KTH Engineering Sciences

Gröna Tåget
Trains for tomorrow's travellers

On Active Secondary Suspension in Rail Vehicles to Improve Ride Comfort

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Abstract

One way to make rail vehicles a competitive means of transportation is to increase running speed. However, higher speeds usually generate increased forces and accelerations on the vehicle, which have a negative effect on ride comfort. With conventional passive suspension, it may be difficult to maintain acceptable passenger comfort. Therefore, active technology in the secondary suspension can be implemented to improve, or at least maintain, ride comfort at increased vehicle speeds or when track conditions are unfavourable.

This thesis describes the development of an active secondary suspension concept to improve ride comfort in a high-speed train. Firstly, an active lateral secondary suspension system (ALS) was developed, including dynamic control of the lateral and yaw modes of the carbody. Furthermore, quasi-static lateral carbody control was included in the suspension system in order to laterally centre the carbody above the bogies in curves at high track plane acceleration and hence to avoid bumpstop contact. By means of simulations and on-track tests, it is shown that the ALS system can offer significant lateral ride comfort improvements compared to a passive system.

Two different control strategies have been studied—the relatively simple sky-hook damping and the multi-variable H_∞ control—using first a quarter-car and then a full-scale vehicle model. Simulation results show that significant ride comfort improvements can be achieved with both strategies compared to a passive system. Moreover, H_∞ control in combination with the carbody centring device is better at reducing the relative lateral displacement in transition curves compared to sky-hook damping.

Secondly, an active vertical secondary suspension system (AVS) was developed, using simulations. Dynamic control of the vertical and roll modes of the carbody, together with quasi-static roll control of the carbody, show significant vertical ride comfort improvements and allow higher speeds in curves. Further, the AVS system compensates for negative ride comfort effects if the structural stiffness of the carbody is reduced and if the vertical air spring stiffness is increased.

Finally, the two active suspension systems (ALS and AVS) were combined in simulations. The results show that both lateral and vertical ride comfort is improved with the active suspension concept at a vehicle speed of 250 km/h, compared to the passive system at 200 km/h. Further, active suspension in one direction does not affect the other direction. The ALS system has been included in two recent orders comprising more than 800 cars.

Keywords: rail vehicle, active secondary suspension, ride comfort, sky-hook damping, H_∞ control, multi-body simulations, on-track tests

Preface

The work presented in this thesis has mainly been carried out at the division of Rail Vehicles at KTH, Stockholm, in close co-operation with Bombardier Transportation, Västerås. The project is part of the Swedish research and development programme Gröna Tåget (Green Train), financed by Trafikverket (Swedish Transport Administration).

The original vehicle model was received from Bombardier Transportation in Västerås. Björn Roos and Anders Brandström have answered many of my questions regarding this model. General questions regarding the simulation tool have been answered by Homan Seyedin and Christoph Weidemann at SIMPACK in Munich.

The actuator model was provided by Liebherr in Lindenberg, Germany. Philipp Kegel assisted me in integrating it with my vehicle model. During the on-track tests, Lothar Klein from Liebherr assisted us regarding software issues.

The measurement data from the on-track tests was partly received from Interfleet Technology.

Paul Dreik has made valuable comments and suggestions regarding the simulations with H_∞ control.

I am really grateful for the support and assistance from my supervisors Sebastian Stichel and Rickard Persson. It has been a pleasure working with you!

I would also like to thank my colleagues at the division of Rail Vehicles for creating a nice working environment. Also big thanks to all ”lunch friends” on the fifth floor for the interesting and fun discussions.

Last, but not least, I would like to thank my dear wife Lena for her moral support as well as linguistic advice on this thesis. Music sounds better with you!

Anneli Orvnäs

Stockholm, November 2011

Outline of thesis

The scope of this thesis is to develop an active secondary suspension concept for a high-speed rail vehicle. The thesis includes an introduction and the following appended papers:

Paper A Orvnäs, A., Stichel, S., and Persson, R., 2008. *On-Track Tests with Active Lateral Secondary Suspension: A Measure to Improve Ride Comfort*. ZEVrail Glasers Annalen 132, No. 11–12, pp. 469–477.

Paper B Orvnäs, A., Stichel, S., and Persson, R., 2010. *Ride Comfort Improvements in a High-Speed Train with Active Secondary Suspension*. Journal of Mechanical Systems for Transportation and Logistics (JSME), Vol. 3, No. 1, pp. 206–215.

Paper C Orvnäs, A., Stichel, S., and Persson, R., 2009. *Development and Test of Active Secondary Suspension in a Rail Vehicle*. Proceedings of the 21st International Symposium on Dynamics of Vehicles on Roads and Tracks, IAVSD'09, Stockholm, August 17–21.

Paper D Orvnäs, A., Stichel, S., and Persson, R., 2011. *Active Lateral Secondary Suspension with H_∞ Control to Improve Ride Comfort: Simulations on a Full-Scale Model*. Vehicle System Dynamics, Vol. 49, No. 9, pp. 1409–1422.

Paper E Orvnäs, A., Stichel, S., and Persson, R., 2011. *Aspects of Using Active Vertical Secondary Suspension to Improve Ride Comfort*. Proceedings of the 22nd International Symposium on Dynamics of Vehicles on Roads and Tracks (extended version), IAVSD'11, Manchester, August 14–19. Submitted for international journal publication.

Paper F Orvnäs, A., Stichel, S., and Persson, R., 2011. *An Active Secondary Suspension Concept to Improve Lateral and Vertical Ride Comfort*. Submitted for international journal publication.

Division of work between authors

Planning of the preparatory simulations and the subsequent on-track tests has been carried out by Anneli Orvnäs, Sebastian Stichel and Rickard Persson. All simulations have been performed by Orvnäs. All papers have been written by Orvnäs and reviewed by Stichel and Persson.

Other publications not presented in this thesis

Report: Orvnäs, A., 2008. *Active Secondary Suspension in Trains – A Literature Survey of Concepts and Previous Work*. Publ. 0803, ISBN 978-91-7415-144-2, Division of Rail Vehicles, Department of Aeronautical and Vehicle Engineering, Royal Institute of Technology (KTH), Stockholm, Sweden.

Licentiate thesis: Orvnäs, A., 2009. *Active Lateral Secondary Suspension in a High-Speed Train to Improve Ride Comfort*. TRITA AVE 2009:22, ISBN 978-91-7415-300-2, Division of Rail Vehicles, Department of Aeronautical and Vehicle Engineering, Royal Institute of Technology (KTH), Stockholm, Sweden.

Report: Orvnäs, A., 2010. *Methods for Reducing Vertical Carbody Vibrations of a Rail Vehicle – A Literature Survey*. Publ. 1002, ISBN 978-91-7415-631-7, Division of Rail Vehicles, Department of Aeronautical and Vehicle Engineering, Royal Institute of Technology (KTH), Stockholm, Sweden.

Conference proceedings: Orvnäs, A., Stichel, S., and Persson, R., 2010. *Ride Comfort Improvements of a Regina Train with Active Lateral Secondary Suspension*. Proceedings of the 8th International Conference on Railway Bogies and Running Gears, ISBN 978-963-313-024-7, pp. 285–294.

Thesis contribution

This thesis presents an active secondary suspension concept that provides significant ride comfort improvements, but still at a reasonable cost.

This thesis is believed to make the following contributions to the present research field:

- Two literature studies regarding active secondary suspension in the lateral and vertical directions have been compiled. They describe various active suspension concepts, actuator types, control strategies and measures to improve ride comfort.
- An existing vehicle model, originally developed at Bombardier Transportation, has been modified for this study and validated against measurement results. The agreement between measurements and simulations is good up to 10 Hz approximately.
- The complete control strategy needed for running on straight track, in curves and in transition curves has been developed and optimized through simulations, using properties of a real existing actuator.
- Comparisons of the performance of single-variable sky-hook damping and multi-variable H_∞ control for active lateral secondary suspension in trains have been carried out.
- Lateral ride comfort improvements have been proven by multi-body simulations as well as on-track tests.
- Vertical ride comfort improvements have been proven by multi-body simulations.
- Suggestions have been made to simplify air springs and to reduce the frequencies of carbody flexible modes after the introduction of active secondary suspension.

Contents

1 Introduction

1.1 Background

Increased speed is a way for rail vehicles to compete with other means of transportation. However, higher speeds usually generate increased forces and accelerations on the vehicle, which negatively affect ride comfort. Therefore, active technology in the secondary suspension system can be a solution to achieve ride comfort improvements in cases where the conventional passive suspension cannot be further optimized.

The concept of active technology in rail vehicles has been studied theoretically and experimentally for several decades, generally showing significant ride comfort improvements compared to a passive suspension system [1–4]. However, the active suspension system has not yet had a convincing breakthrough in service operation (except for the tilting train technology [5]), as has been experienced in, for example, aircraft and automotive industry. The main reason for the lack of success is most likely that the offered solutions have been too expensive in relation to the benefits gained. Compared to the passive solution, the active suspension system must prove to be at least as reliable and safe, in order to be considered as an option. However, if a concept can be found that simultaneously manages good performance and acceptable costs, there is significant potential for future implementation.

Active secondary suspension in rail vehicles can be implemented in order to achieve one or more of the following goals:

a) improve passenger ride comfort,

b) maintain good ride comfort even when vehicle speed is increased,

c) maintain good ride comfort although track conditions are unfavourable,

d) increased carbody width,

e) reduced carbody weight.

