On Derailment-Worthiness in Rail Vehicle Design

Analysis of vehicle features influencing derailment processes and consequences

by

Dan Brabie

Doctoral Thesis

TRITA AVE 2007:78
ISSN 1651-7660
Doctoral Thesis in Railway Technology
ISSN 1651-7660
TRITA AVE 2007:78
DAN BRABIE

On Derailment-Worthiness in Rail Vehicle Design
Analysis of vehicle features influencing derailment processes and consequences
© DAN BRABIE, 2007

Royal Institute of Technology
School of Engineering Sciences
Department of Aeronautical and Vehicle Engineering
Division of Rail Vehicles
SE-100 44 Stockholm
Sweden
Phone +46 8 790 6000
www.kth.se
Preface and acknowledgements

The work reported in this doctoral thesis has been carried out at the Division of Rail Vehicles, Department of Aeronautical and Vehicle Engineering, at the Royal Institute of Technology (KTH), Stockholm.

The research project, called “Robust Safety Systems for Trains”, was initiated by the former Swedish State Railways (SJ), triggered by observations of some “successful” high-speed derailments involving the Swedish tilting train X 2000.

The project was funded by combined efforts of Banverket (Swedish National Rail Administration), Vinnova (Swedish Governmental Agency for Innovation Systems) and the Railway Group of KTH. The financial support of the above named companies and organisations is gratefully acknowledged.

First and foremost, I am most grateful to my supervisor, Prof. Evert Andersson, for all his guidance in the choice of directions, sincere dedication and critical comments throughout this time. His vast expertise in the railway field has been a valuable asset to this work.

I would also like to acknowledge the Swedish rail operator SJ AB (Dept. of Traffic and Vehicle Safety), Interfleet Technology AB and Bombardier Transportation for providing an open access to valuable incident information and necessary vehicle data.

The commitment and practical advice received from the members of the reference group is also greatly appreciated: Tohmmy Gustad, Christer Ljunggren, Hugo von Bahr, Tomas Persson and Stefan Sollander.

Special thanks also to Prof. Mats Berg for his involvement along the years and constructive comments on the manuscripts, as well as to all my former and present colleagues at the Rail Vehicle Division for a pleasant working environment.

I would also like to thank Dr. Johan Bäckman for allowing access to the initial incident/accident database. Furthermore, special thanks deserves Ingemar Persson for all the help received with the simulation software package GENSYS and Dr. Anders Ansell for a fruitful discussion on concrete impact issues. Likewise, I wish to thank Dr. Per-Anders Jönsson for always taking time to solve my puzzles on the powerful Andromeda computer system.

I also wish to thank Dr. Andrew Peplow for his comprehensive proofreading of the manuscripts.

On a personal level, I must thank my wife, Catharina, for her persistent patience and understanding. At the same time I must apologise to Jannick and Theodor, our wonderful boys, for not being around them as much as I would have wanted.

I also acknowledge the role of my parents, Cecilia and Voicu, for my education as well as their encouragement along these years.

Stockholm, November 2007
Dan Brabie
Abstract

This thesis aims at systematically studying the possibilities of minimizing devastating consequences of high-speed rail vehicle derailments by appropriate measures and features in the train design including the running gear.

Firstly, an empirical database is established containing as much relevant information as possible of past incidents and accidents that have occurred at substantial running speeds due to mechanical failure close to the interface between the running gear and the track. Other causes that ultimately brought the train in a derailed condition are also covered. Although various accidental circumstances make each derailment a unique event, certain patterns appear to emerge which lead to several critical vehicle parameters capable of influencing the outcome of a derailment or preventing a derailment to occur.

Secondly, the possibility of preventing wheel climbing derailments after an axle journal failure is studied by implementing mechanical restrictions between wheelsets and bogie frame. In this respect, a multi body system (MBS) computer model is developed to account for such an axle failure condition, which is successfully validated on the basis of two authentic passenger car events.

In order to study the overall post-derailment vehicle behaviour, in particular the wheelsets’ vertical motion and lateral deviation on sleepers, a comprehensive MBS post-derailment module is developed and implemented in the commercially available software GENSYS. The model detects wheel-sleeper impact conditions and applies valid force resultants calculated through linear interpolation based on a pre-defined look-up table. The table was constructed through exhaustive finite element (FE) wheel to concrete sleeper impact simulations utilising the commercially available software LS-DYNA. The MBS post-derailment module has been validated successfully in several stages, including a correct prediction of the derailing wheelset’s trajectory over ten consecutive sleepers in comparison with an authentic passenger vehicle derailment event.

An extensive simulation analysis on the feasibility of utilizing alternative substitute guidance mechanisms attached to the running gear on rail vehicles is presented, as means of minimizing the lateral deviation. Three low-reaching guidance mechanisms attached onto the running gear (bogie frame, brake disc and axle journal box) are analysed in terms of geometrical parameters for a successful engagement with the rail in order to prevent large lateral deviations after twelve different derailment scenarios.

Three conventional coupled passenger trailing cars are investigated in terms of lateral deviation and vehicle overturning tendency after derailments on tangent and curved track. This is performed as a function of various vehicle design features and parameters such as: maximum centre coupler yaw angle, carbody height of centre of gravity, coupler height and additional running gear features. In a similar manner, the articulated train concept is investigated in terms of the post-derailment vehicle behaviour as a function of different inter-carbody damper characteristics and running gear features.

Keywords: derailment, bogie design, simulation, wheel-sleeper impact, deviation, vehicle inter-connection, guidance mechanism, substitute guidance, articulated train, railway safety
Outline of Thesis

The thesis consists of a summary and the following six appended papers:

**Paper A**

**Paper B**
Brabie D. and Andersson E.: *Rail vehicle axle failure on the outside of the wheels - means of minimizing the risk of derailment*, submitted for publication to Journal of Rail and Rapid Transit.

**Paper C**

**Paper D**

**Paper E**
Brabie D. and Andersson E.: *Alternative substitute guidance mechanisms - means of minimizing catastrophic lateral deviation after derailments at high speed*, to be submitted for publication.

**Paper F**
Brabie D. and Andersson E.: *Means of minimizing catastrophic consequences after derailments at high speed - vehicle inter-connections and running gear design features*, to be submitted for publication.

For all papers Brabie has gathered information on incidents and accidents, developed simulations models, carried out calculations, analysed results, and written the manuscripts.

The simulations models, analyses and manuscripts have been discussed with and reviewed by Andersson.

Andersson supervised the work and planned the papers in collaboration with Brabie.
Contribution of Thesis

This thesis is believed as being a pioneering work in the area of railway safety aiming at reducing the lack of knowledge on derailment dynamics and its consequences, in particular the influence of vehicle features and design parameters.

The thesis makes the following contributions to the field of railway safety and also to simulation methodology itself:

• A compilation of accidents and incidents is presented on which basis several vehicle features and train design parameters are identified as being able to limit the consequences associated with train derailments at higher speeds.

• A comprehensive vehicle model is developed and successfully validated with two authentic events in terms of the pre-derailment sequence of events after axle journal failures.

• It presents and analyses in detail one method to limit flange climbing derailments caused by axle journal failures by inserting mechanical restrictions between the wheelset and the bogie frame.

• A finite element (FE) model is developed for studying the impact phenomenon between a derailed rail vehicle wheel and concrete sleepers. The model is tentatively validated with good results based on one authentic accident event.

• A multi body system (MBS) post-derailment module is developed which detects contact with sleepers and/or rail fasteners and applies valid force resultants based on existing impact conditions. The accuracy of the MBS code in terms of a wheel’s three-dimensional trajectory over 24 sleepers is successfully validated by comparing with its FE counterpart for an arbitrary impact scenario.

• A complete MBS vehicle model employing the post-derailment module is successfully validated in terms of a wheelsets trajectory over 10 consecutive sleepers, based on an authentic passenger car derailment with subsequent on-site measurements. A further successful validation of a complete MBS vehicle model is made based on an authentic passenger car derailment in terms of the general post-derailment vehicle behaviour.

• An extensive simulation analysis is presented on the feasibility of utilizing alternative substitute guidance mechanisms on passenger rail vehicles in order to minimize lateral deviations after derailment at high speeds.

• A study is presented on the lateral deviation and carbody overturning tendency for conventionally coupled passenger cars as well as for an articulated passenger train, as a function of various vehicle design features and parameters.

• Suggestions are given on rail vehicle design features that should reduce the consequences of derailment.
On Derailment-Worthiness in Rail Vehicle Design

Contents

Preface and acknowledgements ................................................................. i
Abstract ................................................................................................. iii
Outline of Thesis ................................................................. v
Contribution of Thesis ............................................................. vii
Symbols and Abbreviations .............................................................. xi

1 Introduction ......................................................................................... 1
  1.1 Background .................................................................................. 1
  1.2 Previous research ......................................................................... 2

2 Empirical observations ......................................................................... 5
  2.1 The incident/accident database ...................................................... 5
  2.2 Additional cases ............................................................................. 6
  2.3 Empirically based conclusions ...................................................... 10

3 Post-derailment vehicle simulations ................................................. 13
  3.1 Methodology .................................................................................. 13
      3.1.1 Wheel-concrete sleeper impact: FE simulations ...................... 13
      3.1.2 MBS post-derailment module ............................................... 14
  3.2 Validation stages ........................................................................... 16
      3.2.1 Phase I (FE model => Authentic derailment) ...................... 16
      3.2.2 Phase II (FE model => MBS model) .................................. 18
      3.2.3 Phase III (MBS model => Authentic derailments) ............. 19

4 Minimizing consequences of axle journal failure .............................. 21
  4.1 MBS model of an axle journal failure .......................................... 21
      4.1.1 Methodology ........................................................................ 21
      4.1.2 Validation ............................................................................. 21
  4.2 Mechanical restrictions .................................................................. 23

5 Minimizing derailment consequences .............................................. 25
  5.1 Derailment scenarios under consideration .................................. 25
  5.2 General vehicle modelling considerations .................................. 26
  5.3 Conventional train configuration .................................................. 27
      5.3.1 Lateral substitute guidance mechanisms ............................ 27
      5.3.2 Vehicle coupler restrictions ............................................... 30
      5.3.3 Height of centre of gravity and other vehicle design features 33
  5.4 Articulated train configuration ......................................................... 34

6 Discussion and Conclusions ................................................................. 39

7 Future directions of study ................................................................. 43

References ............................................................................................. 45

Appendix A - Concrete material modelling details ................................. 49

