Train–Track–Bridge Interaction for the Analysis of Railway Bridges and Train Running Safety

THERESE ARVIDSSON

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Cover photo: Andreas Andersson, 2015. An X62 train passing the Aspan bridge on the Bothnia Line during a KTH measurement campaign with a load-controlled exciter.

Tryck: Universitetsservice US-AB
Abstract

In this thesis, train–track–bridge interaction (TTBI) models are used to study the dynamic response of railway bridges. A TTBI model considers the dynamics of the train in addition to that of the track–bridge system. The TTBI model enables the assessment of train running safety and passenger comfort. In the bridge design stage, a moving force model is instead typically used for the train load. The main aim of this thesis is to use results from TTBI models to assess the validity of some of the Eurocode design criteria for dynamic analysis of bridges.

A 2D rigid contact TTBI model was implemented in ABAQUS (Paper II) and in MATLAB (Paper III). In Paper V, the model was further developed to account for wheel–rail contact loss. The models were applied to study various aspects of the TTBI system, including track irregularities. The 2D analysis is motivated by the assumption that the vertical bridge vibration, which is of main interest, is primarily dependent on the vertical vehicle response and vertical wheel–rail force.

The reduction in bridge response from train–bridge interaction was studied in Papers I–II with additional results in Part A of the thesis. Eurocode EN 1991-2 accounts for this reduction by an additional damping $\Delta \zeta$. The results show that $\Delta \zeta$ is non-conservative for many train–bridge systems since the effect of train–bridge interaction varies with various train–bridge relations. Hence, the use of $\Delta \zeta$ is not appropriate in the bridge design stage.

Eurocode EN 1990-A2 specifies a deck acceleration criterion for the running safety at bridges. The limit for non-ballasted bridges ($5 \text{ m/s}^2$) is related to the assumed loss of contact between the wheel and the rail at the gravitational acceleration $1 \text{ g}$. This assumption is studied in Paper V based on running safety indices from the wheel–rail force for bridges at the design limit for acceleration and deflection. The conclusion is that the EN 1990-A2 deck acceleration limit for non-ballasted bridges is overly conservative and that there is a potential in improving the design criterion.

**Keywords:** dynamics, railway bridge, bridge deck acceleration, train–bridge interaction, vehicle model, wheel–rail force, running safety.
Sammanfattning


Preface

The research presented in this thesis has been financed by the KTH Railway Group with additional support from Trafikverket and the Division of Structural Engineering and Bridges at KTH Royal Institute of Technology. The work has been conducted at the Department of Civil and Architectural Engineering, KTH. The simulations were in part performed on resources provided by the Swedish National Infrastructure for Computing (SNIC) at PDC Centre for High Performance Computing (PDC-HPC) at KTH.

I express my sincere gratitude to my supervisor Prof. Raid Karoumi for his support and professional guidance. Thank you Dr. Andreas Andersson, my co-supervisor, for your devoted interest and guidance during my research work, all endless discussions and all laughs. Many thanks also to my second co-supervisor Adj. Prof. Costin Pacoste, and to Prof. Jean-Marc Battini for reviewing this thesis. Assoc. Prof. Daniel Cantero, NTNU Norwegian University of Science and Technology, is gratefully acknowledged for his part in establishing the numerical model which in the present thesis has been further developed.

Many thanks to all friends, colleagues and former colleagues at the Department of Civil and Architectural Engineering for providing such a joyful and encouraging work environment.

I also wish to thank my parents for their support and encouragement throughout my studies. Finally, I thank my family, Patrik and Arvid, for all love, happiness and support that you give me.

Stockholm, April 2018

*Therese Arvidsson*
Publications

This thesis is based on the work presented in the following appended papers:


Papers I, II, IV and V are planned, implemented and written by Arvidsson. The co-authors have provided guidance throughout the work and reviewed the drafts before submission. Arvidsson has taken part in planning, implementing and writing Paper III together with the first author Assoc. Prof. Daniel Cantero.

In addition, Arvidsson has contributed to the following related publications:


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<tr>
<td>ADM</td>
<td>Additional damping method</td>
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<tr>
<td>DOF</td>
<td>Degree of freedom</td>
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<tr>
<td>FE</td>
<td>Finite element</td>
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<tr>
<td>HSLM</td>
<td>High-speed load model</td>
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<td>MF</td>
<td>Moving force</td>
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<tr>
<td>PSD</td>
<td>Power spectral density</td>
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<td>RB</td>
<td>Rigid beam</td>
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<tr>
<td>RMS</td>
<td>Root mean square</td>
</tr>
<tr>
<td>SIM</td>
<td>Simplified interaction model</td>
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<tr>
<td>TBI</td>
<td>Train–bridge interaction</td>
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<td>TTBI</td>
<td>Train–track–bridge interaction</td>
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Part A

Introduction and general aspects
Chapter 1

Introduction

Increasingly high demands are being placed on railway systems. The strategic plan for transportation within Europe (European Commission, 2011) outlines goals of a well-developed high-speed rail network by 2050. The intention is that a majority of all medium-distance passenger traffic should be conducted by rail. At the same time, it aims at a shift from road-based freight traffic to rail transportation.

In Sweden, the Bothnia Line (Botniabanan) was completed in 2010 serving both passenger and freight traffic along the northern coastline. The Norrbotniabanan is now being planned as a continuation of the line. In southern Sweden, the project Ostlänken (the East link) is under planning – a high-speed passenger railway line between Järna and Linköping. In a later stage the Ostlänken may serve as a part of the European Corridor (Europakorridoren). The European corridor, which is still at a conceptual stage (Trafikverket, 2012), is intended to provide a high-speed connection to the European rail network; see Figure 1.1.

With the limited space for new infrastructure, and to meet the high comfort requirements at high-speed lines, bridges or viaducts may very well form an increasing part of the railway infrastructure. Up to 80–95% of new high-speed tracks in China are built on bridges; in Japan the ratio is up to 60% (Montenegro, 2015). To meet the urge for new railway lines, the design rules need to be continuously developed for a more efficient design of the infrastructure. To this end, research on railway infrastructure is vital.

1.1 Background

From a design point of view there are basically three objectives for dynamic calculations on railway bridges: (1) dynamic amplification of load effects (2) passenger comfort and (3) running safety. The European bridge design codes EN 1991-2 (CEN, 2010b) stipulate that dynamic analyses should generally be conducted for
design speeds above 200 km/h. The speed and regularly spaced train axles imply that the dynamic effect can be considerable. In principle, the calculated dynamic amplification should be applied to all load effects. For lower speeds, EN 1991-2 specifies dynamic factors for the amplification of load effects.

The passenger comfort is of concern since there is a risk for reduced comfort as the train traverses the vibrating bridge deck. Passenger comfort assessment is generally based on measurements of the car body acceleration over a certain track length. EN 1991-2 allows for an indirect verification of the passenger comfort, from limits on the bridge deck deflection given in EN 1990-A2 (CEN, 2005). These limits are intended to ensure a very good comfort with a maximum car body acceleration of 1 m/s$^2$.

According to the European bridge design requirements for dynamic analyses, the running safety is assessed based on the bridge deck acceleration criterion. The bridge deck acceleration is typically decisive for bridge spans up to 30 m while the deflection limit for passenger comfort is decisive for longer spans (Svedholm and Andersson, 2016; Arvidsson and Andersson, 2017).

The acceleration limit originates from the risk of ballast instability. Displacements in the ballast can lead to track misalignment and potentially derailment. Following the introduction of high-speed rail traffic in Europe, excessive bridge vibrations were observed at the Paris–Lyon line. Especially short span bridges showed vibration problems (Zacher and Baeßler, 2009) after the opening of the line in 1981. Increased maintenance was needed to avoid deterioration of the track quality. Shake table tests undertaken in connection to the work of the European Rail Research Institute (ERRI D214, 1999a) showed that ballast loses its interlock at accelerations exceeding 0.7 g. A safety factor of 2 led to the acceleration limit 3.5 m/s$^2$ for ballasted bridges in EN 1990-A2. According to EN 1990-A2, the deck acceleration should be calculated considering bridge frequencies up to $max\{30 \text{ Hz}; 1.5 \times 1^{st} \text{ eigenfrequency}; 3^{rd} \text{ eigenfrequency}\}$. 
For non-ballasted bridges (bridges with slab track) the Eurocode limit is instead 5 m/s² (0.5 g). This limit is related to the assumed loss of contact between the wheel and the rail at the gravitational acceleration 1 g, again with a safety factor of 2 (Zacher and Baeßler, 2009). However, the contact loss at 1 g has not been verified by simulations or observed in measurements. The physical background to the assumption is vague. There is no obvious relation between the wheel–rail contact and the bridge deck acceleration; the wheel being separated from the bridge deck by means of the rail, fastenings and track slab.

In the field of vehicle engineering, the running safety is instead assessed based on quasi-static and dynamic vehicle response and wheel–rail forces. European design codes related to the dynamic analysis of the train–track–bridge system are listed in Table 1.1.

In the bridge design stage, a moving force (MF) model with each train axle represented as a constant force is typically used. A train–bridge interaction (TBI) or train–track–bridge interaction (TTBI) model introduces the train mechanical system. Typically, the train components are modelled as rigid masses connected by springs and dampers. These models make the evaluation of passenger comfort possible based on the car body acceleration. Furthermore, TTBI models enable the analysis of train–bridge response at the presence of track irregularities, as well

<table>
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<th>Table 1.1: European design codes for dynamic analysis of the TTBI system.</th>
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<td><strong>Bridge</strong></td>
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<tr>
<td>1991-2 CEN (2010b)</td>
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<td><strong>Track</strong></td>
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<td>13848-5 CEN (2017)</td>
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<td><strong>Vehicle</strong></td>
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CHAPTER 1. INTRODUCTION

as running safety assessment based on the wheel–rail forces. The inclusion of the train in the theoretical model generally results in a reduction in the bridge deck response as compared to an MF analysis. According to EN 1991-2 a certain amount of additional damping can be introduced in bridges with span up to 30 m to take into account the reduction in bridge response from TBI.

The TTBI models describe the coupled system in more detail compared to the MF models typically used in the design stage. A relevant application for the TTBI models is to evaluate the assumptions behind the bridge design rules.

1.2 Aims and scope

The overall aim of this thesis is to study bridge response from high-speed trains and assess the validity of some of the Eurocode design criteria for the dynamic analysis of bridges. A specific aim is to study the effect of TBI in terms of reduced bridge response, with comparison against the Eurocode additional damping. Another specific aim is to study how TTBI analyses can be applied to study the train running safety on non-ballasted bridges. Particular focus is dedicated to the relation between bridge response and running safety indices obtained from wheel–rail forces. In relation to running safety, the main objective is to evaluate the EN 1990-A2 bridge deck acceleration limit.

