2013  Researcher Training Programme Grant from the Basque Government of Spain (Grant No. BFI10)
What I also did and learnt during my PhD

- Present my work
- Conference papers & Scientific writing
- Set-up computer cluster
  Parallel computing
- Peer to peer communication
- European Collaboration
- UIC Young Researcher Award 2014

Conference papers & Scientific writing

- Set-up computer cluster
  Parallel computing
CM2012
Chengdu, China
Fight for myself
Team player
Team leader
Learn Dutch
Open to new ideas
Section Activity Organiser
Collaborate with companies
Lab & Field tests
In collaboration with Joan Maymi
UIC Young Researcher Award

International Union of Railways (UIC) is the worldwide organisation for railway cooperation. The overall mission of UIC is to promote an increased use of rail transport at world level and to help its members to make rail transport more attractive, effective, sustainable and economically viable. In order to stimulate rail research towards the necessary innovations and step-changes, UIC’s IRRB (International Rail Research Board) holds the biennial UIC Innovation Awards. The ambition of the UIC Innovation Awards is to support the development of rail transport as the sustainable backbone of a transport system that is cost-efficient, reliable, safe and secure and will therefore become the mode of choice for passengers and freight forwarders. The Young Researcher Award is part of the UIC Innovation Awards to recognize young engineers and researchers who have made significant contributions in rail research and innovation. The UIC Young Researcher Award 2014 was awarded to Maider Oregui.


Figure UIC 1: UIC Young Researcher Award 2014
International Railway Research Board hereby certify that Maider Oregui received the UIC Research & Innovation Award for her outstanding contribution to the innovation of the global railway system in the Young researcher category on 3 December 2014 in Paris.

Jean-Pierre Loubinoux
UIC Director General
Boris Lapidus
IRRB Chairman

UIC Innovation Awards

Figure UIC 2: UIC Innovation Award Ceremony
2nd “UIC Research and Innovation Awards” presented in Paris on the occasion of the General Assembly of the worldwide railway association

(Paris, 8 December 2014) The second edition of the UIC Railway Research and Innovation Awards was held in Paris on 3 December, during the General Assembly of the International Union of Railways (UIC).

The award ceremony took place in the presence of UIC Chairman Vladimir Yakunin, Vice Chairman Michele Elia, as well as Director-General Jean-Pierre Loubinoux. The awards were presented by Mr Boris Lapidus, Director-General of the Russian Railway Research Institute (VNIIZhT), Chairman of the Scientific Committee at Russian Railways (RZD), and Chairman of the UIC International Railway Research Board (IRRB) as well as by Jerzy Wisniewski, Director of Fundamental Values at UIC.

The awards went to the following six categories:

- **Safety / Security**: Masamichi Sogabe, from Rail Technical Research Institute – Japan
  Mr Sogabe worked on Risk Assessment for Train-Running Safety during Seismicity on Railway Lines. He developed an analytical method to analyse dynamic interaction between railway vehicles and large structures to assess running safety during seismicity.

- **Sustainable Development**: Byung-Song Lee, from Korea Railroad Research Institute – South Korea
  Mr Byung-Song Lee worked on Wireless Power Transfer (WPT) for Railways. He developed a method for supplying power to electric trains by using Wireless Power Transmission, based on the magnetic resonance and near-field coupling of two coils.
  A new 180 kW single-phase wireless power transfer system for a tram was demonstrated.

- **Rail System Technology**: Christoph Tyssen / Marco Tami, from SBB – Swiss Federal Railways – Switzerland
  Mr Christoph Tyssen and Mr Marcus Völker worked on the Adaptive Control (ADL) Project. They developed an adaptive control system to optimise driving speed profiles by minimising unplanned halts and breaking. ADL saves energy amounting to 2.1% at current rail traffic density.

- **Rail Freight Services**: P. Mortimer / C. O’Neil, from NewRail / TruckTrain Developments Ltd – United Kingdom
  Mr P. Mortimer and Mr C. O’Neil worked on the TopHat(R) project. They identified practical and effective opportunities for rail to make a sustained inroad into growing volumes of HVTS
traffic moving in trailers. This supports EU aspirations on significant modal shift with full size top-lifting tri-axle semi-trailers used in European domestic and international inter-modal traffic.

-Passenger Services: Jaeho Kwak, Korea Railroad Research Institute – South Korea
Mr Jaeho Kwak worked on the Wireless (catenary free) Battery Tram. The wireless battery tram with 5 modules, 32m in length, 42 tonnes and 162 kwh lithium-ion battery has been developed. The tram had travelled 53 km on a single charge without load. The wireless battery tram avoids high current stress within urban areas and requires less powerful substations, saving construction costs.

-Cost reduction and Sustainable Development: Brigita Altenbaher, from Eelpa d.o.o – Slovenia
Ms Brigita Altenbaher worked on DRYproANNSYS “bs” – wheel flange lubrication system. She developed an innovative on-board wheel flange lubrication system which doesn’t need any compressed air and applies the material onto the wheel flanges/rims. This is an environmentally-friendly solution with less wear, noise/squealing and screeching.

Two other prizes were delivered:

-Life Time Achievement: François Lacôte, from France
Mr Lacôte joined SNCF in 1974, where he occupied various positions. In 1981 he joined the Rolling Stock Division, first as Head of Locomotives and Trainset Programmes, then from the end of 1982 onwards, as Head of TGV Programmes. In this capacity, he supervised the design and creation of successive generations of TGV trains, and oversaw the test campaign which on 18 May 1990 set a world rail-speed record of 515.3 kph. In November 2000 he joined the Board of Alstom Transport as Senior Vice President, Technical Division. Co-managing with SNCF and RFF the test programme which led to a new world rail-speed record of 574.8 kph being set in April 2007, he has sat since September 2009 on the Executive Committee of Alstom Transport where he occupies the function of Senior Vice President and Technical Adviser to the President.