Paper A-F
### Symbols and Abbreviations

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Significance</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>b, B, B_{ajb}</td>
<td>lateral gap between guidance mechanism and its adjacent wheel</td>
<td>mm</td>
</tr>
<tr>
<td>D</td>
<td>track cant</td>
<td>mm</td>
</tr>
<tr>
<td>f'_c</td>
<td>concrete unconfined compressive strength</td>
<td>N/m²</td>
</tr>
<tr>
<td>h, H, H_{ajb}</td>
<td>vertical distance between the lowest point on the guidance mechanism and top of the rail</td>
<td>mm</td>
</tr>
<tr>
<td>h_{ccg}</td>
<td>carbody height of centre of gravity</td>
<td>m</td>
</tr>
<tr>
<td>h_{cpl}</td>
<td>coupler height from top of rail</td>
<td>m</td>
</tr>
<tr>
<td>h_{tb}</td>
<td>vertical distance bogie transversal beam to top of rail</td>
<td>mm</td>
</tr>
<tr>
<td>h_z</td>
<td>vertical wheel position at the instant of impact with the sleeper</td>
<td>m</td>
</tr>
<tr>
<td>P</td>
<td>pressure</td>
<td>N/m²</td>
</tr>
<tr>
<td>R, R_S</td>
<td>track curve radius</td>
<td>m</td>
</tr>
<tr>
<td>s_d</td>
<td>sleeper spacing</td>
<td>m</td>
</tr>
<tr>
<td>V</td>
<td>train speed</td>
<td>km/h</td>
</tr>
<tr>
<td>v_x</td>
<td>longitudinal wheel velocity, i.e. train speed, at the instant of impact with sleeper</td>
<td>km/h</td>
</tr>
<tr>
<td>v_y</td>
<td>lateral wheel velocity at the instant of impact with sleeper</td>
<td>m/s</td>
</tr>
<tr>
<td>v_z</td>
<td>vertical wheel velocity at the instant of impact with sleeper</td>
<td>m/s</td>
</tr>
<tr>
<td>y_{pc-max}</td>
<td>maximum inter-carbody lateral deflection</td>
<td>m</td>
</tr>
<tr>
<td>ε_p</td>
<td>plastic strain</td>
<td>(-)</td>
</tr>
<tr>
<td>η</td>
<td>damage scale factor in concrete</td>
<td>(-)</td>
</tr>
<tr>
<td>λ</td>
<td>damage function in concrete</td>
<td>(-)</td>
</tr>
<tr>
<td>σ_{cut}</td>
<td>tensile strength in concrete</td>
<td>N/m²</td>
</tr>
<tr>
<td>σ_{failed}</td>
<td>minimum, damaged strength of concrete</td>
<td>N/m²</td>
</tr>
<tr>
<td>σ_{max}</td>
<td>maximum, undamaged strength of concrete</td>
<td>N/m²</td>
</tr>
<tr>
<td>ψ</td>
<td>wheelset’s yaw angle relative to sleeper at the instant of impact with a sleeper</td>
<td>(°)</td>
</tr>
</tbody>
</table>
**Abbreviations**

A1  Articulated trainset configuration  
AJF  Axle Journal Failure  
AWS  Automatic Warning System  
C1, C2, C3  Conventionally coupled trainset configurations  
CEB  Comité Euro-Internationel du Béton  
CoG  Centre of Gravity  
FE  Finite Element  
DB  Deutsche Bahn  
DoF  Degrees of Freedom  
DMU  Diesel Multiple Unit  
EMU  Electric Multiple Unit  
EN  European Norm  
ERA  European Rail Agency  
HRF  High Rail Failure  
KHST  Korean High-Speed Train  
KTH  Kungliga Tekniska Högskolan (Royal Institute of Technology)  
Max  Maximum  
MBS  Multi Body Systems  
RAIB  Rail Accident Investigation Branch (UK)  
RSSB  Rail Safety and Standards Board (UK)  
SJ  Formerly Statens Järnvägar (Swedish State Railways)  
Std  Standard  
TGV  Train à Grande Vitesse  
ToR  Top of Rail  
UIC  Union Internationale des Chemins de fer  
WFOR  Wheel Flange on Rail
1 Introduction

1.1 Background

Rail travel is undoubtedly one of the safest modes of transportation. However, accidents and incidents continue to occur and, as with any technical system, there is a continuous demand for improvements. Moreover, bearing in mind the ever increasing speed of trains, a further enhanced safety in railway operation is desired. The railway industry assesses this issue by minimizing the risk, which is often defined as a product of the probability that a certain hazardous condition occurs and its possible consequence [23]. Through the years, two complementary railway safety branches have emerged:

- active safety: methods that reduce the probability of accidents (automatic train protection and control, signalling, wheel/rail crack inspection etc.)
- passive safety: methods that aim at minimizing consequences once an accident or incident does occur.

The introduction of active safety measures, in particular automatic train protection and control, has practically eliminated derailment events due to over-speeding or high-speed train collisions [22]. However, despite refined inspection techniques as well as more reliable vehicles [37], derailments due to mechanical failures affecting the wheelset guidance on rails continue to occur. Moreover, certain types of derailment causes such as vandalism/terrorism (objects on the line or other deliberate actions) or environmental factors (landslide, earthquake etc.) are not likely to be diminished by active measures in the near future.

The passive safety area has received much attention in recent years, especially in the European Union through projects like Traincol, Safetrain, Trainsafe [14], Safeinteriors [32] etc. Often, passive safety is presented as synonymous with crashworthiness [38][20], implying the ability of the vehicle to protect passengers and crew members in events following collisions. In this respect, two standards exist or are under completion in Europe: the High Speed TSI [40] and the Euronorm draft [29].

Statistics on British Rail for the time period 1973-1992, indicate that the second largest proportion of fatalities and injuries, after end-on collisions, arises as a consequence of derailments [21]. However, the vehicle behaviour immediately following a derailment is, to the best of the author’s knowledge, not considered in any current standard.

In 2002, the project “Robust Safety Systems for Trains” began at the Royal Institute of Technology (KTH) in Sweden. The focus has been to identify and to perform parameter analyses on those vehicle design features that would minimize any possible catastrophic sequence of events following derailment at higher speeds. The causes have been limited to mechanical failures affecting the running gear as well as other events that immediately lead to derailments. The results from this project stand as a basis of the current thesis.

In analogy with the well-known crashworthiness concept mentioned above, a new parallel branch within passive safety has emerged along this work. It could be entitled derailment-worthiness, with the following definition: the ability of a rail vehicle or
trainset to avoid collisions and overturning of vehicles following a derailment, so that passengers and crew members are protected.

The question whether derailment-worthiness, or simply, “good” post-derailment vehicle behaviour, could be incorporated into the vehicle design has not received much consideration in the literature.

Recently, two derailment incidents from Japan and UK, have initiated these issues in these countries. The Shinkansen derailment at Niigata in 2004 [13] has determined the Japanese railway JR-East to start mounting lateral guidance devices on the axle boxes, so-called ‘L-shaped car guides’, on all Shinkansen bogies in order to minimise the deviation after derailment [12]. In conjunction with a derailment in 2005 at Moy in UK, the Rail Accident Investigation Branch (RAIB) made a recommendation to Rail Safety and Standard Board (RSSB) to commence studies of design elements that would limit the degree of deviation from the track [31].

There appears to be a belief within the railway community that some train designs can cope better with derailments, thus having incorporated some kind of ‘last barrier’, that ultimately may lead to less catastrophic consequences [27][11][1]. One such example is the articulated train design, i.e. carbody ends sharing the same bogie, which have empirically proven a high degree of derailment-worthiness even during an event occurring at approximately 300 km/h. In the same manner and also based on empirical observations, the Swedish operator SJ AB and Interfleet Technology, describe some current non-articulated Swedish trains (X 2000 and X10) as having favourable properties once these vehicles are brought into a derailed condition.

A natural question which may be asked what means are there to quantify the derailment-worthiness of a train design? Clearly, derailments which come to a halt with no injured passengers or staff members should qualify as “good” post-derailment vehicle behaviour. However, in many derailments, the lack of serious consequences could be due to “lucky” accidental circumstances, such as: no steep embankment, no scheduled train on the opposite track etc. Indeed, following a derailment, wheelsets that are maintained at a minimal lateral deviation from the track centre line, are beneficial. Even a higher degree of derailment-worthiness is incorporated in those train designs that can continue maintaining a minimal lateral deviation despite encountering aggravating factors such as curves or track discontinuities (switches and crossings). Consequently, such behaviour enables the vehicles to remain on the track-bed, in-line, upright and connected, which common sense as well as empirical evidence [39] suggests that few serious casualties would occur even at high-speed events.

In conjunction with vehicle crashworthiness, the strive for keeping vehicle on-line, upright and connected is not new. Such behaviour is often mentioned as very desirable, so that various vehicle energy absorption features engage properly following collisions [6][14].

1.2 Previous research

The research and development disclosed in the area of railway safety is rather scarce. Especially the disproportion between the amount of articles written on crash safety on
one side, and train stability after a derailment on the other, is striking. There is, to the
author’s knowledge, no research results published that systematically analyses the
relationship between the seriousness of an event when a vehicle leaves the rails and the
respective train design, in particular the design of the running gear.

The oldest references found in the field of post-derailment assessment date back to 1972
[46] [47], where the equations of motion for tank wagons, with three degrees of freedom
(DoF) for each car in the horizontal plane, are coupled with a simplified system of
constraints. The motion of each derailed vehicle is governed by a horizontal ground
friction vector, inversely directed to the velocity vector, and the couplers, which are not
allowed to fail. Several dependencies are sought such as the influence of ground friction
coefficient, number of cars in the train, train speed, coupler moment etc. The model is
validated with good results in terms of the number of derailed cars for an authentic case,
chosen to best match the two-dimensional assumption. The results follow a pattern
according to accepted mechanical principles. In this context, one finding in particular is
interesting to mention: a mixed consist of vehicles, two loaded followed by one empty,
leads to a substantial increase of the lateral deflection from the track centre line.

In an attempt to improve the safety of freight cars, a computer program was developed to
predict different catastrophic scenarios related to tank wagon accidents [2][3] (liquid
spill, fire effects, explosions etc.). One of the sub-models in the program considers the
derailment mechanics, which allows motion with four DoF per vehicle as well as coupler
separation. However, roll is only included in the equations of motion for uncoupled
vehicles. Derailment is initiated at a pre-defined vehicle in the train consist. All the
following vehicles are considered as derailed, implying that Coulomb friction forces act
in reverse direction to the velocity vector at the two bogie locations of the vehicles. This
program is not reported to be validated, but an example of a hypothetical derailment
prediction is presented.

In reference [17] the main focus is train impact on adjacent structures. A mathematical
model describes the vehicle’s motion after derailment. As in the previous work, once a
derailment state is postulated, a simplistic approach to the wheel-ground interface is
implemented. The two-dimensional equations of motion in the horizontal plane are then
solved iteratively using the principle of virtual work. A parametric study is then
presented which involves the speed of the train at the instant of derailment, the friction
coefficients and the so-called derailment angle. The authors conclude that the lateral
train velocity component is highly affected by the wheel-ground friction coefficient.
Meanwhile, the friction coefficients are reported to have a negligible effect on the
longitudinal velocity component.

