Doctoral Thesis in Vehicle and Maritime Engineering

Mechatronic aspects of an innovative two-axle railway vehicle

ROCCO LIBERO GIOSSI

Stockholm, Sweden 2023
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Doctoral Thesis in Vehicle and Maritime Engineering
KTH Royal Institute of Technology
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Ciao Dede
Abstract

Within the Shift2Rail research program the goals for a sustainable growth of the railway sector are set. Among these are substantial reduction of Life Cycle Costs, improved reliability and energy efficiency, the reduction of noise emissions, and the achievement of full interoperability of the rolling stock. Therefore, a new generation of running gear is envisioned.

An innovative two-axle vehicle that can reduce weight, initial investment and maintenance cost, and emissions is proposed for a metro line system. The vehicle proposed will have only one suspension step. To further reduce the weight and incorporate the otherwise missing anti-roll bar, a composite material connection frame is introduced. The two-axle configuration suffers from poor ride comfort, due to the lack of a second suspension step acting as filter, and from poor steering capability, due to the long distance between wheelsets. Active suspensions are therefore introduced to improve both ride comfort and steering capability.

This Ph.D. thesis showcases the key activities undertaken during the development of the innovative vehicle, building a simulation framework where the vehicle can be virtually tested. Several modelling environments are used such as: SIMPACK for vehicle dynamics, Abaqus for finite elements modelling, Simscape for hydraulic physics simulations, and Simulink for control logic development. During the Ph.D. time two elements of the mechatronic vehicle have been designed and manufactured, i.e. the carbon fiber connection frame and the steering active suspension. The two components models have been experimentally validated and introduced into the simulation environment. A ride comfort and a wheelset steering control strategy have been designed to overcome the limitations introduced by the two-axle configuration. The proposed solutions aim at being applicable in the whole operational scenario of the innovative vehicle.

The present work emphasises the possibility of introducing innovative mechatronic solutions as an alternative to standard bogie solutions aiming at reducing costs and emissions, blurring the boundaries between academic view and possible industrial applications.
**Sammanfattning**

Inom forskningsprogrammet Shift2Rail sattes målen för en hållbar tillväxt av järnvägssektorn. Dessa mål innefattar en avsevärd minskning av livscykkelkostnader, en ökad tillförlitlighet och energieffektivitet, minskning av bullerutsläpp och full driftskompatibilitet för den rullande materielen. För att uppnå målen föreslås här en ny generation av lopwerk.


Detta arbete betonar möjligheten att introducera innovativa mekanotroniska lösningar som ett alternativ till vanliga boggilösningar som syftar till att minska kostnader och utsläpp, och sudda ut gränserna mellan den akademiska synen och möjliga industriella tillämpningar.
A visual representation

At the link embedded on this page and accessible through the provided QR code, it is possible to get a very condensed summary of the results achieved in the Ph.D. thesis. Also visualizations of some important results are provided.

https://github.com/RoccoKTH/Two-axle-Vehicle
List of appended papers

Paper A:

Paper B:

Paper C:

Paper D:

Paper E:

Paper F:
R. L. Giossi, R. Persson, S. Stichel, “Wheel wear reduction of a mechatronic two-axle vehicle controlled with feedforward wheelset steering approaches”, Accepted for journal publication in Vehicle System Dynamics
Division of work between authors

Paper A:
The study has been initiated by both Fu and Giossi that performed the literature research and wrote the paper. In particular the writing work splits following the thematic areas below:

- **Fu**
  - Secondary suspensions: semi-active
  - Secondary suspensions: fully active lateral control
  - Primary suspensions: solid axle steering

- **Giossi**
  - Tilting trains
  - Secondary suspensions: fully active vertical control
  - Primary suspensions: independently rotating wheels

- **Fu & Giossi**
  - Abstract
  - Introduction
  - Key principle separation
  - Summary and outlook

Persson, Stichel, Bruni, and Goodall supervised the work and reviewed the paper.

Paper B:
Giossi initiated the study, performed simulation, and wrote the paper. Persson and Stichel supervised the work and reviewed the paper.

Paper C:
Giossi initiated the study, performed simulation, and wrote the paper. Persson and Stichel supervised the work and reviewed the paper.

Paper D:
Giossi initiated the study, performed simulation, and wrote the paper. Shipsha developed the finite element frame model. Persson, Wennhage, and Stichel supervised the work and reviewed the paper.

Paper E:
Giossi initiated the study, performed simulation, and wrote the paper. Shipsha developed the finite element frame model. Persson, Wennhage, and Stichel supervised the work and reviewed the paper.

Paper F:
Giossi initiated the study, performed simulation, and wrote the paper. Persson and Stichel supervised the work and reviewed the paper.

Thesis contribution
The main contributions of this thesis are:

- A literature survey on mechatronic suspensions in railway vehicles, considering secondary and primary suspensions, and active and semi-active technologies.

- The general development of an innovative two-axle railway vehicle with only one suspension step.

- Inclusion of several modelling aspects into a unique vehicle model, considering flexible structural connection elements and actuator dynamics.

- Development of a novel feedforward steering control strategy to control an innovative two-axle vehicle in order to overcome stability and measurement issues normally involved in solid wheelset steering of railway vehicles.

- Development of a ride comfort control strategy based on better knowledge of the vehicle interconnected components that can adapt with vehicle speed.

- Performance validation of active steering strategies with respect to wheel wear evolution, showing the great impact it can have on maintenance.
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Part I

OVERVIEW
Chapter 1

Introduction

1.1 Background

The Shift2Rail research program, funded by the European Union’s research and innovation funding program Horizon 2020, sets a vision for the future of European trains. The vision is focusing on the substantial reduction of Life Cycle Costs, improved reliability and energy efficiency, the reduction of noise emissions, and the achievement of full interoperability of the rolling stock. These goals have been studied in over 105 projects following the five asset-specific Innovation Programs (IPs) (Figure 1.1) to achieve sustainable growth of the rail sector.

Figure 1.1: The five asset-specific Innovation Programs (IPs) as structured by the Shift2Rail program [1]
CHAPTER 1. INTRODUCTION

In the definitions of the Shift2Rail IPs, the main background for the development of the presented work can be found in IP1. As stated by the IP1 challenge [2],

[...] A combination of rail customers’ ever-evolving requirements for rail passenger vehicles is generating a new wave of challenges to rail vehicle development: quality of service, time constraints, mounting energy costs, more stringent emission standards, and increasing stress on the economics of rail operation. If rail is to compete more effectively with other modes and attract more passengers in the future, it needs a future generation of passenger trains that will be lighter and more energy- and cost-efficient, while at the same time providing a comfortable, safe and affordable travel experience for all passengers. These innovations should therefore not be limited to the traditional, incremental approach to vehicle development, but should also derive from an entirely new way of thinking on product development. [...] 

Therefore, the development of a new generation of running gear is essential to achieve the goals set by Shift2Rail. Within the four projects Run2Rail, Pivot, NEXTGEAR, and Pivot2, a concept for an innovative two-axle vehicle with single axle running gear is developed to reduce vehicle weight, investment and maintenance cost, and energy consumption. Due to large axle distance and only one suspension step, the two-axle vehicle performance is inferior to the one of a standard bogie vehicle. To maintain the weight, investment, and energy savings expected from the innovative vehicle concept and improve the poor performance, active suspensions are incorporated in the vehicle concept making it a mechatronic vehicle.

With KTH being partner of the above mentioned projects, Metro Madrid requirements for line 10 are considered for the vehicle dynamics design due to the unique challenges that the requirements offer: a high speed for a metro system (120 km/h) and high concentration of small curve radii. The performance of the mechatronic innovative vehicle can showcase the potential of innovative mechatronic solutions as alternatives to standard vehicle design.

1.2 Scope

In this frame work, the scope of this thesis is to carry forward the design and development of the innovative mechatronic two-axle vehicle. To do so, a series of aspects are analysed. Among those, innovative components, such as a composite material frame and a suspension-actuator combination,
and control strategies to improve ride comfort and steering capability. The aim is to provide a competitive alternative to standard vehicles in terms of dynamic performance.

1.3 Research questions

The research questions that this thesis will try to answer are the following:

- Is it possible to bring the ride comfort of a two-axle rail vehicle to acceptable levels with active suspensions? (Paper D & Paper E)

- Is it possible to improve the performance of modal control for active ride comfort? (Paper E)

- Given the above, is it possible to determine a control strategy optimized for different speeds? (Paper D & Paper E)

- Can the wheelset steering capability of a two-axle vehicle be improved by active steering control? (Paper C)

- Is it possible to overcome stability issues of feedback control in controlling wheelset steering with simple approaches? (Paper B & Paper C)

- Given the above, is it possible to reduce maintenance cost due to wheel wear with the actively steered two-axle vehicle? (Paper F)

1.4 Thesis structure

In Chapter 2, the methods used in this thesis work to evaluate vehicle performances are introduced. Only a few of the possible evaluation perspectives are considered in the vehicle design. The reasons for the selection of the used criteria are given.

The innovative vehicle concept is introduced and described in Chapter 3. Subsequently, the developments of the key aspects defining the vehicle are expanded in more detail. These can be divided into a “hardware” part and a “software” part. The hardware part contains physical components that define the innovative vehicle, while the control logic belongs to the software part.

Concerning the hardware part, the vehicle structure and its modelling is presented in Chapter 4 where the carbody modelling and the carbon fiber
CHAPTER 1. INTRODUCTION

connection frame are shown. The actuation system is introduced in Chapter 5 where both dynamic actuator models to control ride comfort, and the newly developed steering actuator are presented.

In the software part, the major difficulties affecting the development of the two-axle vehicle are described and the control approaches implemented to solve them are presented. Firstly, in Chapter 6 the ride comfort problem is addressed and the developed blended control strategy is described. Subsequently, in Chapter 7 the steering strategy is described, highlighting the robustness of the proposed feedforward steering approach. The benefits in terms of wheel wear are presented too.

Finally, in Chapter 8 the appended papers are summarized, in Chapter 9 the conclusions are drawn and in Chapter 10 suggested future work is displayed.
Chapter 2

Railway Vehicle Performance Evaluation

The definition of performance evaluation tools directly influences the development of a vehicle. Depending on the operating scenario, the different performance criteria, such as running safety, passenger ride comfort, and wheelset steering capability have different weights in the vehicle design. For a high speed line, where the track is characterized by a large amount of tangent sections and significant travelling time, ride comfort plays a greater role than steering capability. In contrast, for a metro line, where curved track sections are generally prominent and travelling time is generally short, wheelset steering capability can influence the design more than ride comfort. Nevertheless, critical criteria need to be met. Vehicle stability is one of the safety requirements that needs to fulfil the European EN14363 standard \[3\] (generally in contradiction with the steering capability of the vehicle \[4\]), good ride comfort is essential for passenger experience, and steering capability is influencing wheel and rail wear, which is directly related to maintenance cost.

Despite the importance of vehicle safety issues, which includes vehicle stability, the focus of the present study is on ride comfort and steering capability. Therefore, hereafter, only ride comfort and steering capability evaluation criteria are summarized. Safety requirements are studied for the vehicle development in the Shift2Rail project Run2Rail \[5\] and by Persson et al. in \[6\].

2.1 Ride comfort

Ride comfort can be divided into two categories. The first one focuses on motion sickness and concerns low frequency vibrations between 0.1 and 0.5
Hz, while the second one concerns frequencies between 0.5 and 80 Hz [7]. The latter is the most relevant for the present study, while the former one is more relevant for tilting trains [8].

Several methods exist to evaluate ride comfort. In Europe, two main methods are used, Wertungszahl $W_z$ (leading the evaluation up to the 90s [7]), and the one defined by the EN12299 standard [9]. The two methods have a similar approach but the latter is the one most often used in rail vehicle approval and therefore of more interest. EN12299 is often used as reference to develop control strategies ([10], **Paper D**, **Paper E**), study different mechatronic schemes [11], optimize suspension parameters ([12], [13]), and evaluate field test performances [14].

As per the EN standard, continuous comfort index in one direction (dir),

$$C_{c,\text{dir}} = a_{\text{dir},p}^{W_{z,\text{rms}}},$$

is defined as the r.m.s. value of the acceleration at carbody floor level $p$ over a measuring time period $T = 5s$. Depending on the direction, the acceleration is weighted in the frequency domain with the weighting functions $W_d$ for longitudinal and lateral directions, or $W_b$ for vertical direction (Figure 2.1). These functions represent the human sensitivity to excitation with different frequencies in each direction.

The mean ride comfort, mandatory for vehicle approval,

$$N_{MV} = 6\sqrt{\left(a_{x,p,95}^{W_{d,\text{rms}}}\right)^2 + \left(a_{y,p,95}^{W_{d,\text{rms}}}\right)^2 + \left(a_{z,p,95}^{W_{b,\text{rms}}}\right)^2},$$

Figure 2.1: Comfort weighting functions according to EN12299 [9]
2.2. WHEELSET STEERING CAPABILITY

is a combination of the accelerations in the three directions $x$, $y$, and $z$, where the accelerations are the 95th percentiles of at least 60 r.m.s. values. In Table 2.1 the indications to judge the ride comfort quality are shown for both the mean comfort value $N_{MV}$ and the continuous comfort indices $C_{c,y}$, and $C_{c,z}$.