If ride comfort is already good, further improvements at unchanged vehicle speed and track conditions are generally not justified due to the high costs of implementing the active system. However, goal b) has large possibilities for cost-efficient improvements, since vehicle speed can be increased. Moreover, goal c) has good potential of being worth the investment of active technology, since track maintenance costs can be reduced. Further, increased carbody width allowing more seats (goal d) as well as reduced carbody weight (goal e) can be very interesting economically for the operator.

In general, the principle of an active suspension system is based on the idea of controlling certain signals (e.g. carbody accelerations) with the signals themselves, i.e. by means of a closed loop (Fig. 1.1). In order to achieve this control loop in the suspension of a rail vehicle, actuators, sensors and a controller must be included in the vehicle system. The actuators can replace conventional passive dampers, for example between the carbody and bogies. They should actively generate a required control force according to the force demand from the controller. The force demand is calculated based on the vehicle output signals measured by the sensors. These signals, in turn, depend on the generated actuator force influencing the mechanical system. Hence, the control loop is closed.

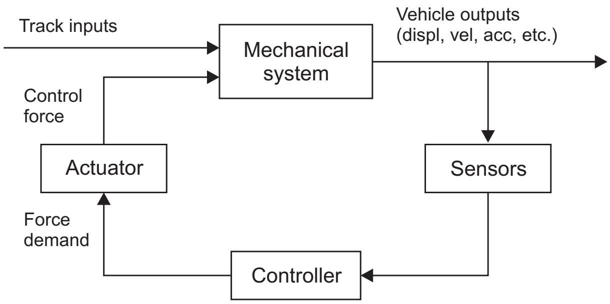

Figure 1.1: The principle of an active suspension system.

How well the control force of the actuator agrees with the force demand depends on the characteristics of the actuator. The ideal actuator generates exactly the same force it is told to generate over an infinite bandwidth and without delay. In reality, this is not achievable and working with active suspension is always a matter of trade-offs between different parameters, such as actuator performance and cost.

1.2 Objectives of this work

This study is part of the Swedish research and development programme Gröna Tåget [6], aiming at developing the next generation of high-speed trains for Nordic conditions. The overall focus is to increase vehicle speed from today's 200 km/h to 250 km/h on existing conventional lines and up to around 300 km/h on new dedicated high-speed lines. To make this possible, a co-operation within the railway industry will be needed, in order to improve vehicle dynamics, carbody tilting, energy consumption, winter climate reliability, aerodynamics and acoustics, among other things. Market needs, capacity and economics, as well as passenger issues will also need to be addressed. A two-car Regina train has been used for the on-track tests performed within the Gröna Tåget research programme (Fig. 1.2).

Figure 1.2: The Regina test train within the Gröna Tåget research programme. Photo courtesy of Bombardier Transportation.

The objective of this study is to develop an active secondary suspension concept to improve ride comfort. The concept should offer good ride comfort improvements—both laterally and vertically—but still at a reasonable cost. Dynamic as well as quasi-static control of various carbody modes are considered. The final goal is to offer a good solution for serial production. The active suspension concept presented in this thesis is developed in co-operation with Bombardier Transportation, Västerås.

The outline of the introductory part is as follows:

- Chapter 2 describes the active secondary suspension along with the investigated sky-hook and H_∞ control methods.
- Chapter 3 presents the applied simulation model and the on-track test conditions.
- Chapter 4 introduces the two methods used for ride comfort evaluation.
- Chapter 5 gives a summary of the appended papers.
- Chapter 6 presents the conclusions and proposals for future work.

2 Active secondary suspension

Many different concepts of active suspension to improve ride comfort in rail vehicles have been investigated throughout the years. Usually, simultaneous control of several modes is examined. This section gives an overview of other research studies applying dynamic and/or quasi-static control to improve ride comfort, followed by practical implementations of such concepts. Further, investigation and comparison of two control methods, sky-hook damping and H_∞ control, has been performed within this work. The concepts of these two methods are presented, referring to previous research studies.

2.1 Active suspension to improve ride comfort

There are various alternatives for how to implement the actuators in the secondary suspension. One way is to fit the actuators into the bogie environment in combination with the existing passive components, in series or in parallel. Fitting the actuator in parallel with a passive spring enables reduced actuator size, since the spring can be principally responsible for taking up the required quasi-static loads, either vertically or laterally. Connecting the actuator in series with passive components can be beneficial if the actuator performance is not sufficient to take care of high-frequency vibrations. A second alternative is to replace the passive components by actuators altogether. This requires reliable actuators which ensure the ability to work in passive mode in case of actuator failure. An overview of different actuator types has been compiled by the author, where the concept of how they work and their advantages and disadvantages is presented [7].

There are two general concepts of active suspension, so-called fully-active and semi-active, basically governed by the required amount of external power, as described by Jalili [8]. The *fully-active* suspension offers high performance control and gives the best response in a wide frequency spectrum. It is able to generate a force in the opposite direction as the relative damper velocity, which means that energy is both transferred to and dissipated from the suspension system. On the other hand, it requires many sensors and an external power supply, as well as a sophisticated control method, described by Kjellqvist [9].

Between the passive and the fully-active systems there is the *semi-active* suspension system. The actuator force depends on the relative damper velocity, i.e. the velocity difference between the two bodies where the actuator is situated. Compared to the fully-active system it is less complex, more low-priced and does not require an external power supply [8]. However, energy cannot be transferred to the system, but only dissipated from it. This means that the actuator in a semi-active system cannot develop a force in the opposite direction to the relative damper velocity. Large actuator forces cannot be generated at low velocities [10]. Despite these drawbacks, the semi-active suspension may still work in passive mode if failure of the control system occurs.

2.1.1 Dynamic control

Dynamic control is concerned with reducing vibrations of the carbody due to imperfections of the track. The actuators are usually designed to control vibrations between approximately 0.5 Hz and 4–10 Hz, depending on the application.

A comprehensive introduction to active suspensions in a rail vehicle has been published by Pratt [11], focusing on the application in high-speed rail vehicles in order to maintain or improve ride comfort at higher speeds. Both lateral and vertical directions of a full-scale vehicle model were considered. The study concerns the investigation of how actuator dynamics influence the overall operation of an active suspension system. Simulation results show that active secondary suspension applied to a single rail vehicle can improve ride comfort by 30 % in the vertical direction, and 45 % in the lateral direction. However, with actuator dynamics included, ride comfort benefits may be degraded by up to 15 % with certain actuator types.

A Japanese study by Hirata *et al.* [12] describes an active secondary suspension system to reduce lateral, yaw and roll motions of the carbody. The study describes simulations and running tests performed with an experimental vehicle. In another Japanese study, experiments on a roller test rig were performed using active lateral and vertical secondary suspensions to improve ride comfort [13]. Up to 50 % reduction of these particular modes of vibration could be shown with the active system.

In an Italian study performed by Pugi *et al.* [14], the vertical passive dampers in the secondary suspension were replaced by semi-active magneto-rheological actuators. The actuators were independently controlled to reduce vertical, pitch and roll accelerations of the carbody. Compared to a passive system, vertical accelerations are reduced by 10–20 %.

In a study performed by Foo and Goodall [15], the first vertical bending mode of the carbody is suppressed by adding an electro-magnetic actuator between the centre of the carbody and an auxiliary mass of one tonne. Additionally, hydraulic actuators were placed across the secondary suspension in the front and rear of the vehicle. Three different configurations of so-called sky-hook modal control—to separately handle the bounce and pitch modes of the carbody—were investigated. The central actuator effectively reduces the flexibility effect on the ride comfort. Compared to a passive system, one of the considered control configurations with actuator dynamics included reduces vertical accelerations by approximately 60 % in the rear part of the vehicle.

Both the primary and secondary suspensions can be considered when using active damping to suppress the modes that significantly affect ride comfort. In a study performed by Sugahara *et al.* [16, 17], semi-active damping is applied to the vertical axle-box dampers in the primary suspension to suppress the vertical vibrations of the bogies, which, in turn, reduces the bending vibrations of the carbody (around 8–10 Hz). Furthermore, active damping in the secondary suspension is applied to the air spring to suppress the rigid vibration modes bounce and pitch (around 1 Hz). The study includes simulations and running tests with a Shinkansen vehicle at speeds up to 315 km/h. The investigated damping system shows significant ride comfort improvements and has reached the stage of practical application.

2.1.2 Quasi-static control

When travelling in curves at high speeds (high track plane accelerations), the carbody moves laterally outwards in relation to the bogie, generating a quasi-static displacement between the carbody and bogies. This lateral displacement of the carbody increases the risk of hitting the bumpstops, which significantly affects ride comfort in a negative way (Fig. 2.1). If the quasi-static displacement can be reduced, not only can good ride comfort be maintained, but a wider carbody profile is also possible, since the play between the carbody and bogies does not have to be as large as before. Furthermore, if the position of the bumpstops is changed and hence the play between the carbody and bogies is decreased, crosswind stability can be improved.

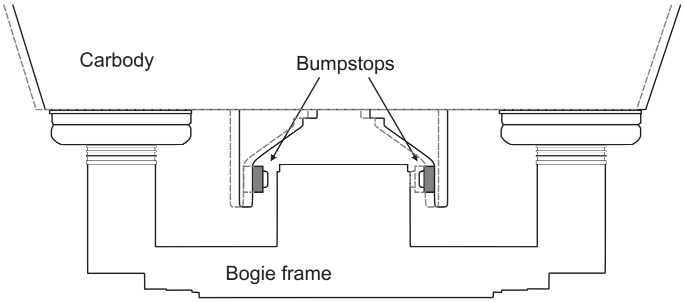

Figure 2.1: Bumpstops limit the lateral carbody displacement in relation to the bogie.