Also with focus on freight cars, a special purpose derailment computer model was
developed in [43]. As in previous work, the model is planar with lateral and yaw DoF for
each car. Moreover, bogies and wheelsets are ignored. Advancements were made on the
inter-vehicle couplers reaction forces including possible separation, car to ground
reaction forces, brake forces and car to car collision forces. The model was validated
successfully based on a well known and documented event in terms of the number and
relative placement of derailed cars. The variation of train speed, car mass and trainset
length had most effect on the derailment severity.
Section 1 - Introduction

The possibility of applying three-dimensional multi body system (MBS) simulations, instead of finite element (FE) simulations in crash analysis is studied in reference [18]. The model accounts for six DoF for each relevant rigid-body part of the vehicle. Although the main focus is the possibility to determine the gross motion of trains after a crash impact, the authors state that derailment dynamics should also be incorporated for crash scenarios. In this respect, the implemented wheel-rail contact model allows wheels to lift from the rail surface. However, little is revealed regarding the wheel-ground contact. In order to study the possibility of derailment, a side crash simulation involving the Korean High Speed Train (KHST) is performed. The lateral displacements of the overridden cars await experimental validation.

In order to investigate possible detection algorithms and sensor positioning for derailing freight cars, MBS simulations and field experiments are performed in [4]. The MBS derailment model was mainly developed for a preliminary assessment of the derailing motion of the wheelsets. Accordingly, the wheelset rolling over concrete sleepers is mathematically modelled as vertical sinusoidal irregularities corresponding to the sleeper frequency. In addition, the lateral motion of the derailed wheelset is not considered. The experimental work consisted of derailing a trailing wheelset in an instrumented four-axle freight car at speeds up to 30 km/h. The power spectral density of the vertical acceleration at the carbody centre reveals a peak corresponding to the sleeper frequency.

A planar derailment model was developed and implemented in the commercially available MBS software ADAMS [26]. As in most of the previous cases, the intention here is to analyse the gross motion of freight cars following derailments. Each car is modelled as one rigid body with three DoF (longitudinal and lateral translation as well as yaw rotation). The actual derailment is prescribed by an angular and lateral velocity on the leading car. In a derailed condition, frictional forces are applied to the car at the position of the bogie centres. A parametric study is performed in terms of derailment severity by varying trainset length and speed, car to ground friction coefficient as well as additional inter-vehicle coupler characteristics. Among others, it is concluded that motion of the derailed cars is strongly dependent on the number of cars in the train, coupler separation and the initial angular velocity imposed on the leading car.
2 Empirical observations

2.1 The incident/accident database

A narrative description, from the point of view of the current study, is presented in Paper A, for all the incidents and accidents (35 as of February 2007) included in the database. Initially, a Swedish database [8] was used and condensed according to the following criteria: (i) passenger trains running at a speed above 70 km/h and (ii) with the primary cause of derailment being axle or wheel failure, track defects or objects on track. Successively more cases have been added to the database, including a larger variety of causes that immediately led to derailments. The database contains events from 1980 and onwards, which have been grouped in five categories according to their primary cause:

- broken rails or other track defects - 16 events
- axle failure on inside of the wheel - 5 events
- axle failure on outside of the wheel - 4 events
- wheel defects - 4 events
- other causes (impact with objects on track, unidentified causes etc.) - 6 events

Based on the country event location, the distribution is as follows:

- Sweden - 12 events
- USA - 9 events
- UK - 6 events
- France - 4 events
- Germany - 2 events
- Canada - 1 event
- Japan - 1 event

The number of accidents and incidents collected outside Sweden is relatively limited, likely because events involving no injuries or no loss of human life, have usually not been made public.

Based on the available information, the sequence of events immediately following the derailment has been studied in an attempt to answer the following questions:

- for incidents - What stopped it turning into a catastrophe?
- for accidents - What could have changed the outcome?

Unfortunately, the amount of detailed factual information in the subsequent accident/incident reports is often proportional to the number of fatalities or the degree of property damage. This may often lead to valuable information being omitted, especially for minor incidents.
2.2 Additional cases

Since Paper A was finalized, derailment events conforming with the initial criterions still occur. Moreover, as of 1 May 2007, the European Rail Agency (ERA) made a great leap forward in the safety area by opening public access to the Public Database of Safety Documents following the European Transport Safety Council recommendations from 2001 [15].

Paper D includes a description of a derailment that occurred at Bomansberget, Sweden, in the context of validating a MBS post-derailment module. As the incident report is not yet finalized (November 2007), no empirical observations can yet be drawn as to explain the “good” post-derailment behaviour of the involved vehicles.

Paper E includes an additional derailment at Moy, UK, due to an encountered landslip at a speed of 90 km/h [31].

In the following, some additional relevant derailment cases are presented which have not been included in the augmented papers (Paper A-F). In a similar manner as in Paper A, a summary of factual information found to be relevant for the studied topic, is given. Moreover, the number of deceased or injured passengers or crew members has been deliberately omitted.

Gröbenzell, Germany

On 24 July 2007, a DB Inter-City train derailed at Gröbenzell, Germany, on the Salzburg to Frankfurt am Main line [25]. The leading driving trailer encountered some object(s) on a tangent track section and derailed with the leading bogie, see Figure 2-1. According to available information, the trainset continued in a derailed condition for a distance of approximately 300 m. It can therefore be assumed that at the time of impact, the train must have had a substantially lower speed than 200 km/h, which is mentioned in the short newspaper notice.

Figure 2-1 The derailment at Gröbenzell showing the derailed leading driving trailer
As the trainset was brought to a stop, a substantial lateral deviation from the track centre line is observed from the photos in Figure 2-1. No further information could be found regarding the presence of switches and crossings.

Judging by the photographs, favourable accidental circumstances such as a lateral deviation away from the other track, the tangent track as well as a moderate embankment slope, have probably kept this event at an incident level.

Croxton level crossing, Thetford, UK

On 12 September 2006, a two-car Class 170 DMU passenger train derailed at the Croxton level crossing on the Ely to Norwich line in UK, when travelling at a speed of 145 km/h [16]. The leading bogie of the front-end driving trailer became derailed upon striking a displaced elastomeric level crossing panel. The train continued to run in a derailed condition for approximately 415 m, stopping with the derailed leading bogie at a substantial lateral deviation from the track centre line, towards the opposite track according to Figure 2-2.

It is worth mentioning that the same type of train was involved in a derailment at Moy, UK, in 2005. In that incident, evidence suggests that lateral contact was made between parts of the automatic warning system (AWS) support bracket and the inner side of the rail. This might have limited the degree of lateral deviation to approximately 0.7 - 0.8 m. It would be interesting to investigate whether the same mechanism has limited the lateral deviation in the Croxton derailment. Presently (November 2007), no official accident investigation report has yet been released.

![Croxton level crossing derailment; a) laterally deviated leading driving trailer and b) derailed bogie seen in the direction of travel](http://andiday.fotopic.net)

Figure 2-2  Croxton level crossing derailment; a) laterally deviated leading driving trailer and b) derailed bogie seen in the direction of travel (photo: Andrew Day, available at http://andiday.fotopic.net)
Section 2 - Empirical observations

Grayrigg, UK

On 23 February 2007, a Class 390 Pendolino trainset consisting of nine vehicles derailed at a speed of 150 km/h at Grayrigg, UK, on the London to Glasgow double-track line [30]. The derailment was initiated at a facing switch, by an improperly positioned left-hand switch rail, see Figure 2-3, that was not in the open position. The train came to rest on the left-hand side of the embankment, at a maximum distance of 320 m from the point of derailment, with all vehicles derailed and some overturned, see Figure 2-4. The track geometry on site consists of a left-hand curve with radius $R = 1487$ m and cant $D = 95$ mm. Moreover, a trailing switch is located at a distance of 95 m further from the point of derailment. According to the preliminary report, it is believed that the trailing bogie of the leading vehicle and the leading bogie of the subsequent vehicle were running close to or on the other adjacent track in the early stages of the derailment. The ongoing investigation will attempt to determine the most likely path taken by the derailed vehicles.

![Figure 2-3](image)

**Figure 2-3** Grayrigg derailment; the faulty switch that initiated the derailment, seen in the direction of travel (photo: RAIB)
On Derailment-Worthiness in Rail Vehicle Design

Figure 2-4   Aerial view of the Grayrigg derailment; cars numbered according to their position in the trainset; train’s direction of travel according to the arrow (photo: RAIB)

Gnesta Station, Sweden

On 26 July 2007, an X 2000 trainset derailed at a speed of $V = 180$ km/h on the Gothenburg to Stockholm line, when passing through Gnesta station, where the train was not scheduled to stop [24]. The actual derailment cause is currently (November 2007) under investigation by the Swedish Accident Investigation Board. Just as the train started to negotiate a left-hand curve section, the leading bogie of the leading driving trailer derailed with its trailing wheelset. The derailment occurred towards the right, on the outside of the curve with radius $R = 1000$ m and cant $D = 140$ mm. Despite a cant deficiency of 245 mm in the curve, the trailing wheelset ran in a derailed condition after approximately 800 m, until stop, at a relatively constant lateral deviation of approximately 150 mm [42]. The derailed wheels of the trailing wheelset in their final positions can be seen in Figure 2-5.

Derailment of any wheelset in the trainset’s leading bogie towards the outside of a curve that leads to high unbalanced lateral track plane acceleration (high cant deficiency) is inherently a very treacherous situation. In this event, it is the author’s opinion that the minimal lateral deviation was obtained through a combination of beneficial factors. The vertical contact the left-hand bogie frame and the rail, see Figure 2-5a) could have reduced the vertical bouncing of the derailed wheelset on sleepers. Excessive dynamic movements of the trailing wheelset could, in a “worst case”, have brought about the derailment of the leading wheelset too. Moreover, the vertical contact with the railhead induced a slight concave surface on the bogie frame which may have provided a certain resistance against lateral movement. This phenomenon has previously been observed in similar situations. The relatively high bogie yaw stiffness could also have had a positive contribution to the current event.
It can be further speculated that an additional lateral deviation of the bogie, could have been prevented by lateral contact between the outer rail and the right-hand brake disc, seen in Figure 2-5b).

![](image)

**Figure 2-5** *The derailed trailing wheelset and parts of the low-reaching bogie frame at Gnesta station derailment, pictures taken in train’s direction of travel: a) left-hand side and b) right-hand side (photos: Ulf Tolérus/ Interfleet Technology Sweden)*

### 2.3 Empirically based conclusions

In **Paper A**, a correlation between vehicle design and its ability to cope with derailments has been attempted, bearing in mind the differences in accidental circumstances which make each derailment a unique event. A correct empirical assessment of the vehicle *derailment-worthiness* for a specific train design could be made once the following two conditions are fulfilled:

- a detailed factual information of the incident/accident
- a detailed knowledge on the respective vehicle design and parameters such as: features in the running gear (gear case, brake disc, transversal beam, bogie frame etc.) and their geometrical measures, inter-vehicle connections, carbody height of centre of gravity etc.