Table 2.1: Indication scale for ride comfort indices

<table>
<thead>
<tr>
<th>$N_{MV}$</th>
<th>$C_{c,y}, C_{c,z}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\langle \cdot \rangle &lt; 1.5$</td>
<td>Very comfortable</td>
</tr>
<tr>
<td>$1.5 \leq \langle \cdot \rangle &lt; 2.5$</td>
<td>Comfortable</td>
</tr>
<tr>
<td>$2.5 \leq \langle \cdot \rangle &lt; 3.5$</td>
<td>Medium</td>
</tr>
<tr>
<td>$3.5 \leq \langle \cdot \rangle &lt; 4.5$</td>
<td>Uncomfortable</td>
</tr>
<tr>
<td>$4.5 \leq \langle \cdot \rangle$</td>
<td>Very uncomfortable</td>
</tr>
<tr>
<td>$\langle \cdot \rangle &lt; 0.2 \text{ m/s}^2$</td>
<td>Very comfortable</td>
</tr>
<tr>
<td>$0.2 \text{ m/s}^2 \leq \langle \cdot \rangle &lt; 0.3 \text{ m/s}^2$</td>
<td>Comfortable</td>
</tr>
<tr>
<td>$0.3 \text{ m/s}^2 \leq \langle \cdot \rangle &lt; 0.4 \text{ m/s}^2$</td>
<td>Medium</td>
</tr>
<tr>
<td>$0.4 \text{ m/s}^2 \leq \langle \cdot \rangle$</td>
<td>Less comfortable</td>
</tr>
</tbody>
</table>

2.2 Wheelset steering capability

As mentioned above, the second important aspect to be considered concerns the steering capability. Wheelset steering capability can be defined as the ability of the vehicle to negotiate a curve. Among other aspects, such as traction and braking, steering capability influences the wheel and rail wear. A vehicle with poor steering capability will produce more wear and thus a higher maintenance cost. Therefore, evaluation of the steering capability is of paramount importance in the vehicle design.

In contrast to the ride comfort, no standard exists to evaluate steering capability. Two major approaches are used in research studies to evaluate steering capability through wear estimation. The first one, called wear number or $T_\gamma$, is a direct measure of the energy dissipated in the contact patch between the wheel and the rail. The second estimates the wear evolution as lost volume of wheel material over time.

2.2.1 Wear number: $T_\gamma$

The first and most used approach to evaluate steering capability is the $T_\gamma$ method. This method is a simple engineering approach determining the energy dissipated in the contact between wheel and rail. For each contact point, the dissipated energy per unit length,

$$T_\gamma = |F_\xi \nu_\xi| + |F_\eta \nu_\eta| + |M \phi|,$$

(2.3)

is given by the sum of the creep forces and spin moment multiplied by their respective creepages. Here, $\xi$ is the longitudinal direction on the contact.
patch and \( \eta \) is the lateral direction. The spin and spin moment are oriented as the normal direction to the contact patch. \( T_\gamma \) is measured in \( J/m \) expressing the energy that is being dissipated by the vehicle (or more precisely by one contact patch) per travelled distance.

\( T_\gamma \) is a simplified method that nevertheless provides relevant information when curve negotiation is considered. A high \( T_\gamma \) value indicates that a significant amount of energy is wasted in more sliding than necessary. Often, high values of \( T_\gamma \) coincide with flange contact. Its fast computation puts the \( T_\gamma \) as one of the most used methods in steering capability evaluation. It represents a fast tool that can quantitatively compare steering capability of different vehicle configurations on different curve negotiation conditions. The method has been extensively used in active suspension studies both as validation tool (from the early stages \[15\], \[16\], to more recent studies \[17\], \[18\]), and as designing tool (\[19\], Paper C).

### 2.2.2 Uniform wheel wear

A more advanced but more computationally demanding method to evaluate steering capability, and more generally wheel-rail interaction, involves the study of the wheel evolution due to wear. Several methods exist and the KTH method is among the ones that provide a good compromise between damage model complexity and simulation computational requirements \[20\].

The method, firstly introduced by Jendel in \[21\], is based on the wear model defined by Archard. The height of the lost material \( \Delta z(x, y) \) due to wear can be calculated as,

\[
\Delta z(x, y) = K_n \frac{p(x, y)s(x, y)}{H},
\]

where, \( p(x, y) \) is the local normal pressure, \( s(x, y) \) is the slip distance, \( H \) is the hardness of the wheel, and \( K_n \) is the Archard’s map coefficient. Archard’s map is divided into four zones that depend on the combination of pressure and slip velocity between the two bodies in contact. The coefficients were originally determined by pin on disk tests.

The KTH method unfolds into several steps. Initially, load collectives that represent a specific railway line are defined based on statistical analysis. Simulations are then performed using the defined load collectives. For each load case, time step, and contact point, the lost volume due to wear is calculated by Eq. \[2.4\]. Finally, the lost volume is applied on the wheel (or rail) profiles considering the distribution of curve radii in the load collectives. The process is repeated until a desired mileage is attained. A visualization of how the output from one simulation on one contact patch is translated to material loss is schematized in Figure 2.2. Here, Hertz elliptic contact
2.2. WHEELSET STEERING CAPABILITY

is applied to calculate normal pressure and FASTSIM is used to calculate tangential stresses. Optionally, the KTH method can be applied with non-hertzian contact and other methods to calculate tangential stresses.

The method has proved to be effective in reproducing experimental results ([21]–[23]), where tuning of the Archard’s map coefficients is generally involved. This process requires several iterations and it is case dependent. Nevertheless, the KTH method showed good performance in comparison with other methods [24] and has been used in maintenance optimization for both wheels [25] and rails [26].
Chapter 3

The Innovative Vehicle

The development of a new generation of running gear is essential to achieve the goals set by Shift2Rail for future European trains. These focus on the substantial reduction of Life Cycle Costs (LCC), improved reliability and energy efficiency, the reduction of noise emissions, and the achievement of full interoperability of the rolling stock. Consistently, many technological advancements have been proposed or achieved in the last decades that have the potential to improve the running gear of railway rolling stock.

3.1 Two-axle vehicles overview

Two-axle vehicles with single axle running gears are efficient and low cost, but are currently associated with low performing vehicles such as freight cars, having poor ride comfort (due to the single suspension step) and poor steering capability (due to the increased wheelset distance). Nevertheless,

Figure 3.1: Two-axle vehicles not in operation anymore: Uerdingen railbus model VT 98.9 [27] (Left), and Pacer class 142 [28] (Right)
the concept of two-axle passenger vehicles in Europe is not new. In the mid 20th century, two-axle vehicles were used thanks to their low cost (due to a simpler structure) and easy maintenance. Examples of these vehicles are the German Uerdingen railbus model VT 98.9 \cite{27} (Figure 3.1 (Left)) and the recently phased out UK vehicle Pacer class 142 \cite{28} (Figure 3.1 (Right)).

Noticeable modern examples of single suspension step vehicles are the Copenhagen S-train \cite{29} (Top), and the German Next Generation Train (NGT). The first one is in service on the local network in Copenhagen and can reach a maximum service speed of 120 km/h \cite{29, 30}. The second one, is a high speed independently rotating wheels single suspension step two-axle vehicle in development by the German Aerospace Center (DLR) institute. This project has found great research interest in both control \cite{31, 32} and estimation processes \cite{33, 34}. In Figure 3.2 (Top) the structure of the Litra SE is shown \cite{29}, while in Figure 3.2 (Bottom) a view of the NGT \cite{35} is displayed.

3.2 Concept

As described above, two-axle vehicles with single suspension steps have the potential to be a substitute of standard bogie vehicles due to an expected reduced investment and maintenance cost. One of the most prominent aspects associated with two-axle vehicles is the reduced vehicle weight driven by the absence of a more standard bogie that generally has a mass of approximately two to three tons considering equipment. A reduced vehicle weight should guarantee significant energy savings. Therefore, the development of new two-axle vehicles with only one suspension step is gaining attention.

The vehicle proposed in the Shift2Rail project Run2Rail \cite{5} is one of this
3.2. CONCEPT

Figure 3.3: S8000 bogie vehicle design (Top) and Innovative vehicle design (Bottom)

kind. It is meant to be a substitute for the class S8000 vehicle currently in use on Metro Madrid line 10 with a maximum speed of 120 km/h. A reduced vehicle length of 12 m is foreseen to avoid excessive distance between the wheelsets and too high axle load. In Figure 3.3, a visual comparison between the S8000 vehicle and the innovative two-axle vehicle is given using the same scale for both vehicles, showing the key distances characterizing the vehicles. Due to the reduced vehicle length, more vehicles are needed to achieve the required passenger capacity, increasing from three vehicles per unit for the S8000 to five vehicles per unit for the innovative vehicle.

Additional weight savings are achieved by the running gear design itself. Axleboxes are placed in the inner side of the axle (between wheels), allowing for a shorter wheelset and thus a reduced weight. The frame connecting the wheelsets and the carbody incorporates the structural support function of carrying the vehicle and the indispensable function of the anti-roll bar. The latter is achieved exploiting the flexibility of the frame itself. The U-shaped frame (analyzed in more details in Subsection 4.2) is made in carbon fiber reinforced polymer fulfilling the requirements of structural reliability, anti-roll flexibility, and a low weight of only about 70 kg [36]. Moreover, coil springs are supposed to be used as vertical suspensions, opening for a possible air-free vehicle. In Figure 3.4 a same scale scheme comparison of
CHAPTER 3. THE INNOVATIVE VEHICLE

Figure 3.4: S8000 (Left) and Innovative vehicle (Right) running gears showing the mass difference of the two bogies considering equipment. For the innovative vehicle 600 kg of equipment is foreseen.

the S8000 running gear and the innovative vehicle one is shown.

By keeping the same payload per meter of 1000 kg/m, the presented design is foreseen to have a substantial weight reduction of 400 kg/m in terms of tare weight per meter. This design is expected to save between 5 to 9 % energy per vehicle run [37]. In Table 3.1, the introduced characteristics of the two vehicles compared to each other are shown.

Table 3.1: Summarizing parameters for the S8000 vehicle and the two-axle innovative vehicle

<table>
<thead>
<tr>
<th></th>
<th>S8000</th>
<th>2-axle</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max speed</td>
<td>120 km/h</td>
<td>120 km/h</td>
</tr>
<tr>
<td>Bogie distance</td>
<td>11.1 m</td>
<td>n.a.</td>
</tr>
<tr>
<td>Wheelset distance</td>
<td>2.2 m</td>
<td>8 m</td>
</tr>
<tr>
<td>Vehicle length</td>
<td>19/17 m</td>
<td>12 m</td>
</tr>
<tr>
<td>Vehicles per unit</td>
<td>3</td>
<td>5</td>
</tr>
<tr>
<td>Train length</td>
<td>55 m</td>
<td>60 m</td>
</tr>
<tr>
<td>Frame mass</td>
<td>3000 kg</td>
<td>70 kg</td>
</tr>
<tr>
<td>Payload per meter</td>
<td>1000 kg/m</td>
<td>1000 kg/m</td>
</tr>
<tr>
<td>Tare weight per meter</td>
<td>1900 kg/m</td>
<td>1500 kg/m</td>
</tr>
</tbody>
</table>
3.3 Mechatronic configuration

The innovative vehicle offers significant weight savings, but to be attractive, the inherent design issues must be addressed. The first one concerns ride comfort. Due to the removal of one suspension step, a higher amount of vibrations coming from the rail are transferred to the carbody, worsening the ride experience of the passengers. The second involves the steering capability due to an increased wheelset distance. Therefore, active systems are implemented to maintain the significant reductions in cost as well as reductions in emissions and energy consumption due to the reduced mass and to achieve adequate vibration comfort levels for passenger vehicles.

Active and semi-active systems have found an increasing number of applications in the railway sector throughout the years. Major reviews were published in 1983, 1997, 2003, 2007, and 2020 ([38]–[41], Paper A) showing the improvements and the trends of railway active suspension systems. Mechatronic suspensions are mostly used for dynamic ride comfort (focusing on carbody vibrations) and steering capability improvement (focusing on wheelset behaviour and wheel rail interaction). Nevertheless, tilting trains represent a fully developed exception having the focus on the quasi-static carbody inclination during curve negotiation to allow increased vehicle speed and reduced running time [42]. Another exception is represented by active hunting stability [43]–[45] where active torque is applied to reduce yaw and lateral movement of the wheelset in absence of longitudinal suspensions. A combination of active steering and active stabilization is represented by active yaw relaxation [15] where the longitudinal suspension is actively modified to have a high stiffness in tangent track to prevent hunting and a low stiffness in curves to let the wheelset adopt its unconstrained rolling position. Independently rotating wheels constitute a shift of paradigm with respect to the aforementioned trends. This configuration significantly differs from the solid axle solution by the absence of the stiff torsional connection between left and right wheel on a single axle. By doing so, on the one hand, the dependency between stability and curving capability is removed opening up for the removal of the hunting instability problem and potentially improving curving capability [46], [47]. On the other hand, the self centring capability is absent and needs to be restored, either mechanically (Talgo’s solution [48] or other types of linkages between wheels [49]–[51]) or actively with actuated independently rotating wheels (AIRW) [52], driven independently rotating wheels (DIRW) [53], or directly steered wheels (DSW) [54]. A noticeable modern example of DIRW is the German NGT ([35]).

Stability control may conflict with some requirements of the safety standard EN14363 [3]. Independently rotating wheels are a promising alternative to solid axles that nevertheless involves more complex arrangements.
and development. Therefore, the proposed mechatronic vehicle uses only dynamic comfort control, in vertical and lateral direction, and steering of solid axle wheelsets. A preliminary investigation of the vehicle equipped with independently rotating wheels was performed in [5].

The mechatronic configuration scheme of the vehicle is given in Figure 3.5. In this configuration, globally 10 actuators are needed. Four take care of the dynamic response of the vehicle in vertical direction and two in lateral direction aiming at improving ride comfort (Table 2.1). The remaining four actuators aim at providing wheelset guidance including preventing hunting instability and steering the wheelset in curves. Dynamic actuators for comfort improvement are discussed in Subsection 5.1 while steering actuators are discussed in Subsection 5.2.
Chapter 4

Vehicle Structure

As introduced in Chapter 3, the innovative vehicle is characterized by several components that need to be studied and modelled. Hereafter, two structural parts are considered, i.e. the carbody, and the U-shaped frame.