Quasi-static lateral control is applied to centre the carbody above the bogies in curves. This application is also called *low-bandwidth control*, since it detects the low frequencies of deterministic track inputs, i.e. curves, in order to minimise the lateral displacement of the carbody in relation to the bogies. This Hold-Off Device concept was introduced in the early 1990s by Allen [18]. However, the concept of low-bandwidth control was mentioned already in 1983 by Pollard [19]. In a Spanish research study by Conde Mellado *et al.* [20], the carbody centring is achieved by a lateral pneumatic actuator connected to the air spring. In curves, the actuator is controlled by the pressurized air of the inner air spring.

Carbody tilting inwards in curves at high track plane accelerations is applied to allow higher speeds without negatively affecting ride comfort (Fig. 2.2) [5]. One common way to implement tilt control is to use *precedence control* [21]. This means that information from the leading bogie is used for the rest of the vehicle in order to get a more precise control. The control design must take the vehicle speed and length into account as well as delays introduced by the filters. The objective of tilt control is to reduce the lateral acceleration felt by the passengers to zero, which is referred to as nulling tilt. However, one problem with tilting trains is the not ignorable number of passengers that experience motion sickness. Therefore, it is normal not to fully compensate for the acceleration to minimise this motion sickness phenomenon. This is referred to as partial tilt compensation with a typical compensation around 60–70 %.

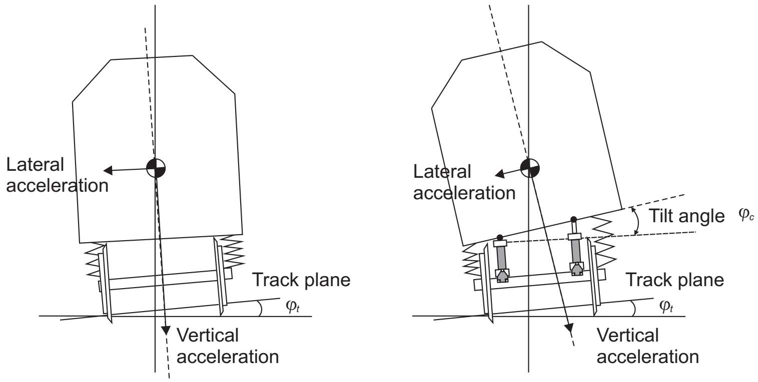

Figure 2.2: The concept of carbody tilting by vertical actuators. The lateral acceleration is decreased due to tilting of the carbody (right) [5].

The air spring can be used to control the quasi-static roll of the carbody and hence to generate tilting. In a study performed by Alfi *et al.* [22], active air springs combined with a lateral actuator in the secondary suspension are used to achieve carbody tilting and centring in curves. Experimental and numerical simulation results show that the active suspension system improves ride comfort and allows for higher speeds in curves.

Asano and Kajitani [23] and Kajitani *et al.* [24] describe a research study where ride comfort is improved in a Shinkansen vehicle at high speeds by means of dynamic lateral control in combination with carbody tilting. Running tests have been performed at speeds up to 365 km/h.

Further, carbody tilting can be achieved by adding an active element to the anti-roll bar in the secondary suspension (Fig. 2.3) [25]. The active anti-roll bar can be integrated with a lateral actuator in the secondary suspension to simultaneously achieve carbody tilting and dynamic lateral control [26].

In the late 1990s, a comprehensive active secondary suspension system was developed by Stribersky *et al.* [27–29]. The suspension control system included semi-active lateral and vertical dampers that were able to achieve control of the lateral and vertical carbody vibrations as well as carbody centring and tilting in curves. The simulation results were confirmed by field tests performed with prototype bogies equipped with the active damping system.

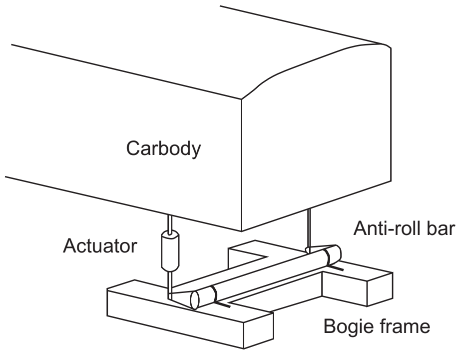

Figure 2.3: The principle of an active anti-roll bar [25].

2.1.3 Practical implementations

As mentioned before, tilting technology is the most successful application within the area of active suspensions in trains when it comes to service implementation. In the early 1990s, the tilting train *X2000* was developed by Adtranz (today Bombardier) and taken into service in Sweden. A few years later, a semi-active secondary suspension working in lateral and yaw modes was tested in combination with the tilting technology [30]. However, the active lateral secondary suspension stayed on the experimental stage, whereas the tilt control is still in operational use today (2011).

In Italy, the Fiat Pendolino started using lateral carbody centring in combination with tilt technology in service operation in the late 1980s [31]. However, during the latest years, lateral centring of the carbody has been achieved only by tilting below the secondary suspension, without a particular carbody centring device. Spanish and British Pendolinos still use the fully-active carbody centring device in combination with the tilt system [4, 31].

The development of active lateral secondary suspension systems in Japan has been successful. In 1997, JR West implemented active lateral suspension in service operation on a Shinkansen Series 500 [32]. Further, JR Central has tested a semi-active lateral suspension system on the Shinkansen Series 300. This suspension system was later installed in the Shinkansen Series 700, which started commercial operation in 1999 [33]. The Shinkansen train Series E2 and E3 with a fully-active lateral suspension system have been in service for JR East since 2001, reducing yaw and roll vibrations of the carbody [4]. This concept was further developed for the Shinkansen Fastech 360 S and Z by JR East [34]. The Shinkansen Series 700 was extended to the Series N700, using a semi-active lateral suspensions system in combination with carbody tilting. The trains were put into commercial operation in 2007 after two years of test runs performed by JR Central and JR West [35].

In 2000, an application concerned with control of the roll mode of the carbody was implemented in Bombardier's regional Talent train. An active anti-roll bar is used to achieve carbody tilting. The component is a transversely-mounted torsion rod on the bogie with vertical links to the carbody. At least one of the links is replaced by a hydraulic actuator, which applies carbody tilting via the torsion rod [10, 36].

Although many control systems for reducing vertical carbody vibrations of a rail vehicle have been studied, very few systems have been applied to actual rail vehicles and tested on commercial lines. In Japan, running tests with a Shinkansen vehicle equipped with semi-active primary and secondary vertical suspensions have been performed [16, 17]. The system with active primary vertical suspension has now reached an advanced development stage and is close to practical implementation in Japanese service operation [37, 38].

2.2 Control methods

2.2.1 Sky-hook damping

A straight-forward and widely used control strategy in the area of active technology in rail vehicles is the so-called *sky-hook damping* [39]. In the conventional suspension system, the passive damper reacts on the relative velocity between the carbody and bogie. In the sky-hook damping system, the absolute velocity of the carbody is damped by an actuator, since the required force is generated independently of the bogie velocity (Fig. 2.4).

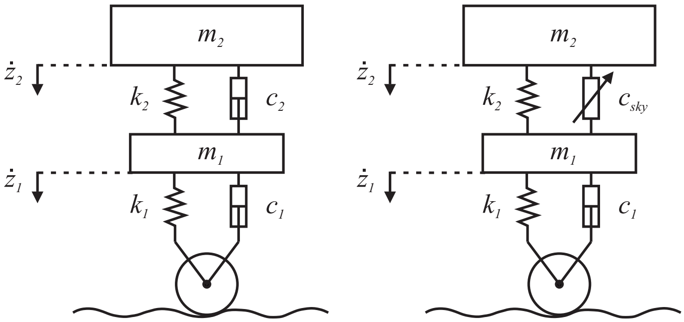

Figure 2.4: Conventional suspension model (left) and sky-hook damping model (right).

For the conventional passive damping, the equation of motion is expressed as

$$m_2\ddot{z}_2 = (\dot{z}_1 - \dot{z}_2)\,c_2 + (z_1 - z_2)\,k_2, \tag{2.1}$$

whereas the corresponding equation for the sky-hook damping can be expressed as

$$m_2\ddot{z}_2 = -\dot{z}_2 c_{sky} + (z_1 - z_2)\,k_2. \tag{2.2}$$

The equations of motions for the two different damping systems can be rewritten as the following transfer functions between the bogie and carbody:

$$G_{passive} = \frac{c_2 s + k_2}{m_2 s^2 + c_2 s + k_2} \tag{2.3}$$

and

$$G_{sky\text{-}hook} = \frac{k_2}{m_2 s^2 + c_{sky} s + k_2}. \tag{2.4}$$

The absence of bogie velocity influence in the numerator is the most important difference between the two transfer functions. This also allows the sky-hook damping to be set larger than the conventional damping, thus further reducing the vibration transfer from the bogie to the carbody.

In practice, sky-hook damping is usually implemented as shown in Fig. 2.5. In fact, the required absolute velocity signal is normally an integrated acceleration signal measured by a sensor in the carbody. Further, the velocity signal is high-pass filtered and multiplied by the sky-hook damping coefficient in order to generate the required actuator force (described in [10]).

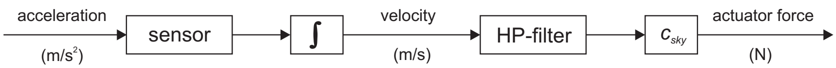

Figure 2.5: Practical implementation of sky-hook damping.