For most of the studied cases outside Sweden, the above-mentioned conditions are not completely fulfilled. Nevertheless, some distinct tendencies emerge and are presented in **Paper A**:

- Once an axle failure on the outside of the wheels occurs, observations made on the *X 2000* power car indicate that mechanical restrictions in the running gear may prevent an actual derailment to occur.
- The low-reaching bogie frame design of superior strength on the Swedish high-speed train *X 2000* and commuter train *X10* is believed to be advantageous when the vehicle comes into a derailed condition. This is derived from a couple of empirical observations where low-reaching bogie frame parts have retrieved the...
lateral guidance as well as provided a vertical support by contact with the rails. Moreover, the X 2000 bogie frame has on several occasions successfully traversed switches and crossings by, literally, breaking or bending the rails.

- Brake discs, low-reaching parts on the axle box as well other low-reaching running gear features have acted as a lateral substitute guidance mechanism, preventing further wheelset lateral deviation after derailment.

- Trains with an articulated configuration (TGV, Eurostar) have showed favourable properties even for high-speed derailments, at least for track sections with no potential aggravating elements, i.e. switches and crossings. Whether this is a result of the articulated train architecture or it is due to other properties of the articulated vehicles or related to the running gear design, it is not possible to determine with the scarce detailed information being available at this stage.

- Events involving double-decker cars, mostly from North America, indicate that these vehicles seem to be more prone to overturning.

- For a train running with derailed wheelsets, the sole factor having the most devastating potential is encountering track switches and crossings.

The additional incident/accident events presented in Paper D-E and in Section 2.2 above, strengthen some of the theories put forward in Paper A.
Section 2 - Empirical observations
3 Post-derailment vehicle simulations

In order to conduct a more systematic study on the empirically found critical vehicle design features and parameters, a thorough understanding of the dynamic behaviour of wheelsets rolling on sleepers is required.

Most of the previous post-derailment models, see Section 1.2, were developed mainly for large lateral deviations from the track centre line and are limited by simple two-dimensional constraints. For the current work, the need to predict the vehicle dynamic motion correctly, immediately after a derailment, has been identified. Accordingly, the simplistic approach to the wheel-sleeper contact cannot be utilized in the current work.

The current section summarises the necessary steps that have led to the development of a new MBS post-derailment module as well as its validation.

3.1 Methodology

3.1.1 Wheel-concrete sleeper impact: FE simulations

In order to acquire impact data of rail vehicle wheels rolling on concrete sleepers, the general-purpose FE code LS-DYNA [45] is employed, widely used for impact simulations.

As a first step, an FE model of one wheel and a representative upper part of a Swedish A9P concrete sleeper are modelled, according to Figure 3-1a. The wheel is modelled as a rigid body, with two fixed rotational DoF, roll and yaw. The concrete sleeper rests on a rigid plate permitted to translate in the vertical direction only. The rigid plate is connected to the ‘ground’ through stiffness and damper elements in parallel with characteristics corresponding to a quite stiff track. The nodes of some surfaces have translational constraints in order to capture the resistance imposed by rail fasteners and surrounding ballast.

The concrete volume is given a fine mesh consisting of 8-node solid elements and the Pseudo-Tensor material model characteristic [7]. The input parameters include the unconfined compressive strength, material density, shear modulus and Poisson’s ratio. Since concrete is such a widespread building material, many empirical relations defining its behaviour have been correlated based on the unconfined compressive strength alone [33]. Additional information on the material model and further FE simulation prerequisites can be found in Appendix A and Paper C.

The force resultants, as functions of time, are collected from large numbers of FE impact simulations for various combinations of five initial impact state parameters, according to Table 3-1 and Figure 3-1b.

The output of the FE impact simulations is collected into a look-up table for further transfer to the MBS code.
Section 3 - Post-derailment vehicle simulations

Figure 3-1  

a) Schematics of the FE wheel-concrete sleeper impact model and b) initial impact state parameters that are assumed to characterise each wheel to concrete sleeper impact sequence

Table 3-1  
Initial impact state parameters on which basis wheel-concrete sleeper resultant forces are calculated through FE simulations.

<table>
<thead>
<tr>
<th>Notation</th>
<th>Definition</th>
<th>Values</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$h_z$</td>
<td>vertical distance from the lowest point on the wheel to the upper sleeper surface</td>
<td>0, 10, 20, 28, 38 mm</td>
<td></td>
</tr>
<tr>
<td>$v_x$</td>
<td>longitudinal wheel velocity (train speed)</td>
<td>100, 200 km/h</td>
<td></td>
</tr>
<tr>
<td>$v_y$</td>
<td>lateral wheel velocity</td>
<td>-1.0, 0.0, 1.0 m/s</td>
<td></td>
</tr>
<tr>
<td>$v_z$</td>
<td>vertical wheel velocity</td>
<td>-1.0, 0.0, 1.0, 1.5, 2.5, 3.5 m/s</td>
<td></td>
</tr>
<tr>
<td>$\psi$</td>
<td>wheel(set) yaw angle relative to the sleepers</td>
<td>(-4, -2) 0, 2, 4°</td>
<td></td>
</tr>
</tbody>
</table>

3.1.2 MBS post-derailment module

The post-derailment module is implemented in the commercially available MBS software GENSYS [28]. However, the scheme can be readily adopted to any other MBS code that allows the user to build up their own mathematical functions.

A special-purpose rigid sleeper is modelled under each wheelset with only one DoF in the longitudinal direction. In GENSYS, the wheel-track interaction is based on the concept of a “moving track” model, implying that the global coordinate system of the train is stationary in the longitudinal direction. The sleeper is continuously translating, opposite to the train’s direction of travel, with the speed of the train relative to the local coordinate system of the wheelset, between $-s_d/2$ and $+s_d/2$, where $s_d$ is the assumed...
sleeper spacing. The sleeper’s upper surface profile is captured in the code as a function of the wheel’s lateral position.

Once a wheel to concrete sleeper impact condition is triggered, see Paper C for details, the code registers the initial impact state parameters, which act as input points to the look-up table previously generated through FE simulations. Thereafter, valid force resultants are calculated in the MBS code through a multi-dimensional linear interpolation algorithm, and applied to the wheel.

The model can also take into consideration possible impact with rail fasteners of Pandrol type that are oriented, relative to the train’s direction of travel, such that the wheel first contacts the front arch of the clip, see Figure 3-2.

![Figure 3-2 Pandrol rail fastener; wheel impact direction and component terminology](image)

The fasteners are incorporated in a special-purpose sleeper that is modelled under each wheelset, with three DoF (longitudinal, vertical and roll). The sleeper is connected to the ‘ground’ by a spring and damper in parallel, with values corresponding to the chosen track flexibility. The ‘ground’ is fixed in all other directions but allowed to translate longitudinally together with the fastener/sleeper body at the speed of the train relative to the wheelset’s local coordinate system. The fasteners upper shape is captured in the code as a function of the wheel’s lateral position.

Once an impact condition is triggered, see further details in Paper D, the wheel and the fastener body become coupled by a spring of piecewise linear stiffness that allows plastic deformation and failure of the centre leg, see Figure 3-2. The stiffness property has been calibrated to match the rail fastener damage in four consecutive sleepers after an authentic Swedish event with results from the MBS post-derailment module simulations.

Briefly, the MBS post-derailment module is valid under the following premises:

- derailment on ballasted track with equally spaced undamaged concrete sleepers of constant properties
- constant post-derailment train speed (no applied braking)
- wheel to ballast contact is not considered
Section 3 - Post-derailment vehicle simulations

- impact with rail fastening system of Pandrol type; currently valid only for situations where the fastening system orientation and the train’s direction of travel coincide in such a manner that the front arch of the clip is pushed out of the centre leg upon impact
- additional impact with other infrastructure parts such as switches and crossings, signalling devices etc. are not considered.

3.2 Validation stages

The FE wheel-sleeper impact model and the MBS post-derailment model have been validated in the sequence summarised here.

3.2.1 Phase I (FE model => Authentic derailment)

Relevant information found in a Swedish accident report from 1980, Upplands Väsby [34], facilitated a first validation of the proposed FE wheel-concrete sleeper impact model. The objectives of the current validation procedure is to compare the authentic sleeper indentation mark depth made by the right-hand wheel of the derailing wheelset, see Figure 3-3, with the results from a continuous impacting sequence predicted by a FE model, according to Figure 3-4.

Figure 3-3 Derailment marks at Upplands Väsby 1980; arrows indicate the first contact points of the derailing wheelset with sleepers. Left-hand frame shows in more detail damage from the right-hand wheel for the first three consecutive sleepers
The FE validation model consists of two lumped masses representing one eighth of the carbody and a quarter of the bogie with vertical and longitudinal DoF. The masses are linked with each other through two, vertical and longitudinal, linear stiffnesses and one vertical linear damper, all representing the secondary suspension. The bogie is connected with the centre of the wheel by one vertical and longitudinal linear stiffness and damper. The values given to the primary and secondary suspension elements are consistent with nominal values found for that type of passenger car. The wheel is modelled as a rigid body with two fixed rotational degrees of freedom, roll and yaw. Furthermore, the wheel profile is unworn and the inertial properties correspond to half of the wheelset’s mass. The twin-block type of concrete sleepers are modelled in a similar manner as described in Section 3.1.1.

Figure 3-4  Schematics of FE wheel to concrete sleeper impact validation model

Detailed analysis of the accident report in combination with a special MBS derailment simulation with nominal parameters consistent with the involved vehicle type, indicated initial values for a number of critical FE model input parameters. A detailed description on these matters can be found in Paper C.

Figure 3-5  Indentation depth obtained from FE simulations for various sleeper-ground flexibility values and with initial impact state parameters found as most plausible
Section 3 - Post-derailment vehicle simulations

The FE results in terms of the concrete indentation depth for the three subsequent sleepers with initial impact state parameters judged as most plausible, are shown in Figure 3-5. As the sleeper-ground properties remain uncertain, three formulated track flexibility parameters ‘soft’, ‘standard’ and ‘stiff’ are included in the diagram. The obtained values are in good correspondence with the estimated indentation depth ranges, deduced from the accident photo, which are shown in Figure 3-5.

3.2.2 Phase II (FE model => MBS model)

The second validation stage aimed at testing the ability of the MBS post-derailment module to correctly predict the wheel to concrete sleeper impact forces based on the look-up table, previously generated through FE simulations.

A rail vehicle wheel linked to a simplified suspension system consisting of two lumped masses, bogie frame and carbody, are implemented in both FE and MBS models. Additionally, 24 concrete sleepers are modelled for the FE simulations. The models receive identical vehicle and track flexibility parameters as well as initial starting parameters. Further validation methodology information can be found in Paper C.