4.1 Carbody

Modelling of the carbody is significant in terms of ride comfort evaluation. Carbody flexible modes may appear in the human body’s most sensitive frequency region between 6 and 20 Hz. The situation is often worsened in light weight vehicles, caused by reducing the carbody mass and thereby reducing the carbody structural stiffness. The importance of including carbody flexibility in simulation models for calculation of ride comfort has been confirmed throughout the years with carbody models of increasing complexity (Zhou et al. [55] using a two dimensional beam model, Bokaeian et al. [56] using a three dimensional beam model, and Ling et al. [57] using a finite element (FE) model).

The proposed innovative vehicle does not exist and the carbody model needs to be created following design guidelines. Therefore, a simplified FE model is developed in Abaqus to fulfill the design criteria of carbody mass and topology, and the most foreseeable structural (or flexible) eigenfrequencies and eigenmodes.

Firstly, topology and mass properties are addressed. The topology of the carbody is designed to have a length of 12 m (Table 3.1) with a simplified squared cross section of 3x3 m. To mimic a realistic shape of a metro vehicle, three doors and four windows for each side, and a hole in the roof for the ventilation unit are considered. In the design phase, additional masses are added at the bottom of the carbody to represent equipment masses and to lower the centre of gravity (C.o.G.) to meet the design location of 1.7
Figure 4.1: Bending, torsion, and rhombic carbody flexible mode shapes compared between a simple beam model \([56]\), a FE model \([58]\), an experimental testing \([59]\), and the proposed FE model.

In Table 4.1, the achieved mass properties highlight the good agreement between the design values and the ones obtained with the FE model.

Table 4.1: Designed carbody mass properties: comparison between design values and achieved with FE model

<table>
<thead>
<tr>
<th>C.o.G. [m]</th>
<th>Mass [kg]</th>
<th>(J_x) [kg(\cdot)m(^2)]</th>
<th>(J_y) [kg(\cdot)m(^2)]</th>
<th>(J_z) [kg(\cdot)m(^2)]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design</td>
<td>-1.7</td>
<td>13000</td>
<td>32000</td>
<td>256000</td>
</tr>
<tr>
<td>FE model</td>
<td>-1.74</td>
<td>12959</td>
<td>31858</td>
<td>227151</td>
</tr>
<tr>
<td>Difference</td>
<td>2.3 %</td>
<td>0.3 %</td>
<td>0.4 %</td>
<td>11.3 %</td>
</tr>
</tbody>
</table>

The second relevant aspect in the carbody design is to match foreseeable carbody eigenfrequencies and mode shapes. A literature survey on the carbody flexibility is performed to identify typical carbody eigenfrequencies and eigenmodes. In Table 4.2, the survey is summarized highlighting the type of model used and/or experimental validation. To achieve good coherence with the average eigenfrequency per mode shape, the material properties of the Abaqus model are tweaked while maintaining the mass properties previously obtained (Table 4.1). The eigenfrequencies for verti-
4.2 U-SHAPED FRAME

cal bending modes and torsion agree well with the average frequencies from literature. Deviations are found for rhombic and lateral bending likely due to the shortness of the proposed vehicle body compared to standard vehicles. In Figure 4.1 a visual representation is given for the most common eigenmodes (vertical bending, torsion, and rhombic mode). Here, the beam model from [56], the FE model from [58], the experimental testing from [59], and the proposed vehicle are compared.

Table 4.2: Eigenfrequencies [Hz]: § beam model, ♯ FE model, * experimental validation, (?) unclear result, (·) possible alternative, / missing result

<table>
<thead>
<tr>
<th></th>
<th>Vertical Bending</th>
<th>Torsion</th>
<th>Rhombic/Diamond</th>
<th>2nd Vertical Bending</th>
<th>Lateral Bending</th>
</tr>
</thead>
<tbody>
<tr>
<td>Zhou et al. § [55]</td>
<td>12.3</td>
<td>/</td>
<td>/</td>
<td>17.0</td>
<td>/</td>
</tr>
<tr>
<td>Miao et al. ♯ [60]</td>
<td>10.7</td>
<td>6.1</td>
<td>/</td>
<td>/</td>
<td>11.3</td>
</tr>
<tr>
<td>Kozek et al. ♯ [61]</td>
<td>8.5</td>
<td>9.5</td>
<td>7.1</td>
<td>/</td>
<td>/</td>
</tr>
<tr>
<td>Tomioka et al. ♯ [62]</td>
<td>9.0</td>
<td>12.0</td>
<td>10.0</td>
<td>/</td>
<td>/</td>
</tr>
<tr>
<td>Akiyama et al. * [63]</td>
<td>13.0 (10.2)</td>
<td>/</td>
<td>7.8 (8.9)</td>
<td>/</td>
<td>/</td>
</tr>
<tr>
<td>Andersson et al. ♯ [7]</td>
<td>10.9</td>
<td>11.3</td>
<td>/</td>
<td>/</td>
<td>11.9</td>
</tr>
<tr>
<td>W. Sun et al. ♯ [64]</td>
<td>9.7</td>
<td>14.2</td>
<td>/</td>
<td>/</td>
<td>11.7</td>
</tr>
<tr>
<td>Y. Sun et al. ♯ [58]</td>
<td>11.6</td>
<td>12.4</td>
<td>9.7</td>
<td>/</td>
<td>14.2</td>
</tr>
<tr>
<td>Ling et al. ♯ [57]</td>
<td>10.1</td>
<td>13.6</td>
<td>9.1</td>
<td>12.1 (12.8)</td>
<td>(?)</td>
</tr>
<tr>
<td>Wang et al. * [65]</td>
<td>11.5</td>
<td>15.8</td>
<td>10.5</td>
<td>(21.9)</td>
<td>/</td>
</tr>
<tr>
<td>Bokaeian et al. § [56]</td>
<td>9.8</td>
<td>13.4</td>
<td>/</td>
<td>15.2</td>
<td>/</td>
</tr>
<tr>
<td>Akiyama et al. * [59]</td>
<td>11.2</td>
<td>16.2</td>
<td>9.9</td>
<td>(?)</td>
<td>/</td>
</tr>
<tr>
<td>Wu et al. * (+) [66]</td>
<td>10.1</td>
<td>/</td>
<td>8.8</td>
<td>15.7</td>
<td>/</td>
</tr>
<tr>
<td>Yang et al. * (+) [67]</td>
<td>13.2</td>
<td>/</td>
<td>9.5</td>
<td>/</td>
<td>/</td>
</tr>
<tr>
<td>Appearance number</td>
<td>14 (15)</td>
<td>10</td>
<td>9 (10)</td>
<td>4 (6)</td>
<td>4</td>
</tr>
<tr>
<td>Average</td>
<td>10.8</td>
<td>12.5</td>
<td>9.1</td>
<td>16.5</td>
<td>12.3</td>
</tr>
<tr>
<td>Proposed</td>
<td>11.0</td>
<td>12.3</td>
<td>13.2</td>
<td>17.6</td>
<td>21.0</td>
</tr>
</tbody>
</table>

4.2 U-shaped frame

The U-shaped frame is of paramount importance in designing the innovative vehicle. This component serves the purpose of a structural element as well as a suspension element. Conventional bogies have the anti-roll bar as a separate element, here this function is incorporated into the frame by its flexibility. The frame concept development started in the Shift2Rail project Run2rail [5], and a first model was developed in SIMPACK where two rigid halves are connected by a torsional stiffness of 250 kNm/rad without damping. Afterwards, the U-shaped frame went through a more detailed design phase in the Shift2Rail project NEXTGEAR [36], where a prototype was
CHAPTER 4. VEHICLE STRUCTURE

Figure 4.2: U-shaped frame evolution through out the projects Run2Rail and the subsequent NEXTGEAR manufactured. The evolution of the frame and its implementations are summarised in Figure 4.2.

The frame is made of Carbon Fibre Reinforced Polymer (CFRP), a high-performance composite material that can improve the frame structural efficiency thanks to its excellent mechanical performance-to-weight ratio and high fatigue strength \( [68, 69] \). The frame was analysed in Abaqus, where the FE models were developed based on multi-layer shell-elements, and failure predictions were performed considering the local stresses in every individual layer of the composite material. The topology of the frame was developed with help of several iterations between Abaqus and SIMPACK to achieve the desired vehicle dynamic characteristics, and sufficient static and fatigue strength. The frame has been tested in a KTH laboratory showing agreement between the model and experimental results \[36\]. In Figure 4.3 the frame at the KTH facility is shown. Three major tests were performed. The first one to evaluate the strength of the frame in withstanding exceptional vertical loads, the second one to evaluate the twist and thereby the anti-roll stiffness, and the third one to evaluate the dynamic properties in terms of eigenfrequencies and eigenmodes. The last one was performed before and after the load tests showing no significant changes in the dynamic response due to the frame loading.
4.3. Carbody-frame interaction

Once the carbody and frame models are defined in Abaqus, the models are imported and connected in SIMPACK. Here, the wheelsets can be connected to the frame through the axleboxes and dynamic analysis can be performed. The frame is connected to the carbody not only through the actuators (Figure 3.5) but also through mechanical connections as shown in Figure 4.4. Here, the key dimensions of the frame are given too.

Figure 4.3: U-shaped frame under test at the KTH facility: exceptional vertical load test (Left), twist load test (Centre), and modal analysis with hammer test (Right) [36]

Figure 4.4: U-shaped frame: connection points to carbody and wheelset, main dimensions (top view), height above the rail (side view) [36]
To avoid excessive SIMPACK computational time in the dynamic simulations, only the first five eigenfrequencies of the frame are retained and imported in SIMPACK. The same has been done for the carbody model.
where the frequencies are the ones shown in Table 4.2. The carbody-frame interaction gives rise to 10 important modes. The first five concern rigid carbody modes (Figure 4.5) and showcase the importance of the frame flexibility in balancing the roll motion. The next five concern carbody flexible modes (Figure 4.6).

The model so defined represents a starting point in the development of the vehicle virtual twin. This helps in taking forward the vehicle design, trying to approximate a real world implementation of the innovative vehicle.
Chapter 5

Actuation Systems

After the definition of structural components (Chapter 4), the remaining key components to be defined are the actuators. As defined in Subsection 3.3, a major distinction can be done between two types of actuators, the ones aiming at controlling carbody motion and therefore ride comfort, and the ones aiming at improving the wheelset steering capability. The first ones can be referred as high frequency actuators, aiming at mitigating oscillatory behaviours in the frequency range between 1 to 30 Hz, while the second ones can be referred as low frequency actuators, aiming at controlling quasi-static behaviours.

5.1 High frequency actuator - Comfort

In the railway sector, high frequency actuators are mainly used to improve ride comfort by actively reducing vibrations transmitted to the carbody. Given a certain operational scenario, the mutual relationship between ride comfort, vehicle speed, and track quality can place limitations on the advancements that can be made through passive vehicle design Paper A. Active or semi-active suspension systems offer a solution to this issue by relaxing the requirements on track quality and passive vehicle design.

Carbody vibrations can be mitigated with two main methods. The first one focuses on controlling only carbody flexible modes, while the second, more common, considers primarily carbody rigid modes. The first category aims at improving already acceptable solutions. Piezoelectric actuators and equipment suspended mass control can be considered to control the first flexible carbody modes. Piezoelectric actuators have shown good performances in scale models experimental testing (61, 70), while suspended equipment control (71, 72) showed reduction in carbody vibrations, improving suspended equipment passive frequency splitting effect (73, 74).
CHAPTER 5. ACTUATION SYSTEMS

When the improvement requirements are considerable, the carbody rigid modes must be controlled to reach the ride quality target. In this second category, actuators are placed between the bogies (or wheelsets) and the carbody in vertical and/or lateral direction replacing conventional dampers. Exceptions are inter-vehicle actuators [75], [76] and lateral-vertical coupled actuators [77]. Fully active actuators or semi-active dampers are generally implemented. Among other possibilities, hydraulic actuators and magneto-rheologic dampers are often used. The first ones output force and displacement by changing the pressure inside two chambers of the same hydraulic cylinder. The second ones change the magneto-rheologic fluid damping coefficient by providing an external magnetic flux. Semi-active dampers require negligible external power supply but the performance enhancements are limited ([11], [78]).

Due to the absence of the bogie, acting as a filter for the vibrations transmitted to the carbody from the rails, the innovative vehicle is subjected to substantial vibrations in the carbody, experiencing a poor passenger ride comfort ($N_{MV}$ above 5 at 100 km/h) when standard dampers are equipped. Therefore, among the above presented solutions, fully active systems need to be considered. Piezoelectric actuators and/or controlled suspended equipment may still have beneficial effects, but they are subdue to a more detailed development of the carbody itself. Several actuator technologies can be implemented when fully active systems are foreseen for comfort control. Electro-mechanical actuators ([79]–[81]) have good efficiency and high frequency band-width, but they suffer from high stiffness at high frequency and jamming possibility. If jamming occurs, the connection between bogie/wheelset and carbody becomes stiff, which will significantly worsen the ride comfort. Electro-magnetic actuators ([71], [82], [83]) have good performances and a cut-off frequency that can make it possible to directly control flexible modes. At the same time, electro-magnetic actuators have limited output force and stroke length. Moreover, they suffer from a strongly non-linear behaviour. Modified controllable versions of air springs can be used as actuators ([84], [86]) and have the large advantage of being a commonly implemented solution in conventional bogie vehicles. Nevertheless, like pneumatic actuators ([87], [88]), the high air compressibility significantly limits the operational bandwidth. Hydraulic actuators ([10], [71], [89], [90]) have a bandwidth in between electro-mechanical and pneumatic actuators, low to no jamming possibility, and a good weight to force ratio. However, hydraulic actuators present high maintenance cost and risk of environment contamination in case of leaking.