The sky-hook damping concept was first introduced by Karnopp in the late 1970s, leading to a comprehensive study published in 1983 [39]. Sky-hook damping has since been thoroughly investigated and analysed by various researchers. Stribersky *et al.* [28] have shown through simulations that sky-hook damping significantly reduces resonance peaks and *rms* accelerations, thus improving ride comfort, both vertically and laterally. The simulation results have also been confirmed by field tests performed with prototype bogies equipped with active damping. Moreover, a Swedish study by Roth and Lizell [30] in the late 1990s also show improved ride comfort through simulations and field tests using semi-active sky-hook damping in the lateral direction.

A difficult problem, and hence a big challenge with the sky-hook damping is to be able to optimize the trade-off between improved comfort and suspension deflection during curving. Nevertheless, acceptable results can be achieved by optimizing the filtering of the absolute velocity signal. Li and Goodall [40] have theoretically analysed three linear and two non-linear approaches to sky-hook damping in the vertical direction, with different filtering solutions. The linear method with a so-called complementary filter improves ride comfort by nearly 23 %, while keeping suspension deflection at the same level as for a passive system. The two non-linear methods, based on Kalman filtering, show over 50 % ride comfort improvement, however, with larger suspension deflection than as for the passive case.

Hohenbichler and Six [41] have analysed the mentioned trade-off between comfort and suspension deflection through simulations with slightly different approaches of sky-hook damping. The conclusion was drawn that, for the considered track conditions, sky-hook damping offers no more than 10 % comfort improvement compared to a passive case.

Baier *et al.* [42] have performed simulations using preview data (accelerations) in combination with sky-hook damping in order to optimize the actuator control, and thus improve ride comfort in the vertical direction. Low-pass filtered accelerations, i.e. deterministic track input without stochastic irregularities, from the first bogie were subtracted from measured accelerations on the following bogies (integrated to velocity according to the sky-hook principle). Hence, the actuators in the bogies using preview data compensated only for track irregularities and not the deterministic track curvature.

2.2.2 H_∞ control

Compared to sky-hook damping, H_∞ control can be designed for more than one goal, for example to reduce carbody vibrations and the relative lateral displacement in transition curves simultaneously. H_∞ control is concerned with finding a controller (K) for an open-loop system (G_0) such that the closed-loop system (G_{ec}) has good performance, stability and robustness (Fig. 2.6). G_{ec} is the transfer matrix from the external disturbance vector $\boldsymbol{w}$ to the error signal vector $\boldsymbol{z}$. Moreover, the measurement vector $\boldsymbol{y}$ is used in K to calculate the control input vector $\boldsymbol{u}$. In order to achieve secured stability and robustness of the system, the signal $\boldsymbol{z}$ should be minimised.

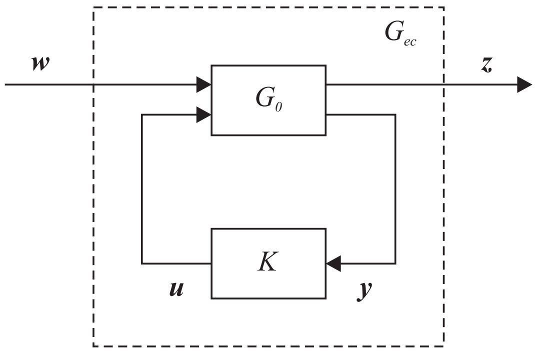

Figure 2.6: Typical configuration of a general control system.

The relations of the signals in the system can be described as

$$\begin{bmatrix} \boldsymbol{z} \\ \boldsymbol{y} \end{bmatrix} = G_0(s) \begin{bmatrix} \boldsymbol{w} \\ \boldsymbol{u} \end{bmatrix} \tag{2.5}$$

$$\boldsymbol{u} = K(s)\boldsymbol{y} \tag{2.6}$$

$$\boldsymbol{z} = G_{ec}(G_0, K)\boldsymbol{w} \tag{2.7}$$

The task is to find the controller K such that

$$\|G_{ec}\|_\infty = \max_\omega \bar{\sigma}(G_{ec}(i\omega)) < \gamma, \tag{2.8}$$

where γ is a chosen boundary criterion and G_{ec} the transfer matrix that describes the closed-loop system. The idea is to find the controller K that minimises the so-called H_∞ norm, i.e. the largest singular value $\bar{\sigma}$ of the plant G_{ec} at a certain frequency ω. If a solution can be found it can either be accepted or the boundary criterion can be decreased in order to find an even better solution. Hence, it is an iterative process to optimize the boundary criterion, as described e.g. by Glad and Ljung [43]. In H_∞ control design, weight functions are applied to the input and output signals of the system in order to define the amount of influence each signal should have in the calculation of the controller.

A general advantage with H_∞ control is its ability to address a multi-variable system. A drawback is that the H_∞ control model tends to reach a rather high order number, since the order number of the weight functions is included. Model reduction is often necessary to make the controller realisable.

In a Japanese study performed by Hirata *et al.* [12], H_∞ control is applied to the secondary suspension of a rail vehicle to reduce lateral, roll and yaw motions of the carbody. The study describes simulations and running tests performed with an experimental vehicle. The experimental test results show that the controlled motions could be significantly reduced. Hence, low-frequency vibrations caused by suspension resonance could be damped.

In the late 1990s, theoretical and experimental tests were performed by Zeng *et al.* [44], applying H_∞ control to the lateral secondary suspension in order to improve stability and dynamic behaviour of a high-speed railway carbody. The active suspension shows improvements in ride comfort compared to a passive case, both through simulations and initial tests on a roller rig.

In a study performed by Orukpe *et al.* [45], a modified H_∞ control approach for vertical vibration control of a rail vehicle was compared to sky-hook damping and a passive system. A four degrees of freedom simulation model was used with actuators placed across the secondary suspension to achieve good vertical ride comfort while maintaining acceptable suspension deflections. The modified H_∞ control proved somewhat more efficient in minimising accelerations and suspension deflection by the tuning of control parameters compared to sky-hook damping and the passive system.

3 Simulations and on-track tests

This chapter describes the simulation model used in this study. Further, the actuator and the on-track test conditions are presented.

3.1 The vehicle model

The vehicle model used in the present study, originally developed by Bombardier Transportation, is built up in the multi-body system (MBS) simulation tool SIMPACK. It models a one-car Swedish Regina vehicle with two motor bogies (Fig. 3.1). However, the original Regina bogies are replaced by new high-speed bogies developed within the Gröna Tåget research programme (Fig. 3.2). In the simulations with active lateral control (ALS), the carbody is rigid, whereas flexible in the simulations when active vertical control (AVS) is applied.

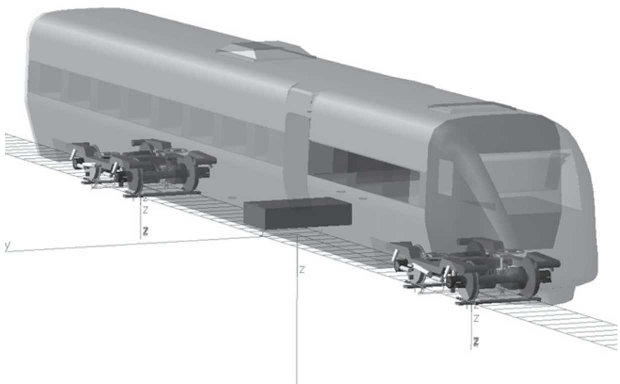

Figure 3.1: The vehicle model in the simulation programme SIMPACK.

The secondary suspension between the carbody and the bogies consists of two non-linear air springs and two non-linear yaw dampers per bogie. Additionally, there is one longitudinal traction rod and one anti-roll bar per bogie.

In the first phase of this research project, the two conventional lateral dampers in each bogie were replaced by one lateral actuator, placed diagonally in relation to the actuator in the other bogie (**Paper A**, **B** and **D**). In the next phase of the project, active vertical secondary suspension was considered, replacing the two conventional vertical dampers in each bogie by two vertical actuators (**Paper E** and **F**).

The wheelsets and the bogie frame are interconnected by the primary suspension. Between each end of the wheelset (i.e. the axle journals) and the bogie frame there are non-linear spring elements with damping. Furthermore, the primary suspension consists of four separate non-linear axle-box dampers per bogie.

Wheel geometry is modelled with the unworn wheel profile S1002 and rail geometry with the unworn rail profile UIC 60. The rail inclination is 1:30. Creep forces are calculated with the FASTSIM algorithm. Measured track irregularities are applied as excitations in the simulation model and scaled according to the definition of comfort track [46].

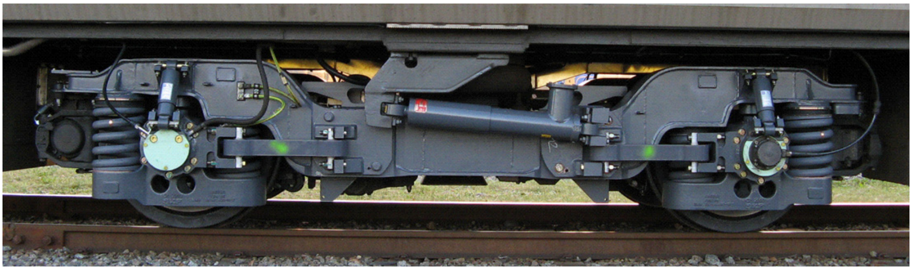

Figure 3.2: The new high-speed Regina bogie. Photo courtesy of Bombardier Transportation.