The wheel’s trajectory generated by the FE simulations is compared with its MBS counterpart, for 24 sleepers, approximately 14.3 m at a sleeper spacing distance of 0.65 m. The vertical and lateral positions of the wheel as a function of its longitudinal location above the track sleepers are presented in Figure 3-6. Due to an initial yaw angle, the wheel also starts to deviate laterally.

![Figure 3-6](image)

*Figure 3-6 Vertical and lateral wheel trajectory over 24 sleepers, 14.3 m, generated by FE and MBS simulations for the same set of arbitrary starting values*

The curves generated by the MBS code seem to correlate well with the FE solution in terms of the wheel’s trajectory, both in the vertical and lateral directions. This implies that the linear interpolation methodology based on the data acquisition procedure described in Section 3.1.1 and the MBS post-derailment module code described in Section 3.1.2, are in good agreement with the FE impact simulations as well as real behaviour.
### 3.2.3 Phase III (MBS model => Authentic derailments)

A Swedish passenger train derailment event from 2006 at Bomansberget, permitted the third validation step to be reached, see further information in Paper D.

Due to a wheel rim failure, the leading wheelset of the trailing bogie of an intermediate passenger car, derailed at a speed $V = 130$ km/h.

The damage produced by the derailing left-hand wheel on the first ten consecutive sleeper and fastener components is shown in Figure 3-7. The right-hand wheel of the same wheelset, with greatly reduced diameter due to failure, had limited interaction with parts of the track for the first ten sleepers.

On-site concrete indentation measurements and fastener damage documentation, permits an assessment of the wheel’s vertical and lateral position relative to the sleepers, for most of the sleepers, see Figure 3-8. In the vertical direction, neither concrete nor fastener centre leg damage can be observed for sleeper #6 and 7. Nevertheless, a plausible wheel trajectory can still be deduced: the tread did not pass below the centre leg upper level and not above the clip level, as both clips are out of place. Similarly, an exact lateral location of the flange can not be indicated above sleeper #5, 6 and 7.

A vehicle with nominal parameters consistent with the involved passenger car at Bomansberget was modelled in the MBS software GENSYS. Based on detailed factual circumstances information for this event [19], MBS validation simulation could be performed.

![Figure 3-7](image.png)

*Figure 3-7  Damage to the first ten sleepers from the point of derailment made by the left-hand wheel at Bomansberget; (the right-hand wheel had limited interaction with the track due to damage) (Photos: Dan Brabie)*
Initially, the observed damage to the first four sleepers served as a calibration tool for the stiffness characteristics describing the wheel tread to fastener impact. However, the MBS time-domain simulation was continued, by registering the wheel’s motion up to the tenth sleeper. This is shown in Figure 3-8, concluding that the wheel’s vertical and lateral trajectory matches successfully the positions deduced from measurements and observations.

A second MBS validation in terms of the post-derailment vehicle dynamics was performed, based on an authentic passenger car derailment that occurred at Tierp, Sweden in 2001 [36]. In this event, an axle journal failure on a trailing wheelset derailed the leading wheelset on an X 2000 power car, travelling at a speed of 200 km/h. The derailed leading wheelset deviated slightly towards the inner side in a curve, at a cant deficiency of approximately 150 mm, and continued to roll on the Pandrol rail fasteners and concrete sleepers until stop. Similarities with the Bomansberget incident in terms of the rail fastening type and orientation relative to the wheel’s longitudinal impact direction, imply that the proposed MBS post-derailment module can readily be applied.

Just as in the authentic case, the MBS simulation correctly predicts that the trailing wheelset continues to remain on the rails, even as the leading wheelset rolls and bounces on rail fasteners and sleepers. Moreover, a 30% increase of the stiffness that describes the wheel to fastener impact, actually leads also to a derailment of the trailing wheelset. Further information can be found in Paper D.
4 Minimizing consequences of axle journal failure

Although not the most frequent derailment cause in the incident/accident database, see Section 2 and Paper A, axle failure at the journal bearing, i.e. on the outside of the wheel, possesses an imminent danger to vehicle safety. The current section summarizes this issue, which is covered extensively in Paper B.

4.1 MBS model of an axle journal failure

A technical report [41], supported by on-train observations, indicated that mechanical restrictions limiting the wheelset’s movement relative to its bogie frame might be beneficial in cases of axle failure on the outside of the wheels, at the journal bearing. In order to assess the feasibility of implementing such mechanical restrictions in a bogie, an axle journal failure model has been developed and implemented in the MBS software GENSYS as well as validated against two authentic events.

4.1.1 Methodology

The actual failure is modelled by removing the longitudinal and vertical primary suspension elements at the involved axle side, for an arbitrarily specified location along the track. The lateral stiffness is maintained as long as the wheelset is pushed towards the fractured axle journal. In addition, linear longitudinal and vertical viscous damper elements are activated at the interface of the failed axle to approximate possible contact between the broken side of the axle journal with the wheelset.

Once the axle journal has failed, the surface of the rotating axle may come into contact with stationary parts of the bogie frame. In the computer model, this is approximately accounted for by applying a set of longitudinal and vertical force vectors to both the wheelset and bogie frame. The magnitude of these force vectors are calculated by registering the simulated forces emerging during contact, and scaled by an arbitrarily chosen friction coefficient.

4.1.2 Validation

Two Swedish incidents at Tierp [36] and Gnesta [35] from 2001, both affecting the X 2000 power car, were chosen for the purpose of validating the axle failure computer model.

At Tierp, the leading wheelset in the leading bogie derailed over the low rail in a circular curve section with cant $D = 110$ mm (cant deficiency 153 mm) at a speed $V = 200$ km/h. This occurred as a result of an axle journal failure on the trailing wheelset in the same bogie, on the low side relative to the curve.
Section 4 - Minimizing consequences of axle journal failure

In Gnesta, no derailment occurred after a similar failure on the leading wheelset in the trailing bogie, despite negotiating a series of left and right-hand curves, an ‘S-curve’, with cant $D = 140$ mm (cant deficiency 243 mm) at a speed $V = 180$ km/h.

The MBS axle failure model is validated by comparing the simulated tendency of derailment resulting from combinations of various conditional, i.e. unknown, parameters with the authentic sequence of events. Detailed factual information found for these events enabled to narrow relatively well the range of conditional parameters, which include: the exact axle failure location along the track as well as wheel-rail and axle-bogie frame friction coefficients. An earlier study indicated that the damping coefficient describing the possible contact between the failed axle and its broken side, had a marginal effect on the derailment tendency [5]. Accordingly, this parameter is excluded from the list of conditional parameters.

The results are presented as derailment maps in Figure 4-1, as a function of a broad range of vertical and longitudinal play, i.e. mechanical restrictions, between the axle and the bogie frame. For the Tierp case, the derailment map indicates the lowest combination of mechanical restrictions (out of all combinations of conditional parameters) for which the leading or trailing wheelset of the affected bogie derails first. For the Gnesta case, the results are presented in a similar manner, however, since no derailment occurred in reality, no distinction is made as which wheelset derails first. The figures also indicate the plausible range of mechanical restrictions found in an actual X 2000 power car bogie. The derailment maps in Figure 4-1 are based on a set of 540 and 720 computer simulations for the Tierp and Gnesta validation cases, respectively.

For the Tierp case, the computer simulations clearly show that for certain conditional parameters in the plausible range of mechanical restriction, the leading wheelset of the vehicle derails towards the low (inner) rail in the curve, consistent with the authentic case. Furthermore, these simulations successfully predict that the leading wheelset leaves the rails within a few metres from entrance in the circular curve section, in
On Derailment-Worthiness in Rail Vehicle Design

accordance with authentic observations from the first damaged sleeper. At the same time, certain combinations of conditional parameters and mechanical restrictions may also lead to a derailment of the trailing wheelset, unlike the sequence of events of the authentic case. This is specially for cases involving a higher axle to bogie frame friction coefficient, where the trailing wheelset derailment line passes through the plausible range of mechanical restrictions. For other combinations of conditional parameters the simulations agree with authentic observations.

For the Gnesta case, the results agree with the outcome of the authentic case, where no derailment occurred for most of the conditional parameter combinations and plausible mechanical restrictions. However, certain conditional parameter combinations, may also lead to a derailment.

4.2 Mechanical restrictions

A parameter analysis is performed for alternative axle failure locations in a bogie, in order to test the feasibility of minimizing the derailment tendency after such events. In this respect, axle journal failures are initiated on each of the four axle journals of the leading bogie. The vehicle models used have general characteristics resembling the X 2000 power and passenger trailer cars. The analysis includes various conditional parameters such as: cant deficiency in the curve \( D = 110 \) and \( 140 \) mm at a speed of \( V = 200 \) and \( 180 \) km/h, respectively), track irregularities, friction coefficients wheel to rail and axle to bogie frame.

The results are presented in Figure 4-2 as derailment maps for the two vehicle types. Each line corresponds to the location of the axle failure, denoted relative to the closest wheel position in the bogie and in the curve. The lines indicate the lowest combination of mechanical restrictions at which a derailment occurs, out of all tested conditional parameters. Each line in the diagrams corresponds to a set containing 600 MBS simulations.

![Figure 4-2 Derailment maps for an X 2000 power and trailer car for different axle failure locations; the lowest combination of mechanical restrictions which leads to a derailment](image-url)
The simulation results indicate clearly that mechanical restrictions may prevent a derailment to occur after an axle journal failure, even on curved track with high cant deficiency. For this to happen, relatively tight vertical and longitudinal restrictions need to be accommodated for in the bogie frame. Further detailed information, as well as a parameter analysis containing different wheelset guidance stiffnesses are presented in Paper B.
5 Minimizing derailment consequences

5.1 Derailment scenarios under consideration

For derailment-worthiness analyses of various rail vehicle design features and parameters, the following derailment scenarios are considered and modelled in the MBS software GENSYS, see Table 5-1:

- **Axle journal failure (AJF):** Previously, the MBS model of an axle journal failure has been successfully validated based on two authentic passenger car incidents, see Section 4 and Paper B. The failure of the trailing wheelset journal (inner wheel), affects the leading bogie of a passenger car. As a result, the leading wheelset starts to deviate laterally on the high rail of a circular curve section, following flange climbing.

- **‘Wheel flange on rail’ (WFOR):** Such a hypothetical condition is initiated on the leading wheelset of a rail vehicle by an initial lateral displacement of 80 mm in combination with a yaw angle of 1.58°. The leading bogie and carbody midpoints are also displaced laterally 40 and 20 mm, with yaw angles of 1.58 and 0.13°, respectively. No defects, neither to the vehicle nor to the track, are considered. This derailment scenario would correspond to a sequence of events subsequent to the vehicle encountering relatively small objects on the track, as well as other possible running gear or track malfunctions.

- **High rail failure (HRF):** Failure of the high (outer) rail in a curve is automatically a treacherous situation as the wheelsets lose all lateral guidance. The failure is modelled by removing subsequently the lateral and vertical wheel-rail contact elements once the train reaches the start of a circular curve section.