The following limitations apply to the research work:

- The TTBI system is modelled in 2D which allows us to perform parametric analyses at a relatively low computational cost. As a consequence, the lateral dynamic effects are neglected. The 2D analysis is motivated by the assumption that the vertical bridge vibration, which is of main interest, is primarily dependent on the vertical vehicle response and vertical wheel–rail force.

- The running safety assessment is based on safety indices in the literature and the fact that short-time contact loss does not impose a risk for derailment. Derailment from flange climbing due to high lateral loads is not considered as the analyses are performed in 2D.

- The car body acceleration from the rigid multi-body vehicle model serves as a simplified estimate of the passenger comfort with no consideration of the car body flexibility or the dynamic properties of the passenger seats.

- No measurements have been conducted within the scope of this work. Comparisons against previously measured data have been performed to verify the theoretical models.
1.3 Research contribution

The work presented in this thesis have resulted in the following scientific contributions:

- A summary of conclusions from the vast number of studies on TTBI available in the literature with specific focus on train load modelling alternatives.

- A demonstration of how TTBI models can be applied in assessing the relevance of the design requirements for dynamic analysis of high-speed railway bridges. The EN 1991-2 additional damping and the EN 1990-A2 deck acceleration limit for non-ballasted bridges have been thoroughly examined. The EN 1991-2 track irregularity factor and the EN 1990-A2 deck deflection limits have been briefly studied.

- A methodology to isolate the effect of the bridges on the running safety from the effect of track irregularities. To this end, simulations with the same track profile are performed on a track section with and without bridge.

- The application of a TTBI model to explain the effect of the sleeper passing frequency observed in measured bridge response.

1.4 Outline of the thesis

This thesis consists of two parts of which Part A provides an extended summary of the work presented in the papers appended in Part B. Part A, Chapter 1 gives an introduction and demonstrates the relation between the appended papers. Chapter 2 presents the TTBI models together with verification examples. The main topics of the research work in the papers are discussed in Chapter 3 together with new results on additional damping, the factor for track irregularities and a sensitivity study. Conclusions and a discussion of further research are given in Chapter 4.

The extended summary is followed by five appended papers. The relation between the papers is illustrated in Figure 1.2 together with the main research question for each paper and the studied design criteria. The literature review (Paper I) treats various subjects within bridge dynamic analyses and TTBI. The review is the starting point for the subsequent papers that each treats specific aspects of the TTBI system. Paper II is a numerical study on the relative importance of TBI. A 2D TTBI model is developed in Paper III. The computational model is further developed in Paper IV and V. In Paper IV, the TTBI model is applied to replicate measured bridge response in a case study where an ordinary MF model proved insufficient. Paper V studies the relation between bridge response and train
running safety and comfort on non-ballasted bridges. Figure 1.3 summarizes the computational framework for each paper.

Papers I, II, IV and V are planned, implemented and written by Arvidsson. The co-authors have provided guidance throughout the work and reviewed the drafts before submission. Arvidsson has taken part in planning, implementing and writing Paper III together with the first author Assoc. Prof. Daniel Cantero. Specifically, Arvidsson assisted in developing the rigid wheel–rail contact formulation in the TTBI model; Arvidsson also validated the model against a commercial software and performed the case study on track irregularity wavelengths. A description of each appended paper is as follows:

**Paper I** presents a literature review with a particular focus on TBI models for the evaluation of vertical bridge deck acceleration. The review is complemented by numerical examples comparing different TBI models. Furthermore, general
1.4. OUTLINE OF THE THESIS

<table>
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<th>Paper I</th>
<th>Literature review</th>
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<td>● TRAIN</td>
<td>Moving force/Vehicle</td>
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<td>Running safety</td>
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<td>Passenger comfort</td>
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<td>Articulated/Conventional</td>
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<td>Axle load &amp; configuration</td>
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<td>▲ TRACK</td>
<td>Ballasted/Slab track</td>
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<td>Transition zones</td>
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<td>Track stiffness</td>
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<td>Sleeper passing freq.</td>
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<td>■ BRIDGE</td>
<td>Soil–structure interaction</td>
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<td>Stiffness, mass, damping</td>
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<td>Simply supp./Continuous/Portal frame</td>
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<td>Paper II</td>
<td>Numerical study</td>
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<td>● Car body acc.</td>
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<td>■ Wheel–rail force</td>
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<td></td>
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<td>■ Deck acc.</td>
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<td>■ Bridge eigen-frequency</td>
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| Paper III | Model framework | ● Wheel–bogie–car |
|           |                 | ● Rigid contact |
|           |                 | ▲ Ballasted |
|           |                 | ■ Portal frame |
|           |                 | ▲ Track irregularities |
|           |                 | ■ Change in bridge frequency during train passage |
|           |                 | ● Car body acc. |
|           |                 | ■ Wheel–rail force |
|           |                 | ■ Deck acc. |
|           |                 | ■ Bridge eigen-frequency |

| Paper IV | Numerical study | ● Wheel |
|          |                 | ● Hertz contact |
|          |                 | ▲ Ballasted |
|          |                 | ■ Simply supp. |
|          |                 | ● Moving force/Vehicle |
|          |                 | ■ Sleeper passing freq. |
|          |                 | ▲ Track stiffness |
|          |                 | ■ Measurement |
|          |                 | ■ Deck acc. |

| Paper V | Numerical study | ● Wheel–bogie–car |
|         |                 | ● Hertz contact |
|         |                 | with contact loss |
|         |                 | ▲ Slab track |
|         |                 | ■ Simply supp./Continuous |
|         |                 | ● Running safety |
|         |                 | ■ Passenger comfort |
|         |                 | ▲ Track irregularities |
|         |                 | ■ Range of bridge parameters |
|         |                 | ■ Wheel–rail force |
|         |                 | ■ Duration of contact loss |
|         |                 | ■ Car body acc. |
|         |                 | ■ Deck acc. and displ. |

Figure 1.3: Computational framework for TTBI in Papers I–V.
aspects of the dynamic analysis of bridges are treated, as well as track models and the effect of track irregularities.

**Paper II** provides a screening of key parameters in the dynamic analysis of beam bridges subjected to passenger train loads. A two-level factorial experiment is applied to highlight the relative influence of TBI as compared to variations in other key parameters: bridge stiffness, bridge mass, bridge damping ratio, bridge rotational support stiffness and train axle load. The EN 1991-2 additional damping criterion is studied, with additional results in Chapter 3.

**Paper III** presents the framework for a 2D coupled TTBI model for ballasted railway bridges, implemented in MATLAB (The MathWorks, Inc., 2012). Each component of the model is presented in detail and the model is validated against the commercial software ABAQUS (Dassault Systèmes, 2011). The effect of different wavelengths of track irregularities on the bridge and vehicle response is studied together with the effect of the vehicle on the bridge’s fundamental frequency.

**Paper IV** presents a case study where the TTBI proves to be essential in simulating the measured bridge response. The periodic loading from the wheels passing the sleepers introduces the sleeper passing frequency. It is demonstrated that the deck vibration in a portal frame bridge can be greatly amplified if the sleeper passing frequency matches a bridge frequency. This effect can be captured in a theoretical model including the wheel mass and the track structure.

**Paper V** presents a comprehensive parametric study on the vehicle and wheel–rail response on non-ballasted bridges at the design limit for acceleration and deflection. The running safety of trains is assessed based on safety indices from the wheel–rail contact forces. The passenger comfort is studied based on the car body acceleration. Comparisons are made against the EN 1990-A2 limits for safety and comfort. The 2D coupled TTBI model from Paper III is here further developed for the non-ballasted track application with articulated trains. Moreover, a linearized Hertzian contact model that allows for wheel–rail contact loss is introduced.
Chapter 2

Model and model validation

Figure 2.1 depicts the theoretical TTBI models used in Papers III-V. These coupled finite element (FE) models in 2D include: the vehicle, the bridge and the ballasted track or the (non-ballasted) slab track. The vehicle model includes car body, bogie and wheel. In Paper IV, the vehicle wheel masses are modelled travelling over a ballasted track to simulate the sleeper passing frequency; the remaining vehicle bodies are represented by a constant force. A model omitting the track system and with a simplified wheel–bogie vehicle model is used in Paper II where the focus is on the bridge response and not the vehicle or wheel–rail forces. The coupled equations of motion for the TTBI models are presented in Section 2.1, together with descriptions of each subsystem in Sections 2.2–2.5. Examples are provided to explain and validate several aspects of the models.

2.1 Coupled equations of motion

Common for all TTBI models is that they require the solution of two coupled systems of equations: the train subsystem and the track–bridge subsystem. The two systems are coupled via the dynamic interaction force, which depends on time as the vehicles move over the bridge, as well as on the bridge and vehicle displacements. There are two fundamental approaches to solve the interaction system: the iterative and the coupled approach.

Through an expression for the interaction force, the system can be transformed into two uncoupled equations and solved iteratively by enforcing self-consistency between the track–bridge and the vehicle at each time step (Yang and Fonder, 1996; Liu et al., 2014). The two subsystems can also be solved in turns for the whole time sequence, repeatedly, until convergence is reached for the interaction forces (Zhang and Xia, 2013). The discrete equations of motion are written (Yang and Fonder,
CHAPTER 2. MODEL AND MODEL VALIDATION

where subindex TB refers to the track–bridge system and V refers to the vehicle system, M, C and K are the mass, damping and stiffness matrices, respectively, and $u$ is the displacement vector. The track–bridge force vector, $f_{TB}$, is composed of the static (gravity) load from the vehicle as well as the dynamic interaction forces. The vehicle force vector, $f_{V}$, contains the dynamic interaction forces.

In this thesis, the coupled approach is instead adopted. This results in a single equation system which has larger matrices but eliminates the iterative process between two equation systems. Depending on the assumption for the wheel–rail coupling the coupled equations take different forms. One assumption is that of rigid contact where the degree of freedom (DOF) of the wheel is eliminated and assumed to follow that of the point of contact with the rail (or bridge); see schematic sketch in Figure 2.1 (a). Another assumption is a spring representing the wheel–rail contact, as depicted in Figure 2.1 (b).