3.2 The actuator

The hardware used for the on-track tests is an electro-hydraulic actuator developed by Liebherr in Germany. It is designed to meet the requirements in the lateral direction. The actuator is a cylindrical damper with two chambers separated by a movable piston (Fig. 3.3). The chambers are provided with hydraulic fluid by means of two pumps, which are driven by an asynchronous motor fed by power from the train. Pressure valves control the outflow from the cylinder, generating a pressure difference between the two chambers. The valves are controlled by varying voltage, determined by the force demand fed to the actuator. The pressure difference enables the actuator to create the required force response—both in push and pull directions (a maximum of 30 kN for a relative speed of 50 mm/s). The actuator performs well in the frequency range up to about 6 Hz. For the simulations, the actuator is modelled in Simulink with its actual characteristics.

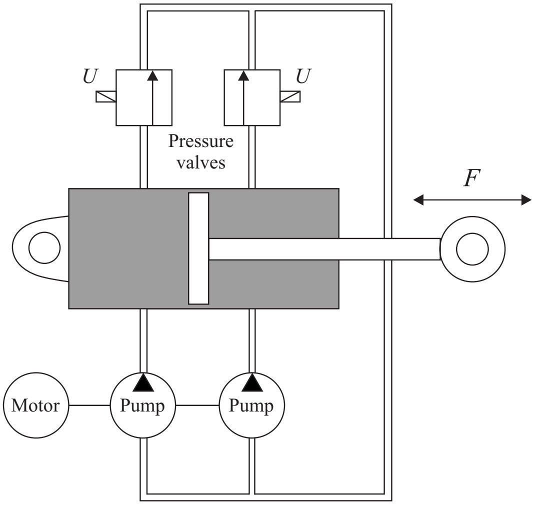

Figure 3.3: Schematic picture of the actuator developed by Liebherr.

3.3 The on-track tests

On-track tests to evaluate the active lateral secondary suspension (ALS) were performed during three summers (2007–2009) with a two-car Regina test train (Fig. 1.2). In one of the cars, the four conventional lateral dampers were replaced by two electro-hydraulic actuators, as described in Section 3.1. The other, unchanged car, was used as reference when evaluating the measurements (Fig. 3.4). Measurement results are presented in **Paper A** and **B**.

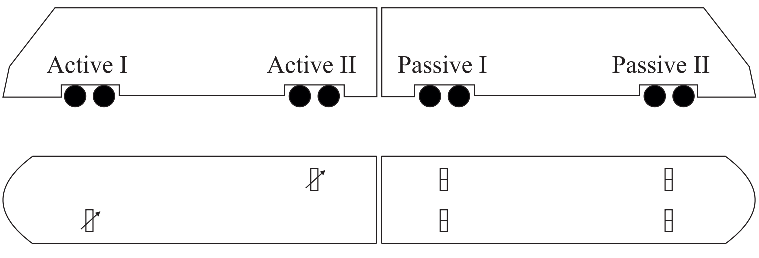

Figure 3.4: Test train configuration for the on-track tests with lateral actuators in one of the cars.

The tests have been performed according to UIC requirements for rail vehicle certification [47]. The general conditions of the tracks are considered as average to good for a 200 km/h track in Sweden. Apart from tests on straight tracks, three different curve radius classes have been included: small-, medium- and large-radius curves, corresponding to 250–600 m (divided into 250–400 and 400–600 m according to UIC 518), 900–1 500 m, and 3 000–5 000 m, respectively. The different test sections are illustrated in Fig. 3.5.

Curve class	Section	*R* (m)
Small-radius	Ånge – Sundsvall Hudiksvall – Sundsvall Järna – Nyköping	300 500 600
Medium-radius	Järna – Töreboda	1 000
Large-radius	Örbyhus – Skutskär	3 000
Straight track	Skövde – Töreboda	---

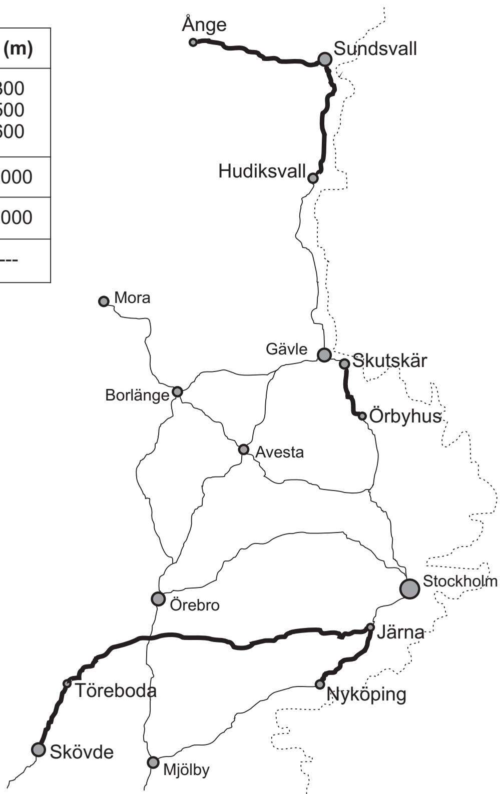

Figure 3.5: Test sections for UIC approval of the Regina test train.

In September 2008, a new Swedish speed record of 303 km/h was achieved, with the active lateral secondary suspension system (ALS) integrated in the vehicle. The test track was an ordinary Swedish track otherwise used for 160–200 km/h (Skövde-Töreboda). The track quality was close to the limit of what is accepted for 200 km/h.

4 Ride comfort evaluation

Ride comfort is normally evaluated by measuring and weighting accelerations on the carbody floor. The two methods used for comfort evaluation in the present study, ISO 2631 and *Wz*, are presented in the following section. The differences between the two evaluation methods are mainly the frequency and time/distance evaluation intervals. Additionally, in curves, the permissible vehicle speed is limited by the lateral track plane acceleration to stay within the range for acceptable ride comfort, which is described in the second section.

4.1 Dynamic comfort evaluation

4.1.1 ISO 2631

Ride comfort evaluation according to ISO 2631 is well described in the European standard EN 12299 [48], which is a railway application of ISO. The *rms* values of frequency-weighted accelerations on the carbody floor level are evaluated as

$$a^{wrms} = \left[\frac{1}{T} \int_0^T [a^w(t)]^2 dt \right]^{0.5}, \tag{4.1}$$

where $a^w(t)$ is the frequency-weighted acceleration as a function of the time t. $T = 5$ s is the duration of the measurement.

The weighting filter for horizontal comfort evaluation is a product of band-limiting and transition filters: $H_{horiz}(f) = H_h(f) \cdot H_l(f) \cdot H_{th}(f)$, where $H_h(f)$ is a second order high-pass filter with cut-off frequency at 0.4 Hz, $H_l(f)$ is a second order low-pass filter with cut-off frequency at 100 Hz, and $H_{th}(f)$ is a transition filter where weighting is proportional to acceleration at lower frequencies and to velocity at higher frequencies. The corresponding transfer functions are [49]

$$H_h(s) = \frac{s^2}{s^2 + \frac{2\pi \cdot 0.4}{0.71} s + (2\pi \cdot 0.4)^2}, \tag{4.2}$$

$$H_l(s) = \frac{(2\pi \cdot 100)^2}{s^2 + \frac{2\pi \cdot 100}{0.71} s + (2\pi \cdot 100)^2}, \tag{4.3}$$

and

$$H_{th}(s) = \frac{(s+2\pi\cdot 2)4\pi}{s^2 + \frac{2\pi\cdot 2}{0.63}s + (2\pi\cdot 2)^2}. \tag{4.4}$$

The corresponding weighting filter for vertical comfort evaluation is a similar product of filters: $H_{vert}(f) = H_h(f)\cdot H_l(f)\cdot H_{tv}(f)$, where the transition filter, also taking the upward gradient into account, is expressed as the transfer function [49]

$$H_{tv}(s) = \frac{(s+2\pi\cdot 16)}{s^2 + \frac{2\pi\cdot 16}{0.63}s + (2\pi\cdot 16)^2} \cdot \frac{(s^2 + \frac{2\pi\cdot 2.5}{0.8}s + (2\pi\cdot 2.5)^2)}{s^2 + \frac{2\pi\cdot 4}{0.8}s + (2\pi\cdot 4)^2} \cdot 32.768\pi. \tag{4.5}$$

Equations 4.2–4.5 are used to plot the weighting curves in Fig. 4.1 (together with the corresponding transfer functions for *Wz* comfort evaluation). According to ISO 2631 evaluation, human subjects are considered to be most sensitive to accelerations in the 0.5–2 Hz frequency range in the horizontal direction and in the 4–10 Hz range in the vertical direction; cf. Fig. 4.1. The ride comfort levels for the individual horizontal and vertical directions according to ISO 2631 are presented in Table 4.1. The levels are the same for the horizontal as well as the vertical direction.

Table 4.1: Ride comfort levels (horizontal and vertical) evaluated by ISO 2631 [48].