<table>
<thead>
<tr>
<th>#</th>
<th>Abbreviation</th>
<th>Derailment cause</th>
<th>Speed (km/h)</th>
<th>Curve radius (m)</th>
<th>Cant (mm)</th>
<th>Cant deficiency (mm)</th>
<th>Lateral track-plane acceleration (m/s²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>AJF-16</td>
<td>Axle journal failure in a circular curve section</td>
<td>200</td>
<td>1200</td>
<td>150</td>
<td>245</td>
<td>1.6</td>
</tr>
<tr>
<td>2</td>
<td>AJF-10</td>
<td></td>
<td>100</td>
<td>300</td>
<td>30</td>
<td>153</td>
<td>1.0</td>
</tr>
<tr>
<td>3</td>
<td>WFOR-10</td>
<td>Wheel flange on rail (impact with objects on track, etc.)</td>
<td>200</td>
<td>2120</td>
<td>70</td>
<td>153</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>WFOR-00</td>
<td></td>
<td>100</td>
<td>530</td>
<td>0</td>
<td>0</td>
<td>0.0</td>
</tr>
<tr>
<td>5</td>
<td>HRF-16</td>
<td>High rail failure in a circular curve section</td>
<td>200</td>
<td>1200</td>
<td>150</td>
<td>245</td>
<td>1.6</td>
</tr>
<tr>
<td>6</td>
<td>HRF-10</td>
<td></td>
<td>100</td>
<td>300</td>
<td>30</td>
<td>153</td>
<td>1.0</td>
</tr>
</tbody>
</table>

Table 5-1 Overview of derailment simulation scenarios used in conjunction with vehicle derailment-worthiness studies.
5.2 General vehicle modelling considerations

The rail vehicle formations employed in derailment simulations of the current section, as well as in Paper E-F, consist of three conventionally coupled and one articulated vehicle configurations according to Table 5-2.

Table 5-2  Train configurations employed in the MBS derailment simulations for vehicle derailment-worthiness studies.

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Train configuration</th>
<th>First derailing wheelset</th>
</tr>
</thead>
<tbody>
<tr>
<td>C1</td>
<td>Power car + intermediate passenger car</td>
<td>Wheelset 1 in intermediate passenger car</td>
</tr>
<tr>
<td>C2</td>
<td>Driving trailer + intermediate passenger car</td>
<td>Wheelset 1 in driving trailer</td>
</tr>
<tr>
<td>C3</td>
<td>Three intermediate passenger cars</td>
<td>Wheelset 1 in second intermediate passenger car</td>
</tr>
<tr>
<td>A1</td>
<td>Power car + semi-articulated passenger car + 2 articulated passenger cars + 1 semi-articulated passenger car, see further Figure 5-10</td>
<td>Wheelset 1 of the second median bogie (fifth bogie in the trainset), see further Figure 5-10</td>
</tr>
</tbody>
</table>

In the conventional design (C1 to C3 in Table 5-2), each vehicle consists of one carbody, two bogie frames and four wheelsets. In the articulated design (A1 in Table 5-2), adjacent carbody ends share the same bogie.

The axle loads for the power, trailer and intermediate passenger cars in configuration C1-3 are 180, 148 and 131 kN, respectively. The axle load of the median bogie in the articulated section of the train configuration A1 is 169 kN.

All parts are modelled as rigid bodies with six degrees of freedom each. The primary and secondary suspension is modelled by linear and non-linear springs and dampers. Furthermore, gaps between various running gear elements or between running gear and carbody are introduced as “semi-flexible” stops. These mechanical stops are redundant under normal operational conditions, but may play an important role in some extreme conditions, such as derailments. Each bogie includes a roll bar to produce a linear roll stiffness between bogie and carbody, as well as two non-linear yaw dampers acting primarily in the longitudinal direction between carbody and bogie frames.

The track flexibility model is based on the concept of a so-called “moving track piece” which follows under each wheelset and incorporates two distinct rigid bodies: two rails (UIC60) rigidly connected to a track piece, and a fixed ‘ground’. The track piece is connected in the lateral and vertical directions with the ground by linear springs and dampers. The wheel-rail contact is described by a linearized stiffness in parallel with a linear viscous damper. The wheels are allowed to lift from the rail head.
5.3 Conventional train configuration

5.3.1 Lateral substitute guidance mechanisms

The concept of utilizing substitute guidance mechanisms is not new. For instance, on the Swedish network it is common practice to install two additional rails in-between the running rails, on track sections where a lateral deviation following an accident would have disastrous effects. Such enhanced passive safety aspects can currently be found on viaducts, bridges and along a certain distance prior to a tunnel opening as well as in the tunnel. The concept is that, in a derailed condition, the lateral deviation of the wheelset would be prevented by means of these additional rails.

The guidance mechanisms studied in this thesis are connected to the vehicle. They could be divided into two groups, depending on to which vehicle part they are attached:

- on the sprung mass, as low-reaching bogie frame parts, see Figure 5-1a.
- on the unsprung mass, as low-reaching brake discs or axle journal boxes, see Figure 5-2a.

For both groups, their intended purpose is to guide laterally and stabilise a derailed running gear by simply engaging with the appropriate running rail. This principle is shown in Figure 5-1b for a bogie frame and Figure 5-2b for a brake disc and an axle journal box.

In order to initiate as well as to maintain a successful lateral guidance, geometrical and strength requirements need to be fulfilled. In addition, mechanisms should be able to cope with track discontinuities, i.e. traversing switches and crossings, in a derailed condition so that a further aggravating situation is avoided.

In this thesis (Paper E), the study has been mainly focused on the geometrical feasibility requirements. In this respect, the mechanism should be positioned sufficiently low relative to top of the rails (ToR) as to overcome the vertical dynamic movements induced by the derailed running gear in combination with a sufficient lateral gap to accommodate the width of the rail. Moreover, the positioning of the guidance mechanisms should not interfere with applicable gauging standards for low-reaching parts.

![Figure 5-1](image)  Guidance mechanism attached to the sprung mass:

a) geometrical feasibility parameters under study
b) the intended sequence of events in derailed condition
Section 5 - Minimizing derailment consequences

Some geometrical feasibility results for vehicles running at a speed of 200 km/h are presented in Figure 5-3, 5-4 and 5-5, for a low-reaching bogie frame, brake disc and axle journal box, respectively. Each line corresponds to a different derailment scenario and indicates the highest vertical distance \( h \), \( H \) or \( H_{\text{ajb}} \) for different lateral gaps \( b \), \( B \) or \( B_{\text{ajb}} \) that leads to a successful engagement with the rail. Furthermore, the lines actually indicate an averaged successful vertical distance which is computed among the results of an additional simulation set containing four different initial sleeper locations relative to the first derailing wheelset. Accordingly, a sensitivity analysis is also performed for one of the parameters that seriously affects the wheel’s vertical motion on sleepers, namely the vertical distance between the wheel’s lowest point and the sleepers upper surface \( h_z \) in Table 3-1), at the initial impact with a sleeper.

Consequently, guidance mechanisms with geometrical parameter combinations located on or on the right-hand side of the line, would stop or stabilise the lateral deviation of the bogie as an average. Accordingly, some specific geometrical combinations located on the right-hand side of the lines might also lead to a non-guiding condition. In this context it is worth recalling that the MBS simulations are performed for certain ‘worst-case’ conditions, such as an excessive rail wear and new wheels, among others. It can therefore be assumed that a successful guidance condition is obtained for most of the cases of geometrical combinations found in the proximity of the lines.

The diagrams show the approximately lowest possible vertical distance according to the Swedish and the more restricting interoperable European gauging standards. It is assumed that all parts of the wheelset (except the wheels themselves) must comply with gauging requirements for low-reaching parts, even if the wheelset is lowered 30 mm due to wheel wear. It is also assumed that all parts mounted on the bogie frame must nominally be located a further 30 mm higher due to possible vertical motions in the primary suspension.

Further information on the MBS simulation prerequisites as well as results for vehicles derailing at 100 km/h are further presented in Paper E.
Figure 5-3  Low-reaching bogie frame geometrical feasibility results for derailments at 200 km/h; scenario WFOR-00 is on tangent track, all others are on curved track; see Table 5-1 and 5-2

Figure 5-4  Low-reaching brake disc geometrical feasibility results for derailments at 200 km/h; scenario WFOR-00 is on tangent track, all others are on curved track; see Table 5-1 and 5-2
Section 5 - Minimizing derailment consequences

5.3.2 Vehicle coupler restrictions

The current sub-section studies the feasibility of restraining the maximum coupler yaw angle of a centre coupler, i.e. maximum inter-carbody lateral deflection, in order to minimise vehicle lateral deviation after derailment. In particular, the possibility of maintaining a limited bogie lateral deviation is investigated, so that wheels can continue to roll on the relatively ‘safe’ sleeper area, and not fall into the ballast. Such lateral coupler restrictions can only be feasible if normal running operation of the trainset in tight horizontal curves is not adversely affected.

In conjunction with these studies, the additional effects of the following vehicle design parameters are quantified:

- vertical position of a transversal beam in the bogie relative to ToR
- bogie yaw stiffness
- wheelset guidance stiffness

A standard (‘Std’) trailer car centre coupler configuration that permits a maximum yaw rotation at the pivot centre of ±13°, is assumed. Three additional configurations are modelled and labelled ‘A’, ‘B’ and ‘C’, which limit the maximum inter-carbody lateral deflection, \( y_{pc-max} \), by approximately 59 to 64%, relative to the assumed standard configuration, see Figure 5-6.
In the MBS computer simulation model, the semi-permanent centre coupler is modelled as a rigid beam, and consequently, carbody-coupler separation can not occur. For modern semi-permanent couplers involved in the derailment scenarios considered here, this assumption is considered viable, although not further investigated in the present study.

An important feature that might influence the post-derailment sequence of events is the transversal beam attached onto bogie frames. Depending on its vertical position in the bogie frame, the transversal beam may come into contact with the rail either immediately after a derailment or once the wheels lose the vertical support of sleepers. In the MBS vehicle model, two transversal beams are modelled for each bogie, symmetrically located at a longitudinal distance from the bogie centre of 0.7 m, see Figure 5-7.

Despite zero cant deficiency in the tangent track derailment scenario ‘WFOR-00’(#7 in Table 5-1), the derailing leading wheelset, and subsequently even the trailing wheelset,
Section 5 - Minimizing derailment consequences

continue to deviate laterally until the carbody reaches the limits imposed by the coupler. Once the wheel(s) on one side of the bogie leaves the sleeper surface, the bogie transversal beam may act as a substitute vertical support mechanism, preventing further sinking of the wheels into the ballast.