![Figure 2.1: 2D train–track–bridge coupled model with: (a) ballasted track and rigid wheel–rail contact and (b) slab track and linearised Hertz contact.](image)

$$M_{TB} \ddot{u}_{TB} + C_{TB} \dot{u}_{TB} + K_{TB} u_{TB} = f_{TB}$$  \hspace{1cm} (2.1a)  
$$M_{V} \ddot{u}_{V} + C_{V} \dot{u}_{V} + K_{V} u_{V} = f_{V}$$  \hspace{1cm} (2.1b)
2.1. COUPLED EQUATIONS OF MOTION

2.1.1 Rigid wheel–rail coupling (Papers II and III)

In the rigid contact assumption, the wheel displacement, $u_w$, is restrained to follow the point of contact with the track plus the track irregularity $r_w$. The wheel DOF is therefore eliminated. The vehicle DOFs are thus reduced to those of the suspended bodies. The constraint equations for displacement, velocity and acceleration are (Lin and Trethewey, 1990; Olsson, 1985):

\[ u_w = Nu_{T,i} + r_w \]  
\[ \dot{u}_w = N\dot{u}_{T,i} + vN'u_{T,i} + vr'_w \]  
\[ \ddot{u}_w = N\ddot{u}_{T,i} + 2vN'\dot{u}_{T,i} + aN'u_{T,i} + vr'_w + v^2r''_w \]  

where $[N]_{1\times4}$ is the cubic shape function of the 4-DOF beam element evaluated at the point of contact with the $i$th wheelset, $u_{T,i}$ is the vertical displacement vector of the beam element in contact and $v$ and $a$ are the horizontal train speed and acceleration.

Through the rigid contact assumption the TTBI system can be described with a coupled equation of motion with time-dependent matrices:

\[
\begin{bmatrix}
M_V & 0 & 0 \\
0 & M_T + M_w & 0 \\
0 & 0 & M_B
\end{bmatrix}
\begin{bmatrix}
\ddot{u}_V \\
\ddot{u}_T \\
\ddot{u}_B
\end{bmatrix}
+ \begin{bmatrix}
C_V & C_{V,T} & 0 \\
C_{T,V} & C_T + C_w & C_{T,B} \\
0 & C_{B,T} & C_B
\end{bmatrix}
\begin{bmatrix}
\dot{u}_V \\
\dot{u}_T \\
\dot{u}_B
\end{bmatrix}
+ \begin{bmatrix}
K_V & K_{V,T} & 0 \\
K_{T,V} & K_T + K_w & K_{T,B} \\
0 & K_{B,T} & K_B
\end{bmatrix}
\begin{bmatrix}
u_V \\
u_T \\
u_B
\end{bmatrix}
= \begin{bmatrix} f_V \\
f_T \\
f_B
\end{bmatrix}
\]

where the sub-indices V, T and B indicate vehicle, track and bridge subsystems, and w the coupling terms from each vehicle wheel. The terms in the diagonal band of the matrices are the FE representation of each subsystem. The coupling of the subsystems is expressed with the off-diagonal terms and additions to the diagonal terms. The track–bridge coupling is composed of the terms from the spring-dashpot layer between the track components and the bridge. The track–bridge coupling terms remain constant since there is no change in their configuration during one simulation. Their additions to the diagonal terms are included in $K_T$, $K_B$, $C_T$ and $C_B$.

On the other hand, the vehicle–track coupling depends on the vehicle’s position in time and has to be updated at each time step. As the wheel nodes are eliminated, the coupling is composed of terms from the forces in the primary suspension spring and damper as well as the forces from the wheel mass travelling the rail. With constant vehicle speed ($a = 0$) it follows from the constraints, Eq. (2.2), that the
additions to the diagonal terms for the $i$:th wheel are:

$$M_{w,i} = m_w N^\top N$$

$$K_{w,i} = k_p N^\top N + c_p v N^\top N' + m_w v^2 N^\top N''$$

$$C_{w,i} = c_p N^\top N + 2 m_w v N^\top N'$$

where $m_w$ is the wheelset mass and $k_p$ and $c_p$ are the primary suspension stiffness and damping. Each term is a $4 \times 4$ addition to the $4 \times 4$ matrix of the beam element in contact. The $M_w$ term is the mass from the eliminated wheel node that, as the vehicle moves along the track, is added to the mass matrix of the element in contact. The terms including $v$ and $v^2$ derive from the coriolis and centripetal force of the wheel travelling the deflected rail. The terms originate from the differentiation of the path of the wheel mass at its contact point with the rail, Eq. (2.2). The coriolis and centripetal terms are further discussed by Olsson (1985); Lin and Trethewey (1990); Michaltsos and Kounadis (2001) and Lou and Au (2013).

For each wheel, the off-diagonal terms between the 4 DOFs of the beam element and the DOFs of the suspended vehicle bodies can be written:

$$K_{V,T,i} = -k_p N_V N - c_p v N_V N$$

$$K_{T,V,i} = -k_p N^\top V N^\top$$

$$C_{V,T,i} = -c_p N_V N$$

$$C_{T,V,i} = C_{V,T,i}$$

where $[N_V]_{n \times 1}$ is a matrix describing the relation between the $i$:th wheel and the $n$ DOFs of the suspended vehicle bodies.

The effect of the irregular track profile is treated as an external force. The contribution to the load vectors can be formulated (Olsson, 1985; Lou and Au, 2013):

$$f_{V,i} = (k_p r_w + c_p v r_w') N_V$$

$$f_{T,i} = (-k_p r_w - c_p v r_w' - m_w v^2 r_w'') N^\top$$

$$f_{B,i} = 0$$

where $r_w$ is the track irregularity at the point of contact for the $i$:th wheel. The terms including $k_p$ and $c_p$ are the translation of the irregular profile into suspension forces, whereas the term including $m_w$ is the effect of the inertia of the wheel mass travelling the irregular profile, see Eq. (2.2). In addition to the above expressions, the vehicle force vector includes the gravitational load from the vertical vehicle DOF:s.

The full expressions for each term in the rigid contact TTBI formulation are available in Lou (2007). Submatrices for different beam and vehicle idealisations can
2.1. COUPLED EQUATIONS OF MOTION

also be found in, for example, Olsson (1985); Lin and Trethewey (1990); Xia et al. (2000) and Au et al. (2001).

The coupled rigid contact model was in Paper III implemented in MATLAB (The MathWorks, Inc., 2012). The system was solved with direct integration using the Newmark average acceleration method with no numerical damping. The model was verified against a similar model in the commercial FE software ABAQUS (Dassault Systèmes, 2011) from Paper II. The model details and scripting procedure in ABAQUS are described in detail in Arvidsson (2014).

2.1.2 Linearized Hertz contact with wheel–rail contact loss
(Papers IV and V)

In the Hertz contact model, the wheel–rail contact is modelled with a spring between the wheel DOF and the element in contact. In the general case, Hertz contact is described by the force–deformation \( (Q_C, \delta) \) relation:

\[
Q_C = \delta^{3/2}C_H, \text{ where } C_H = \frac{2E (R_w R_r)^{1/4}}{3 (1 - \nu^2)} \tag{2.7}
\]

with the radius of the wheel, \( R_w \), and the radius of the rail, \( R_r \), both with elastic modulus \( E \) and Poisson ratio \( \nu \). The normal wheel–rail force can be linearly represented by a stiffness coefficient (Dinh et al., 2009):

\[
k_C = \frac{dQ_C}{d\delta} = \frac{3}{2} \delta^{1/2} C_H = \left\{ \delta^{1/2} = \frac{Q_C^{1/3}}{C_H^{1/3}} \right\} = \frac{3}{2} Q_C^{1/3} C_H^{2/3} \tag{2.8}
\]

at a given point in the force–deformation relation. The Hertz contact relations are shown in Figure 2.2.

A linearized Hertzian spring that allows for contact loss is used in this thesis. When the wheel loses contact with the rail, the force is set to zero. Thus, we have the relation:

\[
Q = \begin{cases} 
  k_C (u_w - N_{u_{T,i}} - r_w), & (u_w - N_{u_{T,i}} - r_w) > 0 \\
  0, & (u_w - N_{u_{T,i}} - r_w) \leq 0
\end{cases} \tag{2.9}
\]

where \( Q \) is the vertical wheel–rail force depending on the compression in the contact spring from the deflection of the wheel, \( u_w \), the deflection of the rail in contact, \( N_{u_{T,i}} \), and the track irregularity \( r_w \). In each time step, the contact stiffness needs to be updated according to Eq. (2.9). The contact stiffness is initially based on the contact condition from the previous time step (contact or loss of contact). Iterations are performed within each step to update the contact condition; see the flow chart in Figure 2.3.
Figure 2.2: Hertz contact force-displacement relation, where the tangent to the curve is the contact stiffness. A linear approximation is shown at preload 100 kN.

Figure 2.3: Flow chart for the solution of the coupled system with contact loss.
2.1. COUPLED EQUATIONS OF MOTION

The equation of motion for the system with linearized Hertz contact is written:

\[
\begin{bmatrix}
M_V & 0 & 0 \\
0 & M_T & 0 \\
0 & 0 & M_B
\end{bmatrix}
\begin{bmatrix}
\ddot{\text{u}}_V \\
\ddot{\text{u}}_T \\
\ddot{\text{u}}_B
\end{bmatrix}
+ \begin{bmatrix}
C_V & 0 & 0 \\
0 & C_T & C_{T,B} \\
0 & C_{B,T} & C_B
\end{bmatrix}
\begin{bmatrix}
\dot{\text{u}}_V \\
\dot{\text{u}}_T \\
\dot{\text{u}}_B
\end{bmatrix}
+ \begin{bmatrix}
K_V & K_{V,T} & 0 \\
K_{T,V} & K_T + K_C & K_{T,B} \\
0 & K_{B,T} & K_B
\end{bmatrix}
\begin{bmatrix}
\text{u}_V \\
\text{u}_T \\
\text{u}_B
\end{bmatrix}
= \begin{bmatrix}
f_V \\
f_T \\
f_B
\end{bmatrix}
\] (2.10)

As in the rigid contact case, the track–bridge coupling terms remain constant while the vehicle–track coupling depends on the vehicle’s position in time. Both the DOF:s of the suspended vehicle bodies and the wheel DOFs are included in the vehicle part of the matrices. The vehicle–track coupling is now simply composed of stiffness terms from the force in the wheel–rail contact spring. The following expression can be used for the addition to the diagonal terms from each wheel:

\[K_{C,i} = k_C N^T N\] (2.11)

where \([N]_{1 \times 4}\) is the cubic shape function of the 4-DOF beam element evaluated at the contact point with the \(i\)th wheelset and \(k_C\) is the linearized contact stiffness. Then the off-diagonal terms between the \(i\)th wheel DOF and the 4 DOFs of the beam element in contact are:

\[K_{V,T,i} = -k_C N\] (2.12a)

\[K_{T,V,i} = K_{V,T,i}^T\] (2.12b)

As in the rigid contact case, the track irregularities are treated as external forces. The terms in the vehicle and track force vector from the \(i\)th wheel derive from the translation of the track irregularities to a force in the contact spring:

\[f_{V,i} = k_C r_w\] (2.13a)

\[f_{T,i} = -k_C r_w N^T\] (2.13b)

\[f_{B,i} = 0\] (2.13c)

The linearized Hertz contact model with contact loss was implemented in MATLAB as a further development of the rigid contact model in Paper III. It was used in Papers IV and V. The model results are validated against measured wheel–rail forces as well as results from a 3D model in Paper V, and against other 2D model results in Arvidsson and Andersson (2017).