Value (-)	*Description*
$a^{wrms} < 0.2$	Very comfortable
$0.2 \leq a^{wrms} < 0.3$	Comfortable
$0.3 \leq a^{wrms} < 0.4$	Medium
$0.4 \leq a^{wrms}$	Less comfortable

4.1.2 Wertungszahl (Wz)

Wz is a ride comfort number originating from German research in the 1940s and 1950s by Sperling and Betzhold [50, 51]. It is a frequency-weighted *rms* value of the lateral or vertical accelerations on the carbody floor, normally evaluated over a one kilometre distance. *Wz* is defined as

$$Wz = 4.42(a^{wrms})^{0.3}, \tag{4.6}$$

where a^{wrms} is the *rms* value of the frequency-weighted acceleration. The filter functions for the lateral and vertical directions (B_l and B_v, respectively) are described by the following transfer functions [52]:

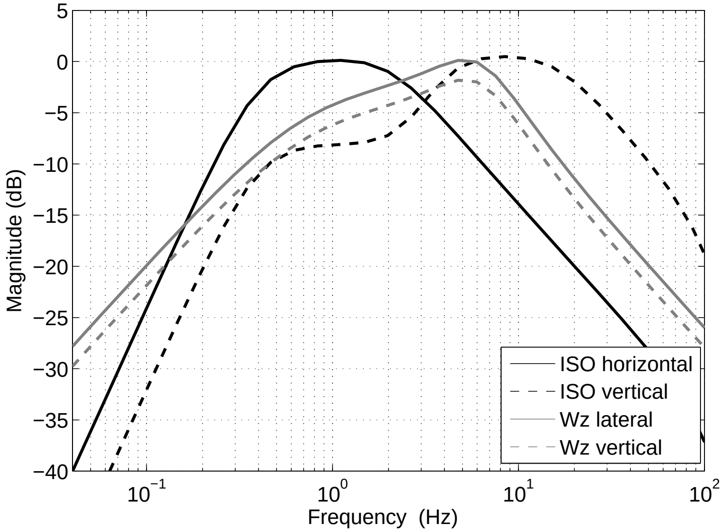

Figure 4.1: Frequency weighting of accelerations according to ISO 2631 and *Wz* ride comfort evaluation.

$$B_l(s) = 0.737 \frac{\frac{0.25}{(2\pi)^2}s^2 + \frac{\sqrt{1.911}}{2\pi}s}{\frac{0.0368}{(2\pi)^3}s^3 + \frac{0.277}{(2\pi)^2}s^2 + \frac{1.563}{2\pi}s + 1}, \tag{4.7}$$

$$B_v(s) = \frac{0.588}{0.737}B_l(s). \tag{4.8}$$

According to *Wz* evaluation, human subjects are considered to be most sensitive to accelerations in the 4–7 Hz frequency range, laterally as well as vertically; cf. Fig. 4.1. Table 4.2 presents the ride comfort levels evaluated by *Wz*. The levels are the same for the lateral as well as the vertical direction.

Table 4.2: Ride comfort levels (lateral and vertical) evaluated by *Wz*.

Value (-)	*Description*
1.0–2.0	Very comfortable
2.0–2.5	Comfortable
2.5–3.0	Less comfortable
> 3.0	Unpleasant

4.2 Ride comfort in curves

In curves, the lateral track plane acceleration, a_y, is an important parameter related to ride comfort (Fig. 4.2). It occurs as the carbody moves laterally outwards in a horizontal curve and is a parameter that decides the maximum permissible vehicle speed in the curve (related to curve radius and track cant). It is defined as

$$a_y = \frac{v^2}{R}\cos\varphi_t - g\sin\varphi_t = \frac{v^2}{R}\cos\varphi_t - g\frac{h_t}{2b_0} \approx \frac{v^2}{R} - g\frac{h_t}{2b_0}, \tag{4.9}$$

where v is the vehicle speed, R the curve radius, φ_t the track cant angle, h_t the track cant and $2b_0$ the track base. The approximations can be made when the track cant angle is small (often less than 7 degrees), which is almost always the case [53]. The corresponding cant deficiency is

$$h_d \approx \frac{2b_0}{g}\cdot a_y. \tag{4.10}$$

The maximum track plane acceleration allowed normally depends on the type of vehicle. In Sweden, the maximum permissible track plane acceleration for trains defined as so-called *category B* is $a_{y,lim}$ = 0.98 m/s^2, corresponding to the cant deficiency $h_{d,lim}$ = 150 mm. For tilting trains (*category S*), the limits are higher: $a_{y,lim}$ = 1.60 m/s^2, equivalent to $h_{d,lim}$ = 245 mm [53].

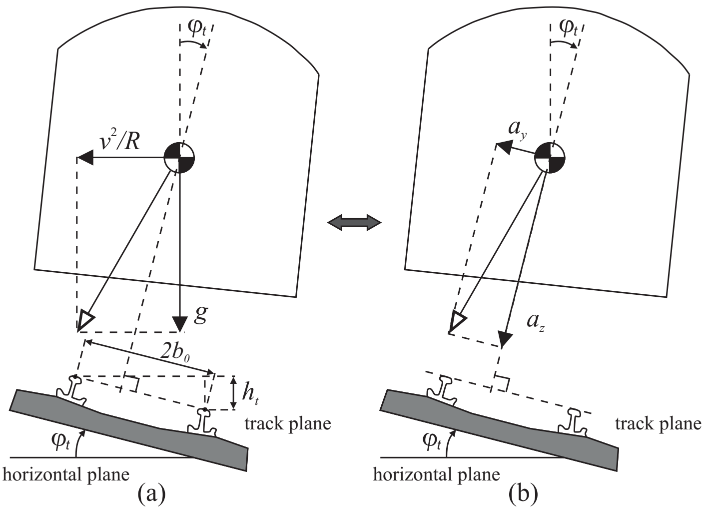

Figure 4.2: Definition of the track plane acceleration, a_y. The subfigures are equivalent to each other [53]. (a) Earth-following coordinate system. (b) Track-following coordinate system.

5 Summary of the present work

Paper A presents the initial development of an active lateral secondary suspension concept (ALS) based on sky-hook damping in order to improve vehicle dynamic performance, particularly on straight tracks. Furthermore, a Hold-Off Device (HOD) was included in the suspension concept in order to laterally centre the carbody above the bogies in curves at high track plane acceleration and hence to avoid bumpstop contact. Preparatory simulations as well as the subsequent on-track tests in 2007 show that the ALS system improves ride comfort compared to a passive system. The on-track tests further show that the carbody centring Hold-Off Device is able to suppress low-frequency periodic motions of the carbody when travelling in large-radius curves at high speed (high track plane acceleration). Furthermore, validation of the simulation model shows good agreement up to 10 Hz approximately, which is considered sufficient for the purpose.

Since results from the first paper showed very good potential of ride comfort improvements by means of the carbody centring Hold-Off Device, this was the main focus for the subsequent work, described in **Paper B**. This paper presents measurement results from on-track tests performed in 2008. The active secondary suspension concept was slightly modified compared to the one presented in the first paper. One modification was the implementation of a gyroscope to detect transition curves and hence to be able to switch off the dynamic damping in these sections. When comparing with corresponding measurement results from the previous year, it is shown that the relative lateral displacement in transition curves is lower when the active damping is not in use. Ride comfort in the actively suspended carbody is significantly improved compared to that in the passively suspended car. The satisfactory results led to implementation of the ALS system in long-term tests in service operation in the beginning of 2009.

In the first two papers, the well-known and relatively simple sky-hook damping was used as control algorithm. The next step was to evaluate a more advanced algorithm to examine whether even better results could be achieved. The choice fell on H_∞ control, and a quarter-car model was built in MATLAB, which is described in **Paper C**. The two control strategies were compared by evaluating how the lateral carbody acceleration and relative lateral displacement between the carbody and bogie were affected by the applied control force. Simulation results show that H_∞ control generates similar carbody accelerations at the same control force as sky-hook damping; however, the carbody displacement is somewhat lower with H_∞ control.

In **Paper D**, a full-scale rail vehicle model was used to investigate how lateral ride comfort is influenced by implementing the H_∞ and sky-hook damping control strategies. The linear quarter-car model from the previous paper is used to design the controller applied in the non-linear full-scale vehicle model. Simulations on a straight track section and in a large-radius curve show that significant ride comfort improvements can be achieved with active suspension—both with sky-hook damping and H_∞ control—compared to a passive system. However, on straight track and in the circular curve section, there is no large difference between the two control approaches regarding ride comfort. The advantage of H_∞ control in combination with the carbody centring Hold-Off Device is its ability to significantly reduce the relative lateral displacement in transition curves. Here, the sky-hook damping degrades the effect of the carbody centring device.

The first four papers focus on improving lateral ride comfort. The next step—described in **Paper E**—was to improve vertical ride comfort. The conventional vertical dampers in the secondary suspension were replaced by actuators. Simulation results show that significant ride comfort improvements can be achieved by dynamic control of the vertical and roll modes of the carbody compared to a passive system. Besides dynamic control, the actuators were able to generate quasi-static roll control between the carbody and bogies in curves. This allows for higher speeds in curves, without negatively affecting ride comfort. Further, the active suspension concept compensates for negative ride comfort effects if the structural stiffness of the carbody is reduced. Hence, considerable carbody weight reduction can be achieved, which compensates for the cost of the active suspension system. Moreover, the anti-roll bar may be removed to reduce the number of components.

In the final paper, **Paper F**, the previously investigated concepts of active lateral and vertical secondary suspensions were combined (ALS + AVS). Dynamic lateral, yaw and vertical control was applied together with quasi-static lateral and roll control. Simulation results show that both lateral and vertical ride comfort is improved for the active system run at a vehicle speed of 250 km/h compared to the passive system run at 200 km/h. Active suspension in one direction is independent of the other direction. Further, the active suspension concept compensates for negative ride comfort effects if the vertical air spring stiffness is increased. Hence, the air spring surge reservoir, requiring a lot of space, can be reduced or even removed.

Table 5.1 provides an overview of the controlled carbody modes (dynamic and quasi-static) along with the control strategies applied in the appended papers.