Figure 5-8 Example of max. lateral displacement of the derailing bogie on tangent track (WFOR-00 #7, see Table 5-1) and train configuration C3 (see Table 5-2);
   a) leading wheelset
   b) trailing wheelset

For a flexible wheelset guidance, the maximum registered lateral displacement from the track centre line of leading and trailing wheelset for a bogie transversal beam vertical position of \( h_{tb} = 170 \text{ mm} \) is shown in Figure 5-8 as a function of coupler restriction configurations. A sensitivity analysis of the bogie yaw stiffness is also included, assuming a standard (‘std’) value of 650 kNm/rad.

All values above the horizontal line (sleeper edge) imply that the wheel’s flange has left the sleepers’ surface, corresponding to a sleeper length of 2.53 m. This occurs with both wheelsets with the coupler configuration ‘Std’. However, the wheels continue to roll on sleepers for almost all the other tested restrained coupler configurations ‘A’, ‘B’ and ‘C’. An increased bogie yaw stiffness tends to further limit the lateral displacement of the derailed wheelsets. However, these changes are not significant. In this context, it is worth mentioning that shorter sleepers might also be used on ballasted tracks. Such situations have not been considered in the current study.

For derailments on a curved track (cant deficiency 153 mm) and due to an axle journal failure (#3 in Table 5-1), the restrained couplers can not keep the wheel of the leading wheelset on the sleeper surface. For the most restrained coupler configuration, ‘C’, and flexible wheelset guidance, the maximum lateral displacement of the leading wheelset can be found in the range of 0.60-0.66 m, depending on the choice of \( h_{tb} \) and bogie yaw stiffness values. Additional results are presented further in Paper F.

Furthermore, the feasibility of restraining the couplers as intended is addressed, in terms of permissible lateral track shift forces and derailment criteria according to UIC 518 [44] and EN 14363 [10]. In this respect, computer simulations are preformed for a “worst-
case” track geometry consisting of a so-called ‘S-curve’ with two circular sections with radii $R_S = 190$ m in opposite directions and with no intermediate section. According to the results, all the tested coupler restrictions are feasible in terms of maximum track shift forces and derailment ratio for vehicles equipped with a flexible wheelset guidance. For vehicle equipped with a stiff wheelset guidance, only coupler configuration ‘A’ should be employed.

5.3.3 Height of centre of gravity and other vehicle design features

In this section, a parameter analysis is performed with focus on the vehicle overturning tendency in relationship to the following:

- carbody height of centre of gravity (CoG)
- coupler height
- vertical position of the transversal beam in the bogie relative to ToR
- maximum coupler yaw angle

The need for such studies has emerged primarily from empirical observations involving North American double-decker cars that would appear more predisposed to overturn after derailments, see further Paper A and Section 2.

The following standard values are assumed: carbody height of CoG $h_{ccg} = 1.61$ m and coupler height $h_{cpl} = 1.025$ m. The results are valid for the standard coupler configuration, ‘Std’, that permits a maximum yaw rotation at the pivot centre of $\pm 13^\circ$, corresponding to a maximum lateral deflection between two adjacent carbodies of approximately 0.64 m.

Figure 5-9 Maximum carbody roll angle as a function of coupler height and carbody height of CoG for:

- a) $h_{tb} = 170$ mm
- b) $h_{tb} = 370$ mm

The maximum carbody roll angle, measured relative to the track plane and towards the outside of the track, after a derailment on tangent track ‘WFOR-00’, (#7 in Table 5-2), is
presented in Figure 5-9, for two vertical height values of the transversal beam in the bogie.

The vertical location of the transversal beam in the bogie, \( h_{tb} \), has a significant influence on the overturning tendency, particularly for cases of high carbody height of CoG in combination with a low coupler height. Once the vehicle’s lateral deviation starts to be obstructed by the coupler, a roll moment is generated on the carbody towards the outside of the track. For cases of relatively large vertical distances between the carbody height of CoG and the coupler, the increased rolling moment overturns the carbody.

Paper F presents additional results obtained after derailments on a curve due to axle journal failure (#3 in Table 5-1) as well as with an additional coupler configuration that allows a larger maximum coupler yaw angle, ‘Extended’ in Figure 5-6.

### 5.4 Articulated train configuration

A trainset consisting of five carbodies with an articulated configuration according to Figure 5-10 is modelled in the MBS software GENSYS. The articulated configuration is based on the “3-point” principle, implying that each median bogie supports one carbody end, i.e. the “supported end”, which in turn carries the adjacent carbody, i.e. the “carrying end”. Accordingly, each carbody in the articulated section of the train is supported by three points, two at the “carrying end” and one at the “supported end”. For the trainset modelled here, the choice of support points can be seen in Figure 5-10.

The carbodies of the articulated configuration are inter-connected by one anti-roll damper acting primarily in the lateral direction and positioned in the middle of the upper corners of the carbodies. Additionally, four longitudinal dampers link the two adjacent carbodies and they are located at each carbody corner. Further information regarding the articulated train design can be found in Paper F.

![Figure 5-10 Sketch of the modelled articulated trainset (configuration A1 in Table 5-2)](image)

For the articulated trainset, a standard carbody height of CoG, \( h_{ccg} \), of 1.61 m has been set. A standard inter-carbody damper is assumed and labelled ‘Std’ with characteristics.
according to Figure 5-11. The lateral deviation and carbody roll after derailments of a train with articulated configuration are studied as functions of the above-named inter-carbody longitudinal dampers characteristics and the following two running gear parameters:

- vertical position of the transversal beam in the bogie relative to ToR
- wheelset guidance stiffness.

Two derailment scenarios are utilized (#3 and 7 in Table 5-1) at a speed of 200 km/h, and both affect the fifth bogie in the trainset, under Cars 3 and 4 in Figure 5-10.

The maximum lateral displacements of the leading and trailing wheelsets of the median bogie are presented in Figure 5-12, for three transversal beam vertical positions. The results correspond to a derailment scenario on tangent track (#7, WFOR-00 in Table 5-1). Stronger inter-vehicle longitudinal dampers than the assumed standard configuration, provide a sufficient resistance against adjacent carbodies’ relative yaw motion, that keep the bogie rolling on the sleepers’ surface at a substantial safety margin from the sleeper edge.

For the case corresponding to the ‘-100%’ damper characteristic, i.e. no dampers are present, the derailing median bogie is relatively “free” to deviate laterally, which leads to overturning.

The maximum carbody roll angle towards the outside of the track of the third vehicle in the trainset, ‘Car 3’, is presented in Figure 5-13, for the same derailment scenario. In case that no dampers are present (‘-100%’), the carbody overturns, as mentioned above, irrespective of the chosen \( h_{th} \) values. For such cases, the derailing median bogie exhibits a considerable roll angle once the wheels leave the sleeper area. This has a direct negative effect on ‘Car 3’ as its carbody is only attached at one support point at the front-end, i.e. the “supported end”. In order to mitigate such relatively large carbody roll angles, certain mechanical restrictions could possibly be incorporated at the bogie frame to carbody interface, at the “supported end”. In the current articulated train model, such restrictions have not been considered.
Section 5 - Minimizing derailment consequences

Derailment scenario: #7, WFOR-00 (tangent track), Train config. A1

Figure 5-12  Example of max. lateral displacement of the derailing bogie on tangent track (WFOR-00 #7, see Table 5-1), train configuration A1 (see Table 5-2) and different height $h_{tb}$ of bogie transversal beam
a) derailing leading wheelset
b) derailing trailing wheelset

Figure 5-13  Example of max. carbody roll angle on third vehicle in the articulated trainset as a function of inter-carbody longitudinal damper configurations and different height $h_{tb}$ of bogie transversal beam

For stronger inter-carbody damper configurations than the assumed ‘Std’ value, the carbody roll angle is kept to a minimum as the wheels of the derailing bogie remain on sleepers, see Figure 5-12.

On a curved track (derailment scenario #3 in Table 5-1), none of the tested inter-vehicle damper configurations can maintain the wheels on the sleepers’ surface. The maximum lateral deviation is found on the trailing wheelset, which in the ‘+150%’ case, reaches 0.65-0.86 m, depending on the chosen $h_{tb}$ value. With damper configurations
corresponding to ‘+50%’ and above, the train does not turn over in any of the simulated cases.

It should be emphasised further that the employed MBS post-derailment module does not take into account the possible interaction occurring between ballast and parts of the running gear. In the conventional configuration, it is believed that this limitation has minor implications for the studied derailment scenarios, as the coupler provides most of the vertical support once wheels deviate outside the sleeper area. Due to the nature of the articulated configuration, the derailing median bogie sinks deeper into the ballast. For such situations, the ballast might, however, provide some vertical support. In this respect, the maximum carbody roll angle results might be overestimated. The general trend, however, should still be valid.

Additional simulations for the derailment scenario on a curve due to axle journal failure (#3 in Table 5-1), as well as a feasibility analysis of the inter-carbody longitudinal dampers in terms of lateral track forces and derailment ratio are further presented in Paper F.
6 Discussion and Conclusions

The empirical study including about 40 accidents and incidents caused by mechanical failures as well as other causes that ultimately lead to derailment, has indicated the following design features able to affect the vehicle *derailment-worthiness*:

- Mechanical restrictions in the running gear
- Low-reaching running gear features
- Vehicle inter-connections
- Vehicle height of centre of gravity
- Articulated train architecture
- Adequate strength of running gear steering parts, e.g. to cope with track switches
- Bogie yaw stiffness

Quantifying the potentially beneficial effects of the above-mentioned vehicle design features and parameters is necessary. In this respect, analyses based on computer simulations are employed.

In order to perform a systematic investigation of the role of mechanical restrictions between a wheelset and the bogie frame after axle journal failure events, an MBS pre-derailment model was developed to account for such sequence of events. The proposed MBS model was successfully validated for two authentic cases of axle journal failure. An axle failure on the leading wheelset, inner side in a curve, appears to be the most sensitive location in a bogie for both *X 2000* power and trailing cars. At such a location, a vertical mechanical restriction just below 50 mm for a power car and 60 mm for an intermediate trailer car in combination with a 10 mm longitudinal restriction is considered sufficient to avoid a derailment. The longitudinal mechanical restriction requirement implies a maximum wheelset to bogie frame yaw angle of 1.1° at static conditions.

In order to continue the analysis of some potential *derailment-worthiness* features identified in the initial empirical study, an MBS post-derailment module was developed to predict the dynamic behaviour of vehicles rolling on sleepers.

In a derailed condition, the forces exerted on the rolling wheels on concrete sleepers are calculated through finite element (FE) analysis of a wheel impacting a concrete sleeper for combinations of initial impact state parameters (approximately 540 FE simulations). The results are stored in a look-up table and then transferred to the MBS model. A mathematical MBS code detects wheel to concrete sleeper or wheel to rail fastener contact and generates compatible impact forces as a function of the initial impact state condition.

The MBS post-derailment module has been validated in the following phases:

- *Phase 1*: An agreement was obtained between the FE wheel-sleeper impact model and an authentic passenger car derailment case in terms of the wheel's indentation in concrete.
Section 6 - Discussion and Conclusions

• **Phase 2**: A successful agreement of the vertical and lateral movements of one wheel rolling over 24 consecutive sleepers in MBS compared with its FE model counterpart

• **Phase 3**: Successful agreement of sleeper impact (indentation) and lateral deviation between a recent authentic passenger car derailment and the outcome of the MBS computer simulation model, including rail fasteners.