Particularly for running stability analyses involving lateral dynamics, more advanced contact theories have been used. Common approaches implement Hertz
contact for the normal contact and Kalker creep theory for the lateral contact; see for example Zhai et al. (2009); Antolín et al. (2013) and Montenegro (2015). Important applications include TBI systems under wind and earthquake load.

Example: rigid contact and Hertz contact

Figure 2.4 shows a comparison between the rigid contact model and the linearized Hertz contact model, with and without contact loss. The results are from the TTBI models in Figure 2.1 for a HSLM-A1 train running over a single span 20 m beam bridge, with properties according to Paper V. The non-ballasted track is modelled with a track profile from the German PSD (see Section 2.4) with wavelengths 1–150 m and $\sigma_{3-25} = 1.0$ mm.

The envelopes of maximum and minimum wheel–rail force is presented for the 1st wheel in the 20th carriage. The deck acceleration is the maximum along the bridge. The results for the whole speed range are maximum envelopes from 24 profile realisations while the time histories are given for

![Graph showing comparison between rigid contact and linearized Hertz contact](image)

Figure 2.4: Wheel–rail force $Q$ and bridge deck acceleration $a_{br}$ for rigid contact and linearized Hertz contact with and without contact loss.
one of the profile samples. The irregular track profile introduces high-frequency content in the wheel–rail force and bridge deck acceleration. Results low-pass filtered at 20 Hz are also given, according to the filter frequency for running safety evaluation in Section 3.5.

The contact models give almost identical results below the speed where contact loss occurs. As the contact spring is very stiff, the dynamics of the wheel–rail contact is in the magnitude of hundreds of Hertz which is above the frequency range of interest for the present model. The reasons for adopting a Hertzian spring are instead: (1) to obtain the duration of contact loss and (2) the Hertz contact model is easier to program as only the stiffness matrix and the force vector are time-dependent and all forces from the wheel masses are inherent in the model and need not be explicitly added (compare Eq. (2.3)–(2.6) and (2.10)–(2.13)). The linearized Hertz model with contact loss will be used in the remaining examples in this chapter. Antolín et al. (2013) discuss the models further for a 3D case with lateral dynamics.

The time history at 400 km/h shows an example of relatively large contact losses (10 ms). Here, there are slight time shifts between the rigid contact and the Hertz contact results and slight differences in maximum amplitude. For results low-pass filtered at 20 Hz, the contact models give close to identical results also for the speeds with contact loss.

2.2 Vehicle models

The most important vehicle characteristics are the mass and inertia of the bodies (car, bogie and wheel) together with the stiffness and damping of the suspension system. A passenger vehicle has typically both primary suspension (bogie–wheel) and secondary suspension (car–bogie).

The rigid beam (RB) model in Figure 2.5 (a) is probably the most common 2D vehicle model. The vehicle is modelled as a rigid multi-body system including the car body (2 DOF), the bogie (2 DOF) and the wheel (1 DOF). The primary and secondary suspension systems are represented by springs and dashpots in parallel. The vertical car body response at the end of the body is given by the vertical DOF plus the contribution from the rotational (pitch) DOF. For a rigid body, the response at the ends is therefore typically higher than at the centre. For conventional bogie (non-articulated) carriages the interaction between adjacent carriages is often neglected. For articulated carriages, sharing a Jacobs’s bogie, a stiff vertical coupling between adjacent car bodies may be considered; see Figure 2.1.
The simplified interaction model (SIM) in Figure 2.5 (c) models the bogie–bridge interaction but neglects the car body and the secondary suspension. The secondary suspension system isolates the car body from much of the vibration and, hence, much of the dynamic interaction is located to the bogie–bridge system. Papers I and II conclude that the SIM is often a relevant idealisation for analysing the bridge response. However, the vehicle car body needs to be modelled for passenger comfort assessment. As discussed in Paper I, half-vehicle models such as Figure 2.5 (b) tend to underestimate the car body acceleration as the pitch frequency is not represented. All 2D vehicle models require similar computational time, which is why the choice is motivated by: (1) which output that is needed and (2) which vehicle data that is available.

3D models are typically adopted for more detailed analyses of vehicle running characteristics and passenger comfort. The bending modes of the car body and the non-linear characteristics of the suspension system can also be considered; see for example Zhai et al. (2009); Ribeiro et al. (2013); Xia et al. (2014); and Montenegro et al. (2014). An example in Paper V shows that the present 2D vehicle model agrees well with a 3D vehicle model in terms of vertical wheel–rail force and car body acceleration.

The geometrical and mechanical input properties depend on the particular vehicle to be modelled. However, selecting these properties is not a trivial issue since they are generally not available, other than to the rolling stock manufacturers. Paper III provides a list of references for train properties of existing trains. In Paper V, theoretical bridge sections were optimised to fulfil the EN 1991-2 dynamic design
2.2. VEHICLE MODELS

Figure 2.6: Assumed HSLM-A vehicle model modes.

requirements. The prescribed dynamic design load is the EN 1991-2 high-speed load model A (HSLM-A). The Eurocode provides no information on mechanical properties for the design load model. Hence, a vehicle model representing HSLM-A was established based on mechanical data from the literature and typical values for the vehicle eigenfrequencies; see Figure 2.6. All assumed properties are given in the appendix in Paper V.

Example: vehicle models

Figure 2.7 shows an example of wheel–rail force and bridge deck acceleration from analyses using vehicle models (a), (c) and (d) from Figure 2.5, for the same case as the example in Section 2.1.2. The results from the SIM and RB model are similar. Hence, the wheel and bogie are more important than the car body for the wheel–rail force and bridge acceleration. The wheel model (where both bogie and car body are represented by a constant force) gives slightly higher response. However, in Paper IV, the wheel model was deemed sufficient giving 10% higher deck acceleration for the case study bridge.

Figure 2.7: Wheel–rail force $Q$ and bridge deck acceleration $a_{br}$ for HSLM-A1 at a 20 m bridge for vehicle model (a), (c) and (d) from Figure 2.5.
2.3 Track model

The track is the system guiding the train and distributing the load to the bridge structure or earthwork. Adequate track stiffness is important for the riding and track stability and to limit the forces on the vehicle and track components (UIC, 2008b). The track stiffness is commonly expressed as the stiffness experienced by one rail and is the combined stiffness of the track components and the substructure. Given the vertical wheel load $Q$ and the vertical rail deflection $z_{\text{rail}}$ (see Figure 2.8), the track stiffness for one rail is defined by:

$$K_{\text{track}} = \frac{Q}{z_{\text{rail}}}$$ (2.14)

A commonly suggested value for the rail deflection under a 100 kN wheel load is 1–2 mm (UIC, 2008b), resulting in a recommended track stiffness of 50–100 MN/m. A track stiffness of 64 ± 5 MN/m is recommended for slab tracks by DB (1999). The Swedish standards (Trafikverket, 2018) recommends similar values for the slab track stiffness. The track stiffness is always higher than the support stiffness under each rail seat as the deformation of the track distributes the load to several rail seats.

The loaded track frequency is estimated as:

$$f_s = \frac{1}{2\pi} \sqrt{\frac{K_{\text{track}}}{m_w}}$$ (2.15)

where $K_{\text{track}}$ is the track stiffness and $m_w$ is the wheel mass.

2.3.1 Ballasted track

The ballasted track system, illustrated in Figure 2.9 (a), consists of: rail, rail fastening with rail pad, sleepers, and ballast. On earthwork, subballast and subgrade are also part of the track. In the FE model, the rails are modelled with either Euler–Bernoulli or Timoshenko beams with a spring–dashpot at each fastening location. The ballast, sleepers and substructure are modelled as a combination of spring–dashpots and mass elements; see Figure 2.1 (a).

The track stiffness as a functions of the rail seat stiffness $k_{\text{seat}}$ is shown in Figure 2.10 (a). The rail seat stiffness $k_{\text{seat}}$ is the series combination of the springs under each fastening location: $k_{\text{rp}}$, $k_{\text{ba}}$ and $k_{\text{sb}}$ according to Figure 2.1 (a). The rail seat stiffness of the ballasted track at the embankment before the bridge in Paper IV was assumed 75/2 MN for one rail, giving a track stiffness of $K_{\text{track}} = 100$ MN/m for sleeper distance 0.6 m. The loaded fundamental track frequency was around 50 Hz for half a wheelset (1000 kg) on one rail. The track was assumed stiffer at the bridge (due to a possibly lower ballast layer and no contribution from the substructure): $K_{\text{track}} = 170$ MN/m with a loaded fundamental track frequency of around 65 Hz. Lists of sources for ballasted track model parameters are compiled in Papers I and III.
2.3. TRACK MODEL

Figure 2.8: Rail deflection \( z_{\text{rail}} \) under a wheel load \( Q \) at the slab track model.

Figure 2.9: Ballasted track (a) and slab track (b). Photos by A. Andersson (Bothnia line, Sweden) and RAIL.ONE (2018) (HSL-ZUID railway line, the Netherlands).

Figure 2.10: Static track stiffness \( K_{\text{track}} \) and rail deflection \( z_{\text{rail}} \) under a 100 kN wheel load: (a) ballasted track (b) slab track with two rail pad stiffness values.
2.3.2 Slab track

The non-ballasted track system, illustrated in Figure 2.9 (b), consists of rail, rail fastening system with rail pad, track slab and substructure. The concrete track slab is about 30 cm thick and is in turn supported by a layer of compacted engineering material or a cement stabilised layer. In some non-ballasted track systems, prefabricated sleepers are integrated in the slab by in-situ infill concrete (e.g. Rheda, Züblin). Altogether prefabricated track slabs are also used (e.g. Bögl, ÖBB–PORR, Japanese Shinkansen tracks, CRTS China Railway Track System). The slab is generally separated from the supporting structure (cement stabilised layer, bridge or tunnel) by means of a bituminous mortar a couple of centimetres thick. Even if it has some degree of elasticity, the main purpose of the mortar is to make replacement possible. For sections where sound or vibration insulation is needed, a rubber mat or elastic bearings can separate the track from the surroundings. The slab track requires less maintenance compared to a ballasted track to achieve high track profile quality. The main disadvantage is the higher cost for construction (UIC, 2008a).