Table 5.1: The control configurations in the secondary suspension in each appended paper.

	Dynamic control	Quasi-static control	Control strategy
Paper A	Lateral, yaw	Lateral	Sky-hook
Paper B	Lateral, yaw	Lateral	Sky-hook
Paper C	Lateral	—	Sky-hook, H_∞
Paper D	Lateral, yaw	Lateral	Sky-hook, H_∞
Paper E	Vertical, roll	Roll	Sky-hook
Paper F	Lateral, yaw, vertical	Lateral, roll	Sky-hook

6 Conclusions and future work

Active technology in the secondary suspension of a rail vehicle can be used to achieve ride comfort improvements in cases where the conventional passive suspension cannot be further optimized. This thesis describes the development of an active secondary suspension concept for high-speed rail vehicles.

Firstly, an active lateral secondary suspension system (ALS) was developed by means of multi-body simulations and on-track tests with a two-car Regina train. Dynamic control of the lateral and yaw modes of the carbody in combination with quasi-static lateral carbody control provide significant lateral ride comfort improvements. By means of the quasi-static lateral control, the carbody is centred above the bogies in curves at high track plane accelerations and bumpstop contact is generally avoided. Ride comfort is thus positively affected and due to the reduced relative lateral displacements between the carbody and bogies, a wider carbody profile is possible, since the bumpstop play can be reduced. With the reduced lateral displacements, crosswind stability can also be improved.

Another important benefit originating from the carbody centring device is its ability to remove the unfavourable low-frequency periodic motions in large-radius curves at high speed (high cant deficiency), experienced in early on-track tests with passive secondary suspension. These motions have a considerably adverse effect on the ride comfort, and if they can be avoided, ride comfort in these curves is significantly improved.

Validation of the simulation model with the ALS system shows good agreement up to 10 Hz approximately, which is considered sufficient, since the simulations were performed with a rigid carbody instead of a flexible one. The test train with the ALS system fulfils the UIC 518 requirements for a maximum speed of 250 km/h and a maximum cant deficiency of 183 mm. Further, lateral ride comfort is better than for the passive vehicle at 200 km/h on the same track. The satisfactory measurement results from the on-track tests with the ALS system resulted in long-term tests in service operation, which have been carried out for more than two years (since 2009). Further, the ALS system has been included in two recent orders comprising more than 800 cars.

Secondly, an active secondary suspension system in the vertical direction (AVS) was developed by means of multi-body simulations with a flexible carbody. Dynamic control of the vertical and roll modes of the carbody in combination with quasi-static roll control provides significant vertical ride comfort improvements compared to a passive system. Furthermore, various modifications of the vehicle are possible without the ride comfort being negatively affected. The AVS system compensates for the negative ride comfort effects if the frequency of the first vertical bending mode of the carbody is reduced. Compared to a passive system with a bending mode frequency of 11 Hz, maintained or even improved vertical ride comfort can be achieved for an actively suspended carbody with a bending mode frequency of 8 Hz. Hence, if a lower structural stiffness of the carbody can be allowed, the number of added elements needed to stiffen up the carbody can be reduced, which, in turn, reduces the total carbody weight and cost. The weight reduction is less in relation to the stiffness reduction, which results in the reduced bending mode frequency. Moreover, the AVS system compensates for negative ride comfort effects if the air spring vertical stiffness is increased. This means that the total air spring volume can be reduced, resulting in a decrease or even a removal of the air spring surge reservoir.

The quasi-static roll control reduces the relative roll angle between the carbody and bogies in curves. The vertical actuators can be used to generate carbody tilting relative to the bogie plane up to around one degree, which reduces the lateral acceleration felt by the passengers. With the conditions in this study, this allows for a speed increase of 5 % with maintained ride comfort.

Finally, the combination of the ALS and AVS systems was studied in simulations. The control strategy included dynamic control of the lateral, yaw and vertical carbody modes in combination with quasi-static lateral and roll control of the carbody. Simulation results show that the active suspension in one direction does not affect the other direction. Further, ride comfort is significantly improved, both laterally and vertically, compared to a passive system, even when vehicle speed is increased from 200 to 250 km/h.

Future work

In transition curves, the carbody tends to be laterally displaced in relation to the bogie, due to the influence of the dynamic control. This effect could be reduced by applying an alternative control algorithm in curve transitions, possibly guided by a preview system based on track geometry data and a positioning system.

The AVS system for vertical ride comfort improvements has so far been investigated by means of multi-body simulations. The next step is to further develop the system so that on-track tests can be performed. A suitable actuator for control in the vertical direction has to be chosen. If only dynamic vibration control is to be implemented in the vertical direction—and not quasi-static control, which requires a higher maximum actuator force—a smaller and less costly actuator can be used. A smaller actuator generally has faster force response and operates in higher frequencies, which may improve the performance further.

Within the Gröna Tåget development programme other approaches of active technology have been investigated. Active radial steering and stability control (ARS) has been applied to meet the challenge of balancing the contradictory requirements of good running behaviour at high speeds on straight tracks on the one hand and good curving performance with low track and wheel damage on the other. Moreover, active control of the pantograph system to regulate the contact force is at an initial development stage. Although some work remains before active technology can be fully integrated in rail vehicles it is very likely that it will successively be used to a greater extent in the coming years.

References

[1] Hedrick, J.K., and Wormley, D.N., 1975. *Active Suspensions for Ground Transport Vehicles – A State of the Art Review*. Mechanics of Transportation Systems, ASME AMD, Vol. 15, pp. 21–40.

[2] Goodall, R.M., and Kortüm, W., 1983. *Active Controls in Ground Transportation – A Review of the State-of-the-Art and Future Potential*. Vehicle System Dynamics, Vol. 12, pp. 225–257.

[3] Goodall, R., 1997. *Active Railway Suspensions: Implementation Status and Technological Trends*. Vehicle System Dynamics, Vol. 28, pp. 87–117.

[4] Bruni, S., Goodall, R., Mei, T.X., and Tsunashima, H., 2007. *Control and Monitoring for Railway Vehicle Dynamics*. Vehicle System Dynamics, Vol. 45, pp. 743–779.

[5] Persson, R., 2011. *Tilting Trains – Enhanced Benefits and Strategies for Less Motion Sickness*. Doctoral Thesis, TRITA AVE 2011:26, ISBN 978-91-7415-948-6, Department of Aeronautical and Vehicle Engineering, Royal Institute of Technology (KTH), Stockholm, Sweden.

[6] Homepage of Gröna Tåget (Green Train): *www.gronataget.se*.

[7] Orvnäs, A., 2009. *Active Lateral Secondary Suspension in a High-Speed Train to Improve Ride Comfort*. Licentiate Thesis, TRITA AVE 2009:22, ISBN 978-91-7415-300-2, Department of Aeronautical and Vehicle Engineering, Royal Institute of Technology (KTH), Stockholm, Sweden.

[8] Jalili, N., 2001. *Semi-Active Suspension Systems*. The Mechanical Systems Design Handbook – Modeling, Measurement, and Control (Chap. 12), ISBN 0-8493-8596-2, CRC Press.

[9] Kjellqvist, P., 2002. *Modelling and Design of Electromechanical Actuators for Active Suspension in Rail Vehicles*. Doctoral Thesis, TRITA-ETS-2002-03, Department of Electrical Engineering, Royal Institute of Technology (KTH), Stockholm, Sweden.

[10] Goodall, R.M., and Mei, T.X., 2006. *Active Suspensions*. Handbook of Railway Vehicle Dynamics (Chap. 11), ISBN 0-8493-3321-0, Taylor and Francis.

[11] Pratt, I., 1996. *Active Suspension Applied to Railway Trains*. Doctoral Thesis, Department of Electronic and Electrical Engineering, Loughborough University.

[12] Hirata, T., Koizumi, S., and Takahashi, R., 1995. *H_∞ Control of Railroad Vehicle Active Suspension*. Automatica, Vol. 31, No. 1, pp. 13–24.

[13] Sasaki, K., Kamoshita, S., and Enomoto, M., 1994. *A Design and Bench Test of Multi-Modal Active Suspension of Railway Vehicle*. IECON'94, 20th International Conference on Industrial Electronics, Control and Instrumentation, Vol. 3, pp. 2011–2016.

[14] Pugi, L., Rindi, A., Bartolini, F., and Auciello, J., 2009. *Simulation of a Semi-Active Suspension System for a High Speed Train*. Proceedings of the International Symposium on Dynamics of Vehicles on Roads and Tracks (IAVSD'09), Stockholm, Sweden, August 17–21.

[15] Foo, E., and Goodall, R.M., 2000. *Active Suspension Control of Flexible-Bodied Railway Vehicles Using Electro-Hydraulic and Electro-Magnetic Actuators*. Control Engineering Practice, Vol. 8, pp. 507–518.

[16] Sugahara, Y., Kazato, A., Koganei, R., Sampei, M., and Nakaura, S., 2009. *Suppression of Vertical Bending and Rigid-Body-Mode Vibration in Railway Vehicle Car Body by Primary and Secondary Suspension Control: Results of Simulations and Running Tests Using Shinkansen Vehicle*. Journal of Rail and Rapid Transit, Vol. 223 (F6), pp. 517–531.

[17] Sugahara, Y., Watanabe, N., Takigami, T., and Koganei, R., 2011. *Vertical Vibration Suppression System for Railway Vehicles based on Primary Suspension Damping Control – System Development and Vehicle Running Test Results*. QR of RTRI, Vol. 52, No. 1, February, pp. 13–19.