Hydraulic actuators are chosen due to their good weight to force ratio and relatively low investment cost. Additionally, if no command is sent to the control unit, the hydraulic actuator will behave as a damper with low
5.1. HIGH FREQUENCY ACTUATOR - COMFORT

Figure 5.1: Ride comfort actuator: Scheme (Left), and its implementation in Simscape (Right)

damping ratio making it a safer choice than electro-mechanical actuators. The actuators are modelled in the Simscape-Hydraulics environment to increase the trustworthiness of the modelled system. During simulations, the Simscape module solves the non-linear equations related to the actuator dynamics and provides the actuators input forces to SIMPACK. The latter provides the actuators connection points velocity to Simscape creating a natural feedback loop in the co-simulation environment between Simulink and the multibody simulation program SIMPACK. The actuator scheme and its Simscape implementation are shown in Figure 5.1. The hydraulic actuators are composed by a double acting hydraulic cylinder in which each chamber pressure is controlled by an independent circuit communicating with the same reservoir. Each circuit is composed by a pump, a check valve and a pressure-controlled valve. The latter can vary the pressure in the circuit proportionally to a command current.

The parameters of the hydraulic actuators are given in Table 5.1. The oil properties used are based on available industrial oils, while the actuator parameters are based on the actuator modelled in the Run2Rail project [5], and the actuators used in field application ([89], [90]). Subdued to further developments of the comfort actuators, more detailed models of each constituting component can be considered. These include several characteristics difficult to determine such as pressure losses, turbulent flow conditions, friction, and characteristics variable with the operating conditions such as oil temperature, oil viscosity, and bulk modulus [91].
CHAPTER 5. ACTUATION SYSTEMS

Table 5.1: Hydraulic comfort actuator key parameters

<table>
<thead>
<tr>
<th>Hydraulic Chamber</th>
<th>Stroke</th>
<th>140 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Chamber Diameter</td>
<td>28 mm</td>
</tr>
<tr>
<td></td>
<td>Rod Diameter</td>
<td>14 mm</td>
</tr>
<tr>
<td></td>
<td>A1</td>
<td>616 mm²</td>
</tr>
<tr>
<td></td>
<td>A2</td>
<td>462 mm²</td>
</tr>
<tr>
<td>Check Valve</td>
<td>Diameter</td>
<td>8 mm</td>
</tr>
<tr>
<td></td>
<td>Crack Pressure</td>
<td>0.5 bar</td>
</tr>
<tr>
<td></td>
<td>Max Opening Pressure</td>
<td>0.6 bar</td>
</tr>
<tr>
<td>Fluid Properties</td>
<td>Density</td>
<td>836 kg/m³</td>
</tr>
<tr>
<td></td>
<td>Viscosity</td>
<td>30 mm²/s</td>
</tr>
<tr>
<td></td>
<td>Bulk modulus</td>
<td>1.8e9 Pa</td>
</tr>
<tr>
<td>Ideal Pump</td>
<td>Flow Rate</td>
<td>2 l/min</td>
</tr>
</tbody>
</table>

5.2 Low frequency actuator - Steering

Low frequency actuators are used in the railway research field to solve the well-known trade-off between wheelset steering capability and hunting stability. These actuators have the main purpose of controlling the wheelset position in curves allowing a stiffer longitudinal primary suspension, thus enhancing the curving capability without compromising the vehicle maximum speed. Being the curve negotiation an almost quasi-static phenomenon, restrictions on the frequency operating range are less relevant in comparison to the comfort actuators. At the same time, requirements on safety and fail-safe mechanisms are more important for steering actuators than for comfort actuators due to derailment risk [92].

Hydraulic ([93]–[95]) and electro-mechanical ([96]–[99]) actuators are generally used in the research field of actuated solid wheelsets. Common configurations of the steering scheme are given in Figure 5.2 where Figure 5.2 (a) and (b) are referred as yaw torque control, while (c) as lateral force control. The scheme of Figure 5.2 (b) is the most favorable scheme that finds more applications in the steering control of solid wheelsets Paper A.

Being a non-commonly implemented problem, wheelset steering has found unique implementations to fit in standard bogie design. Some of these examples are given in Figure 5.3. In the solution provided by Park et al. in [100], a linkage actuated by one electro-magnetic actuator is steering each wheelset of a bogie. The solution proposed by Umehara et al. in [94] uses one hydraulic cylinder to steer both the wheelsets of a bogie, while the solution proposed by Hwang et al. in [96] links the same side of the two wheelsets on the same bogie with an electro-mechanical actuator.

The most common implementation of steering actuators is to have them
5.2. LOW FREQUENCY ACTUATOR - STEERING

Figure 5.2: Possible mechanical arrangements for actuated solid-axle wheelset Paper A

In parallel with passive suspensions. In this way, hunting stability can be studied separately from steering performances. Nevertheless, this approach puts a limit to the improvements that active steering may bring. The problem is particularly apparent for vehicles having a relatively high top speed running on a track with small radius curves. An example of this case is the Metro Madrid line 10 where the vehicle allowable speed is 120 km/h (Table 3.1) and the minimum curve radius is 100 m (with more than 30 % of the track with curve radius lower than 400 m). For such a system, a significant amount of actuation force may be required to steer the wheelset in a correct position due to a relatively high longitudinal suspension stiffness required to guarantee hunting stability at high speed.

Therefore, within the NEXTGEAR project [101], a non-linear spring behaviour is embedded in the more standard hydraulic actuator to create

Figure 5.3: Examples of active steered bogies: steered by two electromagnetic actuators [100] (Left), steered by one hydraulic actuator [94] (Centre), and steered by two electro-mechanical actuators [96] (Right)
CHAPTER 5. ACTUATION SYSTEMS

Figure 5.4: Actuator non-linear stiffness characteristics (Top), and systematic search of non-linear stiffness characteristics comparing $T_{\gamma}$ and critical speed (Bottom) \[6\]. The cyan circle highlights the selected span, high stiffness and low stiffness combination.

A new mechatronic suspension. The two-zone non-linear stiffness shown in Figure 5.4 (Top) is achieved by having disc springs inside the chambers of the actuator. The higher stiffness in the central zone provides hunting stability, while the lower stiffness in the external zone reduces the longitudinal suspension resistance force during curve negotiation. This solution is an alternative to the frequency-dependent bushing \[102\], where the stiffness is frequency-dependent instead of displacement-dependent. The central displacement span together with the high and low stiffness values are studied in \[6\]. Here, a parametric study embracing three spans, ten high stiffness values, and ten low stiffness values, is performed to achieve high critical speed and good steering capability. The critical speed was evaluated by
5.2. LOW FREQUENCY ACTUATOR - STEERING

Figure 5.5: Steering actuator: Scheme (Left), and its implementation in Simscape (Right)

running the vehicle on an ideal tangent track with an 8 mm single excitation event, while steering capability was analysed considering the $T\gamma$ value in the circular section of a 600 m curve.

The hydraulic configuration of the actuator (together with the disc springs arrangement) was designed and manufactured in cooperation with Dellner Dampers. The scheme of Figure 5.5 (Left) shows the cross-coupled actuators acting on the same wheelset. This configuration requires only two pressure-controlled valves to control the pressure in the actuator chambers. The model of the actuator is once more implemented in the Simscape-Hydraulics environment where the non-linear stiffness behaviour and the friction losses in the chambers are modelled too (Figure 5.5 (Right)).

The parameters of the developed model were tuned to reproduce tests and can be found in their entirety in the NEXTGEAR project report [101]. In Figure 5.6 the good agreement between the test results and the simulated actuator can be seen.
CHAPTER 5. ACTUATION SYSTEMS

Figure 5.6: Comparison of experimental testing and simulation results [101]: passive actuator test with imposed sinusoidal displacement of 2 mm with variable excitation frequency (Top-Left), passive actuator test with imposed sinusoidal displacement of 8 mm with variable excitation frequency (Top-Centre), passive actuator test with imposed sinusoidal displacement of 10 mm with frequency 0.1 Hz (Top-Right), and active actuator test with constrained ends and pressure-controlled valves with square waves of variable amplitude (Bottom)

5.3 Fully actuated vehicle model

The models of the actuators selected and described in this chapter are combined in a unique simulation environment in Simulink. Together with the structural characteristics of the vehicle described in Chapter 4[4] they create the fully actuated vehicle model shown in Figure 5.7. The co-simulation module simat guarantees the communication between SIMPACK and Simulink. The vehicle mechatronic model in this configuration can run on curved and tangent tracks, with and without track irregularities, and with vehicle acceleration and breaking. In the following two chapters development of the control algorithms is described, i.e. the dynamic ride comfort control approach, and the steering control approach.
Figure 5.7: Fully actuated vehicle model showing the key components and an overview of the dynamic ride comfort control and the steering control
Chapter 6

Ride Comfort Control

Ride comfort control is among the most studied problems in railway mechatronic systems. Among others, the low safety risk of its implementation \cite{89} and an immediate performance response (possible to evaluate control performance on a single run in experimental testing \cite{89,103,105}) may be found among the primary reasons for the intense studies in this topic.

6.1 Sky-hook control

Several control techniques can be applied to improve ride comfort. Among the model-based controls, Linear Quadratic Gaussian (LQG) and $H_\infty$ are advanced control techniques widely used in the comfort control field \cite{41}, \textbf{Paper A}. Nevertheless, their general complexity limits their application in field tests. In field tests one of the most used approaches is the sky-hook damping, or its modal equivalent. Sky-hook is a simple but effective control

![Sky-hook control principle (absolute velocity feedback).](image)

\textbf{Figure 6.1:} Sky-hook control principle (absolute velocity feedback). \textbf{Paper A}
approach for active vibration isolation and in its modal variation it represents the baseline performance that other types of controllers need to beat. Introduced by Karnopp [106] in 1995, sky-hook damping (also called absolute velocity feedback) solves the trade-off problem of passive suspensions between resonance frequency amplitude attenuation and higher frequencies amplitude magnification. The concept is illustrated in Figure 6.1 (Centre). The damper is moved from the original position (Figure 6.1 (Left)) to a new position connecting the mass to be isolated to a virtual constraint called “sky”. In this way, the dependency of the damping action from the ground (or disturbance) motion is removed. In practice it is obviously not possible to connect the mass to the “sky”. Therefore, a control force is introduced to produce the same effect as the desired “sky” configuration (Figure 6.1 (Right)). Ideally, the control force should be directly proportional to the absolute velocity of the mass to be isolated. In practice, absolute velocity is measured through integrated accelerations. The feedback of quasi-static components of the measured accelerations must be avoided, especially during curve negotiation. Consequently, the accelerations are high-pass filtered after (or together with) the integration.

Being the most simple and generally one of the most effective techniques, sky-hook damping in its modal configuration is applied at first to control the ride comfort of the innovative vehicle.

6.2 Modal sky-hook control

As mentioned above, the sky-hook approach can be extended to a modal sky-hook approach. Modal control is based on the assumption that the system to be controlled can be considered as a linear time invariant system and therefore an eigenvalue problem can be defined. The eigenmodes (or mode shapes) resulting from the eigenvalue problem are the ones targeted by the modal control principle. Depending on the number and placement of sensors and actuators, a certain portion of the eigenmodes of the system can be detected and/or controlled. The example of Figure 6.2 shows a modal control for bounce, pitch, and roll motions of a carbody. In this example, four measurement points (two in the front part of the vehicle and two in the rear part) are combined to isolate bounce, pitch, and roll motions and recombined into four actuation forces through the modes modal decomposition.

Modal sky-hook is attained when the controllers for each decomposed mode (Controller 1, 2, and 3 in Figure 6.2) are defined as the absolute velocity feedback principle shown in Figure 6.1 (Right). This approach has been used to control lateral ride comfort with carbody lateral and yaw motion decomposition in [10], vertical ride comfort with carbody bounce.
6.2. MODAL SKY-HOOK CONTROL

Figure 6.2: Example of modal control for a rail vehicle showing the input/output modal decomposition to detect and control three modes: bounce, pitch, and roll. Paper A

and pitch decomposition in [80], [105], [107], and vertical ride comfort with carbody bounce, pitch, and roll decomposition in [89]. It is important to mention that, as a general control technique, modal control can be extended above the sky-hook approach for vibration control. Regarding the railway application, modal control has been applied for wheelset steering in [108], [109].