The slab track FE model is shown in Figure 2.8. The rails and fastenings are modelled in the same way as in the ballasted track model. The track slab is modelled with Euler–Bernoulli beams and supported by a continuous spring bed representing the substructure (on embankment) or mortar layer (on bridge).

In the slab track, the pad in the rail fastening system must provide the elasticity that the combination of pad and ballast bed gives in a ballasted track. This normally results in a rather soft rail pad with a standard static stiffness of 22.5 MN/m (DB, 1999; UIC, 2008a). Figure 2.10 (b) shows the track stiffness and rail deflection as a function of substructure bed modulus for two values of rail pad stiffness. As seen, there is a threshold substructure bed modulus at about 50 MN/m³ above which increasing substructure stiffness has little effect on the total track stiffness. Above the threshold, the track stiffness is mainly governed by the rail pad stiffness. The slab track model in Paper V has track stiffness $K_{\text{track}} = 60$ MN/m and thus a loaded fundamental track frequency of 39 Hz for a 1000 kg wheel on one rail.

Slab track system descriptions and model parameters can be found in, for example, UIC (2002, 2008a); Thompson (2009); Lichtberger (2005); Dai et al. (2016); Blanco-Lorenzo et al. (2011) and Zhai et al. (2013).

Example: beam theory for rail elements

The rail is often modelled with Euler–Bernoulli beams. The Timoshenko beam element has additional terms that account for the transverse shear deformation. The two element types are compared in Figure 2.11, for the same case as the example in Section 2.1.2. The following can be observed:
2.4 Track irregularities

Track irregularities in the wavelength range around 0.5–150 m and longer are deviations from the ideal track geometry generated from, for example, settlements, the sleeper spacing and irregular track stiffness. The track profile is commonly characterised by the isolated defects (zero to peak values), the standard deviation and the wavelength content.

EN 13848-5 (CEN, 2017) defines wavelength ranges D1 (3–25 m), D2 (25–70 m) and D3 (70–150 m). Zero to peak limit values are given for D1 and D2 with increasingly strict values for higher design speeds. The alert limits for vertical track irregularities are 6–8 mm in D1 and 8–10 mm in D2, for speed range 300–360 km/h. EN 13848-6 (CEN, 2014) defines track quality classes A–E; see Figure 2.12. The classes are from the cumulative frequency distribution of the standard deviation in D1, \( \sigma_{3-25} \).
from measured irregularities in the European rail network. Track class D (70th–90th percentile) or better is recommended as alert limit for standard deviation.

As the unsprung axle masses traverse the irregular profile, variations in the wheel–rail forces arise, providing an additional excitation of the train–track–bridge system. The train suspension system effectively mitigates the short wavelengths. Therefore, mainly the longer wavelengths (D2 and D3) excite the car body modes of vibration and thus the passenger comfort; see Paper III. The results in Paper III moreover show that mainly the short wavelengths (D1) influence the wheel–rail forces (running safety) and the bridge response; see also the example below. The standards do not give more detailed recommendations on the frequency distribution of the track irregularities than what can be implied from the limit values for D1 and D2.

Theoretical descriptions of track irregularities are often based on power spectral density (PSD) functions. The PSD function describes the amplitude of the track profile at each wavelength. The German PSD, $S \text{ m}^2/(\text{rad/m})$, is given by:

$$S(\Omega) = \frac{A_p \Omega_c^2}{(\Omega_r^2 + \Omega^2)(\Omega_c^2 + \Omega^2)}$$  \hspace{1cm} (2.16)

for wavelengths $\Omega \text{ rad/m}$, $\Omega_r = 0.0206$ rad/m, $\Omega_c = 0.8246$ rad/m (Claus and Schiehlen, 1986). The track quality factor $A_p \text{ rad-m}$ can be used to scale the profile to a specific track quality. The German PSD is shown in Figure 2.13 for two values of the track quality factor. The Chinese PSD for high-speed slab tracks (Zhai et al., 2015) is also shown. Measured track irregularities from Swedish ballasted tracks for speeds $<250 \text{ km/h}$ are included for comparison.

Spatial samples are extracted from the theoretical PSD functions by means of the inverse Fourier transform with random phases assigned to each harmonic compo-
2.4. TRACK IRREGULARITIES

The random nature of the profile realisations makes it necessary to make analyses for several realisations. In Paper V, 24 profiles were used, each with the highest zero to peak value out of 1000 random samples.

Theoretical PSD functions have the shortcoming that they cannot reproduce the isolated defects that are present in real track profiles. Therefore, they tend to produce profiles with less variation in maximum deviation. This is illustrated in Figure 2.14, where the variation in running standard deviation for measured and theoretical profiles are given. The standard deviations of the theoretical profiles are scaled to the means of the measured running standard deviations. However, the measured distributions have longer tails with irregularities of large amplitudes. This is seen in the figure as occasional large zero to peak values in the space domain. Thus, the calibration of theoretical profiles involves a compromise between standard deviation and maximum amplitude.

Figure 2.13: German PSD (Claus and Schiehlen, 1986) scaled to \( \sigma_{3-25} = 1.0 \) and 0.6 mm, and Chinese PSD for non-ballasted tracks, \( \sigma_{3-25} = 0.3 \) mm (Zhai et al., 2015). Measured track profiles (speeds <250 km/h) are included for comparison.
Figure 2.14: Measured profiles with wavelengths 3–25 m from tracks for speeds <250 km/h compared to samples from the theoretical German PSD, with standard deviation scaled to fit the mean running standard deviation of the measured signal.

Example: track irregularity wavelengths

Figure 2.15 shows the effect of track irregularities with different wavelength ranges, for the same case as the example in Section 2.1.2.

The wheel–rail force grows larger with increasing speed at the irregular track for two reasons: (1) the increasing forces from the wheel travelling the irregular track profile and (2) that the speed affects which frequencies that are induced by the wavelengths in the track profile.

At speed \( v \) (m/s) the frequency induced by a certain track irregularity wavelength \( w \) (m) is:

\[
    f_{ir} = \frac{v}{w}
\]  

(2.17)

Long wavelengths induce low frequencies while short wavelengths induce high frequencies.
2.4. TRACK IRREGULARITIES

Figure 2.15: Wheel–rail force $Q$ (a, e, f), bridge deck acceleration $a_{br}$ (b), filtered wheel unloading $\Delta Q_{20Hz}/Q_0$ (c) and contact loss $C_{loss}$ (d) at a 20 m bridge for track irregularities with different ranges of wavelengths.
CHAPTER 2. MODEL AND MODEL VALIDATION

The wheel–rail forces are most affected by the shortest wavelengths that induce frequencies around the loaded track frequency already at low speed. For a track frequency of 35 Hz, 1 m wavelengths reach the loaded track frequency at $35 \times 1 \times 3.6 = 130$ km/h. Wavelengths 3 m instead reach the track frequency at: $35 \times 3 \times 3.6 = 380$ km/h. This is confirmed in subfigure (a) where the wheel–rail force at the 1–150 m profile grows from small to large at 130 km/h. At the 3–150 m profile it grows large around 380 km/h.

Subfigure (e) shows the spectral density of the wheel–rail force at 130 km/h. The majority of the energy in the wheel–rail force lies around the loaded track frequency (35 Hz). These frequencies are induced at the 1–150 m profile but not at the 3–150 m profile. The sleeper passing frequency is seen as a distinct peak in the spectra at $f_s = 130/(3.6 \times 0.6) = 60$ Hz. Subfigure (f) shows that at the speed 380 km/h also the 3–150 m profile induces frequencies at the loaded track frequency.

For all speeds above 130 km/h no considerable increase in response is observed from the inclusion of even shorter wavelengths (0.1–150 m). There are two explanations: (1) wavelengths inducing frequencies above the loaded track frequency do not add much energy (2) the amplitude of the irregularities is generally assumed to decrease with decreasing wavelength (cf. Figure 2.13).

From the spectral densities it can be realised that the difference between the track profiles is small for results filtered below the track frequency, e.g. at 20 Hz. The choice of the lowest wavelength is not important for the filtered wheel unloading; see subfigure (c). For the duration of contact loss, the choice is more important; see subfigure (d). This is important to consider in running safety assessments; cf. Section 3.5.

2.5 Bridge model

The bridge is modelled with 2D Euler–Bernoulli beams, either simply supported or continuous over several spans. The 2D model neglects the eccentricity between the support and the bridge neutral axis.

Andersson and Svedholm (2016) studied the difference in bridge response between 2D and 3D models for concrete slab bridges, box bridges and beam bridges in 1–4 spans. They concluded that for many cases the main difference is a frequency shift towards lower bridge fundamental frequency in the 3D model. The frequency shift is often due to the shear lag present in the 3D model where the whole width of the
bridge plate does not contribute to the deck stiffness. The vertical bridge response is thus often similar between 2D and 3D, with somewhat lower resonance speeds in the 3D model.

Support constraints can be introduced in the TTBI model in a simplified way by means of vertical and rotational support springs; see Paper IV. However, detailed modelling of the soil–structure interaction is not considered in this thesis. Soil–structure interaction is briefly discussed in Paper I and have been studied thoroughly by Ulker-Kaustell (2013); Doménech et al. (2016); Svedholm (2017); Östlund et al. (2017); and Zangeneh Kamali (2018).

In this thesis, no special consideration has been taken to model the transition zone between embankment and bridge; see the example below. Transition plates and similar structures are sometimes constructed to minimise relative settlements and produce a smooth stiffness transition between the bridge and the soft substructure before the bridge. If settlements do occur there is a risk for a self-perpetuating process where settlements lead to higher wheel–rail forces, which in turn lead to further settlements (Read and Li, 2006).

### Example: transition zones and track settlement

Figure 2.16 shows an example of the wheel–rail force at a track section with an abrupt change in substructure stiffness. The HSLM-A1 train runs at 400 km/h at a slab track, both with properties according to Paper V. The substructure bed modulus changes from 100 MN/m$^3$ to infinitely stiff ground as shown in the figure. The abrupt change in stiffness is intended to represent an extreme case of an embankment–bridge transition. A joint is typically introduced to the track slab at such a transition; therefore, the slab is modelled as discontinuous.

The abrupt stiffness change introduces oscillation in the wheel–rail force with an amplitude of about 5 kN for a perfectly smooth track. However, due to the discontinuity of the track slab at the transition, also the case with continuous substructure stiffness gives similar oscillations; see the red and blue lines in subfigure (a). However, if a sinusoidal dip is introduced at the transition, the oscillation in the wheel–rail force grows very large; see the black line. The dip (3 mm over 3 m) is intended to represent settlements developed at the transition.