[18] Allen, D.H., 1994. *Active Bumpstop Hold-Off Device*. Proc IMechE Conference Railtech 94, Paper C478/5/013.

[19] Pollard, M.G., 1983. *Active Suspensions Enhance Ride Quality*. Railway Gazette International, pp. 850–853, November.

[20] Conde Mellado, A., Casanueva, C., Vinolas, J., and Giménez, J.G., 2009. *A Lateral Active Suspension for Conventional Railway Bogies*. Vehicle System Dynamics, Vol. 47, No. 1, January, pp. 1–14.

[21] Zolotas, A.C., and Goodall, R.M., 2007. *Modelling and Control of Railway Vehicle Suspensions*. Mathematical Methods for Robust and Nonlinear Control (Chap. 13), ISBN 978-1-84800-024-7, Springer.

[22] Alfi, S., Bruni, S., Diana, G., Facchinetti, A., and Mazzola, L., 2011. *Active Control of Airspring Secondary Suspension to Improve Ride Quality and Safety against Crosswinds*. Journal of Rail and Rapid Transit, Vol. 225, No. 1, January, pp. 84–98.

[23] Asano, K., and Kajitani, Y., 2008. *Efforts for Greater Ride Comfort*. JR EAST Technical Review, No. 12, pp. 40–45.

[24] Kajitani, Y., Kato, H., and Asano, K., 2010. *Efforts for Greater Ride Comfort*. JR EAST Technical Review, No. 16, pp. 69–72.

[25] Pearson, J.T., Goodall, R.M., and Pratt, I., 1998. *Control System Studies of an Active Anti-Roll Bar Tilt System for Railway Vehicles*. Proc Instn Mech Engrs, Vol. 212, Part F, pp. 43–60.

[26] Zhou, R., Zolotas, A., and Goodall, R., 2009. *Integrated Tilt and Active Lateral Secondary Suspension Control*. Proceedings of the International Symposium on Speed-up, Safety and Service Technology for Railway and Maglev Systems (STECH'09), Niigata, Japan, June 16–19.

[27] Stribersky, A., Steidl, S., Müller, H., and Rath, B., 1996. *On Dynamic Analyses of Rail Vehicles with Electronically Controlled Suspensions*. Vehicle System Dynamics, Vol. 25 (Suppl), pp. 614–628.

[28] Stribersky, A., Kienberger, A., Wagner, G., and Müller, H., 1998. *Design and Evaluation of a Semi-Active Damping System for Rail Vehicles*. Vehicle System Dynamics, Vol. 28 (Suppl.), pp. 669–681.

[29] Stribersky, A., Müller, H., and Rath, B., 1998. *The Development of an Integrated Suspension Control Technology for Passenger Trains*. Proc IMechE, Vol. 212, Part F, pp. 33–42.

[30] Roth, P.A., and Lizell, M., 1996. *A Lateral Semi-Active Damping System for Trains*. Vehicle System Dynamics, Vol. 25 (Suppl.), pp. 585–598.

[31] Casini, C., Piro, G., and Mancini, G., 1996. *The Italian Tilting Train ETR 460*. IMechE Conference Transaction, International Conference on Better Journey Time – Better Business, Paper C514/058/96, pp. 297–305.

[32] Norinao, H., 1997. *Active and Semi-Active Suspensions Smooth 300 km/h Ride*. Railway Gazette International, pp. 241–242, April.

[33] Tanifuji, K., Koizumi, S., and Shimamune, R., 2002. *Mechatronics in Japanese Rail Vehicles: Active and Semi-Active Suspensions*. Control Engineering Practice, Vol. 10, pp. 999–1004.

[34] Hughes, M., 2006. *Fastech 360 Twins Herald Speed-Up to the North*. Railway Gazette International, pp. 262–264, May.

[35] Personal e-mail communication with Dr. Eng. Katsuya Tanifuji, Department of Mechanical Engineering, Faculty of Engineering, Niigata University, Japan, 2007.

[36] Düsing, M., Lü, Y., and Jakob, J., 1994. *Wankst ütze für Schienenfahrzeuge*. German Patent DE4311521C1.

[37] Sugahara, Y., Watanabe, N., Takigami, T., and Koganei, R., 2011. *Development of Primary Suspension Damping Control System for Suppressing Vertical Bending Vibrations of Railway Vehicle Car Body*. Proceedings of the 9th World Congress on Railway Research (WCRR 2011), Lille, France, May 22–26.

[38] Personal e-mail communication with Dr. Eng. Yoshiki Sugahara, Vehicle Noise and Vibration Laboratory, Vehicle Structure Technology Division, Railway Technical Research Institute (RTRI), Japan, 2011.

[39] Karnopp, D., 1983. *Active Damping in Road Vehicle Suspension Systems*. Vehicle System Dynamics, Vol. 12, pp. 291–316.

[40] Li, H., and Goodall, R.M., 1999. *Linear and Non-Linear Skyhook Damping Control Laws for Active Railway Suspensions*. Control Engineering Practice, Vol. 7, pp. 843–850.

[41] Hohenbichler, N., and Six, K., 2006. *Potenziale Aktiver und Semiaktiver Skyhook-Regelgesetze in der Sekundärfederstufe von Schienenfahrzeugen* (in English: *Potentials of Active and Semiactive Skyhook Control Laws Applied to the Secondary Suspensions of Railway Vehicles*). Automatisierungstechnik, Vol. 54, No. 3, pp. 130–138.

[42] Baier, M., Hohenbichler, N., Six, K., and Abel, D., 2007. *Benefit of Preview Data for Comfort Improvement in Railway Vehicles*. ISBN 978-80-8070-679-1, EURNEX–Zel, pp. 47–54.

[43] Glad, T., and Ljung, L., 2003. *Reglerteori – Flervariabla och Olinjära Metoder* (in English: *Control Theory – Multivariable and Nonlinear Methods*). ISBN 91-44-03003-7, Second Edition.

[44] Zeng, J., Zhang, W.H., Dai, H.Y., Wu, X.J., and Shen, Z.Y., 1998. *Hunting Instability Analysis and H_∞ Controlled Stabilizer Design for High Speed Railway Passenger Car*. Vehicle System Dynamics, Vol. 28 (Suppl.), pp. 655–668.

[45] Orukpe, P.E., Zheng, X., Jaimoukha, I.M., Zolotas, A.C., and Goodall, R.M., 2008. *Model Predictive Control Based on Mixed H_2/H_∞ Control Approach for Active Vibration Control of Railway Vehicles*. Vehicle System Dynamics, Vol. 46 (Suppl.), pp. 151–160.

[46] Orvnäs, A., Andersson, E., and Persson, R., 2007. *Development of Track-Friendly Bogies for High Speed – A Simulation Study*. Publ 0703, ISBN 978-91-7178-726-2, KTH Railway Group, Stockholm, Sweden.

[47] UIC 518, 2009. *Testing and Approval of Railway Vehicles from the Point of View of Their Dynamic Behaviour – Safety – Track Fatigue – Ride Quality*. UIC, 4th edition.

[48] CEN, EN 12299, 2009. *Railway Applications – Ride Comfort for Passengers – Measurement and Evaluation*. European Committee for Standardization (CEN), Brussels.

[49] SIMPACK Documentation, Version 8903, X–PF:5 Ride Index VDI and ISO.

[50] Sperling, E., 1941. *Verfahren zur Beurteilung der Laufeigenschaften von Eisenbahnwesen*. Organ f.d. Fortschritte des Eisenbahnwesens, 12, pp. 176–187.

[51] Sperling, E., and Betzhold, C., 1956. *Beitrag zur Beurteilung des Fahrkomforts in Schienenfahrzeugen*. Glasers Annalen 80, pp. 314–320.

[52] ORE, C 116/RP 8/D, 1977. *Wechselwirkung zwischen Fahrzeugen und Gleis*. Office of Research and Experiment of the International Union of Railways, Utrecht, The Netherlands.

[53] Andersson, E., Berg, M., and Stichel, S., 2007. *Rail Vehicle Dynamics*. Text book, Division of Rail Vehicles, Department of Aeronautical and Vehicle Engineering, Royal Institute of Technology (KTH), Stockholm.

Appendix A – Notations

Symbols

a^{w}	frequency-weighted acceleration (m/s^2)
a^{wrms}	*rms* of frequency-weighted acceleration (m/s^2)
a_y	lateral acceleration in track-following system (m/s^2)
a_z	vertical acceleration in track-following system (m/s^2)
b_0	half track base (m)
c	damping (Ns/m)
f	frequency (Hz)
F	force (N)
φ_c	tilt angle (rad, °)
φ_t	track cant angle (rad, °)
g	acceleration of gravity (m/s^2)
h_d	cant deficiency (m)
h_t	track cant (m)
k	stiffness (N/m)
m	mass (kg)
ω	angular frequency (rad/s)
R	radius (m)
t	time (s)
T	integration time (s)
U	voltage (V)
v	speed (m/s)
z	vertical displacement (m)
$\dot{z}$	vertical velocity (m/s)
$\ddot{z}$	vertical acceleration (m/s^2)

Abbreviations

ALS	active lateral secondary suspension
ARS	active radial steering and stability
AVS	active vertical secondary suspension
HOD	Hold-Off Device
ISO	International Organization for Standardization
KTH	Kungliga Tekniska Högskolan (Royal Institute of Technology)
MBS	multi-body system
rms	root mean square
UIC	Union Internationale des Chemin de fer (International Union of Railways)
Wz	Wertungszahl