A general formulation of the modal sky-hook control can be expressed as:

\[
\mathbf{u} = - \mathbf{[\phi]} [k] \mathbf{[\phi]}^T \int \ddot{\mathbf{z}} dt, \tag{6.1}
\]

where, \( \mathbf{u} \) is the input vector, \( \mathbf{[\phi]} \) is the mode shape matrix, \( [k] \) is the control gain matrix, \( \ddot{\mathbf{z}} \) is the acceleration measurement vector, while \( \int \cdot \) represents the combination of the integration function \((1/s)\) and the high-pass filter to neglect the accelerations quasi-static contribution. The mode shape matrix for the example of Figure 6.2 is a 4x3 matrix relating the four actuators with the three considered modes, while the control gain matrix is a 3x3 diagonal matrix where each diagonal term represents the control gain for that mode,

\[
[\phi] = \begin{bmatrix}
+1 & +1 & +1 \\
+1 & +1 & -1 \\
+1 & -1 & +1 \\
+1 & -1 & -1
\end{bmatrix}, [k] = \begin{bmatrix}
k_{Bounce} & 0 & 0 \\
0 & k_{Pitch} & 0 \\
0 & 0 & k_{Roll}
\end{bmatrix}. \tag{6.2}
\]
Figure 6.3: Vehicle mechatronic scheme (Top), and mode shape matrix of the vehicle for the measurement points shown in the mechatronic scheme (Bottom). In the latter, the green and black highlights are used to show the phase of each relative amplitude. Paper E

For the innovative vehicle, the first five rigid body modes need to be controlled, i.e. Roll 1 (or Sway), Yaw, Roll 2, Pitch, and Bounce (Figure 38).
6.2. MODAL SKY-HOOK CONTROL

Figure 6.4: Generalized modal control loop as applied to the innovative vehicle. Paper D

This is done with six actuators. The mechatronic scheme shown in Sub-section 3.3 Figure 3.5 is simplified in Figure 6.3 (Top) showing only the comfort actuators and the acceleration measurement points. Given the set of measurement points of Figure 6.3 (Top), the mode shapes shown in Figure 4.5 and Figure 4.6 are calculated from SIMPACK results for each mode $j$ as:

$$\varphi_j = x_{R}^j + R^j x_0 + \delta^j,$$

(6.3)

where, $\varphi_j$ is the mode shape vector, $x_{R}^j$ is the rigid translation vector, $x_0$ is the static coordinates vector, and $\delta^j$ is the flexible deformation vector. The mode shape vectors are scaled with respect to the maximum absolute value amplitude per mode giving the results shown in Figure 6.3 (Bottom). Each column of the mode shape matrix shows the relative movement between the measuring/actuation points where an absolute value equal to 1 shows the points that have the maximum movement while 0 indicates the points that don’t move. Signs express the relative direction between the points. For the same mode, a point showing 1 moves in one direction and another point showing -1 moves in the opposite direction. A colour scale is introduced to facilitate the visualization of the modes.

Given the configuration of the mechatronic vehicle, the modal sky-hook control foresees a mode shape matrix $[\varphi]$ of dimension 6x5 (six actuators and five modes to be controlled), and a control gain diagonal matrix $[k]$ of dimension 5x5. The example block diagram of Figure 6.2 is generalized by applying the formulation of Eq. (6.1) in Figure 6.4. Here, the input vector $u$ consists of the currents to be provided to the comfort actuators described in Section 5.1 and the output measurement vector $\ddot{z}$ is constituted by the actuators’ co-located measurement points on the carbody floor (Figure 6.3 (Top)).

Once the control strategy is defined, the control gain matrix $[k]$ must
be identified. The definition of control gains is commonly a trial and error process. To automate the process, a genetic algorithm (GA) optimization is carried out to identify the control gains that minimize the fit function,

$$F = \frac{1}{3} \left( \frac{N_{MV, front_{left}}}{2} + \frac{N_{MV, front_{right}}}{2} + N_{MV, centre} + \frac{N_{MV, rear_{left}}}{2} + \frac{N_{MV, rear_{right}}}{2} \right). \quad (6.4)$$

The fit function is defined as the mean value between front, centre, and rear positions $N_{MV}$. It is used to evaluate the performance of the control gains combination. GA optimisation seeks to reduce the average comfort index by minimizing the function $F$. A gradient-free optimization procedure is used here since it does not require the knowledge of the relation between the control parameters and the fit function. The optimization procedure is repeated for discrete vehicle speeds from 10 km/h to the admissible speed of 120 km/h with 10 km/h discretisation, giving 12 optimization scenarios. Therefore, for each considered speed a set of five control gains is found. For each set of control gains and vehicle speed, the vehicle is simulated to run on a tangent track of 1000 m with the so-called ERRI High track irregularities (110, 111) which are power spectrum-based signals that are generally accepted in Europe to simulate track irregularity when real data are not available. High level track irregularities are chosen to ensure the performance of the controlled vehicle in unfavourable track conditions.

In Figure 6.5 a schematic view of the Matlab-Simulink-SIMPACK connection used to perform the GA optimization is shown. A central Matlab structure governs the start and stop of the Simulink-SIMPACK co-simulation and collects the results from the SIMPACK results file. Once every control gain combination in one population is tested, the same Mat-
lab structure checks if the maximum number of generations is achieved. If not, it chooses the parents for reproduction, creates offspring, applies mutation, and starts a new set of simulations. If yes, the vehicle speed is set to the next one for which the gains need to be optimized and the procedure restarts.

The effectiveness of the optimized modal control is demonstrated in Figure 6.6 in the frequency domain. Here, the vertical acceleration power spectral density of the two-axle vehicle with passive dampers and the actively controlled one are compared for the running speed of 80 km/h. This speed is the most travelled one on Metro Madrid line 10.

In Figure 6.7 the detailed comfort evaluation applying the EN12299 standard is given for both the $N_{MV}$ ride comfort index and the vertical and lateral continuous comfort values. In Figure 6.7 (Top) the fit function of Eq. (6.4) is shown too. The modal control is capable of keeping the $N_{MV}$ value below the comfortable limit set by the EN12299 standard and drastically improves the performance of the innovative vehicle equipped with passive dampers. This is shown in Table 6.1, where the mean value of the comfort

<table>
<thead>
<tr>
<th>Speed range</th>
<th>10 - 30 km/h</th>
<th>40 - 60 km/h</th>
<th>70 - 90 km/h</th>
<th>100 - 120 km/h</th>
</tr>
</thead>
<tbody>
<tr>
<td>Passive $N_{MV}$</td>
<td>0.66</td>
<td>1.93</td>
<td>4.04</td>
<td>5.39</td>
</tr>
<tr>
<td>Active $N_{MV}$</td>
<td>0.20</td>
<td>0.72</td>
<td>1.35</td>
<td>2.07</td>
</tr>
<tr>
<td>Improvement</td>
<td>70 %</td>
<td>63 %</td>
<td>67 %</td>
<td>62 %</td>
</tr>
</tbody>
</table>
CHAPTER 6. RIDE COMFORT CONTROL

Figure 6.7: Optimization results: comfort index $N_{MV}$ (Top), continuous vertical comfort (Bottom-Left) and continuous lateral comfort (Bottom-Right). Paper D

index between the passive and active vehicles for four speed groups are compared.

The good performance of the modal control cannot only be judged by the $N_{MV}$ index. The continuous comfort value in lateral direction (Figure 6.7 (Bottom-Right)) shows low acceleration levels without any significant increase with speed while the value in vertical direction (Figure 6.7 (Bottom-Left)) increases with the vehicle speed. This is particularly evident in the carbody center. The difference between the vertical and lateral behavior may lead to a perceived poorer comfort in vertical direction due to the absence of an equal lateral counterpart. Therefore a strategy to further improve the vertical continuous comfort needs to be found.

6.3 Blended control

To solve the above mentioned problem of unbalance in vertical and lateral ride comfort, a more in-depth analysis of the vehicle resonance response can be done. As shown in Figure 6.3 for pitch and bounce modes, a strong interaction between the frame and the carbody is found. Specifically, the frame motion amplitudes are approximately half of the carbody motions amplitudes and they are in counter phase. Taking the pitch motion as
example, the front part of the carbody has a negative amplitude of -1, while
the same location of the frame has an amplitude of +0.45. This aspect may
influence the performance of the modal controller in vertical direction.

A blended control approach is therefore introduced. The blended control
is a modification of the modal control introduced in the previous section.
It is based on the assumption that the vertical comfort can be improved by
means of partially acknowledging the frame behaviour in the control loop.

The concept of blended control for vibration isolation purposes was in-
troduced by Alujević [112], [113]. Given two connected bodies, the upper
tone to be isolated and the lower one introducing disturbances, the blended
control proposed by Alujević foresees the possibility of improving the vi-
bration isolation of the upper body by sharing the velocity feedback with
the lower body by a portion $\alpha$. The blending parameter $\alpha$ is used to shift
the velocity feedback from the upper body to the lower body. A value of $\alpha$
equal to 0 produces absolute velocity feedback of the upper body, $\alpha$ equal
to 1 gives absolute velocity feedback of the lower body, while $\alpha$ equal to 0.5
represents relative velocity feedback between upper and lower body.

The blended control concept is modified to be applicable to the inno-
vative vehicle and its modal behaviour. Starting from the modal sky-hook
control described above, a portion $\alpha$ of the modal contribution of the frame
is subtracted from the modal contribution of the carbody. This solution
increases the complexity of the control strategy by two main aspects. The
first one is the need for knowledge of the frame modal contributions (mode
shapes), and the second is the need for additional sensors to measure the
frame accelerations. Given these last two conditions, the modal sky-hook
control of Eq. (6.1) can be rewritten to incorporate the frame contribution,

$$!u! = -[\varphi][k] \int \left( [\varphi]^T \ddot{z} - [\alpha] [\varphi_{frame}]^T \ddot{z}_{frame} \right) dt, \quad (6.5)$$

where, $[\alpha]$ is the blended contribution diagonal gain matrix, $[\varphi_{frame}]$ is the
mode shape matrix of the frame contribution, and $\ddot{z}_{frame}$ is the frame accel-
erations measurement vector. In Figure 6.8 the blended control is schemat-
ized highlighting the differences between the modal control and the blended
control approaches.

The GA procedure previously described needs to be reapplied due to the
redefinition of the control approach. As the number of optimization param-
eters increases significantly, a sensitivity analysis must be performed on the
modal control to identify the control parameters that have the largest impact
on the vehicle’s ride comfort in lateral and vertical directions. The multi-
plicative dimensional reduction method (M-DRM) [114] is applied. Given N
parameters to investigate with M possible variations, the M-DRM requires
only NxM evaluations, while other methods may require more evaluations
Figure 6.8: Schematic block diagram of modal and blended controllers. Addition or removal of the red dashed block let the controller pass from modal control to the blended control, while the addition of the blue filled block let pass blended control to its refined version with filtering effect. The control gain of mode $j \ k_j$ and the blended control gain of mode $j \ \alpha_j$ are function of the vehicle speed. Paper E

up to a maximum of $N^M$ for a full factorial study. The method can suffer from inaccuracies if the number of values $M$ for each parameter is small. Nevertheless, as shown in [14], a general trend can be identified even with a small number of variations $M$.

The M-DRM method is applied to the vehicle running at constant speed with the focus on the weighted vertical and lateral acceleration in the front, centre and rear part of the carbody separately. The results of Figure 6.9 show a clear distinction between the control parameters importance in vertical and lateral directions. The vertical direction is mainly influenced by pitch and bounce control, while the lateral direction is influenced by roll 1, yaw, and roll 2 control.

The results of the sensitivity analysis highlight the possibility of reducing the parameters to be optimized for the blended control approach. Since the focus of the blended control is to improve the performance for the vertical ride comfort while maintaining the already good results for the lateral ride comfort, only the parameters that influence the vertical comfort need to be redefined. Therefore, the number of optimization parameters to be optimized for the blended control reduces to four, i.e. pitch and bounce gains for the modal contribution ($k_4$ and $k_5$), and pitch and bounce gains ($\alpha$) for the blended contribution. Moreover, the fit function of Eq. (6.4) can still be used. Since only the parameters influencing vertical comfort are changed during the optimization procedure, a variation in the fit function...
6.3. BLENDED CONTROL

Figure 6.9: M-DRM sensitivity analysis results. Vertical weighted acceleration cumulative sensitivity (Top), and lateral weighted acceleration cumulative sensitivity (Bottom). Paper E

(of Eq. (6.4)) will be the result of changes in the vertical behaviour only.

In Figure 6.10 the performance of the blended control (BC) is compared with the ones of the modal control (MC) considering the front and rear part of the carbody as well as the comfort index $N_{MV}$ and the weighted vertical and lateral accelerations. As expected from the sensitivity analysis, the influence of the blended control on the lateral weighted acceleration is negligible. Differences appear in vertical direction only. The blended approach shows large improvement with respect to the modal control in the front part of the vehicle with a minimum improvement of 10 % and a maximum improvement of 19 %. In contrast, the central part of the carbody experiences a deterioration of the ride comfort at higher vehicle speeds.

To further investigate the potential of the blended control, the discretised speed varying control gains defined with the GA optimisation procedure are approximated with a continuous polynomial function. On the one hand, this approach will produce sub-optimal control gains at the designed vehicle speeds, but, on the other hand, the polynomial interpolation makes it possible to extend the control range to any vehicle operational speed. In Figure 6.11 the polynomial approximations as function of the vehicle speed are shown.

The performance of the two designed controllers can now be compared over an extended vehicle speed range (from 10 km/h to 120 km/h with
 CHAPTER 6. RIDE COMFORT CONTROL

Figure 6.10: Optimization results comparing modal and blended controllers. The improvements of the blended approach in percent is given below each graph. $N_{M V}$ value (Top), weighted vertical acceleration (Centre) and weighted lateral acceleration (Bottom). Paper E

Figure 6.11: Approximation functions for both modal and blended control. Paper E
Figure 6.12: Modal against blended control in frequency domain (Top), modal against refined blended control in frequency domain (Centre), and modal against refined blended control comparing $N_{MV}$ and weighted vertical acceleration (Bottom). Paper E
a 1 km/h interval). For each considered speed a power spectral density analysis is performed on the carbody accelerations in vertical direction. The difference in dB between blended and modal control power spectral density responses are shown in Figure 6.12 (Top). A negative value of the scale shows where blended control performs better than modal control producing a lower amplitude in the frequency domain at that frequency and speed.

It is clear that for the central part of the vehicle the improvements are limited in the frequency region between 5 and 14 Hz. The blended control therefore needs a refinement. Ideally, the blended controller should be kept active up to 14 Hz while modal control should be required above 14 Hz. Being the major difference between modal and blended control the introduction of the frame contribution in the control loop, a second order low-pass filter is applied to the measured frame accelerations to try to neglect their contribution in the frequency region above 14 Hz. The low-pass filter,

$$\phi = \frac{\omega_c^2}{s^2 + 2\xi \omega_c + \omega_c^2},$$

(6.6)
is applied to each modal contribution of the frame acceleration. Here, $\xi$ is the damping ratio (set to 1/2 to avoid overshoot) and $\omega_c$ is the cut-off frequency which has been set to 18 Hz.