For a track section with random track irregularities (German PSD, $\sigma_{3−25} = 1.0$ mm), the difference between the continuous substructure and the abrupt change in substructure is still rather small. Slightly higher wheel–rail forces
are obtained if a sinusoidal dip at the transition is added to the random track irregularities.

The high influence of the sinusoidal dip and the random track irregularities indicates that, for the present slab track model, the wheel–rail force is governed more by the profile amplitude (settlement and random irregularities) than by the change in substructure stiffness. Recalling Figure 2.10, the slab track stiffness is largely governed by the rail pad. The relatively small effect from a change in substructure stiffness motivates a simple model of the track–bridge transition zone with no consideration of transitional structures. The effect of the transition is possibly larger for a ballasted track as the depth and compaction of the ballast layer may be affected by the transition.

Figure 2.16: Wheel–rail force $Q$ at a track section with and without an abrupt change in substructure stiffness.
Chapter 3

The research work

The TTBI models described in the previous chapter has been applied to study several aspects of railway bridge dynamics. This chapter provides extended descriptions of the main topics from Paper I–V. Section 3.1 treats the reduction in bridge response from TBI and provides some new results not presented in the papers. Section 3.2 focus on the bridge response from track irregularities, again with some new results. Measured bridge response is discussed in Section 3.3, passenger comfort in Section 3.4 and running safety in Section 3.5. Extended results from a model parameter sensitivity study from Paper V are given in Section 3.6.

3.1 Additional damping from TBI

The presence of the vehicle on the bridge contributes to increased energy dissipation from the bridge. Generally, reduced bridge response is obtained compared to an MF analysis. The amount of reduction is mainly governed by: the bogie–bridge frequency ratio, the bogie–bridge mass ratio and the bridge–carriage length ratio as discussed in detail in Paper I and by Doménech et al. (2014). The reduction from TBI is largest at high mass ratios and at a bogie–bridge frequency ratio of 1.0–1.5. The reduction is only relevant at resonance, and can thus be more evident for longer train sets inducing a more pronounced resonance. The presence of the vehicle on the bridge moreover changes the eigenfrequency of the system, as shown in Paper III.

Design calculations are typically performed using a MF load model. TBI is thus neglected and with that the reduction in bridge response compared to the MF model. According to EN 1991-2, a certain amount of additional damping, $\Delta \zeta$, can be introduced in spans up to 30 m to take into account the effect of TBI. The additional damping method (ADM) was derived by ERRI D214 (1999b) from TTBI analyses of simply supported beam bridges. Additional damping in an MF model
was calibrated against results from TTBI analyses. Two trains were considered (ICE 2 and Eurostar) with no consideration of track irregularities. Results where no resonance occurred were removed.

Figure 3.1 (a) shows the calibrated additional damping from the D214 analyses; the Eurocode $\Delta \zeta$-curve was determined from the lower bound of the results. Included in the figure are also additional damping values calculated for the bridges from the parametric study in Paper V. Similarly to the D214 study, an MF model with additional damping was calibrated so that the maximum bridge acceleration matched the TTBI results for smooth track from Paper V. Bridges with no clear resonance peak were removed. The scatter in the calibrated $\Delta \zeta$-values is considerable with highest values up to 0.8%. However, the lower bound is close to zero, with many of the results well below the Eurocode $\Delta \zeta$-curve.

One explanation for the difference between the D214 results and the present results is the distribution of bridge frequencies; see Figure 3.2. The D214 bridges have comparatively low frequencies in relation to the span length; all are below the frequency range recommended in EN 1991-2. The present bridges have a larger distribution of eigenfrequencies with both high and low frequencies for each span length. The train characteristics may also influence the results. However, the ICE 2 and the present HSLM-A vehicle model have rather similar bogie frequency, 6 and 5 Hz, respectively. With similar carriage length, the HSLM train has fewer bogies per carriage being an articulated train.

Figure 3.3 shows an example of two 15-m span bridges with different fundamental frequency (6.1 and 9.1 Hz) and very different amount of calibrated additional damping. Subfigure 3.3 (a) shows the results from the present TTBI model on a bridge from the D214 study. It provides a benchmark of the present model against that of D214 as the calibrated additional damping is the same, $\Delta \zeta = 1.0\%$ for the ICE 2 train. The $\Delta \zeta$ calibrated for a HSLM-A1 train is instead 0.3%. The second bridge in Subfigure 3.3 (b) is from Paper V. It has a higher eigenfrequency that is not close to the train bogie frequencies. The calibrated $\Delta \zeta$ is 0.06% and 0.2% for HSLM-A1 and ICE2, well below the EN 1991-2 additional damping for 15 m bridges (0.65%).

With $\Delta \zeta$-values plotted against frequency instead of span length there is a slightly higher correspondence between the D214 results and the present results; see Figure 3.1 (b). However, the lower bound of the present results is still close to zero. The dependence on various train–bridge relations means that the amount of reduction from TBI is different for different combinations of trains and bridges. Hence, as discussed also by Doménech et al. (2014), the ADM is not appropriate in the design of bridges that are to be inter-operable.
3.1. ADDITIONAL DAMPING FROM TBI

Figure 3.1: Additional damping $\Delta \zeta$ from ERRI D214 (1999b) and Paper V bridges.

Figure 3.2: Eigenfrequencies for bridges in ERRI D214 (1999b) and Paper V.

Figure 3.3: Bridge deck acceleration $a_{br}$ for TTBI, MF and MF with calibrated additional damping $\Delta \zeta$ for two 15 m bridges and the ICE 2 and HSLM-A7 trains.
3.2 Bridge response from track irregularities

As shown in Paper V, both passenger comfort and running safety are directly affected by the track quality level. The additional high-frequency variation in the wheel–rail forces can also lead to higher bridge response, mostly in the higher frequencies.

A MF load model does not account for the additional excitation from track irregularities. Therefore, EN 1991-2 specifies track irregularity factors to be used in bridge design calculations. The factor \((1 + \phi''\)) is used for standard track maintenance, while \((1 + 0.5\phi'')\) for carefully maintained track. \(\phi''\) depends on the bridge span length and fundamental frequency, with increasing magnification for shorter spans (higher fundamental frequencies). The track irregularity factor was introduced based on studies by the Office for Research and Experiments, ORE D23 (1970) and ORE D128 (1976). Their work included measurements and numerical analyses. ERRI D214 (1999a) validated \(\phi''\) for resonance loads from high-speed trains. They concluded that \(\phi''\) can be applied to bridge acceleration in addition to deflection and stresses. The modelling of track irregularities in these studies comprised single sinusoidal dips that represent the effect of a loose sleeper.

The ERRI D214 filtered all results at 20–30 Hz. The motive was to remove the high frequency content from the impacts following loss of wheel–rail contact. It was presumed that contact losses should not occur at a well-maintained track. However, even with no contact loss, much of the energy in the wheel–rail force lies around the loaded track frequency, above 30 Hz; see the example in Section 3.2. Some of this high-frequency content will be transferred to the bridge structure.

Figure 3.4 shows a comparison between the EN 1991-2 \(\phi''\) and calculated factors from the present TTBI model. The calculated factors are the ratio between bridge response with an irregular and a smooth track profile from the parametric study in Paper V. Recalling Figure 3.2, all bridges whose eigenfrequency falls within the EN 1991-2 range \(\{n_{0,\text{min}}, n_{0,\text{max}}\}\) are included. Both a lower and a higher track quality are considered, according to Paper V, and compared to \((1 + \phi'')\) for standard maintenance and \((1+0.5\phi'')\) for carefully maintained track, respectively. In contrast to the sinusoidal dips used in the ORE and ERRI studies, the present results are for realistic track profiles. Results for low-pass filters with cut-off frequency 20 and 60 Hz are shown. The following can be observed:

- The cut-off frequency has a huge impact on the track irregularity factor; compare with the frequency content from track irregularities in Figure 2.15.

- A higher magnification is obtained for the lower track quality compared to the higher track quality, both with 20 and 60 Hz filters.
3.2. BRIDGE RESPONSE FROM TRACK IRREGULARITIES

- The results give an indication of higher magnification for shorter spans (higher fundamental frequencies), as is assumed in the EN 1991-2 $\phi''$.

- The magnification of acceleration is somewhat higher than the magnification of deflection. This is possibly due to the combination of: (1) the track irregularities primarily add high-frequency content to the signal and (2) the acceleration is proportional to the vibration frequency squared, giving more weight to high frequencies for acceleration compared to deflections. The EN 1991-2 $\phi''$ is the same for acceleration and displacement.

Further studies are needed to draw solid conclusions concerning the relevance of the EN 1991-2 $\phi''$. Especially, research is needed on relevant filter frequencies based on the effect of the high-frequency content on the track and bridge structure.

Figure 3.4: Increase in bridge acceleration (a)–(b) and deflection (c)–(d) due to track irregularities for two cut-off frequencies. Lower track quality results are compared to EN 1991-2 standard track maintenance while higher track quality results are compared to carefully maintained track for bridges with frequencies within $\{n_{0,\text{min}},n_{0,\text{max}}\}$, cf. Figure 3.2.
3.3 Measured bridge response

A TTBI model can be applied to explain certain aspects of measured bridge response. Paper IV uses a simple vehicle model (the wheel masses only) and a model of the ballasted track on the bridge to model the bridge vibration occurring at the sleeper passing frequency. The measured data are from the Faresmyran bridge located along the Bothnia Line in the north of Sweden. For this case, an MF model could not replicate the measured bridge deck acceleration since the excitation from the wheels passing the sleepers is not included in the moving force description.

Figure 3.5 presents an example of a comparison between measured bridge deck acceleration and model results. The measurements (Ülker-Kaustell, 2007) are from a concrete beam bridge, Ullbrobäcken, carrying two tracks in two continuous spans. The Green Train runs at 230 km/h and has been modelled with mechanical properties according to Paper II. The 2D bridge model properties are assumed from design documents and updated based on the measured results. A one-level track model with assumed track stiffness represents the ballasted track. The measured track profile at the bridge is used. However, wavelengths 0.1–2.5 m has been added from the German PSD as the measured profile does not include the shortest wavelengths. The results from the TTBI model are in reasonable agreement with the measured
3.4. PASSENGER COMFORT

results, both in acceleration amplitude and frequency content. Even though the model does not fully describe the measured response, the example shows that a TTBI model can be used to explain the high-frequency content in the measured bridge deck acceleration. As seen from subfigures (a) and (c) the MF model cannot predict the high-frequency content. The ballasted track is considered sensitive to vibrations up to 30 Hz (CEN, 2005) which presents a motive to filter the bridge deck acceleration. Up to 30 Hz, the MF and TTBI model give similar results.