The possibility of utilising substitute guidance mechanisms, such as low-reaching bogie frames or axle boxes, or axle mounted brake discs, was investigated as means of minimizing the lateral deviation after derailments. Geometrical feasibility parameters were examined in relationship to the Swedish and interoperable European gauging standards. Successful lateral engagements with the rail were detected after various derailment scenarios on tangent and curved track. The numerical results obtained, support the beneficial effects of substitute guidance mechanisms observed in a number of authentic derailment events.

In the vertical direction, guidance mechanisms should generally be located as close as possible to the top of the rail, without violating the permissible gauging standards. In the lateral direction, guidance mechanisms should be located further away from the adjacent wheel, so that the initial vertical movements, or ‘bouncing’, are attenuated by the time a lateral engagement with the rail is attempted.

The interoperable European gauging standard restricts the amount of possible geometrical positioning configurations for a guidance mechanism placed on the bogie frame. For mechanisms placed on the axles, either between or on the outside of the wheels, such limitations affect to a lesser extent the amount of feasible geometrical combinations.

For conventional vehicle trains with centre couplers, additional means of minimizing the lateral deviations after single bogie derailments of an intermediate car are utilizing the inter-vehicle couplers. MBS computer simulations have revealed the possibility of reducing the maximum permissible coupler yaw angle relative to the carbody by approximately 60% compared to an assumed standard configuration that permits a maximum lateral deflection between adjacent carbody ends of 0.65 m. Such restrictions are feasible in terms of maximum track shift forces and derailment ratio for vehicles equipped with both flexible and stiff wheelset guidance, when running through a track section containing two curves with radii \( R = 190 \) m in opposite directions and with no intermediate section in between.

For derailments on tangent track at a speed of 200 km/h, such coupler restrictions limit the lateral deviation of the bogie below 0.57 m so that all wheels remain on the relatively “safe” sleeper surface, assuming a sleeper length of 2.53 m. For derailments on a curved track with relatively high cant deficiency at the same speed, the largest deviation obtained with the least restraining coupler is approximately 0.71 m from the track centre line, implying that the wheel has fallen into ballast. A low-reaching bogie transversal beam might be beneficial in such situations, as it provides a vertical support mechanism so that the wheel would have limited contact with the ballast. A tighter coupler restriction, a stiffer bogie yaw resistance, a stiffer wheelset guidance and a low-reaching bogie transversal beam, can reduce the lateral deviation further to approximately 0.58 m.
Studies of the influence of carbody height of CoG on the overturning tendency were performed as a function of coupler height, vertical positioning of the transversal beam in the bogie and the maximum coupler yaw angle relative to the carbody. Vehicle overturning occurred both on tangent and curved track with cant deficiency for vehicle designs corresponding to a relatively large vertical distance between the carbody height of CoG and coupler, in combination with a high-reaching bogie transversal beam. It can therefore be concluded that a low carbody CoG and a low-reaching transversal beam in the bogie frame are favourable vehicle features with respect to carbody roll angle after derailment, in particular for railway systems with a low coupler height. For such cases, a low-reaching bogie transversal beam provides a vertical support mechanism that greatly reduces the risk of vehicle overturning.

An initial study was also performed on the post-derailment behaviour of an articulated trainset consisting of one conventional leading power car and four trailer cars, two of which were modelled as fully articulated. Employing similar derailment scenarios at a speed of 200 km/h as with the conventional train design, the lateral deviation of the median bogie and carbody overturning tendency have been studied as a function of the inter-carbody longitudinal damper characteristics. Furthermore, different vertical positions of the transversal beam in the bogie have been tested.

In several derailment scenarios, on tangent and curved track, the articulated configuration lacking inter-carbody longitudinal dampers behaves rather poorly in terms of lateral deviation and overturning tendency in comparison with a two-bogie conventional design. In a conventional configuration, the ultimate barrier against large inter-carbody lateral deviations after a derailment relies on the centre coupler, if sufficiently robust, and the assurance that the adjacent vehicle is safely rolling on rails. Evidently, the articulated configuration, with two carbody ends sharing the same bogie, can not utilize the adjacent vehicle in the same manner.

With lower inter-carbody longitudinal damper characteristics than the assumed standard, the articulated vehicle experiences larger carbody roll angles in comparison with the conventional configuration, on both tangent and curved track.

With incorporated inter-carbody longitudinal dampers, however, the derailment-worthiness of the articulated trainset could be drastically improved. On tangent track and with an assumed standard damper characteristic, the leading wheelset’s maximum lateral deviation reaches approximately 0.70 m, implying the wheel is no longer on sleepers. However, by increasing the damper characteristic with 50%, the maximum lateral deviation of the leading wheelset is now limited to approximately 0.23 m. In this case the wheelset is rolling on the sleepers’ surface at a substantial safety margin from the sleeper edge.

On a curved track with a quite high cant deficiency (153 mm), even an increase by 150% in damper force relative to the assumed standard value, leads to a maximum wheelset lateral deviation of 0.86 m. This is an approximately 0.2 m larger deviation than the value obtained with the most restrained coupler configuration in the conventional train configuration, for otherwise similar running gear features.

It should be emphasized that the present study is performed under the assumption of a number of vehicle and design parameters as well as derailment scenarios that might not always be generally applicable. For a correct assessment of the post-derailment vehicle
behaviour, a detailed MBS model of the vehicle under investigation is necessary, including a correct description of all low-reaching features in the running gear, restrictions in suspension and coupler deflections etc.

It should also be emphasized that some conclusions drawn in the present study are based on vehicle features and parts with sufficiently high strength capabilities. Such assumptions might not either be generally applicable.

However, the principal influence of a number of vehicle design features can still be evaluated on which basic conclusions, that ultimately lead to a more derailment-worthy design, can be drawn. In summary, these basic conclusions are:

- mechanical restrictions that limit the wheelset’s longitudinal and lateral movements relative to the bogie frame
- low-reaching lateral guidance mechanisms attached on to the bogie frame (bogie frame parts or transversal beams with a staggered design) or the axles (journal boxes or brake discs)
- low-reaching vertical support mechanisms (transversal beams or bogie frame parts)
- small vertical distance between carbody height of CoG and centre coupler pivot centre
- small centre coupler yaw angle for the conventional train configuration
- powerful inter-vehicle longitudinal dampers for the articulated train configuration
Future directions of study

Train derailments at high speed are inherently uncontrolled situations which, unlike many other conditions and events, can hardly be subjected to experimental investigations. The opening of a European database of accidents and incidents by the European Rail Agency (ERA) has implied a great leap forward for passive safety research, in particular for derailment-worthiness issues. It would be possible to find out that derailment events have occurred, in particular useful for incidents that otherwise receive limited medial attention outside the respective country. Nevertheless, in order to assess any current derailment-worthiness features, a thorough knowledge of the involved vehicle design is also required, besides detailed event information. In this respect, the ERA should recommend safety authorities and national investigating bodies in Europe to supplement the current practice of focusing on “What went wrong?” with additional questions at issue such as: “What stopped it turning into a catastrophe?”. In this respect a standardized template dealing with the most probable sequence of events should be developed.

In terms of post-derailment simulation methodology, computational tools should be developed to investigate running gears negotiating switches and crossings in a derailed condition. This topic is particularly important for running gear equipped with low-reaching substitute guidance mechanisms. In this context, also the necessary strength of guidance mechanisms should be addressed.

Moreover, computational models should be developed to account for wheel to ballast interaction for a better quantitative analysis of the vehicle overturning tendency. A correct assessment of forces arising at the vehicle inter-connections is also dependent on a proper description of the lateral forces on wheels when ploughing through ballast.

It is the author’s ambition and hope that the outcome of this thesis initiates a debate and raises the derailment-worthiness awareness in the international fora dealing with safety issues of rail vehicles. This might subsequently extend the list of vehicle features influencing derailment processes and consequences.
References


On Derailment-Worthiness in Rail Vehicle Design


Appendix A - Concrete material modelling details

The finite element simulations described in this thesis make use of a concrete material model called *Pseudo-Tensor* or Material Type 16 incorporated in the *LS-DYNA* code. Furthermore, this model can be used in two major modes by means of a tabulated yield stress versus pressure (mode I) or two curve model with damage and failure (mode II). The latter model was chosen with a further option of allowing the code to internally generate various material constants based on the concrete unconfined compressive strength.

The above mentioned material model option provides two yield versus pressure curves, \( \sigma_{\text{failed}} \) (minimum, damaged strength) and \( \sigma_{\text{max}} \) (maximum, undamaged strength), which limit the second invariant of the deviatoric stress \( \sigma_{\text{yield}} \), where

\[
\sigma_{\text{yield}} = \sigma_{\text{failed}} + \eta(\sigma_{\text{max}} - \sigma_{\text{failed}}) \tag{A-1}
\]

with,

\[
\sigma_{\text{failed}} = a_0 + \frac{p}{a_1 + a_2p} \tag{A-2}
\]

\[
\sigma_{\text{max}} = a_{0f} + \frac{p}{a_{1f} + a_2p} \tag{A-3}
\]

where

- \( p \) - pressure
- \( \eta \) - damage scale factor

The internally generated values for the material constants are:

\( a_0 = f'_c/4, a_1 = 1/3, a_2 = 1/3f'_c, a_{0f} = 0, a_{1f} = 0.385, \)

where \( f'_c \) is the concrete unconfined compressive strength.

The damage scale factor \( \eta \) shifts the \( \sigma_{\text{yield}} \) curve between the failed and the maximum state. The scale factor is tabulated against a damage function \( \lambda \) according to recommendations [45], see Figure A-1.

The damage function is a measure of the accumulated plastic strain \( \varepsilon^p \), and takes the form:

\[
\lambda = \int_{0}^{\varepsilon^p} \left(1 + \frac{p}{\sigma_{\text{cut}}}ight)^{-b_1} d\varepsilon^p, \tag{A-4}
\]

where \( b_1 \) is set to 1.25
and

\[ \sigma_{cut} = 1.7 \left( \frac{f'_c}{a_0} \right)^{2/3} \]  

(A-5)

The concrete relative strength increase when subjected to higher strain rates is captured by applying the curve in Figure A-2, which scales the compressive strength parameter according to the tentative recommendations of CEB [9] for an \( f'_c = 50 \text{ MPa} \) concrete.

\[ \frac{\sigma_{cut}}{f'_c} = \eta \lambda \]

**Figure A-1** Tabulated value pairs for the scale factor \( \eta \) vs. the damage function \( \lambda \) presented on a logarithmic scale [45]

**Figure A-2** Tabulated value pairs for the concrete compressive strength increase vs. strain rates [9]