The benefits of the refined version of the blended control can be seen from both the power spectral density analysis and from the perspective of ride comfort. In the frequency domain (6.12 (Centre)), the filtering effect is significant in the centre part of the carbody, reducing the area of positive values (red zone) above 14 Hz. In terms of ride comfort (6.12 (Bottom)), the benefits achieved by the original blended control are kept (with a negligible deterioration) in the front part of the vehicle while both $N_{MV}$ and weighted vertical acceleration are improved in the centre part of the vehicle. The refined version of the blended control has the potential of evening out the difference between vertical and lateral passenger comfort. The difference between vertical and lateral comfort at 80 km/h reduces from 0.08 m/s$^2$ to 0.03 m/s$^2$ in the front location and from 0.12 m/s$^2$ to 0.08 m/s$^2$ in the centre location when the refined blended control is applied instead of the modal version.

6.4 Conclusions on comfort aspects

The definition of a strategy to improve ride comfort is one of the main aspects that needs to be addressed to make the two-axle vehicle a competitive alternative to standard bogie solutions. The blended control in its refined
version can offer this solution. Nevertheless, its implementation is subdue to two main assumptions. The first one is the knowledge of the complete mode shape matrix of the vehicle (including the frame modes), and the second is the possibility of measuring the frame accelerations in the required key position.

Provided these assumptions are satisfied, the blended control can bring the ride comfort of the vehicle to acceptable levels according to the EN12299 evaluation method and reduce the spread between lateral and vertical comfort. Moreover, the definition of polynomial interpolation of the control gains makes it possible to apply the blended controller with any vehicle speed in the operational speed range. This takes forward the design of the vehicle towards its fully actuated configuration.
Steering control of solid axle wheelsets can provide substantial benefits but is affected by some key difficulties that need to be overcome. Among them, as previously mentioned, safety concerns limit the number of conducted track tests and might thus prohibit implementations of wheelset steering technologies. In literature, proposals for processes and strategies to provide a safety environment for wheelset steering can be found \[92\], \[115\]. Moreover, as far as control requirements are concerned, measurement of key variables to control, and stability of the controller itself over a variety of conditions, are aspects that need to be addressed.

### 7.1 Stability of feedback approaches

Several steering approaches can be used. Their effectiveness is demonstrated on linear vehicle models \[15\], \[16\], linear vehicle models with non-linear conicity \[116\], \[117\], and fully non-linear vehicle models \[17\], \[18\] using contact force evaluation or the $T\gamma$ method.

Assuming that the control variables are known or that they can be determined, feedback approaches are generally implemented for steering control purposes \[41\], \textbf{Paper A}. Nevertheless, as described for example in \[118\], feedback approaches may suffer from instability. Due to the non-linear nature of the wheel-rail contact and the large amount of possible operating conditions (for example vehicle speed variation, curved and tangent tracks, track irregularities) that the railway vehicle must be able to cope with, control stability is of primary importance. Several studies have been performed to ensure the stability of feedback approaches concerning the steering control problem. Self-tuning and $\Gamma$-stable linear quadratic regulators (LQR) are introduced in \[119\] respectively \[120\]. $H\infty$ control is applied on bogie and two-axle vehicles in \[121\] respectively \[18\]. Lastly, a robust PI controller
CHAPTER 7. STEERING CONTROL

Figure 7.1: Vehicle mechatronic scheme for steering control. Paper C

is designed for different wheelset configurations in [17].

Given the configuration of the innovative two-axle vehicle (summarized only for steering purpose in Figure 7.1), a simple proportional-integral-derivative (PID) control approach is introduced to verify the applicability of the perfect steering strategy. The perfect steering strategy [122] is achieved by having the longitudinal creep forces on wheels on the same axle equal to zero (if no traction or braking force is applied) and, at the same time, by having equal lateral creep force on each wheelset in the same bogie. Several interpretations can be made to satisfy the given definition. With a linearized wheel-rail contact theory it is possible to derive two conditions on the wheelset position [16],

\[ y_{w1,2} = -b_0 \frac{r_0}{\lambda_{eq} R}, \quad (a), \quad \psi_{w1} - \psi_{w2} = 0 \quad (b), \quad (7.1) \]

where, \( y \) is the lateral displacement of the wheelset from the track centre line, \( b_0 \) is the semi-wheels distance, \( r_0 \) is the nominal wheel radius, \( \lambda_{eq} \) is the equivalent conicity, \( R \) is the curve radius, and \( \psi \) is the wheelset attack angle (wheelset deviation from radial position with respect to the track). Subscripts 1 and 2 refer to the leading respectively trailing wheelset on the same bogie/two-axle vehicle. Fulfillment of these two conditions should guarantee an almost negligible \( T\gamma \) value during curve negotiation. The perfect steering condition of Eq. (7.1) (a) is dependent on the negotiated curve radius. For a solid-axle wheelset and the actuators placement of Figure 7.1, it is not possible to position control simultaneously the lateral displacement and the attack angle due to their intrinsic relation. Therefore, only one condition at a time can be achieved leaving one condition unsatisfied in the perfect steering strategy. Even if simultaneous compliance with Eq. (7.1) (a) and Eq. (7.1) (b) can not be achieved, the application of either Eq.
7.1. STABILITY OF FEEDBACK APPROACHES

(a) or Eq. (7.1) (b) as reference signals can be beneficial bringing the wheelset closer to the perfect steering condition.

Lateral displacement and attack angle are not easily measurable variables, but assuming they are obtainable (Section 7.2), it is possible to design a simple PID control that, during curve negotiation, aims at reaching zero steady-state error \( e(t) \) between the wheelset lateral displacement \( y_w \) and the reference condition of Eq. 7.1 (a) \( (y_w^{ref}) \) by introducing a steering force

\[
F_s = \pm \left( K_p e(t) + K_i \int e(t) dt + K_d \frac{e(t)}{dt} \right), \quad e(t) = y_{w1,2} - y_{w1,2}^{ref}.
\] (7.2)

A set of running cases is defined to investigate the stability of the feedback system. In Table 7.1 the 12 running cases are shown, embracing four curves and three non-compensated lateral accelerations (NLA) each. The minimum speed for all curves is set to 10 km/h while the largest curve radius is calculated to guarantee the NLA requirement for the vehicle maximum speed. The tuning of the PID controller is done with the Ziegler-Nichols’ tuning method [123] on the 600 m curve at the highest speed. Here, the S1002 wheel profile in combination with the UIC60 rail profile, at 1:40 rail inclination, and a track gauge of 1.435 m produces an equivalent conicity of 0.18 at 3 mm wheelset lateral displacement.

Table 7.1: Operational cases for the feedback system evaluation

<table>
<thead>
<tr>
<th>Radius [m]</th>
<th>Cant [mm]</th>
<th>Vehicle speed [km/h]</th>
<th>NLA variable</th>
<th>NLA 0 m/s²</th>
<th>NLA 0.65 m/s²</th>
</tr>
</thead>
<tbody>
<tr>
<td>250</td>
<td>150</td>
<td>10</td>
<td>56</td>
<td>73</td>
<td></td>
</tr>
<tr>
<td>400</td>
<td>100</td>
<td>10</td>
<td>58</td>
<td>82</td>
<td></td>
</tr>
<tr>
<td>600</td>
<td>80</td>
<td>10</td>
<td>64</td>
<td>96</td>
<td></td>
</tr>
<tr>
<td>1066</td>
<td>60</td>
<td>10</td>
<td>74</td>
<td>120</td>
<td></td>
</tr>
</tbody>
</table>

The feedback control so defined is not stable at low speed (Figure 7.3 (Top-left)). A simple proportional reduction of the control action (about 80%) is applied to meet good performance and stability at the lowest speed, without changing the PID control parameters. Despite a solution can be found, reaching a satisfactory behavior at low speed implies degrading the performances at higher ones (Figure 7.3 (Top-centre and Top-right)).

A scaling strategy of the control action can be applied to meet the requirements at low and high speed for the operational cases of Table 7.1. The scaling strategy is split into two contributions, one handling speed variation...
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Figure 7.2: Scaling functions for the adaptive scaling PID control: vehicle speed scaling (Left), curvature scaling (Right). Paper B

\( (G_v) \) and one handling curve variation \( (G_\rho) \) to adapt to the whole defined curve batch. The steering force \( F_s \) is scaled by the scaling functions \( G_v \) and \( G_\rho \) shown in Figure 7.2 to produce a new steering force,

\[
F_{s,\text{scaled}} = G_v G_\rho F_s.
\]  

(7.3)

Results of the adaptable scaling functions can be seen for the 600 m curve in Figure 7.3 (Top), where the scaled PID shows good performance and stability for both low and high speed. On the operational cases defined above, acceptable results are achieved for all the studied cases (Figure 7.3 (Bottom)) showing a zero steady state tracking error (and thus a compliance with the requirement of Eq. (7.1) (a)) in the circular part of the negotiated curves.

The simplified adaptable scaling control law described above is enough to satisfy the robustness requirements in the small sample of curves and speed combinations of Table 7.1. The approach can be extended to any vehicle speed in between 10 km/h and 120 km/h and potentially on an extended curve range. Nevertheless, the robustness of the controller over conicity variation is not guaranteed. This phenomenon is of great importance because wear of wheels and rails tends to change the conicity. In most cases the conicity increases with increased wear. In Figure 7.4, an explicative example is given. Here, the vehicle controlled with the adaptable scaling PID is run over the 400 m curve at two vehicle speeds, 10 km/h (NLA = -0.63 m/s\(^2\)) and 82 km/h (NLA = 0.65 m/s\(^2\)). The conicity is changed by changing rail inclination and track gauge without changing the controller definition. The controller fails in producing reliable results when the conicity is changed and, in the example, for 10 km/h and conicity 0.40 it causes the vehicle to derail. A refined feedback control may manage the
7.1. STABILITY OF FEEDBACK APPROACHES

Figure 7.3: Leading wheelset lateral displacement of the controlled vehicle: results comparing the standard PID and the adaptive scaling PID (Top), and results of the adaptive scaling PID on the cases defined in Table 7.1. Paper B

Figure 7.4: Leading wheelset lateral displacement with the adaptable scaling PID controller designed on equivalent conicity $\lambda_{eq} = 0.18$ applied to conicities $\lambda_{eq} = 0.01$ and $\lambda_{eq} = 0.40$ for the vehicle negotiating a 400 m curve: vehicle speed 10 km/h (Left) and 82 km/h (Right). Paper C
conicity variation. Nevertheless, the example, chosen on purpose, indicates the difficulties that must be overcome when applying a feedback control on a vehicle operating on varying conditions.

7.2 Feedforward controller

As shown, instability of feedback control can significantly affect the performance (and safety in the example of Figure 7.4) of the mechatronically steered vehicle. Nevertheless, as briefly mentioned above, control robustness is not the only problem affecting feedback systems. As in the above example, feedback systems require the knowledge (or measurement) of the variables to be controlled. The wheelset position with respect to the track is among the variables that are most beneficial to be controlled but very difficult to measure. Most of the solutions to overcome measurement issues are based on Kalman filters such as: linear Kalman filter [124], extended Kalman filter [120], unscented Kalman filter [125], and constrained unscented Kalman filter [126]. Another approach suggests camera detection in combination with deep leaning algorithms in [127]. The mentioned methods often require high computational demand or a high number of data points for the training.

A feedforward approach has the potential to overcome the stability and measurement issues by introducing steering forces based on more easily measurable variables. Yaw stiffness and curvature were use by Shen et al. in [116], while cant deficiency and curvature where used by Braghin et al. in [128] to develop a control force look-up table for secondary yaw control of a tilting vehicle. Cant deficiency or non-compensated lateral acceleration (NLA) and curvature are reliable measurements. NLA can be estimated by an accelerometer in the running gear which is widely used in tilting trains [42]. Curvature can be derived by a yaw rate gyroscope in the carbody centre $\dot{\psi}_c$ (Figure 7.1) combined with a tachometer to obtain the vehicle speed $v$,

$$\frac{1}{R} = \frac{\dot{\psi}_c}{v}.$$  \hspace{5cm} (7.4)

To evaluate the performance of the system on a wide range of operating conditions and develop a robust and effective feedforward control, the running cases of Table 7.1 are extended to embrace seven curves and 17 speeds per curve (Figure 7.5 (Top)). The maximum speed for each curve corresponds to an NLA of 0.65 m/s$^2$ while the minimum one is set to 10 km/h for all curves, allowing for a comprehensive study of the vehicle performance during curve negotiation. For each curve the speeds are given by a linear distribution of eight speeds between 10 km/h and the balanced speed (NLA = 0 m/s$^2$), the balanced speed, and a linear distribution of eight
7.2. FEEDFORWARD CONTROLLER

Figure 7.5: Running cases for the development of the feedforward control approach: speed distribution [km/h] as function of curve radius and NLA (Top), and equivalent conicity as function of wheelset lateral displacement (Bottom). Paper C

speeds between the balanced speed and the maximum speed for that curve. To tackle the problem of conicity variation, three conicity profiles are used during the development phase. The equivalent conicity for each profile at 3 mm displacement (0.01, 0.18 respectively 0.4) is obtained by changing rail inclination and track gauge using S1002 and UIC60 wheel and rail profiles. An equivalent conicity of 0.3 is introduced to validate the capability of the feedforward approach to adapt to conicity variation (Figure 7.5 (Bottom)).

The controller has been designed through a comparative study of five different feedback approaches with nine possible combinations of leading and trailing wheelset reference signals. The combinations are a consequence of
the impossibility of simultaneously controlling the solid wheelset lateral position and its yaw angle. In Table 7.2 the nine combinations used during the comparative study are summarized. As per the perfect steering control signal of Eq. (7.1), the nine combinations of Table 7.2 are based on the assumptions that controlling either the wheelset lateral displacement or the yaw angle of the same wheelset will bring the wheelset to a generally better steering position, even if the theoretical requirement is not completely fulfilled.