3.4 Passenger comfort

EN 12299 (CEN, 2009) outlines the general procedure for evaluation of passenger comfort in railway traffic. Mean comfort indices, $N_{MV}$, are calculated from the root mean square (rms) of a car body acceleration signal in all three directions. The 95-percetile from 60 samples, each 5 second long, are used. The weighted 5 second rms samples in each measuring direction are denoted continuous comfort indices. Frequency weighing filters take account of the sensitivity of the human body to different vibration frequencies and directions. EN 12299 provides scales for both mean comfort and continuous comfort. The highest comfort level, “very comfortable”, has $N_{MV} < 1.5$ or continuous rms comfort index $< 0.2 \text{ m/s}^2$.

Passenger comfort at bridges cannot easily be evaluated according to the same procedure, due to the short time that the train is on the bridge. A 50 m span is passed in less than 1 second at 200 km/h. The influence of the bridge is outweighed by the track before and after the bridge in the 5 second rms values. In Paper V, the passenger comfort is therefore evaluated based on absolute maximum vertical car body acceleration (centre and ends of car body) when passing the bridge instead of rms values.

The European bridge design codes allow for an indirect verification of the passenger comfort, based on limits for the bridge deck deflection given in EN 1990-A2 (CEN, 2005). These limits, shown in Figure 3.6 (a), are intended to ensure a very good comfort with a maximum car body acceleration of $1 \text{ m/s}^2$. The maximum relative deflection $L/\delta$ are given for simply supported bridges in three or more spans and have multiplication factors that allow for higher deflection for single and two span bridges, as well as continuous bridges in three or more spans. Earlier deflection limits were developed by ORE D160 (1988) and ERRI D190 (1995) based on simulations and experiments.

A comparison against the Japanese comfort deflection limits (RTRI, 2007) is given in Figure 3.6 (b), along with the Japanese deflection limits for running safety. The Japanese limits are intended for the Shinkansen high-speed trains, while the European limit apply also for static load models. For longer spans, the comfort limits in the two standards are rather similar. However, the Japanese limits are more stringent for spans shorter than 40 m. For short spans, the Japanese limits for
safety are instead rather close to the European comfort limits. The reason for the differences is an interesting subject for further studies. Similar comfort deflection limits are given also in the Chinese standards (CNRA, 2014).

### 3.5 Running Safety

There are several possible reasons for derailment; among them can be mentioned rail breakage, geometrical imperfections in wheels or turnouts and obstacles on the track. Other reasons are related to the horizontal and vertical forces between the rail and the wheels of the running vehicle: lateral track shifting, flange climbing, vehicle overturning, and rail rollover. The wheel–rail force consists of: (1) the static part, (2) the quasi-static part (e.g. additional forces in a curved track section with constant speed and cant), and (3) the dynamic part. The quasi-static and dynamic part of the load may increase under severe running conditions, e.g. over-speeding, extreme winds and at track defects.

Flange climbing occurs if the vertical force is not large enough to prevent the wheel from climbing onto the top of the railhead. Flange climbing is possible due to the frictional lifting force acting on a flange that is rolling against the rail at an angle. The risk is assessed by the derailment factor which is the quotient of the lateral, $Y$, to the vertical, $Q$, wheel–rail force. Nadal (1908) gives a theoretical limit for the
derailment factor:

\[
Y/Q = \frac{\tan(\beta) - \mu}{1 + \mu \tan(\beta)}
\]  

(3.1)

for contact angle \( \beta \) and friction coefficient \( \mu \); see Figure 3.7.

The running safety can also be assessed from the wheel unloading ratio (or offload factor):

\[
\frac{\Delta Q}{Q_0} = \frac{Q_0 - Q_{\text{min,dyn}}}{Q_0}
\]

(3.2)

where \( Q_0 \) is the static vertical wheel load and \( Q_{\text{min,dyn}} \) is the minimum dynamic vertical wheel load. Loss of contact occurs at \( \Delta Q/Q_0 = 1 \). Given a certain duration of contact loss and the presence of a lateral force, the contact loss could pose a risk for derailment.

The relative wheel–rail displacement needs to overcome the flange height, typically 30 mm, in order for the wheel to derail. Field tests and simulations show that derailment from flange climbing occurs only when the safety limit has been exceeded for a certain distance or time duration (Matsudaira, 1963; Ishida and Matsuo, 1999; Iwnicki, 2006; Ishida et al., 2007; Montenegro, 2015). The Japanese National Railways (JNR) allow the derailment factor to exceed the safety limit for time periods shorter than 50 ms (Matsudaira, 1963). The Association of American Railroads use the 50 ms time limit for freight trains (Elkins and Carter, 1993; Iwnicki, 2006). Ishida and Matsuo (1999) suggest 15 ms for the Shinkansen high-speed train; see also Ishida et al. (2007). Using a model of the flange–rail contact they show that the wheel rise is below 1 mm for critical \( Y/Q \) ratio below 15 ms duration. Montenegro (2015) concludes that 15 ms is overly conservative as the relative wheel–rail displacement is small compared to the amount needed for the wheel to climb the flange and derail. A simplified estimate of the wheel rise is given by the relative wheel–rail displacement that develops during contact loss in the present 2D TTBI model. Figure 3.8 shows that the maximum wheel rise is well below a typical flange

![Figure 3.7: Forces at the wheel–rail contact, reproduced from Nadal (1908).](image)
CHAPTER 3. THE RESEARCH WORK

height for durations up to 15 and even 25 ms. The results are from the same case as in the example in Section 2.1.2.

Filtering of the wheel–rail force is an alternative approach to determine whether the wheel–rail contact losses are long-lasting enough to pose a risk for derailment. In EN 14363, the wheel–rail force signal is low-pass filtered at 20 Hz and evaluated with a sliding mean over 2.0 m. The filtering and sliding mean of the signal mitigates the short-time contact losses originating from higher oscillation frequencies. For oscillations at 20 Hz, the maximum possible contact loss is 25 ms (half a period); see Figure 3.9. Therefore a 20 Hz low-pass filter removes short-time contact losses originating from frequencies with half-periods shorter than 25 ms.

![Figure 3.8: Relative wheel–rail displacement and contact loss for HSLM-A1 at 20 m bridge (148–400 km/h), where the duration of contact loss increases with speed.](image)

![Figure 3.9: Maximum contact loss (25 ms) at a specific oscillation frequency (20 Hz).](image)
Limits on the derailment factor and wheel unloading in European, Chinese and Japanese design codes are given in Table 3.1. The derailment factor and wheel unloading are used as running safety criteria in the European design codes related to vehicle engineering. However, the wheel unloading is for quasi-static forces and not for dynamic forces. Both the Chinese and the Japanese bridge design codes use the derailment factor and wheel unloading as running safety criteria for dynamic wheel–rail forces. As an alternative to these safety indices, the Japanese design codes allow for safety assessment from the bridge deflection. The Japanese deflection limits are given in Figure 3.6 and should be used together with limits on end rotations and differential displacements. Neither the derailment factor nor the wheel unloading are considered in the European bridge design codes.

European bridge design codes instead assess the running safety from the bridge deck acceleration criterion together with requirements for end rotations and differential displacements. However, the requirements for end rotations are more related to the quasi-static forces in the track components and are primarily available for ballasted tracks. For ballasted bridges, the acceleration limit is due to destabilisation of ballast and risk for derailment at high accelerations. For non-ballasted bridges the acceleration limit is related to the assumed loss of contact between the wheel and the rail at the gravitational acceleration 1 g. This assumption is studied in Paper V where running safety indices from the wheel–rail force are studied for bridges at the design limit for acceleration and deflection. The focus is on the variation in the dynamic wheel–rail force from bridge vibrations and also from the irregular track profile. Lateral forces are neglected. Thus, the running safety is evaluated only based on the 2D vertical motion of the coupled TTBI system with wheel unloading

Table 3.1: Limit values for derailment factor $Y/Q$ and wheel unloading $\Delta Q/Q_0$.  

<table>
<thead>
<tr>
<th>Source</th>
<th>$Y/Q$</th>
<th>$\Delta Q/Q_0$</th>
</tr>
</thead>
<tbody>
<tr>
<td>EN 14363 (CEN, 2016), vehicle acceptance</td>
<td>0.8 ($20$ Hz filter, 2 m window)</td>
<td>0.6 (quasi-static)</td>
</tr>
<tr>
<td>EN 14067-6 (CEN, 2010a), cross wind assessment (CNRA, 2014), Chinese bridge design code</td>
<td>0.8</td>
<td>0.9 ($2$ Hz filter)</td>
</tr>
<tr>
<td>(RTRI, 2007), Japanese bridge design code</td>
<td>0.8 (normal track maintenance)</td>
<td>0.8 (normal track maintenance)</td>
</tr>
<tr>
<td></td>
<td>0.3 (track assumed smooth)</td>
<td>0.37 (track assumed smooth)</td>
</tr>
<tr>
<td>Matsudaira (1963)</td>
<td>$&lt; 50$ ms (allowed safety limit exceedance)</td>
<td>–</td>
</tr>
<tr>
<td>Ishida and Matsuo (1999)</td>
<td>$&lt; 15$ ms (allowed safety limit exceedance)</td>
<td>–</td>
</tr>
</tbody>
</table>
and contact loss. Relevant safety indices are needed to determine whether wheel–rail contact losses are long-lasting enough to pose a risk for derailment. To this end, two running safety indices were used in Paper V:

1. Filtered wheel unloading $\Delta Q_{20Hz}/Q_0 < 0.6$, with low-pass filter at 20 Hz (in line with the quasi–static limit and filtering in EN 14363).

2. Unfiltered wheel unloading, duration of contact loss $< 15$ ms (time limit for flange climbing suggested by Ishida and Matsuo (1999)).

### 3.6 Sensitivity study

A sensitivity analysis for the train and track parameters was performed for the HSLM-A1 train travelling a 400 m track section without bridge. The maximum response from the speed range 260–400 km/h was extracted for 2000 Monte Carlo samples with variations in the model input parameters following uniform distributions. The mean values for the train parameters were according to Paper V. Variation was introduced in all mechanical data ($\pm 30\%$ in mass and inertia, and $\pm 50\%$ in suspension characteristics). The mean values of the track parameters were according to Paper V with variation in the rail pad stiffness (40–160 MN/m, two pads) and damping (7–13%). A track profile with wavelengths 1–150 m was randomly scaled so that the standard deviation $\sigma_{3–25}$ lies between 0 and 1 mm in each Monte Carlo sample.

The Monte Carlo sample results are plotted against the standard deviation of track irregularities in Figure 3.10. The contact loss, filtered wheel unloading and car body acceleration are shown against the track profile standard deviation $\sigma_{3–25}$ for 2000 Monte Carlo samples with random vehicle and track data.