-Young Research Award: Maider Oregui, from TU Delft - Netherlands
Ms Maider Oregui worked on investigating vertical track dynamics. She investigated rail-pad and fastening deterioration with 3D FE models + laboratory tests.
•Identified characteristic frequencies of damage track:
•Improved defect detection measures.
•Improved maintenance measures.
•Developed preventive maintenance measures.

To finish, two special prizes were awarded to:

-Olivier Grossat, UIC, for developing new functional options of 3D printing of documents. The current project consists of experimenting with large-format 3D concrete printing for engineering and architectural functions. Moreover, this will be the first OpenSource concrete printing project: the machine, the layout diagrams and software used for its operation are available and can be used by all (http://impression-concrete.blogspot.fr/).
Grigore Havrneau, UIC, for his major research and innovation contribution in the RESTRAIL project. He developed a toolbox (http://www.restrail.eu/toolbox/) for decision makers – the first online guide that provides evidence-based recommendations for choosing and implementing cost-effective measures to prevent railway suicides and trespassing accidents and to mitigate the consequences of these incidents.

MEDIA CONTACTS:

Dennis Schut: schut@uic.org
Eva de Valk
Verslag uit
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TU‘ERS ON TOUR

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Kort
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Een ‘food truck’ met IBM’s supercomputer Watson aan boord stond woensdag 3 december voor de ingang van faculteit EWI. Op basis van 35 duizend recepten en een triljoen verschillende combinaties van ingrediënten, bedacht Watson vernieuwende gerechten. Deze werden vervolgens door een chef-kok in de truck bereid. (Foto: Hans Stakelbeek)

Wielertocht
Erasmus Sport, het sportcentrum van de Erasmus Universiteit, houdt op 30 mei 2015 een wielertoertocht langs Leiden, Delft en Rotterdam. Het evenement moet de samenwerking tussen de drie onder de aandacht brengen. delta.tudelft.nl/29219

Uiterlijk
Een aardig uiterlijk en genoeg zelfvertrouwen zijn voor hoogopgeleiden net zo belangrijk als werkervaring en diploma, blijkt uit onderzoek van het Sociaal en Cultureel Planbureau. Het SCP vindt dat de overheid werkzoekenden moet stimuleren aandacht te besteden aan hun ‘esthetisch kapitaal’. delta.tudelft.nl/29218

Overschot
Universiteiten hebben veel geld overgehouden in 2013: samen bijna 150 miljoen euro. Dat meldt de Algemene Onderwijsbond (AOb). Uit haar cijfers blijkt dat de TU Delft het jaar 2013 afsloot met een overschot van 31,2 miljoen euro. Dat is 5,4 procent van de totale baten. delta.tudelft.nl/29217

Verwijderd
Het Massachusetts Institute of Technology verwijdert alle online colleges van de Nederlandse professor Walter Lewin. Hij wordt beschuldigd van seksuele intimidatie. Lewin studeerde in Delft, maar vertrok in de jaren zestig naar MIT. Daar werkte hij tot zijn pensionering in 2009. delta.tudelft.nl/29216

Space
“Expertise op het gebied van ruimtevaart bundelen en creëren voor een lokale, regionale en mondiale impact op onderzoek, onderwijs en valorisatie.” Dat is kort gezegd de taak van het nieuwe TU Delft Space Institute, dat in januari 2015 wordt opgericht. Het instituut zal zich richten op ‘sensing from space’, ‘space robotics’ en ‘distributed space systems’. Hoogleraar space systems engineering Eberhard Gill wordt directeur.

De week van...

‘s werelds meest efficiënte dunne-film zonnecel is van Delftse makelij. Promovendus Hairen Tan (EWI) rekende het record op van 14,5 naar 14,8 procent.

Eind vorige maand kreeg hij de Young Researcher Award voor zijn prestatie tijdens de World Confe-
rence on Photovoltaic Energy Conversion in Kyoto.

“Het klinkt als een kleine verbete-
ring”, zegt Tans begeleider dr. Arno Smets. “Maar in de wereld van de dunne-film zonnecellen verlopen de ontwikkelingen in heel kleine stapjes. Vergelijk het met de honderd meter sprint, daarbij verbeter je het record ook niet meteen met een volle seconde.” De hogere energie-
opbrengst komt onder meer door een betere benutting van het zon-
nespectrum. Tan heeft zijn zonnecel een soort tandemstructuur gegeven, waarbij twee absorptiematerialen – amorf silicium en nanokristallijn silicium – elk verschillende gebieden uit zonnespectrum absorberen. (Foto: Hairen Tan)

Uit de trillingen van rails valt veel op te maken over de staat van onderhoud. Voor die ontdekking ontving promovenda Maider Oregui van de afdeling railway engineering (CITG) op 3 decem-
ber in Parijs de innovatieprijs voor jonge onderzoekers van de interna-
tional railway research board.

Door een combinatie van veldtests en simulaties is ze in staat uit de trillingen van rails bepaalde afwij-
ningen af te leiden. Als voorbeeld noemt ze golfslag as, kleine onef-
fenheden (squats) of losrakende bouten. Voor continue metingen worden de trillingen gemeten met een speciaal assenstel onder de trein. De uitkomsten daarvan kunnen het onderhoud goedkoper en effec-
tiever maken en het treinverkeer veiliger. (Foto: Maider Oregui)
LIST OF PUBLICATIONS

JOURNAL PAPERS


CONFERENCE PAPERS