**Table 7.2:** Leading (subscript 1) and trailing (subscript 2) wheelset steering reference signals defining the nine combinations used for the comparative study defining the feedforward controller. Each row defines one of the five feedback approaches, while each column gives the possible combination. As example, B2 corresponds to perfect steering where the yaw angles of the wheelset are controlled to be equal (Eq. (7.1)).

<table>
<thead>
<tr>
<th>Feedback Approach</th>
<th>Combination 1</th>
<th>Combination 2</th>
<th>Combination 3</th>
<th>Combination 4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Radial steering (A)</td>
<td>$\phi_{w1,2}^{ref} = 0$</td>
<td>$\phi_{w1}^{ref} - \phi_{w2}^{ref} = 0$</td>
<td>$\phi_{w1}^{ref} - \phi_{w2}^{ref} = 0$</td>
<td>$\phi_{w1}^{ref} - \phi_{w2}^{ref} = 0$</td>
</tr>
<tr>
<td>Perfect steering (B)</td>
<td>$y_{w1,2}^{ref} = -b_0 \frac{x_{w1}}{R}$</td>
<td>$y_{w1}^{ref} = f_1 \left( \frac{1}{\pi} \right)$</td>
<td>$y_{w1}^{ref} = f_2 \left( \frac{1}{\pi} \right)$</td>
<td>$y_{w1}^{ref} = f_3 \left( \frac{1}{\pi} \right)$</td>
</tr>
<tr>
<td>$T_\gamma$ based (C)</td>
<td>$\phi_{w1,2}^{ref} = f_1 \left( \frac{1}{\pi} \right)$</td>
<td>$y_{w1,2}^{ref} = f_2 \left( \frac{1}{\pi} \right)$</td>
<td>$\phi_{w1}^{ref} = f_1 \left( \frac{1}{\pi} \right)$</td>
<td>$\phi_{w1}^{ref} = f_2 \left( \frac{1}{\pi} \right)$</td>
</tr>
<tr>
<td>NLA based (D)</td>
<td>$\phi_{w1,2}^{ref} = \frac{\sin}{\pi R_{\gamma}} NLA$</td>
<td>$\phi_{w1,2}^{ref} = \frac{\sin}{\pi R_{\gamma}} NLA$</td>
<td>$\phi_{w1,2}^{ref} = \frac{\sin}{\pi R_{\gamma}} NLA$</td>
<td>$\phi_{w1,2}^{ref} = \frac{\sin}{\pi R_{\gamma}} NLA$</td>
</tr>
<tr>
<td>Combined (E)</td>
<td>$\phi_{w1,2}^{ref} = 0$,</td>
<td>$\phi_{w1,2}^{ref} = 0$,</td>
<td>$\phi_{w1,2}^{ref} = 0$,</td>
<td>$\phi_{w1,2}^{ref} = 0$,</td>
</tr>
<tr>
<td></td>
<td>$y_{w1,2}^{ref} = -\beta \left( \frac{1}{\pi} \right)$</td>
<td>$y_{w1,2}^{ref} = -\beta \left( \frac{1}{\pi} \right)$</td>
<td>$y_{w1,2}^{ref} = -\beta \left( \frac{1}{\pi} \right)$</td>
<td>$y_{w1,2}^{ref} = -\beta \left( \frac{1}{\pi} \right)$</td>
</tr>
</tbody>
</table>

The $T_\gamma$ values in the circular part of the curve and the required actuation force are collected for each of the nine tested combinations. A cost function,

$$C = \frac{1}{\alpha_A} (T_{\gamma,1} + T_{\gamma,2}) + \frac{1}{\alpha_B} (|F_1| + |F_2|) + \frac{1}{\alpha_B} \left( T_{\gamma,1,L}^{2nd} + T_{\gamma,1,R}^{2nd} + T_{\gamma,2,L}^{2nd} + T_{\gamma,2,R}^{2nd} \right),$$

is established to evaluate which feedback signal produces the lowest wear number $T_\gamma$ with the lowest actuation force. The minimum of $C$ among the nine feedback approaches guarantees that for a specific conicity, curve radius, and NLA combination, the lowest combination of $T_\gamma$ values, actuation forces and presence of a second contact point is achieved. In Eq. (7.5), the subscripts 1 and 2 stand for leading respectively trailing wheelset, the subscripts L and R stand for left respectively right wheel, and the superscript $2nd$ represents the second contact point. The $\alpha$ coefficients are hyperparameters that penalize each quantity of the cost function.
The results of the cost function applied to the three different conicities independently are given in Figure 7.6. Generally, the perfect steering approaches (B) show low presence of a second contact point but, at the same time, they require the highest actuation force. A similar behaviour is shown by approaches $T\gamma$ based (C) while the combined approach (E) shows among the smallest actuation forces. In the overall 357 running cases, radial steering (A) has been chosen 1 time, perfect steering (B), 160 times (B1 64 and B2 96 times), $T\gamma$ based approach (C), 77 times (C1, C2, C3 and C4 0, 34, 1 respectively 42 times), NLA based approach (D), 5 times and the combined approach (E), 114 times. For the innovative vehicle, if a feedback approach is to be considered, it is less beneficial to control the trailing wheelset to a pre-specified attack angle than to let it be uncontrolled. Controlling its lateral position (approaches B1, C2, C4, E) or applying the same attack angle as the leading wheelset (B2) gives better results. It is important to underline that there is no best choice of feedback approach that covers all the running cases. The lack of a clear choice for a feedback system strengthens the potential benefits provided by a feedforward control approach for the innovative vehicle.

The cost function is applied separately to each conicity and each running case giving three steering force distributions for the leading wheelset and three for the trailing wheelset as function of curvature and NLA. The three force distributions per wheelset are averaged to create one unique force distribution capable of handling the three conicity cases. A least square approximation is used to approximate the determined force distribution and extend the applicability range to any curve and vehicle speed. The
CHAPTER 7. STEERING CONTROL

Figure 7.7: Feedforward control architecture. Paper C

Figure 7.8: $T\gamma$ values in the circular part of the curve for the studied case comparing the vehicle without active control (NC) and the one with feedforward control (FF): leading wheelset (Top) and trailing wheelset (Bottom). Paper C
steering force surface as function of curvature and NLA is determined by the simple polynomial function,

\[ F = k_1 NLA \frac{1}{R} + k_2 \frac{1}{R} + k_3 NLA + k_4, \]  

(7.6)

where the coefficients \( k_j \) are different for the leading and trailing wheelset producing a different command steering force for the two wheelsets.

In Figure 7.7 the feedforward control strategy is schematized. The two steering forces \( F_1 \) and \( F_2 \) for leading respectively trailing wheelset are determined based on the input NLA and the absolute value of the curvature. The forces are then assigned a direction based on the direction of the curve, positive for a right-hand curve \((1/R>0)\) and negative for a left-hand curve \((1/R<0)\). Additionally, two conditions guarantee that no actuation force is delivered if the curve radius is greater than 2000 m and inverse actuation is avoided by imposing \( F>0 \) for all conditions.

The feedforward controller so defined is sub-optimal for each of the design points (conicity, speed, and curve) but shows great robustness against a variety of conditions. In Figure 7.8 the controller performance is shown on the design conicities showing great improvement with respect to the innovative vehicle with passive longitudinal suspension. Moreover, the controller, without any further modifications, is capable of running with the validation

![Figure 7.9](image-url): Time simulation results for the vehicle running on a double s-curve track with speed, curvature and cant variation with the validation equivalent conicity 0.3.
conicity of 0.3. In Figure 7.9, the controller is shown to be robust against the track irregularities disturbance, curve radius, cant, and speed variation. Here, the $T\gamma$ values for the leading and trailing wheelsets are presented with and without track irregularities with the vehicle running with the validation conicity 0.3 showing overall low energy dissipation.

The feedforward steering can now be integrated with the steering actuator developed for the innovative vehicle (Figure 7.10). Here, the command forces are limited to 40 kN. In the control module (Centre of Figure 7.10), the required steering command forces are translated to current signals to be provided to the actuators pressure-controlled valves.

### 7.3 Effects on wheel wear

Despite the good performance in terms of energy dissipation $T\gamma$ and robustness of the developed feedforward controller, the performance of the controlled vehicle in terms of wheel wear evolution is unknown. The latter, defines a milestone in the evaluation of the designed vehicle, representing a performance check of the vehicle design applicability.

As shown, $T\gamma$ is an extremely useful engineering quantity that can be used during control development and generally as performance evaluation tool due to its simplicity of calculation given standard outputs from multi-body simulations. However, $T\gamma$ can suffer in predicting the wheel wear evolution because of the lack of information about the local stress conditions in the contact between wheel and rail [20], [24].

Among other methods, the KTH wear model based on Archard’s wear map [21], [129] can be used. The method involves an iterative process...
7.3. EFFECTS ON WHEEL WEAR

Figure 7.11: Statistical analysis for the curve cases of Metro Madrid line 10. Nominal track curvature (A), s-curve (B), curve families and occurrence for curve radii greater then 200 m (C), NLA clustering (D). Paper F

where a specific railway line is analysed, and Archard’s wear coefficients are tuned. The method is effective in reproducing experimental results and it was remarkably used by Fu et al. in [95] in combination with active steering.

The first step in wear evaluation with the KTH wear model is to select and analyse a specific railway line. Being the designed mechatronic vehicle supposed to be an alternative for the metro vehicle S8000, it appears reasonable to apply the KTH wear model on Metro Madrid line 10. The line is statistically analysed to define load collectives. The load collectives (or load cases) are needed to reduce the computational burden required to perform the wear evaluation. At the same time, the load collectives give the possibility of analysing the results over a reduced number of cases. The track is first divided into tangent sections and curve sections. The curve sections statistical analysis is shown in Figure 7.11 where curve families are defined. In Figure 7.11 (C), the number of curves with a curve radius within the curve families bound is shown. Line 10 is characterized by a large amount of small radius curves (R ≤ 400 m). The K-means clustering method [130] is used to determine subcategories in each curve family to capture the information related to vehicle speed on curve (Figure 7.11 (D)). The S-curve of Figure 7.11 (B) represents an outlier that is considered separately. The tangent track section has been split into two halves, one with the vehicle travelling at constant speed and one with braking. For each load case, the representative share is calculated as the ratio between the cumulative
length of the load case (total distance that the vehicle travels on line 10 for a specific load case) and the total length of line 10 (approximately 40 km).

**Table 7.3:** Summary of load collective and representative shares of Metro Madrid line 10

<table>
<thead>
<tr>
<th>Cumulative length [m]</th>
<th>Representative share [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>S-curve</td>
<td>132</td>
</tr>
<tr>
<td>200 &lt; R ≤ 300</td>
<td>5788</td>
</tr>
<tr>
<td>300 &lt; R ≤ 400</td>
<td>7045</td>
</tr>
<tr>
<td>400 &lt; R ≤ 600</td>
<td>3893</td>
</tr>
<tr>
<td>600 &lt; R ≤ 800</td>
<td>2193</td>
</tr>
<tr>
<td>800 &lt; R ≤ 1200</td>
<td>1005</td>
</tr>
<tr>
<td>1200 &lt; R ≤ 2000</td>
<td>804</td>
</tr>
<tr>
<td>Tangent</td>
<td>18418</td>
</tr>
</tbody>
</table>

The results of the track analysis are summarized in Table 7.3 for the curve families and tangent track. The proportion of line 10 sections that have a curve radius smaller than 600 m is 42.9 % while the tangent track is 46.8 %. Therefore, it is clear that the steering capability of the mechatronic vehicle can significantly influence the performance.

The feedforward controller described in Section 7.2 is robust but not dedicated to a specific problem. A dedicated feedforward steering approach is considered to investigate possible benefits of a line dedicated controller that relies on vehicle-infrastructure communication instead of on-board vehicle measurements. The dedicated control relies on the assumption that perfect knowledge of the track is known beforehand. For each of the curve load cases, it is possible to find steering command signals for both leading and trailing wheelset that can optimize the steering performance of the vehicle. Such a dedicated controller loses the generality of the robust feedforward controller in favour of an on-purpose single-objective controller.

For each curve load case and each wheelset, three parameters of the command signals are optimized with firefly algorithms [131], [132]. These are, entering transition curve preview (the distance that the force command needs to be activated before the actual start of the entering transition curve), the exiting transition curve preview, and the actuation force in the circular curve (Figure 7.12 (Left)). The optimization function is once again derived from the $T\gamma$ values. The sum of the integral of the leading and trailing wheelsets $T\gamma$ values over the simulation time is used as optimization function (Figure 7.12 (Right)). A low value of the function means low energy dissipation during curve negotiation, and therefore, a better steering capability. A procedure similar to the one illustrated in Figure 6.5 is used to
7.3. EFFECTS ON WHEEL WEAR

Figure 7.12: Overview of the optimization process for the dedicated steering controller: optimization parameters (Left), and optimization function showing a schematic of the leading and trailing wheelsets wear number (Right). Paper F

create the Matlab-Simulink-SIMPACK connection where instead of genetic algorithms, firefly algorithms are used.

After an initial tuning phase of Archard’s wear coefficients to fit measured worn wheels of the S8000 vehicle, the feedforward controller developed in the previous section and the on-purpose optimized controller are evaluated using the KTH wear model. As shown in Figure 7.13, the wear depth (defined as the difference in vertical direction between the original S1002 wheel and the worn wheel) is significantly lower for the active vehicle in comparison to the S8000 vehicle. This is valid for both controllers. The original feedforward steering controller shows once more its robustness being capable of handling previously undefined wheel profile variation. All the worn profiles are compared for the wheel re-profiling distance of the S8000 vehicle of 150000 km.