![Figure 3.10](image-url)

**Figure 3.10**: Contact loss $C_{loss}$ (a), filtered wheel unloading $\Delta Q_{20Hz}/Q_0$ (b) and car body acceleration $a_c$ (c) plotted against the track profile standard deviation $\sigma_{3–25}$ for 2000 Monte Carlo samples with random vehicle and track data.
acceleration are all sensitive to the track quality level and increase with increasing track irregularities. The remaining scatter is due to the variation in rail pad and vehicle mechanical data.

Figure 3.11 provides a graphical method (Au and Wang, 2014; Tell, 2017) to estimate the effect of each input parameter. The prior distribution of the stochastic variables are plotted together with the conditional distribution given that the 90th percentile of response is exceeded. The method shows which parameters are important based on the change in their distribution for the 90th percentile sample. The figure shows 7 of the 12 stochastic variables.

Figure 3.11 confirms that the amplitude of track irregularities is the most important factor for contact loss, wheel unloading and car body acceleration. The high influence of the track profile complicates the interpretation of running safety and passenger comfort results for trains on bridges. This is why a method was developed in Paper V to isolate the effect of the bridge from the effect of the track profile. To this end, simulations with the same track profile were performed on a track section with and without bridge.

The rail pad stiffness has considerable effect on the contact loss (from the unfiltered wheel–rail force). This agrees with the discussion on the increase in wheel–rail force with increasing track stiffness in Paper IV. However, the effect of increased track stiffness is negligible for the filtered wheel unloading as much of the wheel–track dynamic interaction lies above 20 Hz.

Among the vehicle parameters, only the car body mass and the wheel mass have notable effect on the wheel–rail forces. The explanation for decreasing wheel unloading and contact loss for increasing car body mass is simple: the dynamic part of the wheel–rail fore is comparatively smaller for higher static loads; cf. Eq. (3.2). A relevant lower limit car body mass should therefore be assumed for running safety assessments. In the event that the wheel unloading is also affected by vibrations from passing a bridge this effect is counteracted by the increase in bridge response from higher axle loads.

Increasing wheel mass increases the wheel–rail forces. This can be realised from the coupled equations of motion for the rigid contact model, Eq. (2.3)–(2.6), where \( m_w \) occurs in the coupling terms and force terms. A relevant upper limit wheel mass should therefore be assumed for running safety assessments.

The car body acceleration is very little affected by the wheel mass, but is instead sensitive to the suspensions and the suspended vehicle bodies. The figure shows the most important parameters: car body mass, car body inertia and secondary suspension system stiffness, but also the remaining vehicle parameters have some effect.
Figure 3.11: Sensitivity study: prior distribution of stochastic variables and conditional distribution given that the 90th percentile of response is exceeded. The notations are according to Figure 2.1.
Chapter 4

Conclusions and further research

The work presented in this thesis has contributed to an enhanced understanding of the effect of TTBI on railway bridge dynamic response. The following conclusions are based on the main findings in the work; more specific conclusions can be found in the appended papers.

4.1 Theoretical conclusions

The application of TTBI models for the analysis of railway bridge dynamic response has led to the following conclusions:

- The EN 1991-2 additional damping $\Delta \zeta$ was shown non-conservative for many train–bridge systems as calibrated additional damping well below the code values were obtained for span lengths 10–20 m.

- The choice between different 2D vehicle models can be made based on the needed output and the available input data. The SIM (wheel–bogie) was shown to give similar results to the RB (wheel–bogie–car) model for both bridge response and wheel–rail forces. However, the car body needs to be modelled for passenger comfort assessment.

- The choice of the shortest modelled track irregularity wavelengths has a large influence on the unfiltered wheel–rail forces. This proved important for running safety assessments based on duration of contact loss. In this work, wavelengths down to 1 m were found sufficient. These wavelengths were shown to induce frequencies at the loaded track frequency; for wheel–rail forces filtered below this frequency the effect of the low wavelengths was removed. The Swedish requirements for track profile quality (Trafikverket, 2015) considers wavelengths down to 1 m, in contrast to 3 m considered in EN 13848-5.
- A TTBI model may prove to be essential in explaining measured bridge response as signals may include high-frequency content due to track irregularities and the sleeper passing frequency.

The running safety on non-ballasted bridges was assessed based on the wheel–rail forces in a comprehensive parametric study in Paper V. The work led to the following conclusions:

- The unfiltered wheel–rail force was shown to have a considerable high-frequency content from the wheel running over the irregular track. However, previous research has shown that short-time wheel unloading is not relevant for derailment. Consequently, safety indices based on duration of contact loss or filtered wheel unloading should be adopted to determine whether the variations in the wheel–rail force pose any risk for derailment.

- The assumption behind the present Eurocode acceleration limit is that wheel–rail contact loss occurs at 10 m/s² (1 g). The present results showed that the contact loss was more related to the track quality than the bridge response and that a deck acceleration of 1 g did not in itself lead to loss of contact.

- The running safety indices were not exceeded for bridge deck accelerations up to 30 m/s²; the EN 1990-A2 serviceability deflection limit for passenger comfort was reached before the safety indices were compromised. Noteworthy is that the Japanese design codes have bridge deflection limits for running safety that are similar to the Eurocode comfort deflection limits.

4.2 Practical implications

The work in this thesis shows that a TTBI model can be applied to assess the validity of some of the Eurocode design criteria for dynamic analysis of bridges. The results have led to the following practical implications:

- The EN 1991-2 additional damping $\Delta \zeta$ is not appropriate in the design of bridges that are to be inter-operable since the effect of TBI varies with the train–bridge system characteristics.

- The EN 1990-A2 deck acceleration limit for non-ballasted bridges (5 m/s²) is overly conservative and there is a potential in improving this running safety criterion.

The running safety is ideally assessed from indices based on the wheel–rail forces. In the bridge design stage however, this may require unnecessarily complicated
models. Preferably, the design requirement should be based on the bridge response available from an MF analysis.

The passenger comfort on bridges was briefly studied in Paper V. The tentative conclusion is that the EN 1990-A2 bridge deck deflection limits are relevant in ensuring passenger comfort by limiting the car body acceleration. However, the car body acceleration was shown sensitive to the vehicle mechanical parameters which is why solid conclusions would require vehicle models with properties that represent an envelope of real trains. From a limited study on the EN 1991-2 track irregularity factor it can be concluded that the magnification of bridge response varies very much with the track quality and the choice of filter frequency. Further studies are needed to draw solid conclusions concerning the relevance of the EN 1991-2 $\phi''$.

4.3 Further research

The author identifies the following further research directions directly following the work in this thesis:

- The correlation between wheel–rail forces and deck response at non-ballasted bridges needs to be further studied. A deeper knowledge of the full TTBI system is vital in order to deduce running safety requirements based on bridge response. The work within this thesis shows tentatively that bridges complying with the EN 1990-A2 serviceability deflection limits also comply with the running safety indices based on wheel–rail forces. Further studies are needed to show, for example, if a criterion based on bridge deflection is applicable, or if an increased deck acceleration limit should be adopted. Comparisons against the Japanese deflection limits for safety are interesting.

- The motives behind the differences between running safety requirements on bridges in, for example, European, Chinese and Japanese design codes need to be identified. Running safety indices from wheel–rail forces are used in Chinese and Japanese bridge design codes and in European requirements for vehicle engineering, but not in European bridge design codes. In this context, it is also interesting to compare and harmonize the running safety requirements at bridges with those at plain track sections.

- TTBI studies in 3D are interesting to determine whether there exists conditions when the running safety is limited by 3D effects. With reference to the work by Andersson and Svedholm (2016) the vertical bridge deck response of the studied bridges was deemed relatively well represented by a 2D model. However, 3D effects such as deck twist, lateral deformation and end rotations from eccentricities have not been studied. Experiments are also an important step to further validate both the theoretical models and the adopted running safety indices.
CHAPTER 4. CONCLUSIONS AND FURTHER RESEARCH

- The limited amount of studies on bridge response from track irregularities motivates further studies on the EN 1991-2 track irregularity factor $\phi''$ for application to bridge deck acceleration from high-speed trains. Such studies are dependent on two aspects: (1) the track quality levels for simulations must be well-defined, preferably in line with the EN 13848 track maintenance requirements and (2) relevant filter frequencies must be defined based on the effect of high-frequency content on the bridge and track structure. Both points may very well be different between ballasted and non-ballasted bridges.

- As the Eurocode additional damping was shown non-conservative, lower limit additional damping values could be calibrated for specific bridges based on ranges of train parameters. This could be useful in the analysis of, e.g., existing bridges as a simplified way of accounting for the reduction in response from TBI. To this end, the analysis of complex eigenvalues as discussed by Svedholm (2017) and Zäll (2018) can be useful.

- The passenger comfort on bridges is an interesting topic to study in more detail, accounting for factors such as lateral vibrations, flexible car bodies and passenger seat suspensions. The short time that the train is on the bridge raises questions on how to treat the car body acceleration signal in relation to the passenger sensitivity to short-term discomfort.

Closely related to the present work is also further research on:

- The 3.5 m/s$^2$ deck acceleration limit for ballasted bridges due to ballast instability. EN 1990-A2 states that frequencies up to $\max\{30 \text{ Hz}; 1.5 \times 1^{\text{st}} \text{ eigen-frequency}; 3^{\text{rd}} \text{ eigen-frequency}\}$ shall be considered. Especially for short-span bridges, this may result in frequencies up to 100 Hz or more and very high simulated deck accelerations (Johansson et al., 2011). The upper frequency limit need to be further studied based on experimental testing. Since the dynamic response of short-span bridges often shows peak acceleration from transient loading rather than resonant loading, it is also of interest to study the ballast behaviour under different loading conditions.

- An enhanced definition of track profile quality that is applicable both for track maintenance and running safety simulations. Li et al. (2012) have stressed the importance of the second-order derivative in characterising track irregularities, which is not considered in the present design codes. There is also a lack of recommendations on the frequency distribution of the track irregularities. As shown in this thesis, profile samples from theoretical PSD functions may have less isolated defects compared to measured profiles, resulting in a trade-off between calibration of standard deviation and maximum amplitude.
Bibliography


BIBLIOGRAPHY


BIBLIOGRAPHY


Part B

Appended papers
Paper I

Train–bridge interaction – a review and discussion of key model parameters

Paper II

Statistical screening of modelling alternatives in train–bridge interaction systems

Paper III

Train–track–bridge modelling and review of parameters

Paper IV

Influence of sleeper passing frequency on short span bridges – validation against measured results

First International Conference on Rail Transportation, Chengdu, China, 10–12 July 2017.
Paper V

Train running safety on non-ballasted bridges