A quantification of the improvement can be achieved by looking at the approximate lost volume due to wear,

\[
Lost\ Volume = 2\pi r_0 \int_{y_{in}}^{y_{end}} W_d dy,
\]  

where \(y_{in}\) and \(y_{end}\) are the initial respectively final profile lateral coordinates, \(W_d\) is the wear depth as function of the lateral wheel coordinate, and \(r_0\) is the nominal wheel radius. In Figure 7.13 (Right) it can be seen that the original controller improves the performance with more than 60 % compared to the performance of the standard vehicle, and the optimized dedicated control improves the performance with above 70 %. By computing the lost volume for each individual case of the load collectives, the reason of this improvement can be understood. As shown in Figure 7.14, the controlled vehicle shows a lower lost volume for all the curve load cases.
In particular, for the first two curve families (row two and three of Table 7.3 and highlighted in Figure 7.14) the benefits of the innovative steered vehicle are substantial. These two families are characterized by a small curve radius (R<400 m), and a high representative share (32.6%).

The results in terms of wheel wear show that the mechatronic vehicle can be of substantial benefit, potentially extending the wheel life above the current 150000 km wheel reprofiling interval. Moreover, it shows that the feedforward controller based on on-board measurements can cope with wheel profile evolution without further modification. Additionally, the results highlight the possibility of adopting a network-dedicated control relying more on vehicle infrastructure interaction than vehicle measurements. The approach can potentially reduce the number of additional sensors in the vehicle. However, being dedicated to a specific network, the on-purpose controller loses generality of applicability introducing the necessity of multiple line-dedicated controllers.
7.4. Conclusions on steering aspects

The last piece that had to be designed was a suitable steering strategy to overcome the poor steering performance of the innovative vehicle. The designed feedforward controller has the potential to overcome the stability issues introduced with feedback approaches. The simple feedforward control can cope with track irregularity excitation and a variety of standard operational variations such as track curvature, cant, and vehicle speed. At the same time it produces low energy dissipation $\gamma$ given few on-board measured inputs. Additionally, the controller may effectively increase the wheel reprofiling interval with respect to the compared bogie vehicle. A line-dedicated control has been shown to produce improvements in comparison to the on-board measurement feedforward steering control. A line controller opens up to the possibility of removing the dependency of the steering controller from the local dynamic behaviour of the vehicle at the expense of generality of application.

Figure 7.14: Lost volume per load case comparing the standard bogie vehicle and the mechatronic one: sum of lost volume (Top), improvements with respect to the standard bogie in percentage (Bottom). Paper F
Chapter 8

Summary of Appended Papers

Paper A:

Active suspension in railway vehicles: a literature survey

A thorough investigation of the existing technologies and trends in active suspension in railway vehicles is performed. The active suspensions are divided into several categories, tilting mechanism, secondary suspension control, and primary suspension control, highlighting the basic concepts and summarizing the new theories and solutions that have appeared over the last decade. Experimental studies and development status are shown too. Active secondary suspensions are categorized into active and semi-active suspensions. Primary suspensions are instead divided between acting on solid-axle wheelsets and independently rotating wheels. Lastly, a brief summary and outlook is presented in terms of benefits, research status and challenges.

Paper B:

Gain Scaling for Active Wheelset Steering on Innovative Two-Axle Vehicle

The innovative single axle running gear is introduced and initial investigation is performed to improve curving performance via active wheelset steering control. The selected control aims to minimize the longitudinal creepage by controlling the lateral wheelset position on the track. A set of running cases is defined to evaluate the wheelset steering capability of the controlled vehicle. A simple PID control can produce good results but it suffers from instability issues. An adaptive scaling approach is introduced to overcome the stability issue arising from the need of the vehicle to run in several conditions.
CHAPTER 8. SUMMARY OF APPENDED PAPERS

Paper C:

Improved curving performance of an innovative two-axle vehicle: a reasonable feedforward active steering approach

To overcome the stability issues of a feedback approach as observed in Paper B, a feedforward steering control strategy based on non-compensated lateral acceleration and track curvature for the mechatronic rail vehicle is designed. To define the controller, a set of 7 curves, 17 speeds per curve and 3 equivalent conicity profiles is used, giving a design scenario of 357 cases. The feedforward control is synthesized starting from the best achievable results of selected feedback approaches in terms of wheel energy dissipation and required actuation force. The robustness of the controller is validated against curve, speed and conicity.

Paper D:

Active Modal Control of an Innovative Two-Axle Vehicle with Composite Frame Running Gear

The second vehicle dynamics related issue regarding the innovative vehicle design is initially investigated. Due to the single suspension step, passenger ride comfort can significantly degrade. Modal control is implemented and optimized with genetic algorithms to bring the ride comfort to acceptable levels as defined by the EN12299 standard. Here, six hydraulic actuators are used to control both vertical and lateral direction. The vehicle model accuracy is increased with the finite element running gear model, a simplified finite element carbody model, and dynamic actuator models.

Paper E:

Towards the realization of an innovative rail vehicle — active ride comfort control

Based on the results achieved in Paper D, the modal control is modified to improve the performance of the active system and to even out the unbalanced behaviour between vertical and lateral continuous comfort. The modifications involve modal control with an additional sensor in the centre of the flexible carbody and blended control. The latter takes advantage of the information coming from the connection frame to enhance the carbody ride comfort by feeding a portion of the frame acceleration into the feedback modal loop. Based on the frequency response of the results, a low-pass filtered blended controller is introduced to neglect the high frequency content of the frame accelerations. Polynomial interpolation functions are derived
to approximate the genetic algorithm optimized gains and make the defined controller applicable to any speed in the vehicle operational scenario.

**Paper F:**

*Wheel wear reduction of a mechatronic two-axle vehicle controlled with feed-forward wheelset steering approaches*

The innovative mechatronic vehicle is evaluated with respect to wheel wear, where the KTH wear model is used. The vehicle model embraces flexible carbody and connection frame models, and ride comfort and steering actuator models. The Archard wear map coefficients are determined to reproduce measured worn wheel profiles of the existing vehicle running on Metro Madrid line 10. The innovative vehicle is controlled with two feed-forward approaches. The first one is the controller developed in Paper C where the controller relies on on-board measurements, while the second is optimized using firefly algorithms assuming perfect vehicle-infrastructure communication. The controlled vehicle reduces the wheel wear above 60% in terms of lost wheel volume due to wear. Good coherence is found between improvements predicted with the wear number and the ones achieved in terms of lost wheel volume.
Chapter 9

Conclusions

The scope of the presented work was to carry forward the development of an innovative mechatronic vehicle that could potentially reduce vehicle weight, together with maintenance and first investment costs. In the thesis several aspects have been introduced and discussed and the choices made are motivated.

The described vehicle represents a case study on the potential of innovative solutions in railway applications. It has been shown that weight reduction is achievable with a novel vehicle configuration. At the same time it has been shown that mechatronic solutions can be designed to overcome the limitations that a non-standard railway vehicle may carry. In this respect, the following conclusions can be drawn:

- Fully active hydraulic actuators can be used to reduce the vibrations transferred from the rail to the carbody of the innovative vehicle making it fulfill today’s performance requirements despite the absence of the second suspension step. Modal sky-hook control applied to the hydraulic actuators is a simple but effective control that can be implemented to reach acceptable ride comfort levels as defined in the EN12299 standard.

- Modal sky-hook control is a simple and effective control technique that is extremely effective in vibration control. Nevertheless, as applied to the vehicle behaviour, it is designed with only the carbody motion in mind. It has been shown that it is possible to further reduce the vibrations transferred to the carbody by considering a portion of the disturbance signal in the control loop, or more precisely the accelerations coming from the connection frame. The so-called blended control has been shown to be effective in the operational speed range of the innovative vehicle, at the expense of a more complex system.
CHAPTER 9. CONCLUSIONS

- It has been shown that optimized versions of both modal and blended control can be defined for different vehicle speeds. This introduces the possibility of having a controller that adapts its control action according to the vehicle speed to achieve the best performance in different conditions. Given the possibility of measuring the vehicle speed, the control gains can be approximated with simple polynomial functions to adapt to any vehicle speed.

- The poor wheelset steering capability of the innovative vehicle can be significantly improved when active suspension is implemented. By steering the wheelset in a more correct position, it is possible to reduce the energy dissipated in the contact between wheel and rail reducing the negative influence of the long wheelset distance.

- It has been shown that a simple PID controller can be scaled to adapt to track curvature and vehicle speed variations. However, the adaptive PID controller can become unstable under wheelset conicity variation. This phenomenon is a key aspect related to wheel wear. Thus, to overcome this last obstacle, a feedforward approach based on reliable measurements has been designed showing robustness against a variety of conditions.

- Finally, the significant effect of the steering strategy applied to the innovative vehicle is demonstrated on wheel wear evolution. As shown, the wheel volume lost due to wear is significantly reduced when comparing to the standard bogie vehicle for the same travelled distance. Thus, it would be possible to run the innovative vehicle for a longer distance before wheel reprofiling, greatly reducing the maintenance cost.

The present work demonstrated the possibility of designing an alternative solution to standard bogie vehicles to meet the requirements set by the Shift2Rail joint venture for a sustainable growth of the rail sector.
Chapter 10

Future Work

Several aspects have been considered in the present work. Two major categories can be distinguished in defining the future work. The first is strictly related to the development of the innovative vehicle, while the second focuses on the development of mechatronic systems in railways in a more general perspective.

Innovative vehicle development

The innovative mechatronic vehicle has been presented and key aspects of its development have been introduced and exploited. Nevertheless, many more aspects than the ones considered need to be investigated in the future.

• Traction and braking equipment needs to be properly modelled in relation to the connection frame. Even if considered during the development of the frame itself in terms of mass and size, the effects of these key components in the dynamic behaviour of the vehicle are not known yet. As shown, vehicle speed variation does not affect the wheelset steering controller that nevertheless needs further investigation to ensure performance in relation to properly modelled traction and braking systems.

• The not existing carbody has been modelled to approximate typical frequency content of existing vehicles. Further development of a more realistic representation of the carbody is clearly important in the development of the vehicle itself. The carbody design can significantly influence ride comfort and dynamic behaviour.

• The composite material connection frame and the steering active suspensions have been developed and tested showing good potential and
good coherence with the expected outcome. Nevertheless, their inter-
action through wheelset and carbody is unknown in real world appli-
cation. Experimental validation of the interaction between these two
components needs to be performed to ensure the applicability of the
concept.

- Dynamic actuators to control ride comfort have been modelled based
  on actuators used in field studies but they differ from these. Therefore,
  the development of the actuators in conjunction with the need of the
  innovative vehicle has key importance. Once more, their relation with
  the frame dynamics needs to be experimentally validated.

- Both steering and ride comfort control strategies have been developed
  in the ideal world of simulations. Even if simulation tools have been
  used to narrow the gap between real world and simulations, many
  components have been considered ideal. Further investigation in con-
junction with experimental validation of actuators and the connection
frame is of paramount importance in the success of the innovative ve-
hicle.

Many more aspects exist and have to be considered during the develop-
ment of a railway vehicle. The innovative vehicle presents a unique opportunity
to investigate standard railway vehicle aspects with a new challenging per-
spective.

**Mechatronic and control aspects**

Several control strategies have been investigated to solve the major issues
affecting the performance of the innovative two-axle vehicle. Nevertheless,
the control and mechatronic aspects considered in the presented study may
have impact on the performances of other types of railway vehicles and need
to be further investigated.

- It has been shown, as by many other authors before, that modal con-
  trol is an effective control to enhance the performance in terms of ride
  comfort of a passenger vehicle. Blended control has shown even better
  performance with respect to the standard modal control for the inno-
  vative vehicle. Its applicability and significance for a more standard
  bogie (or Jacobs bogie) vehicle is unknown. Further investigation of
  its applicability on a more general scale and the conditions for its
  benefits is desirable.

- It has been shown that concerning ride comfort, a vehicle speed adapt-
ing controller can be beneficial. Nevertheless, this effect, once more
applicable to the innovative vehicle, needs further investigation for other types of vehicles. This should help in identifying a possible relationship between vehicle speed and controller behaviour.

- The stability issues of steering feedback approaches have been briefly investigated and are here overcome with a feedforward steering control approach. The methodology used to delineate the feedforward controller was of particular interest for the innovative vehicle. Nevertheless, a general methodology to develop feedforward controllers for the steering of railway vehicles should be defined to systematically identify simple, good, and robust feedforward approaches.

- Due to the mutual relation between vehicle and track, and the increasing need of knowledge of the vehicle state with respect to the railway network, the focus of steering approaches can be moved from vehicle centered to system centered. In conjunction with knowledge of the upcoming track and precise knowledge of the vehicle speed, a steering controller can be identified to minimize energy consumption and maintenance cost.
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Acknowledgements

Well, it was a long journey.

During my Ph.D. a couple of things have happened (that I can remember). The amazonian forest took fire, Canada reached the highest ever recorded temperature of 49.6 degrees (and it also burned down a little), a pandemic started and finished, a close by war started, and a friend of mine died. I surely missed something.

I think things need a change, a bit at least. For me and for other things too. I’ll try my best.

In the meantime, let me thank some of the important persons in my life.

I would like to thank my family first. In particular my mother and my father. Without them I wouldn’t be here, in every possible sense.

I would like to thank my beloved and supporting Lucia. I hope we will stay together down the road of existence.

I would like to thank my friends afterwords, the ones far away and the ones near by. They define in a good portion my experience of the world.

Last but not least, I would like to thank my supervisors that have been the necessary tough, loving, and carrying guide I needed during this time together.

Thank you all, I hope to have you close in any adventure the future will give us.
Part II

APPENDED PAPERS